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Assessment of Desiccant Dehumidification:

Technical Feasibility Report

Prepared by:

Sustainable Design & Consulting LLC

Prepared for:

University of Hawaii at Manoa, Hawaii Natural Energy Institute

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ASSESSMENT OF DESICCANT DEHUMIDIFICATION

Project Deliverable 4:

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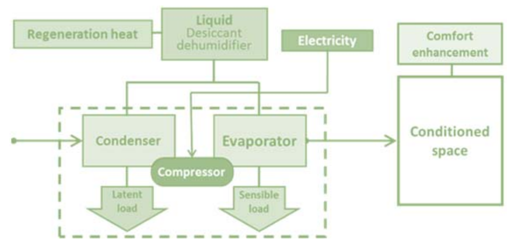
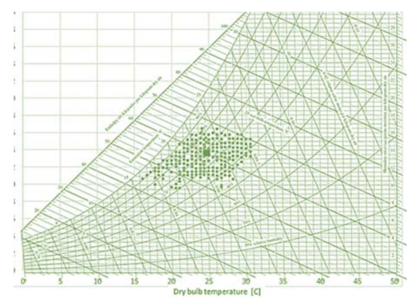
FINAL





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RCUH P.O. #Z10117197

HNEI

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The authors would like to thank Hawaii Natural Energy Institute for the funding of this project. The authors believe that desiccant cooling applications can play a significant part making building conditioning more energy efficient and foster the implementation of more environmentally friendly ways to provide occupant comfort in buildings.

ABBREVIATIONS

AC	Air-conditioning
ACHR	Air Conditioning, Heating and Refrigeration NEWS
AHU	Air-handling Unit
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BET	Brunauer–Emmett–Teller
BRR	Benefit to Risk Ratio
CHP	Combined Heat and Power
CA	conditioned air (CA')
COP	Coefficient of performance
DP	Dew point
DX	Direct Expansion Air Conditioning System
DEAC	Direct Evaporative Air Coolers
DEC	Direct evaporative cooler
DEVAP	Desiccant Enhanced Evaporative Air Conditioning
EER	Energy Efficiency Ratio Engelard Titanium Silicate (ETS)
EAC	Evaporative Air Cooling external source of energy (Q)
GTI	Gas Technology Institute
GCI	Green Cooling Initiative
HNEI	Hawaii Natural Energy Institute
HR	Humidity Ratio
HVAC	Heating Ventilation Air Conditioning
IC	Internal Combustion
IEAC	Indirect Evaporative Air Coolers
LDDX	Liquid Desiccant DX System
IDECOAS	Indirect and Direct Evaporative-Cooling-Assisted Outdoor Air System
IRR	Internal Rate of Return
LD	Liquid Desiccant
LDDX	Liquid Desiccant Direct Expansion
IEC	Indirect Evaporative Cooler
LiCl	Lithium Chloride

LCST	Lower Critical Solution Temperature
M-cycle	Maisotsenko (M) -Cycle
O&M	Operations & Maintenance
OA	Outdoor (or outside) Air
PV	Photovoltaic System, also solar-PV Power System
RH	Relative Humidity
SHR	Sensible Heat Ratio
SAC	Solar air-conditioning
SDC	Sustainable Design & Consulting LLC
SI	International System of Units
TSA	Temperature Swing Adsorption
VAC	Vapor-Compression Air Conditioning
WBT	Wet-bulb temperature
VAV	Variable Air Volume
VLI	Ventilation Load Index

UNITS

atm	Atmosphere pressure
BTU	British Thermal Unit
BTUH/sqft	BTU per square feet
°C	Degree Celsius
CFM	Cubic feet per minute
F	Fahrenheit
K	Kelvin
kW	kilo Watt
kJ	kIlo Joule
MPa	Mega Pascale
square feet	Square Feet

TABLE OF CONTENTS

SECTION 1 - EXECUTIVE SUMMARY AND OVERALL CONCLUSIONS AND RECOMMENDATIONS

The present feasibility report is the deliverable of Part Two of the project “Assessment of Desiccant Dehumidification” which Sustainable Design & Consulting (SDC), LLC is performing for Hawaii Natural Energy Institute (HNEI). Part One of the project presented a literature review and provided an overview of desiccant technology and their potential integration into systems that provide thermal comfort in building conditioning, specifically for the hot and humid sub-tropical climate of Hawaii

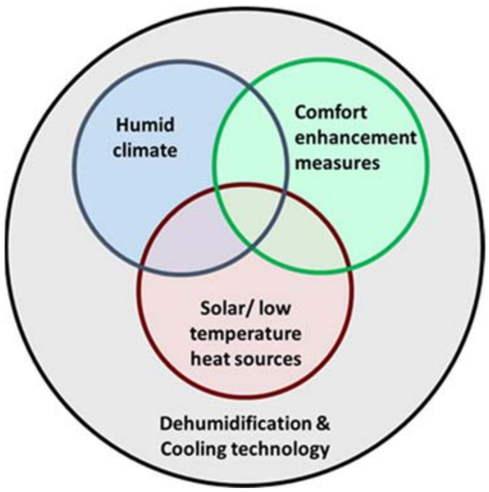
This report presents the feasibility assessment of desiccant dehumidification in conjunction with building conditioning and cooling. Such use of desiccant dehumidification with cooling and/or ventilation in buildings is often referred to as “desiccant cooling”, a term which is used throughout this report. Only those desiccant cooling systems that are suitable and cost effective to work in conjunction with building and space conditioning were considered.

The literature and technology identified desiccant dehumidification and cooling technologies which can be combined to provide effective space conditioning systems. One of the main advantages of using desiccant dehumidification in building conditioning is the ability to separate main functions of the air conditioning (AC) system. Conventional AC systems must perform three main functions, which are the removal of the latent and sensible great load, e.g. dehumidification and cooling, as well as ventilation. Humidity control in conventional AC systems is based on cooling the supply air to below dew point temperatures in order to induce condensation of the moist air on cold coils. This dehumidification process is referred to as cooling based dehumidification.

Optimizing these three functions, while outdoor temperature and humidity changes and indoor heat and humidity loads are continuously changing, is inherently difficult with conventional AC systems. For example, a common challenge in hot and humid climates is overcooling of spaces, which is a result of dedicating high cooling capacity to extract humidity on the cooling coils. A remedy to excessively cold air is reheating the supply air before it reaches the spaces, which requires significant reheat energy.

Desiccant dehumidification avoids excessively cold supply air by using sorption processes which remove water vapor from the supply air without cooling the air below its dew point. Apart from avoiding overcooling and reheat and the associated energy demand, desiccant dehumidification increases the controllability of the indoor thermal conditions and indoor air quality, since the three AC functions can be individually controlled. For example, the thermal (sensible) load of the exterior air and the added internal load might be low, while the latent load is high. A desiccant cooling system can be configured to regulate both the sensible and latent indoor conditions within the specified comfort range, without the need to provide excess cooling capacity. As a result, energy for reheat can be saved.

APPROACH OF THE FEASIBILITY STUDY: The feasibility study used a framework of four assessment categories which were applied on desiccant cooling systems specifically for Hawaii climatic conditions. As the figure below indicates, these categories included the inherent characteristics of desiccant cooling technologies as the encompassing main category and three sub-categories which represent Hawaii specific aspects. The three sub-categories are the hot and humid climate in Hawaii, the use of energy saving solar or low-temperature heat sources and comfort enhancement measures. The figure illustrates that the Hawaii specific subcategories are embedded in the inherent technology performance characteristics and overlap, therefore affecting each other.



Four assessment categories for the feasibility report

Inherent technology aspects of desiccant cooling were discussed. Desiccant dehumidification can be carried out in parallel of in sequential order with the sensible cooling. Liquid or solid desiccant systems were analyzed and both types of desiccant systems are suitable dehumidification technologies for controlling the latent heat load for buildings. Desiccant dehumidification does not remove sensible heat, in fact heat of sorption is added to air passing through the desiccant material, therefore cooling is typically required. This means that the removal of sensible heat from the supply air is an indispensable part of the desiccant cooling systems.

The climate in Hawaii is hot and humid, year round. The present feasibility study used a typical meteorological year (TMY) file of Honolulu to determine the relevant climatic properties to assess feasibility of desiccant cooling. The so-called Ventilation Load Index (VLI) parameter was used to quantity the latent and sensible load of the external air supply on an annual basis. The VLI for Honolulu is one the highest in the nation, with the latent load being six times higher than the sensible load. This high VLI emphasizes the importance to implementing energy efficient desiccant dehumidification instead of the energy intensive cooling based dehumidification. The analysis of the climatic record for Honolulu used two records, one for the entire year and the second for the daytime hours. The results indicated different dehumidification and cooling needs for the year round and for the daytime weather data, with higher dry bulb temperatures and lower relative humidity levels during daytime hours.

The potential of solar heat for desiccant regeneration and thermally driven chillers was evaluated. The potential for the use of solar heat in desiccant cooling is significant in Hawaii. Similarly, heat derived from combined heat and power (CHP) facilities can be a significant and energy saving source for use in space conditioning. Hawaii has one of the highest energy costs in the nation, and therefore high energy savings directly increase the competitiveness of desiccant cooling applications. The report addresses the fact that at present Hawaii derives about 80% of its electricity from petroleum and coal. These fuels can cause logistical challenges for CHP units, and natural gas would be the fossil fuel of choice. There are plans to ship natural gas to Hawaii for energy conversion, and such changes in the energy profile of Hawaii would open up significant application potential for desiccant dehumidification either to fully or partly supply the a clean and reliable heat source for desiccant regeneration. The energy source of choice for desiccant cooling, though, would be solar or other low temperature sources.

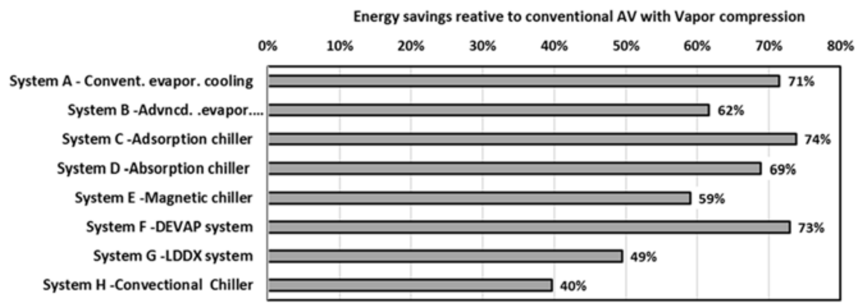
Comfort enhancement measures can add to the feasibility of desiccant cooling, yet comfort issues are typically only marginally considered. Ceiling fans represent a comfort enhancement measure which is specifically suitable for Hawaii. The increased indoor air movement generated by state-of-the-art ceiling fans provide a highly energy efficient cooling effect. Spaces equipped with conventional AC systems can only marginally take advantage of the ceiling fans due to challenges to control humidity at higher indoor air temperatures, which are allowed when using ceiling fans for added convective cooling. Buildings using desiccant cooling systems, on the other hand, can take full advantage of the significant advantages of ceiling fans. The potential contribution of ceiling fans in desiccant cooling was quantified in the present feasibility report and used as a main criterion in determining energy savings.

Eight candidate systems were selected to determine the feasibility of desiccant cooling for Hawaii. These candidate systems were combinations of either solid or liquid desiccant dehumidification and a range of sensible cooling systems. The differentiating factor of the desiccant cooling candidate systems were thus the type of sensible cooling units. The eight systems were as follows:

- Candidate system A. Desiccant and Direct / Indirect Evaporative Cooling downstream of desiccant dehumidification
- Candidate system B. Enhanced evaporative cooling - M-Cycle downstream of desiccant dehumidification
- Candidate system C. Adsorption chiller downstream of desiccant dehumidification
- Candidate system D. Absorption chiller downstream of desiccant dehumidification
- Candidate system E. Magnetic (magneto caloric effect) chiller downstream of desiccant dehumidification
- Candidate system F. Integrated DEVAP membrane system,
- Candidate system G. Liquid-Desiccant Direct-Expansion (LDDX) air-conditioner, integrated dehumidification and sensible cooling
- Candidate system H. Conventional chiller, downstream of desiccant dehumidification

The eight desiccant cooling candidate systems represented technologies with significantly different types of operation, including fuel requirement and capability to use solar or other low temperature sources, and technical and market maturity. Some of the technologies have been commercially available and are tested in actual operation, whereas others are emerging, yet very promising cooling technologies. One of the candidate systems uses the conventional vapor compression cycle in conjunction with desiccant dehumidification. The significant energy savings determined for this system configuration underlines the great potential of desiccant cooling applications and showcase opportunities to implement desiccant cooling in building retrofits.

RESULTS OBTAINED: The main result of the feasibility study, besides identifying viable desiccant cooling system integrations, was determining which candidate system offered most advantages and therefore is recommended to be most suitable for the desiccant cooling applications in Hawaii. The resulting ranking used a wide range of ranking criteria. Costs were not used as ranking criteria since some of the technologies are emerging technologies with no detailed and confirmed first costs, while others technologies analyzed have a cost track record. Instead the projected saved energy relative to a conventional AC system was used as a main criterion. The energy savings were calculated relative to a typical conventional AC-unit using cooling based dehumidification. The resulting energy savings are depicted in the figure to the left.



Energy savings determined for eight candidate systems

The ranking used both discrete and continuous ranking scores. Both types of ranking scores were converted to a percentage based score in order allow a quantitative assessment and comparison between alternatives, and therefore determining the candidate systems with the highest score. The ranking included a risk/benefit analysis, which balanced anticipated benefits with risks and derived a quantitative assessment the benefit to risk ratio (BRR). The values of BRR were used in the ranking.

System	Description	Overall ranking score	Overall Rank	Lead over lowest rank
C	Adsorption chiller	73%	1	32%
G	LDDX system	69%	2	29%
A	Conventional Evaporative cooling	67%	3	26%
D	Absorption chiller	63%	4	22%
B	Advanced Evaporative cooling (M-cycle)	59%	5	18%
F	DEVAP system	58%	6	17%
H	Conventional chiller	54%	7	13%
E	Magnetic chiller	41%	8	0%

Resulting overall ranking for a NEW installation

The overall results of the ranking are depicted in the Table to the left. This ranking assumed a new installation in buildings. The combination of desiccant dehumidification and downstream adsorption chiller was given the highest overall rank, followed by an innovative integrated liquid desiccant cooling system and a conventional evaporative cooling system with upstream desiccants.

The predicted ranking for the eight candidate systems was also determined for building retrofitting. An effective retrofitting can increase the feasibility of desiccant cooling applications, since it would allow existing buildings to be retrofitted to take advantage of the significant energy savings which have been established for the desiccant cooling systems considered in this report. When the ranking criteria were readjusted to account for retrofitting the most promising system was the integrated desiccant and cooling LDDX system, followed by the adsorption chiller with upstream desiccant unit. Interestingly, the system with a conventional chiller and upstream desiccant system was determined to have the third best overall ranking score for retrofitting.

MAIN CONCLUSIONS AND RECOMMENDATIONS:

- Desiccant cooling is a very attractive and energy saving option to provide good indoor comfort for the climatic and logistic conditions in Hawaii. Hawaii's typical climatic conditions create a year round need to cool and dehumidify conditioned buildings and spaces.
- Desiccant cooling technology has been identified that can be powered by solar and other low temperature heat sources, such as combined heat and power (CHP). Since space conditioning has such a high energy demand and causes significant electricity peak demand in Hawaii the achievable energy savings are powerful incentives to foster implementing desiccant cooling in Hawaii.
- The fact that desiccant cooling separates sensible and latent heat removal provides a wide array of opportunities to combine desiccant and cooling technologies and tailor the resulting systems to the specific needs of the building designer and operator.
- Separating the main functions of AC-systems not only create significant energy saving opportunities, but also provide opportunities to improve indoor comfort and healthy indoor air quality. Specifically, desiccant cooling system can provide optimized conditions for using ceiling fans in conditioned spaces, where conventional AC would exclude their optimized use.
- This feasibility study should provide a starting point for more detailed investigations of the use of desiccant cooling systems in Hawaii. The scope of the present feasibility study precluded detail energy and costs analysis, because of the introductory nature of this investigation. In follow-up investigations a closer cooperation with vendors or technology developers would help to add an important practical dimension, possibly culminating in a demonstration or pilot project of desiccant cooling systems in Hawaii.
- The nation and specifically Hawaii has a need to improve the energy performance of building and especially the energy intensive building and space conditioning. The literature reviewed and used in this research project has shown that there are intensive efforts underway, at the government and private level, to depart from the conventional way of providing space conditioning. The goal will be to derive energy efficient working systems of to provide good and healthy indoor

environments. This feasibility study has shown that desiccant cooling systems are good candidates to achieve this important goal.

SECTION 2 – BACKGROUND AND HYPOTHESIS

The conditioning of buildings with appropriate HVAC systems is becoming more prevalent as the demand for cooling and good indoor air quality increases. The increase of the number and capacities of HVAC systems stresses the need for more energy and cost efficient HVAC technologies in order to keep the environmental impact from HVAC as low as is achievable. Conventional HVAC have been providing building and space conditioning for many decades, but the increasing demand for HVAC and the need to curb environmental impact from harmful refrigerants and high energy demand.

Conventional HVAC system processes have been basically unchanged, with optimization of system components striving for more energy efficiency and lower impacts, to provide the same or better occupant comfort.

Many innovative chiller and cooling technologies have been introduced to replace the vapor compression process that is inherent to conventional HVAC systems. Irrespective of what chiller or cooling technology is used, the cooling based dehumidification of HVAC systems that serve residential and commercial building have been mostly regarded as more cost efficient than an alternative desiccant dehumidification approach.

Using desiccant rather than cooling based dehumidification changes not only the cooling process in providing building and space conditioning, but also opens up new HVAC approaches for the built environment. This report will show that HVAC systems with integrated desiccant dehumidification, or more specifically **desiccant cooling**, can provide significant energy and cost benefits and lower the environmental impact. Since mechanical building conditioning represents about 40% of the worldwide energy consumption, significant energy savings will go a long way to contribute to combat energy scarcity and harmful pollutants that are associated with conventional chillers and energy production.

While desiccant cooling is being proposed for various regions and climate zones, this report focuses on determining the feasibility of desiccant cooling applications in Hawaii’s climate.

SECTION 3 - PARADIGM SHIFT OF USING DESICCANT COOLING SYSTEMS

The use of desiccant dehumidification transforms the paradigm of conventional and cooling based HVAC systems. This section describes what the transformation towards desiccant cooling systems entails and what the fundamental process changes can open up opportunities to significant energy and costs savings.

3.1 Primary Functions of HVAC Systems

The acronym HVAC stands for heating, ventilation, and air conditioning. The primary function of the HVAC system is to maintaining comfortable indoor conditions. This is being accomplished by the regulation of indoor humidity, air flow, as well as temperature so that these parameters stay within acceptable limits.

Figure 3.1.1 illustrates the three main functions of HVAC systems. Since the focus of this feasibility study is on typical conditions in Hawaii, temperature and humidity control will only consider cooling and dehumidification, respectively.

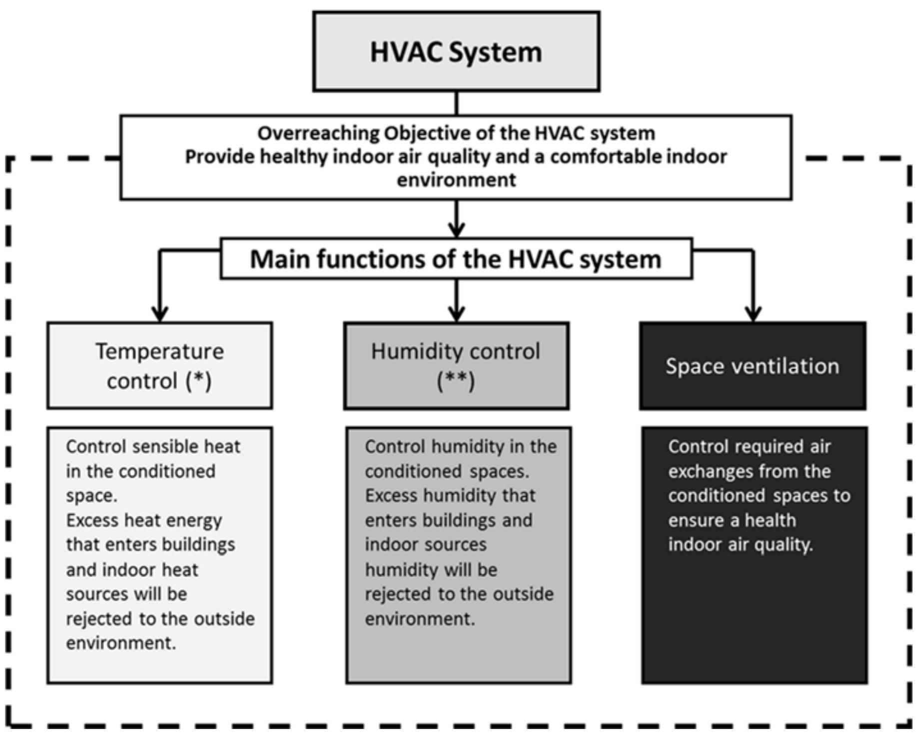


Figure 3.1.1: Main functions of the HVAC systems

(*) Temperature control is only addressed for Hawaii climate, which means no heating will occur.
(**) Humidity control is only addressed for Hawaii climate, which means humidity has to be reduced

As shown in Figure 3.1.1, the three main functions of the HVAC system for a typical Hawaii condition are described as follows.

- The temperature control is done by removing excess sensible heat from the conditions space. Heat energy is added to the interior space through solar gain, warm outside air supply and internal heat sources, such as lighting, electric appliances and occupants. In order to achieve heat balance under steady state conditions the heat entering and leaving has to be the identical.
- The humidity control is accomplished by removing moisture that enters the HVAC system with the supply air or that is generated inside the space by such water vapor sources as occupants and exposed water surfaces.
- Space ventilation assures that enough outside air is provided to expel pollutants from the ventilated space. The minimum ventilation flow rate is required by local code, typically following ASHRAE guidelines.

Figure 3.1.2 shows the primary technologies that are available to accomplish the main function of the HVAC system.

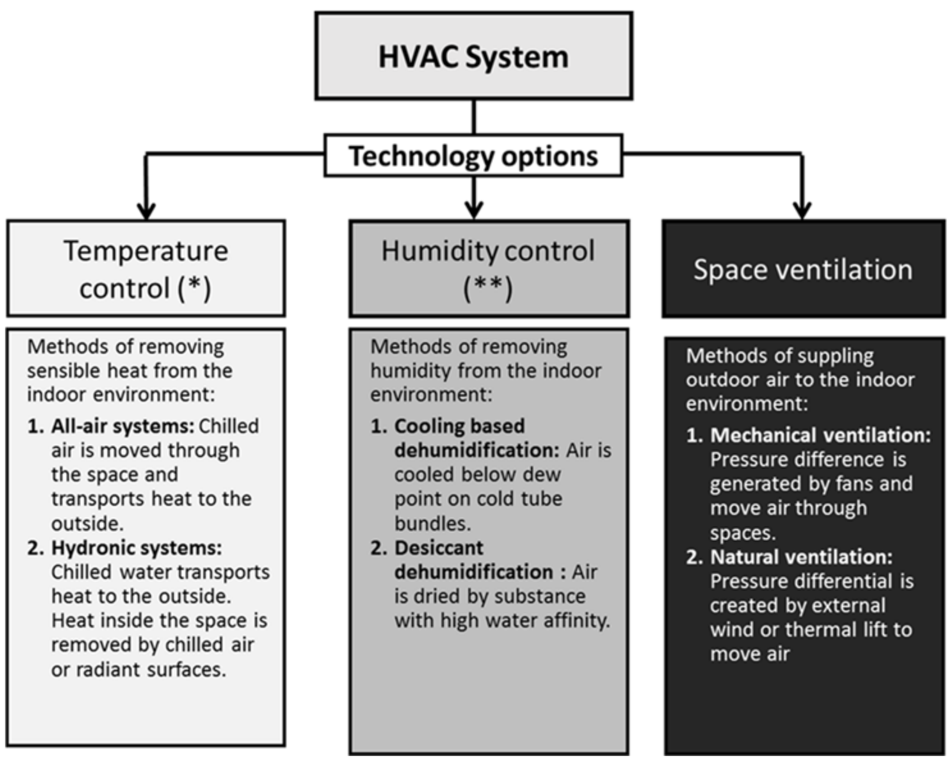


Figure 3.1.2: Primary technologies used in larger scale HVAC systems to accomplish main functions of the HVAC systems

(*) Temperature control is only addressed for Hawaii climate, which means no heating will occur.
(**) Humidity control is only addressed for Hawaii climate, which means humidity has to be reduced

The technologies shown in Figure 3.1.2 are described in more detail:

Temperature control – All air systems: The most common technology to remove heat form a conditioned space is through the use of chilled air circulating through the conditioned space. The indoor air flow is induced by mechanical fans. For most applications the larger portion of internal air for is recirculated air, with a quantity of fresh outdoor air being mixed to the recirculated air and an equal quantity discharged from the space. The air passes over cooled tube bundles that remove sensible heat from the air; cooling of the air below its dew point also removes humidity. The air inside the space cool objects, including occupants, by convective heat transfer between the cold air and the surfaces of these objects. The induced air flow requires significant mechanical work, and therefore electric energy for air fans.

Temperature control – Hydronic systems: Hydronic systems differ from all-air systems in the way heat energy is transported from the inside to the outside. Instead of air, hydronic systems use chilled water or another working fluid to transport heat. Transport of heat through a liquid agent is more efficient than through air because of higher specific heats of liquids and the fact that the liquid pumps require much less electricity than air fans. Hydronic systems typically use radiant cooling surfaces or chilled beams for the primary removal of heat.

Humidity control – Cooling based dehumidification: The overwhelming method of providing dehumidification is through using cooling based dehumidification. In this process air is passing over cold coils (e.g. tube bundles) which have surface temperatures below dew point. As the air passes over the cold surfaces of the tube bundle, water vapor condenses on the cold surfaces and is thus removed by drainage of the condensate.

Humidity control – Desiccant dehumidification: An alternative to the leading cooling based dehumidification is the use of desiccant material to dry the inside air to the desired indoor humidity level. The desiccant material works on the principal of differential partial vapor pressures to remove humidity from the air. Differently from the cooling-based dehumidification, the air temperature does not have to be reduced below the dew point in order to initiate vapor condensation.

Space ventilation – Mechanical ventilation: Air is forced into the conditioned space by means of air fans. The fans collect outside air and provide the required increase of pressure to maintain a certain air flow rate in a system of air ducts. Mechanical ventilation is not dependent on external climatic conditions to provide satisfactory ventilation rates, since the fans are the only driving force of the indoor air movement.

Space ventilation – Natural ventilation: In naturally ventilated buildings or spaces, indoor air movement is induced by external conditions. The two basic mans of natural ventilation are cross ventilation and stack effect ventilation. Under the cross ventilation mode, a pressure differential is generated on the outside skin of the building, which induces air movement through the building from higher to lower pressure opening in the building envelope. Under the stack ventilation mode, buoyancy induced air flow results from warmer air rising inside the building and venting at higher building openings, while colder and fresh outside air enters the building.

Discussion of the technologies described above:

Conventional HVAC systems are of the “all-air” type, which combine all three HVAC functions in one system. These systems provide chilled air to the conditions spaces, thereby removing sensible and latent heat for thermal comfort, and providing outdoor air for required ventilation to assure indoor air quality. As it turns out, the simultaneous performance of all three functions makes it difficult to optimize the system performance. For instance, usually the required ventilation air flow rate is smaller than the flow rate of chilled air for sensible cooling, leading to control problems. Since sensible and latent heat removal is carried out at the same time on the surface of the cooling coils, and with the latent heat removal larger than sensible load, over cooling of the space can occur. This is a problem in particular in humid climates such as Hawaii.

Since the removal of latent load required requires colder coil temperatures that would be required for sensible load removal, the air downstream of the cold coils is too cold and must be reheated to avoid problems with quantities of cold air causing comfort problems. In short, due to the difficulty of optimizing all three main functions of conventional HVAC systems in one setting, the system performance has to be corrected by applying excess energy, for instance reheat the cold air coming from the cold coils.

Faced with the described control difficulty of optimizing the conventional all-air system and improving energy performance alternative processes have been proposed, which separate the three HVAC functions and apply different technologies. Separating sensible and latent load removal is one of the most effective alternatives to the conventional all-air HVAC.

One of the main benefits of deviating from the conventional cooling based dehumidification and using desiccant dehumidification is the avoidance of the below dew point chilled air temperatures which are needed to effective condensation on the coils. While desiccant dehumidification is equally more effective at lower temperatures, these process temperatures do not have to be as low. The first obvious benefit of desiccant dehumidification is the avoidance of reheat.

Since the latent load removal is carried out by the desiccant material, the sensible load removal can occur at higher air temperature or with other than forced air cooling technologies, such has chilled beams or radiant surfaces.

In addition, under the process of separate sensible and latent cooling, the outside airflow rate required for ventilation can be much lower than the make-up air, which means the portion of the internal airflow that is fresh and outside air. Lower air flow rates conveyed in smaller air ducts can be better controlled than larger flow rates in large ducts, as is the case for the conventional all-air HVAC systems.

Options to take advantage of energy savings through separate sensible and latent removal through the use of desiccant material will be presented later on in the report.

3.2 Separation of HVAC Functions in Desiccant Cooling Systems

Desiccant cooling systems allow separation of the three main HVAC functions of sensible cooling, dehumidification (latent load removal) and ventilation. Figure 3.2.1 illustrates the separation of the functions.

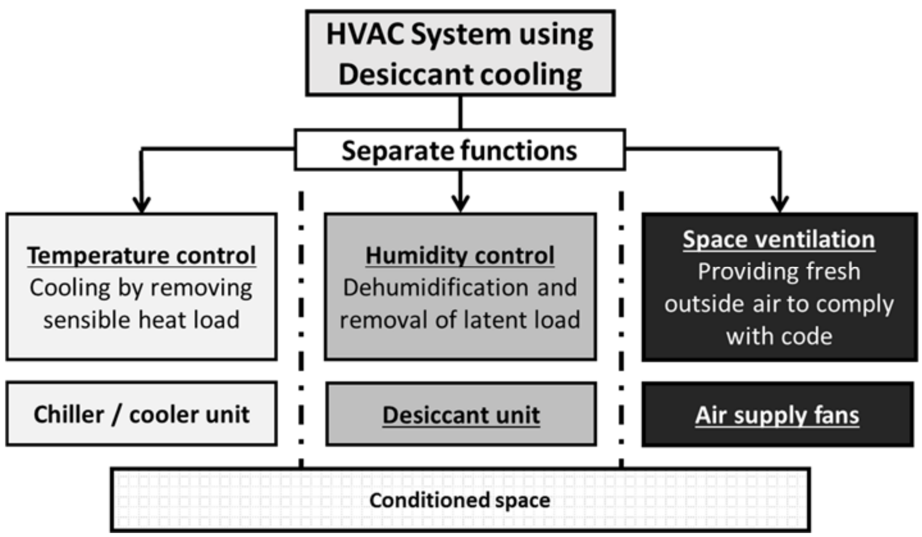
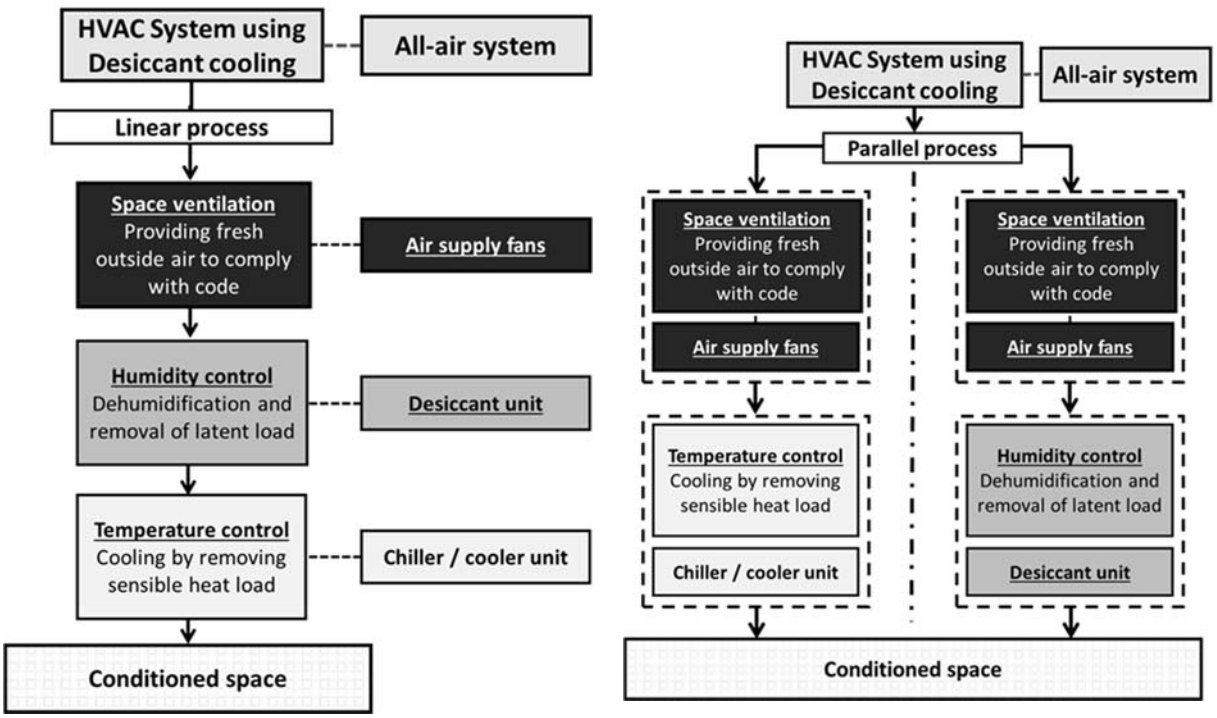


Figure 3.2.1: Separation of main HVAC function in a desiccant cooling systems

The ventilation function can either be combined with the sensible cooling and/or dehumidification functions. Figures 3.2.2 and 3.2.3 illustrate possible integrations of sensible cooling and latent load removal with space ventilation. Basically we can differential two options, all-air systems and hydronic systems.

Figure 3.2.2 shows the possible configurations of an <u>all-air desiccant cooling system</u> with (a) and (b) showing linear and parallel sensible and latent load removal processes. The all-air systems will be described later on in the report. Figure 3.2.2 (a) illustrates the linear systems where one ventilation air flow passing through latent and sensible load removal successively. Figure 3.2.2 (b) illustrates the parallel process using two ventilation air flows, where one is cooled and the other dehumidified and where the flow rate of ventilation air entering the conditioned space can be individually controlled.

Figure 3.2.3 illustrates the possible configurations of a desiccant cooling system with a hydronic, which means radiant of chilled beam sensible cooling. This form of sensible cooling supplies the cooling capacity inside the conditioned space and does not depend on externally chilled air.



(a) Linear process of separate functions in the desiccant based HVAC system

(b) Parallel process of separate functions in the desiccant based HVAC system

Figure 3.2.2: Two basic process schemes of an all-air HVAC system based on desiccant cooling

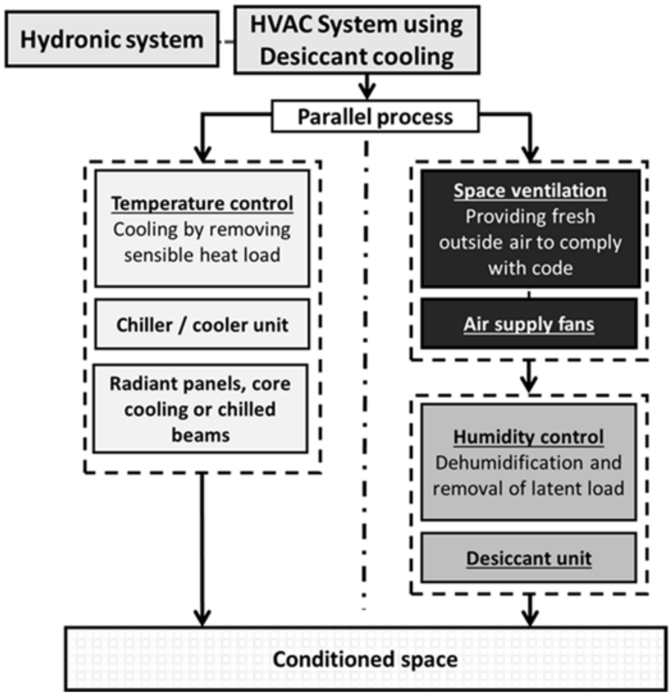


Figure 3.2.3: Basic process scheme for a hydronic HVAC based on desiccant cooling

3.3 Summary of Section Three

Summary: The separation of latent and sensible load removal, e.g. dehumidification and cooling, in HVAC systems based on desiccant cooling processes open up new opportunities for energy efficient space conditioning. The fundamental difference between conventional HVAC and systems based on desiccant cooling is the required air supply temperature, which must be below dew point for the conventional cooling based dehumidification and the warmer desiccant dehumidification. In addition, the separation of basic functions enables more flexible HVAC control models and an optimization of the three basic HVAC functions, which was not achievable to such extent in conventional HVAC systems.

SECTION 4 – HOW CONVENTIONAL HVAC SYSTEM COMPONENTS CAN BE EFFECTIVELY INTEGRATE IN DESICCANT COOLING

This section describes processes and procedures of improving the performance of conventional HVAC technologies, partly in the conventional HVAC setting on in conjunction with desiccant cooling applications.

Many conventional cooling and chiller technologies will not become obsolete when operated under a desiccant cooling process. In fact, the ability to integrate conventional cooling and chiller technologies into desiccant cooling systems as part of retrofitting is important aspect and condition for the broader acceptance of desiccant application in the building industry.

This section describes three operational settings under which efficiency of conventional chillers are increased. These performance improvement measures are made possible by the fact that sensible heat removal in desiccant cooling systems can occur at higher temperatures than for conventional HVAC systems using cooling-based dehumidification.

4.1 Saving Opportunities for Chillers in Conjunction with Desiccant Dehumidification

The fact that in desiccant cooling applications the chilled water temperature can be higher than in conventional cooling based dehumidification opens up opportunities for significant energy savings for the removal of sensible heat.

There are certain aspects of chiller operation in conjunction with desiccant cooling, which can contribute to considerable energy savings. The most pertinent aspects are presented hereafter:

<u>Increase of chilled water temperature:</u> In conventional HVAC systems, chillers typically operate at below dew point temperatures to ensure effective cooling based dehumidification. Since the latent load is removed with desiccants, the chiller can be operated at higher chilled water temperatures. The particular temperature set point of the chilled water is dependent on the type of heat rejection inside the conditioned spaces. The basic fact is that the air flow rate has to be sufficiently high to remove the excess heat for the conditioned space. For this feasibility study, it suffices to state that cooling temperatures could be increased to the mid-50s F, up from the lower 40s F, and still provide effective sensible heat load removal from the spaces.

Figure 4.1.1 shows the basic process diagram of the working principle for the conventional HVAC system with cooling based dehumidification. Outside supply air is forced through a duct system by mean of a fan. A chiller delivers chilled water at around 42 to 44 F to an air handling unit which cools the supply air to below dew point temperatures. Condensation of water vapor occurs on the coil bundles and water

vapor is separated from the air supply flow, thereby drying the air. Typically, a reheat coil raises the cold air temperature upstream of the conditioned space in order to avoid discomfort from very cold air impinging on occupants. Reheating causes a significant energy penalty.

Figure 4.1.2 shows the basic process diagram of the working principle for the conventional HVAC system with desiccant dehumidification. In Figure 4.1.2 the supply air stream is depicted as a parallel process. The chiller delivers chilled water to the air handlers at a chilled water supply temperature, which is well above the temperatures required for cooling based dehumidification. No reheat is required.

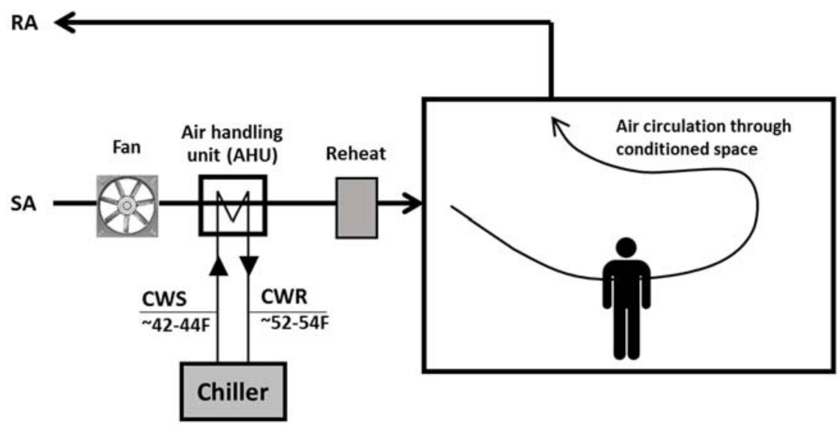


Figure 4.1.1: All-air based heat removal from conditioned space; in conventional HVAC system with cooling based dehumidification

CWS = chilled water supply
CWR = chilled water return
RA = return air
SA = supply air

Indicating low chilled water supply temperature to the AHU

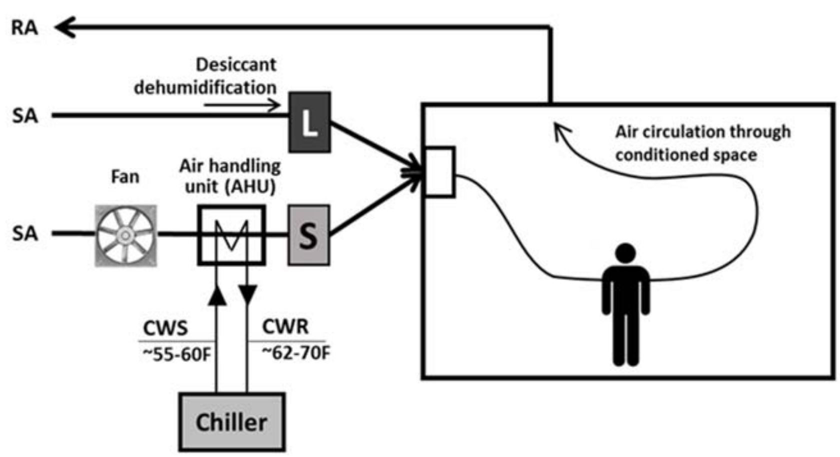


Figure 4.1.2: All-air based heat removal from conditioned space; in conventional HVAC system with **Desiccant based dehumidification**

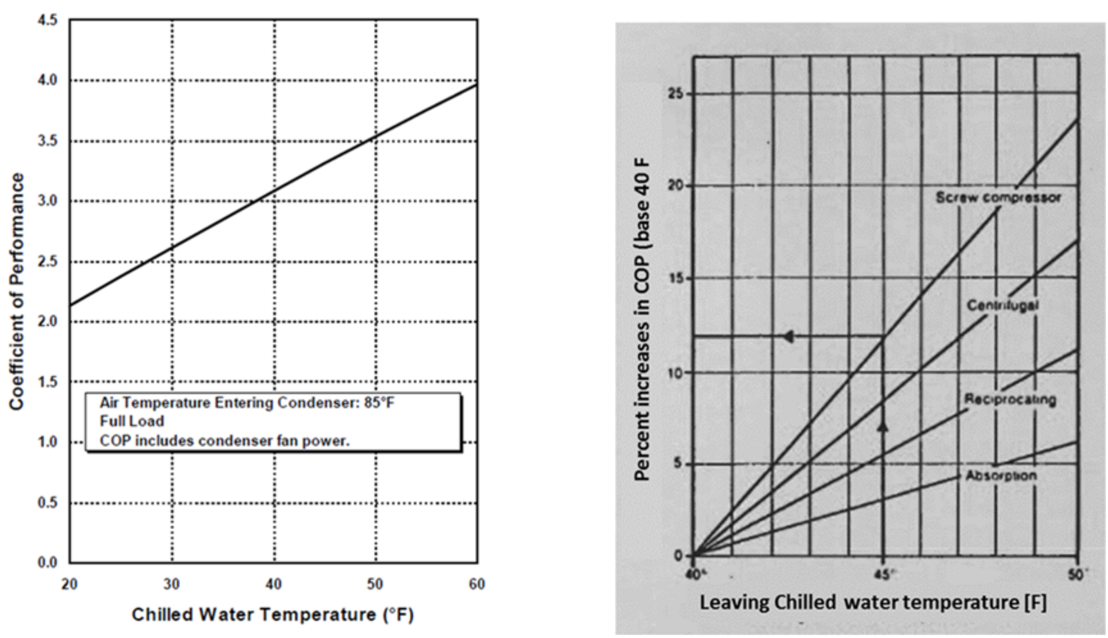
CWS = chilled water supply
CWR = chilled water return
RA = return air
SA = supply air
L = latent load removal
S = sensible load removal

Indicating medium chilled water supply temperature to the AHU

The reviewed literature reports on increases of efficiency as a function of chilled water temperature. While the assessment of the increase in chiller performance expressed as coefficient of performance (COP) was not the same among authors, estimates can be used to identify the underlying processes.

Two estimates of increases of chiller efficiency are presented in Figure 4.1.3 (a) and (b). Figure 4.1.3 (a) shows a graph that shows a generic chiller increasing COP by about 1 % for each degree of increased chilled water temperature. Figure 4.1.3 (b) suggest that the degree of increases in COP depends not only on the temperature increase of chilled water, but also on the type of chillers used. The data presented in Figure 4.1.3 (a) suggests that increasing the chilled water temperature by 12 F would increase the COP of a generic chiller by approximately 25%. Figure 4.1.3 (b) suggests that for a centrifugal compressor, which consumes the least power of the indicated at full load operation, the COP would increase by about 21%.

The data indicates that significant energy savings can be obtained by raising the chilled water temperature. While energy for the chiller operation is saved, more energy has to be provided for the air fans, since for an equal amount of heat energy, the decrease in temperature differential of the supply and return air must be compensated by an increased flow rate of cool air through the conditioned spaces. The literature reviewed indicated that the energy savings obtained from the more efficient chiller performance clearly outweighs the incremental energy spent or the fans.



(a) Generic chiller (Wulfinghoff, 1999)

(b) Different types of chillers (Gidwani,1984)

Figure D: Effect of Chilled Water on Chiller Coefficient of Performance

<u>Adjust to condenser water temperature:</u> The condenser water loop rejects heat of the chiller to the cooling tower. A drop in condenser water temperature increased the energy efficiency of the chiller.

Figure 4.1.5 shows the estimated increase in COP through a lower condenser water temperature. Figure 4.1.4 uses a condenser water baseline temperature of 85 F and shows increases of COP as the condenser water temperature drops below the baseline. The magnitude of COP increases linearly depending on the drop of the condenser temperature and the type of compressor used for the chiller unit. Using again the centrifugal chiller for the most conservative estimate of achievable increases in efficiency the figure suggests an increase of 0.5 % per degree of condenser water temperature.

Figures 4.1.3 and 4.1.4 show basic process diagrams for the condenser re-cooling for HVAC systems using cooling based and desiccant dehumidification, respectively.

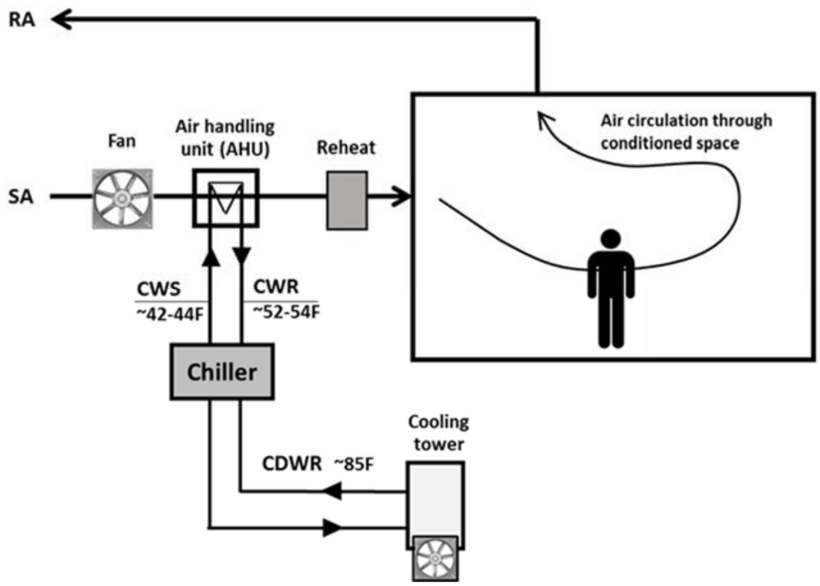


Figure 4.1.3: All-air based heat removal from conditioned space; in conventional HVAC system with **cooling based dehumidification**

CWS = chilled water supply
CWR = chilled water return
CDWR = condenser return
RA = return air
SA = supply air

Indicating cooling tower re-cooling

Conventional process control of chillers works best with a constant condenser temperature, but more advanced chillers and controls can accommodate changes in condenser water without negatively affecting the performance of the chiller. The condenser water is adiabatically cooled in a cooling tower. Achievable condenser water temperatures in conventional cooling towers are limited by the wet bulb temperature and a 5 to 10 F approach. There are newer evaporative cooling technologies that are not limited by the wet bulb temperatures of the outside air, but by the dew point. This can significantly lower achievable condenser water temperature. (these new dew point reduction evaporative cooling processes, such as the so called “M-cycle”, are discussed later in this report.

Since under desiccant cooling processes the chilled water and therefore the chiller performance can be better controlled for purely sensible load removal, adjusting the chillers to the available loads can be more easily achieved than with the conventional cooling based dehumidification.

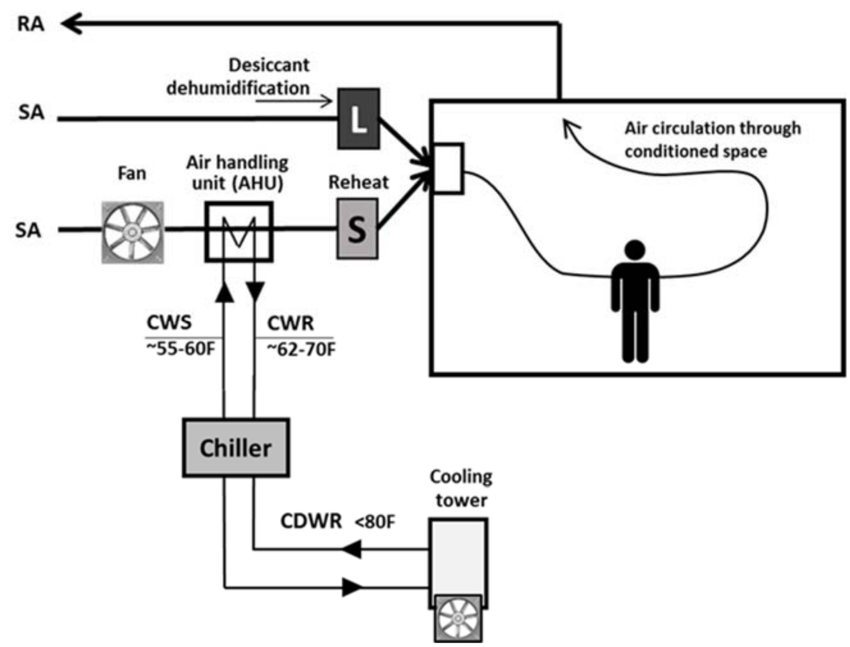


Figure 4.1.4: All-air based heat removal from conditioned space; in conventional HVAC system with **Desiccant based dehumidification**

CWS = chilled water supply
CWR = chilled water return
CDWR = condenser return
RA = return air
SA = supply air

Indicating cooling tower re-cooling

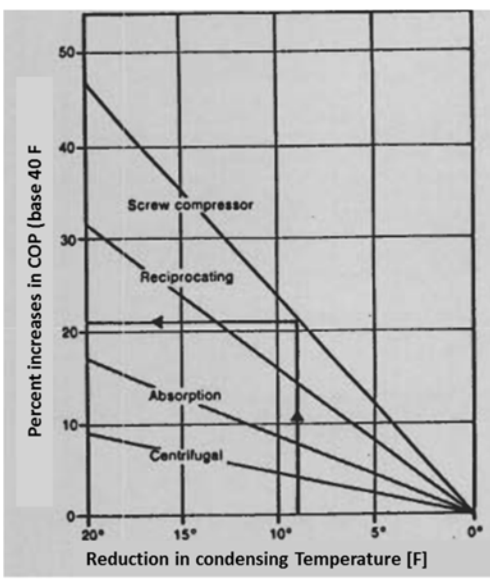


Figure 4.1.5: Effect of reduction in condenser water temperature on Chiller Coefficient of Performance (Gidwani,1984)

<u>Thermal storage of chilled water:</u> The conventional chilled water system produces chilled water as required to meet the building cooling load. With peaks in cooling demand, chillers have to be sized to meet peak demands, which increases the system complexity and initial costs. In addition, periods of peak cooling demand typically occur during daytime when electricity demand charges apply. A technical solution for conventional HVAC systems is to shave off cooling peaks by installing some form of chilled water storage. Since the conventional systems use cooling based dehumidification, the type of thermal

storage used is often ice storage. The benefits of ice storage include a small footprint for the unit since heat energy is stored as latent heat of fusion. Figures 4.1.5 and 4.1.6 show basic process diagrams for the thermal storage options for HVAC systems using cooling based and desiccant dehumidification, respectively.

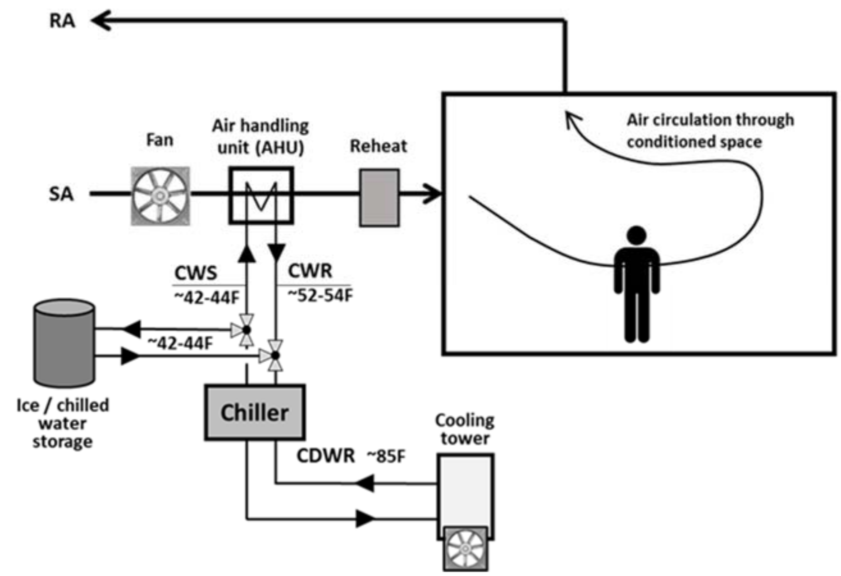


Figure 4.1.5: All-air based heat removal from conditioned space; in conventional HVAC system with **cooling based dehumidification**

CWS = chilled water supply
CWR = chilled water return
CDWR = condenser return
RA = return air
SA = supply air

Indicating ice storage or chilled water storage

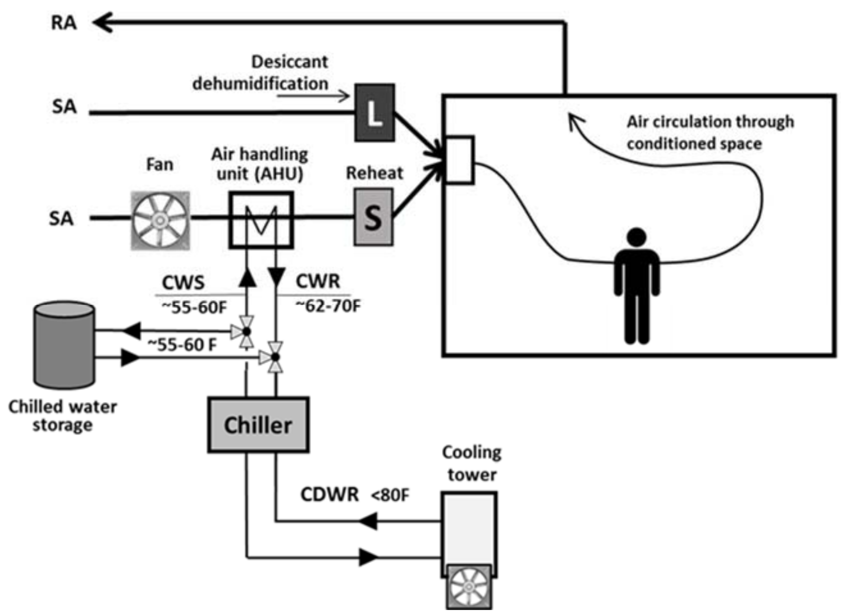


Figure 4.16: All-air based heat removal from conditioned space; in conventional HVAC system with **Desiccant based dehumidification**

CWS = chilled water supply
CWR = chilled water return
CDWR = condenser return
RA = return air
SA = supply air

Indicating chilled water storage

For desiccant cooling systems, the chilled water storage requirements will be reduced since only sensible heat will be rejected to the chilled water and at higher temperature than the below dew point temperatures of conventional HVAC applications. The energy requirements to produce the chilled water is less than in ice storage, since the COP of the chiller operating at higher chilled water temperatures is higher and the latent heat of fusion can be avoided. Depending on the load profile of the building, a chilled water tank can be installed and the chilled water storage can be charged during off-peak hours.

Summarizing: This section discussed technical solutions for significant energy savings that can be accomplished with conventional HVAC equipment when they work in conjunction with desiccant dehumidification. In these applications, chillers only have to remove the sensible heat load and can operate at higher chilled water temperatures, since the chilled water does not have to support cooling based dehumidification in the cold coils. As can be seen later in the report, a higher chilled water temperature has significant benefits and opens up innovative and energy saving solutions for space conditioning.

4.2 Technologies of Sensible Heat Rejection inside the Conditioned Spaces

The manner in which heat rejection by occupants is accomplished inside conditioned spaces can have a significant effect on energy efficiency as well as on indoor comfort. Three basic technologies of sensible heat rejection from the conditioned space to the outside are presented hereafter.

Circulation of cool air inside the conditioned spaces:

Working principle: Cold air is forced into the conditioned space and an equal amount of warm air leaves the space. The overall quantity of heat energy removed from the conditioned space is a product of air flow rate, temperature difference between the supply and discharge air flow and the specific heat of air. Inside the space, heat is transferred from objects to the moving air through convective heat transfer. External heat energy enters the space through a combination of conduction, convection and radiation. Internal heat gain is from occupants and heat producing equipment. The heat that enters and/or is generated inside the space has to be rejected to the outside in order to maintain thermal equilibrium.

Figure 4.2.1 illustrates the conventional process of circulating cold air through the conditioned space in order to reject the heat that enters or is generated inside the conditioned space. Cold air is forced into the conditioned space. Heat is transferred from objects inside the space to the moving air by convective heat transfer processes (e.g. Q'_{conv}.), thereby cooling the objects, including walls, ceiling, floors and all equipment inside the space. The occupants reject sensible heat through mostly convective ($Q'_{conv.}$) and, to a limited quantity, through radiant ($Q'_{trans.}$) heat losses. The air flow leaving the space is split into two parts, the portion that is recirculated and the portion that is discharged to the exterior, which account for about 15% and 85%, respectively.

Figure 4.2.2 shows a typical configuration of an air supply duct system. In the all-air HVAC system, the duct system conveys the air which transports heat from the conditioned spaces to the exterior. Transporting heat energy through air is inefficient due to the relatively high fan power that has to be used to pressurize the air and force it through the spaces.

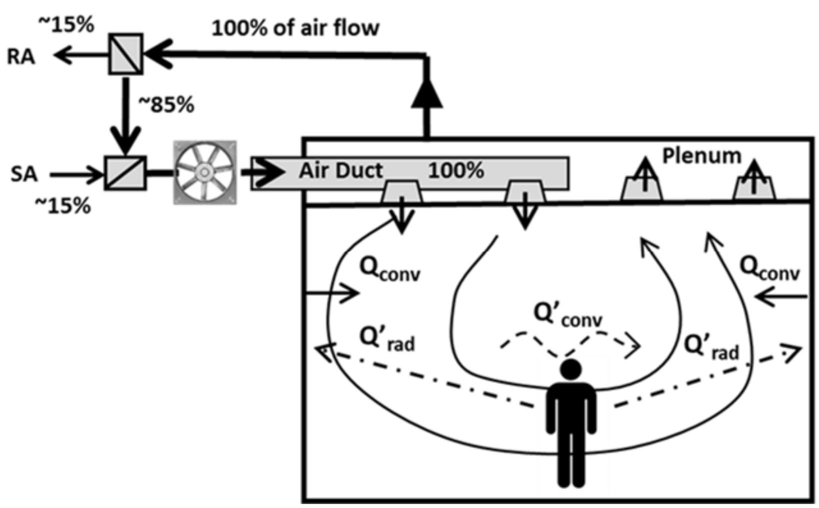


Figure 4.2.1: All-air based heat removal from conditioned space

Qconv = convective heat transfer
Q'conv. = convective heat loss occupant
Q'rad. = radiative heat loss occupant
RA = return air
SA = supply air

Indicating mechanisms of heat transfer in space and

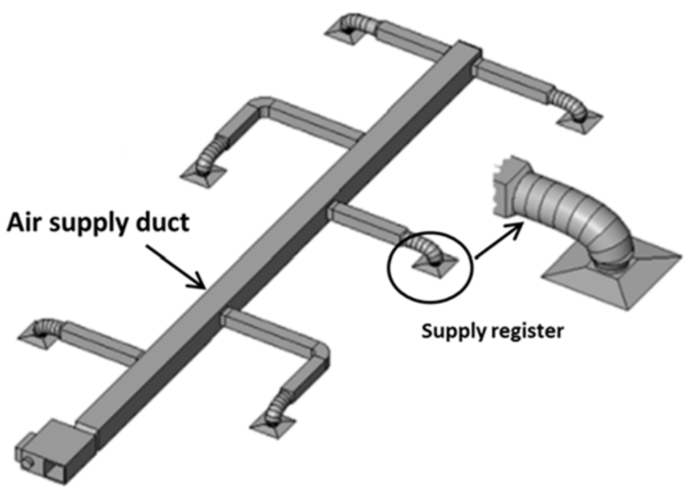


Figure 4.2.2: All-air based heat removal from conditioned space; schematics of air duct system

The air supply is conveyed to the space through a series of air ducts.

Benefits: This process of heat rejection has been applied successfully over many decades and is the conventional method in "all-air" systems. The system is simple since only a supply fan has to provide sufficient pressure to force the cold supply air through a duct system to the interior of the space and provide a return path for the return air.

Disadvantage: Because of the low specific heat of air, large volumes of air have to be moved through the space to reject heat from the space to the outside. This requires considerable energy for the air fans. The distribution of cooling capacity in the form of cold air requires the establishment of sufficient supply pressure in the air ducts, as well as balancing the pressure losses for the desired distribution of cold air inside the space. The control of the correct distribution of air flow can create significant challenges.

Actively cooled Radiant surfaces:

Working principle: Cold water is circulated through radiant surfaces (e.g. ceiling panels) thereby removing heat in a closed loop. The radiant surfaces gain heat from the inside of the spaces by mostly radiative heat transfer from objects to the radiative surfaces. The heat is transported from the conditioned spaces to the outside by water. The amount of heat removed from the space is the product of chilled water flow rate, temperature difference between chilled water supply and return and the specific heat of water.

Figure 4.2.3 shows a basic flow diagram of the radiant cooling systems with its hydronic heat transfer process. Most of the sensible heat is rejected to the radiant ceiling by radiative heat transfer to the actively cooled ceiling panels. A small amount of sensible heat is also removed from the dedicated outdoor air supply. The occupants reject more sensible heat through radiative heat transfer ($Q'_{rad.}$) than through convective ($Q'_{trans.}$) heat losses.

Figure 4.2.4 shows a conventional radiant ceiling panel. Radiant heat is received by the ceiling panels and the heat is transferred to the chilled water that is flowing through cooper tubes. The chilled water return is finally pumped to the chiller where the heat is rejected to the external environmental.

Benefits: The transport of heat through chilled water is significantly more effective than through air. The amount of energy required to pump chilled water through the radiant surfaces is significantly smaller than the fan energy required to transport the same amount of heat with air. Besides energy savings radiant cooling is considered as more comfortable way to cool spaces than using only cooled air.

Disadvantage: Radiant cooling applications still represent a small portion of the overall air condition applications. Radiant cooling is often considered a non-conventional cooling technology and is therefore not chosen. Radiant cooling requires effective dehumidification since cold temperatures of the radiant surfaces, typically aluminum panels, could otherwise result in condensation on the panels. The response time of cooling capacity is relative long and it depends on the thermal mass of the radiant panels. The cooling capacity of radiant cooling devices is affected by the form of installation and typically the cooling capacity has limits, requiring effective reduction of heat gain to the spaces.

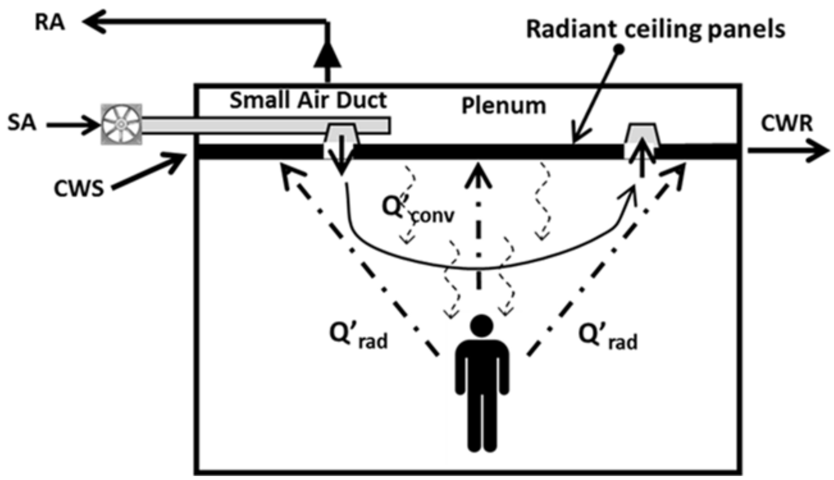


Figure 4.2.3: Radiant ceiling based heat removal from conditioned space

Q'conv. = convective heat loss occupant
Q'rad. = radiative heat loss occupant
CWS = chilled water supply
CWR = chilled water return
RA = return air
SA = supply air

Indicating mechanisms of heat transfer in space and

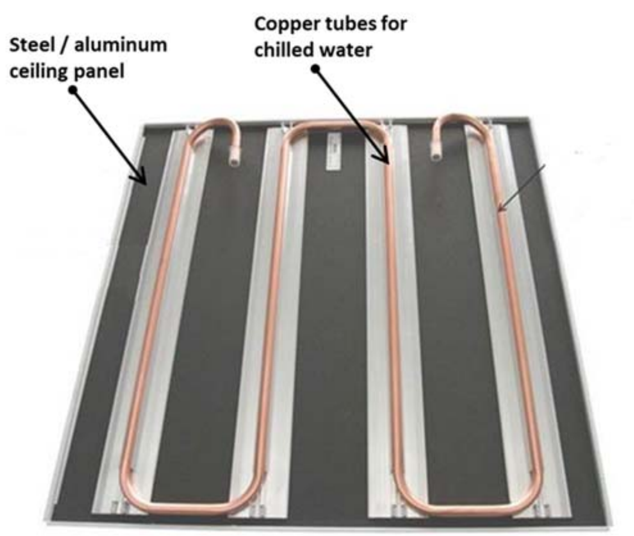


Figure 4.2.4: Radiant ceiling based heat removal from conditioned space

Radiant ceiling panels

The ceiling panels receive radiant heat from inside the conditioned space and transfer the heat to the chilled water that flows through the copper tubes

Chilled Beam:

Working principle: Chilled beams provide sensible cooling through mainly convective heat transfer and limited radiant cooling. Chilled water flows to the chilled beams units where heat is transferred from the inside air to a heat exchanger plate and from there to the chilled water flow. By cooling air and thereby lowering its density the chilled beans beams create localized vertical circulation cells of air. In passive chilled beams the cooled air flows downwards due to larger density of cold air and initiates warmer air to be pulled into the chilled beam. In active chilled beams small fans increase the air flow. Figure 4.2.6 illustrates the active and passive chilled beams.

Figure 4.2.5 shows a basic flow diagram of the chilled beam systems with its hydronic heat transfer process. The sensible heat is rejected to the cold air that exits the chilled beam. The chilled beam receives chilled water and the air passing through the chilled beam is cooled through a water-air heat exchanger. The occupants reject sensible heat through convective heat transfer ($Q'_{con.}$) and, to a limited extent, through radiant ($Q'_{trans.}$) heat losses. The limited radiant heat loss is towards the cold frame of the chilled beam. There is, however, a popular misconception that the chilled beam performs radiant heat transfer. This is not correct and the portion of radiant heat loss to the chilled beam is only about 10 to 20% of the total heat loss.

Figure 4.2.6 shows the configuration of a chilled beam with integrated air supply duct. There are two type of chilled beans, active and passive. For the active of chilled beam, the air flow is supported by some form of fan assist, which increases the flow through the beam, thereby increasing the chilling cooling capacity of the chilled beam unit.

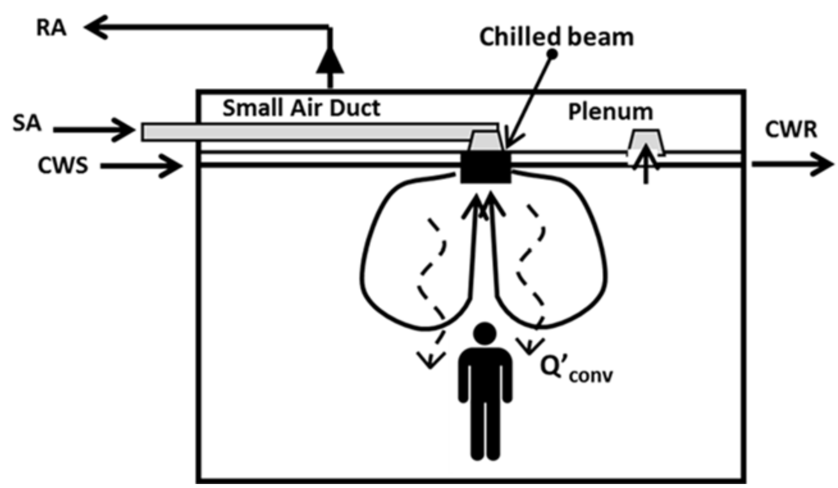
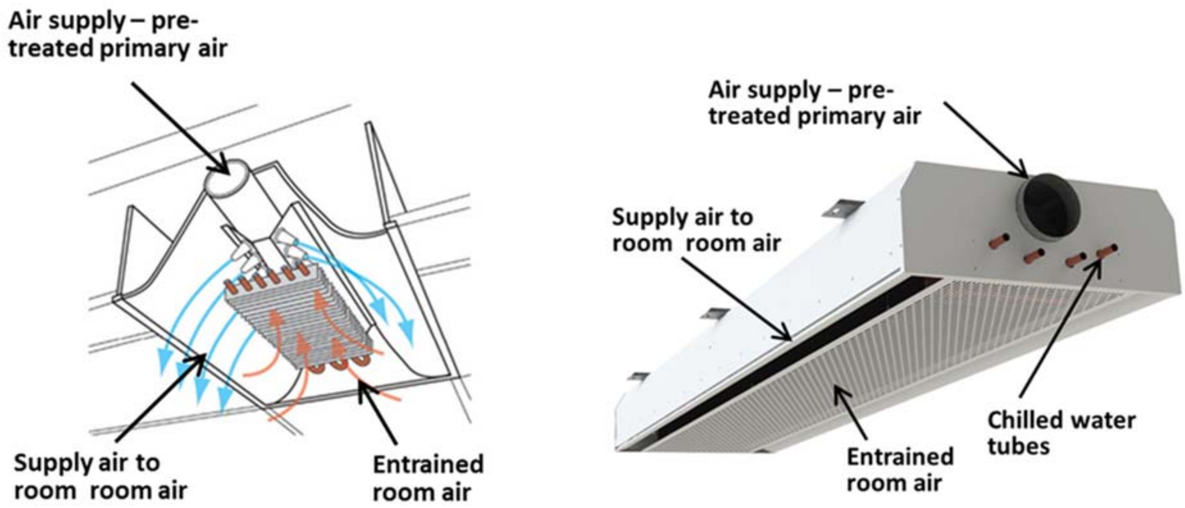


Figure 4.2.5: Chilled beam based heat removal from conditioned space

Q'conv. = convective heat loss occupant
CWS = chilled water supply
CWR = chilled water return
RA = return air
SA = supply air

Indicating mechanisms of heat transfer in space and



(a) Chilled beam – schematics of air flow (b) Chilled beam - unit

Figure 4.2.6: Chilled beam based heat removal from conditioned space

Benefits: The benefits of heat transport are the same as for radiant cooling, since chilled beams transport heat from the space to the chiller by means of water and not air. This saves significant energy. The response time of chilled beams is short and since the chilled water flow to the chilled beam units can easily be regulated and the buoyancy induced vertical circulation cells have a relative small inertia. Chilled beams, while considered an innovative cooling technology, is closer to the current mainstream HVAC paradigm of the all-air system than radiant cooling.

Disadvantage: Chilled beams require space at the space ceiling for installation. Like radiant cooling the systems feature chilled water pipes through occupied space, thereby creating the risk of condensation if the dehumidification does not reduce the humidity level effectively.

Summarizing: The three basic technologies of rejecting sensible heat from inside the conditioned space all require that the dehumidification of air entering the space is effective and assures that dew point is reduced sufficiently to avoid increased relative humidity as air temperature is reduced and, as a significant extreme case, condensation on colder than dew point surfaces can occur. The radiant cooling and chilled beam technologies are especially prone to problems with condensation and only work with effective dehumidification.

SECTION 5 – FRAMEWORK FOR A HAWAII SPECIFIC DESICCANT COOLING SYSTEM

This section presents important aspects that have to be considered to derive optimum and Hawaii specific desiccant cooling applications.

5.1 The Need for System Optimization

The strength but also the weakness of conventional HVAC systems is that they can provides the same type of indoor environment, irrespectively of the location. This means conventional HVAC technology basically is only concerned with the end result of creating standard indoor thermal environments.

Figure 5.1.1 shows the basic heat and mass balance for a generic conditioned space defined by a control volume model. Heat energy and moisture enters the space through the envelope and openings. In addition, heat energy and moisture is also generated inside of the space. The heat energy and moisture intake has to be balanced by the heat reduction and dehumidification. As noted before, the focus of this report are cooling and dehumidification applications required for the Hawaii climate conditions, therefore it is assumed that heat energy and humidity has to be extracted from the conditioned space and not added.

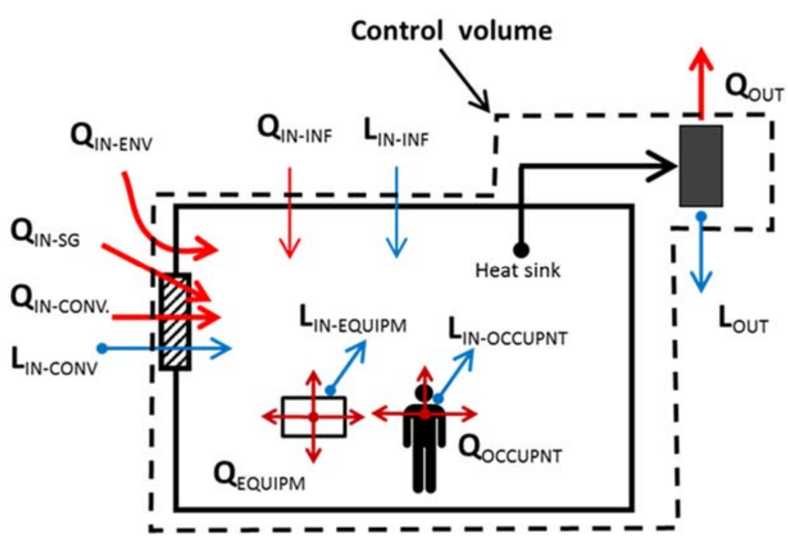


Figure 5.1.1: Heat and mass balance for a generic conditioned space

Depiction of the control volume of a generic conditions room.

For a balanced and steady-state heat energy and moisture balance, in the control volume the amount of heat energy and moisture that has to be extracted from the conditioned space is the same as is entering into and generated within the space. Heat extraction is sensible cooling and moisture extraction is dehumidification, which require heat and humidity sinks, respectively. Therefore, in order to minimize the required heat and humidity sinks, the amount of heat and humidity entering the space and generated inside the space must be minimized by appropriate actions. Some of which are described in the following.

The amount of energy heat entering into and generating inside the conditioned space can be minimized be the following measures:

- Reduce the convective heat gain through the envelope by means of thermal insulation.
- Reduce the radiative heat gain (e.g. solar gain) by means of low emissivity and/or high reflexivity fenestration and suitable shading of openings.
- Reduce the amount of convective heat gain through openings and through a porous envelope, by sealing the envelope and reducing the amount of ventilation to the required flow rate.
- Reduce the amount of internal gain by choosing energy efficient equipment and conveying internal gain to the exterior.

The amount of humidity entering into and generating inside the conditioned space can be minimized be the following measures:

- Reduce the amount of convective humidity gain through openings and thro ugh a porous envelope, by sealing the envelope and reducing the amount of ventilation to the required flow rate.
- Reducing the humidity sources inside the conditioned space. This has limits since a major humidity source are occupants.

The selection of suitable solutions for desiccant cooling applications has to be based on a systems approach. The selection of suitable candidate systems and assessing their benefits and drawbacks requires the definition of a suitable framework of criteria. For the present feasibility analysis, the framework of will be grouped into four high level categories. These criteria will be used to assess and quantify the importance under which the different desiccant cooling systems will perform under Hawaii specific climatic and logistic conditions.

Factors which make Hawaii conditions unique and create challenges as well as opportunities are as follows:

- Hot and humid weather conditions year round
- High levels of available solar irradiation
- Requirement of space cooling and conditioning is year round
- The specific attitude of Hawaii residents to adaptive cooling
- Very high electricity rates, which enhance the economic gain of energy savings.

Figure 5.1.2 illustrates four high level categories in assessing the feasibility of desiccant cooling systems for Hawaii specific conditions. These four high level categories are:

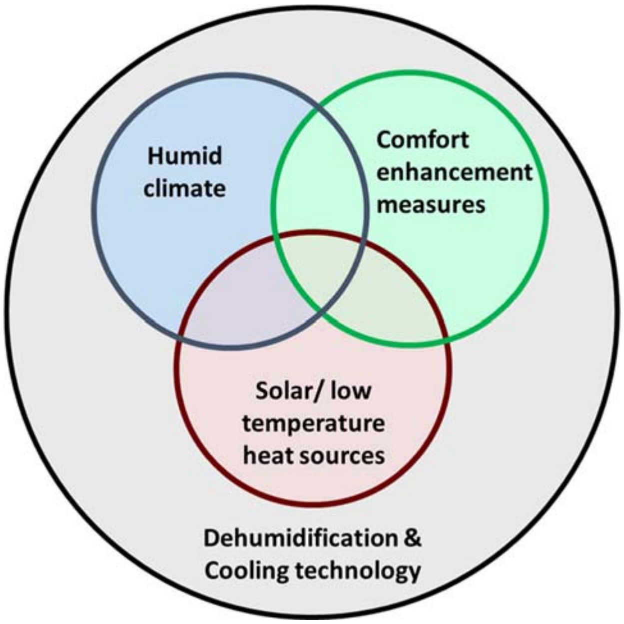


Figure 5.1.2: High level categories of criteria for feasibility assessment of desiccant cooling technologies

The categories of criteria combine technology and climatic issues which are of importance to Hawaii.

This figure is referred to as the “Criteria Category Chart”

The specific desiccant and cooling technologies: The specifics of the considered desiccant & cooling technologies is the main determinant category, with three subsets of categories addressing different scenarios under which the technologies must perform in Hawaii.

The three subsets of categories are as follows:

Humid climate: Hawaii has a hot and humid climate. In Hawaiian climate cooling applications will require some degree of dehumidification. Whereas, for example, open-cycle evaporative cooling works well in dry desert climate, the high humidity in Hawaii has can make evaporative cooling only marginally feasible, or not at all. The high humidity also places a higher burden on the desiccant performance since regeneration is significantly affected by the high humidity level in the outdoor air. In the case of conventional AC-applications with vapor compression the effectiveness of performance is affected to a lesser degree by the high humidity since even high dehumidification needs can be handled by increasing the capacity of the vapor compression process.

Solar & low temperature heat sources: Solar irradiance rates are high in Hawaii year round. Solar heat can be used for desiccant regeneration and thermally powered chillers. The high electricity costs in Hawaii boost incentives to use combined heat and power (CHP) applications, where waste heat can be used for desiccant regeneration. Heat derived from indigenous biofuels can be equally used as heat sources for regeneration.

<u>Comfort enhancement measures:</u> The comfort enhancement measures indicate opportunity to lower the required cooling capacity through increased convective and radiative heat loss by the building occupant. Ceiling fans have significant benefits of providing cost-effective and energy saving cooling effect through an increase of air flowing of the human bodies and therefore create a cooling effects due to increased convective heat transfer. By using radiant cooling, the operative temperature is reduced since the operative temperature is the weighted mean of the air and radiant temperatures. While these comfort enhancement measures do not provide lower space air-temperature they benefit the human comfort experience. Most buildings in Hawaii used to be naturally ventilated and a certain level of acceptance of a more natural indoor environment versus fully conditioned buildings is still prevalent among Hawaii's residents. This facilitates the implementation of comfort enhancement through mixed mode space conditioning which lowers the requirement for cooling capacity and facilitates application potential for alternative cooling systems.

5.2 Desiccant and Cooling Technologies

Part 1 of the present research project featured a literature review of desiccant cooling technologies. The review identified the scope of desiccant technologies in regard to their application to building conditioning. The review considered generic key characteristics of desiccant and cooling technologies, which were somewhat independent of specific locations. In the present feasibility assessment of desiccant cooling applicants in Hawaii the three high level sub criteria indicated in Figure 5.1.2 will be considered to arrive at Hawaii' specific solutions.

Figure 5.2.1 illustrates that the following discussion of generic key characteristics of desiccant and cooling technologies related to the framework introduced in Section 5.1. It must be kept in mind that cooling technologies discussed hereafter are an essential part of the desiccant cooling systems, since they provide the required sensible cooling.

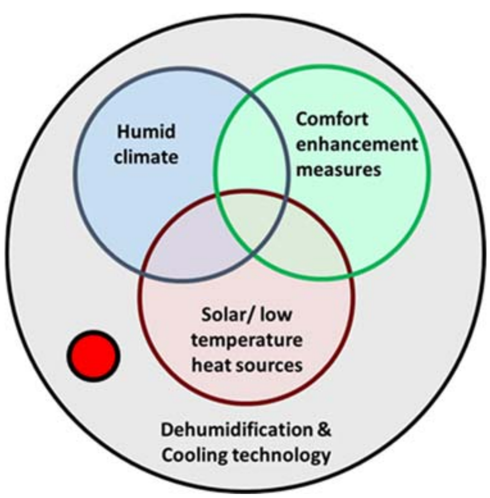


Figure 5.2.1: Key characteristics of desiccant and cooling technologies

Key characteristics of Desiccant and sensible cooling technology is considered on the basis of inherent technical merits, such as level of innovation, successful market entry, etc. The consideration in Section 5.2 are NOT Hawaii specific

The red dot in the figure indicates which of the four categories are being discussed – here it is desiccant and cooling technology in general

The key technology characteristics considered for the present feasibility study are as follows:

Key characteristics	Description
Maturity of technology	The maturity of the technology reflects the readiness of the technology to perform as an integral component of a HVAC systems in a commercial setting. States of readiness considered reflect if the technology has been successfully operated as a commercial product, if prototypes have been operated in multiple settings or if only proof-of-concepts installations have been completed.
Reach of technology	The reach of technology is the anticipated range of application, which includes the use of the technology in different climatic and commercial settings.
Complexity of product	The issue is how complex is the technology as a product and how much change is required by the building operator of commercial buildings or the residential end user.
Estimated costs	How high will be the anticipated first costs for installation and setting up innovative operation. While detailed costs are not available for many innovative technology products, projection are based on generic facts.
Status of market acceptance	The status of market acceptance refers to an estimate of economic readiness and acceptability of the technology option. Acceptability is closely linked to the state of technology maturity and its commercialization in various applications. The acceptance will be relevant not only from manufacturers but also from advocates of consumers and users of the new technology.
Projected energy savings	As with any new technologies the costs or other important benefits must outweigh the pain of changing from conventional technology to the new technology. Energy savings play a very important role in supporting implementation of the new technology.
Ability to operate with solar or other low temperature sources	The question whether the desiccant and/or sensible cooling technology can take advantage of solar or other low temperature sources is an important qualifier for Hawaii. It makes the technology especially suitable for hot weather or for applications where thermal resources are managed in a responsible way.

These key characteristics will be used in the assessment of the feasibility of candidate systems in Section 8 and 9.

5.3 Humid Climate of Hawaii

This section shows how the typical climatic conditions in Hawaii affect the performance of space conditioning. Figure 5.3.1 illustrates that the following discussion of generic key characteristics of desiccant and cooling technologies related to the framework introduced in Section 5.1 is concerned with the humid climatic conditions in Hawaii. Hawaii's climate is sub-tropical with a year-around need for space conditioning.

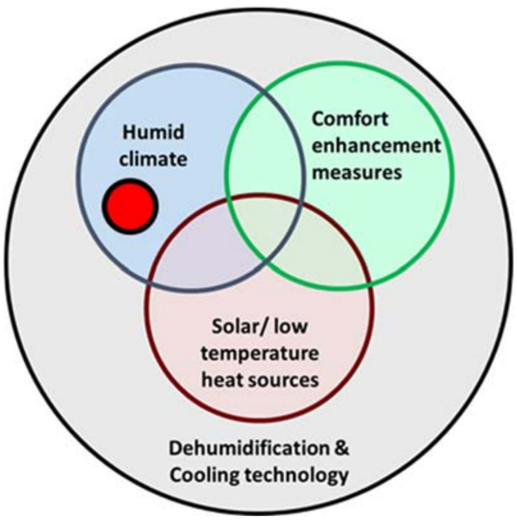


Figure 5.3.1: Key characteristics of desiccant and cooling technologies – Humid climate of Hawaii

Hot and humid climate represents a considerable challenge for space condition. This section discusses the typical Hawaii climatic conditions, which are an important consideration for space conditioning.

The red dot in the figure indicates which of the four categories are being discussed – here it is the humid climate in Hawaii which presents a special challenge for desiccant dehumidification

Time series and probability functions for the present analysis were obtained from the typical meteorological year (TMY) type 3 file for Honolulu. The four main weather parameters are as follows:

1. Dew point
2. Dry bulb temperature
3. Wet bulb temperature
4. Relative humidity with the TMY3 file for Honolulu.

Of the TMY3 file hourly weather data was used for two conditions, (1) a "full record", for all 8760 hours per year, and a "11-h" record, for the time from 8:00 am (start) to 6:00 pm (end). The full and 11-h record portray the conditions under which constant space conditioning occurs or only during the day hours. It should be noted that the 11-hour record was applied 7-days per week and not on a 5-days business schedule basis. A detailed representation of the TMY analysis is presented in **Appendix A.**

Dew point: The dew point averages, maximum and minimum for the full record and the 11-h record are presented in Figures 5.3.2 through 5.3.5.

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.0**	**4.3**
Jan	60.6	70.0	48.0	4.9
Feb	64.8	73.9	50.0	5.8
March	62.3	72.0	51.1	4.3
April	61.7	70.0	50.0	3.7
May	64.8	72.0	57.0	2.9
Jun	65.7	69.4	60.1	1.7
July	67.1	75.0	60.1	2.5
Aug	67.3	73.9	61.0	2.1
Sept	68.7	75.9	61.0	2.5
Oct	67.4	73.0	61.0	2.1
Nov	65.5	71.1	55.9	2.9
Dec	63.8	73.0	51.1	4.9

Figure 5.3.2: Dew point for Full record

Annual and monthly averages, min and maxvalues of Dew Point for the full record

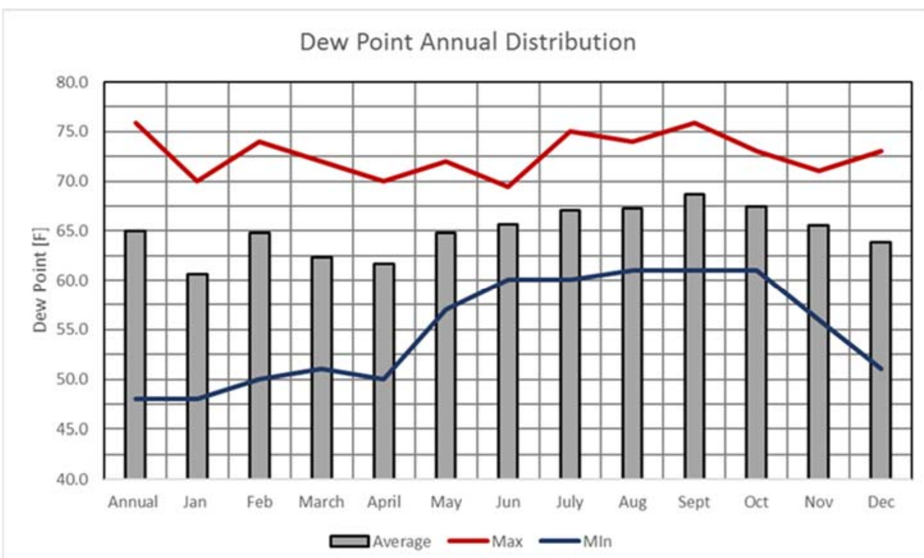


Figure 5.3.3: Dew point for Full record; Annual and monthly averages, min and max values of Dew Point for the full record

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.9**	**4.4**
Jan	61.0	70.0	48.9	4.9
Feb	65.1	73.9	50.0	6.1
March	62.5	72.0	51.1	4.5
April	61.6	69.1	52.0	4.0
May	64.6	71.1	57.0	3.0
Jun	65.5	69.4	60.1	1.9
July	66.9	75.0	60.1	2.7
Aug	66.9	73.9	61.0	2.3
Sept	68.4	75.9	61.0	2.8
Oct	67.3	73.0	61.0	2.4
Nov	65.7	71.1	55.9	2.8
Dec	64.1	73.0	52.7	5.1

Figure 5.3.4: Dew point for 11-h record

Annual and monthly averages, min and max values of Dew Point for the11-h record

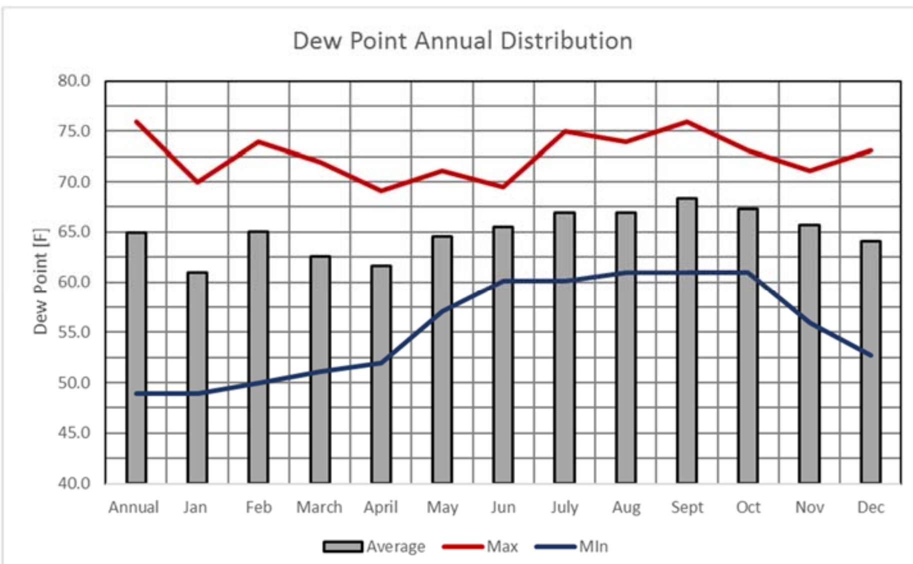


Figure 5.3.5: Dew point for 11-6 record; Annual and monthly averages, min and max values of Dew Point for the full record

The monthly average the dew point for the full and 11-h record are compared in Figure 5.3.5. The results suggest that the dew point averages of both time series are basically identical. This indicates that the absolute humidity, which determines the dew point, stays about the same for both records.

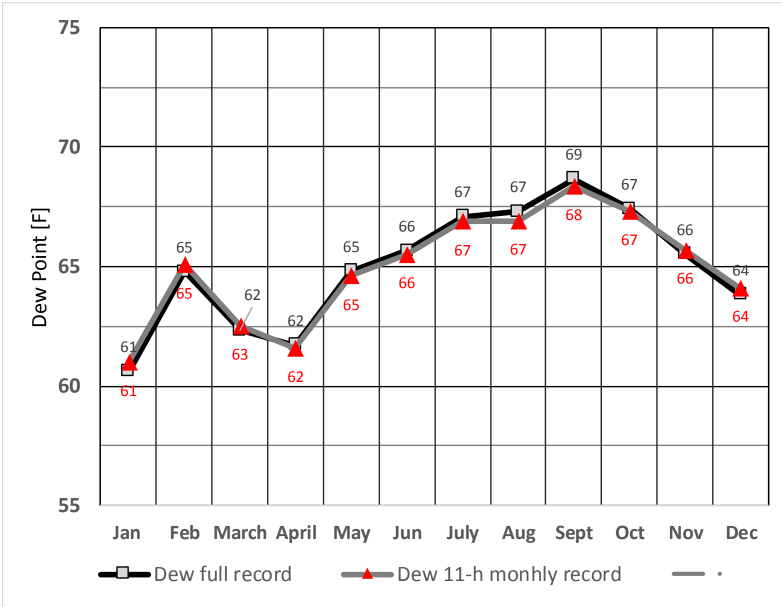


Figure 5.3.6: Comparison of dew point records – between the full and the 11-h record

Figures 5.3.7 and 5.3.8 show probability of exceedance determined by the probability density and cumulative functions for the full and the 11-h record. The figures indicate that the average dew points are distributed closely around the average, with the coefficient of variation (CV) (e.g. standard deviation divided by the average) of 6.6%. The figures provide the dew point values of the 80% probability of exceedance.

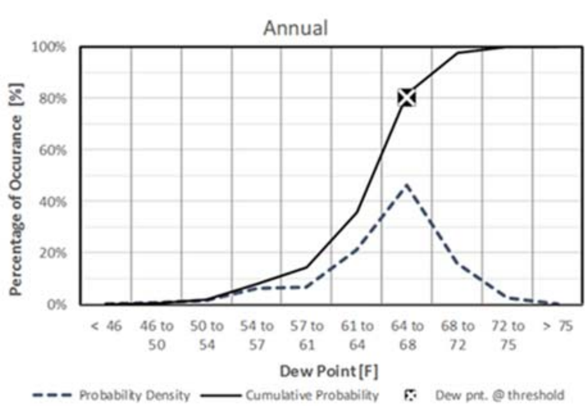


Figure 5.2.7: Probability of exceedance for **Annual** dew point data – **Full record**

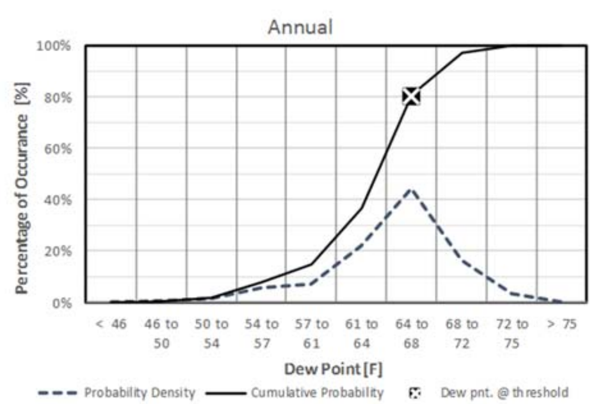


Figure 5.2.8: Probability of exceedance for **Annual** dew point data – **11-h record**

Relative humidity: The averages, maximum and minimum of Relative humidity (RH) for the full record and the 11-h record are presented in Figures 5.3.9 through 5.3.12.

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.9**	**4.4**
Jan	61.0	70.0	48.9	4.9
Feb	65.1	73.9	50.0	6.1
March	62.5	72.0	51.1	4.5
April	61.6	69.1	52.0	4.0
May	64.6	71.1	57.0	3.0
Jun	65.5	69.4	60.1	1.9
July	66.9	75.0	60.1	2.7
Aug	66.9	73.9	61.0	2.3
Sept	68.4	75.9	61.0	2.8
Oct	67.3	73.0	61.0	2.4
Nov	65.7	71.1	55.9	2.8
Dec	64.1	73.0	52.7	5.1

Figure 5.3.9: Relative humidity (RH) for Full record

Annual and monthly averages, min and max values of RH for the **full record**

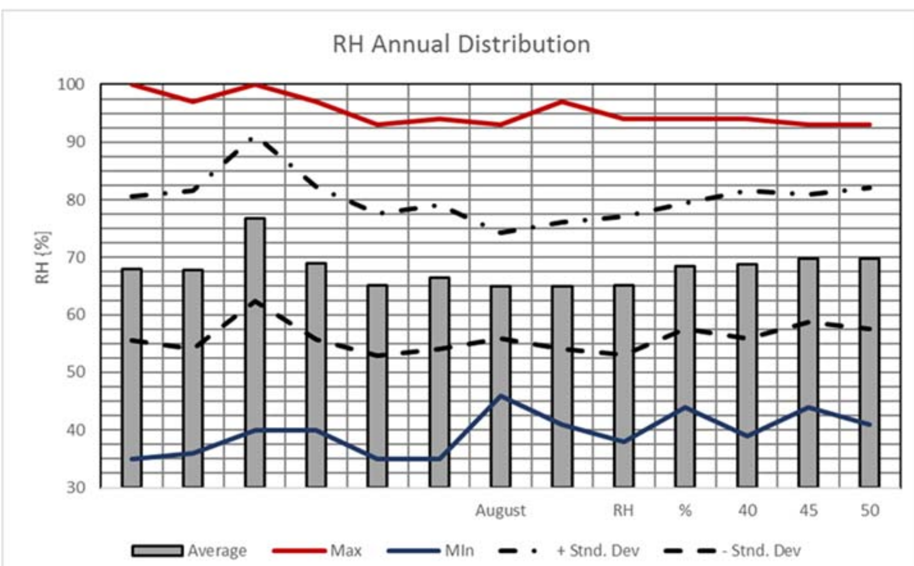


Figure 5.3.10: RH for Full record; Annual and monthly averages, min and max values of RH for the full record

	average %	Max %	Min %	stnd. Dev. %
Annual	59.3	97.0	35.0	10.4
Jan	60.7	93.0	36.0	11.9
Feb	67.6	97.0	40.0	13.0
March	59.3	93.0	40.0	11.0
April	56.4	87.0	35.0	10.2
May	56.5	90.0	35.0	9.0
Jun	57.7	76.0	46.0	6.5
July	56.4	88.0	41.0	8.9
Aug	54.7	82.0	38.0	7.9
Sept	60.5	91.0	44.0	9.3
Oct	57.7	88.0	39.0	8.6
Nov	61.7	90.0	44.0	8.9
Dec	63.4	93.0	41.0	11.2

Figure 5.3.11: RH for 11-h record

Annual and monthly averages, min and maxvalues of RH for the **11-h record**

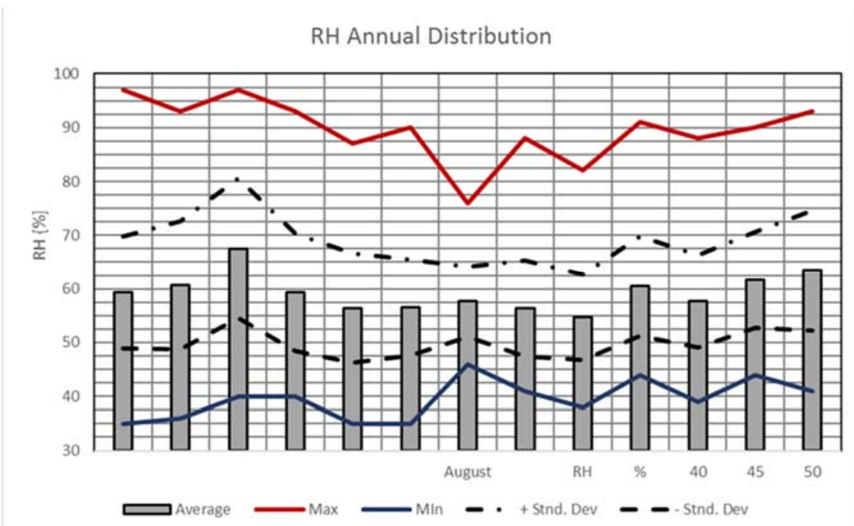


Figure 5.3.12: RH for 11-h record; Annual and monthly averages, min and max values of RH

The monthly average RH for the full and 11-h record are compared in Figure 5.3.13. The results suggest that the RH monthly averages are higher for the full record than for the 11-h record. Figure 5.2.14 indicates the monthly values of the RH values of the full record exceeding those of the 11-h record. This indicates that RH during daylight hours are significantly below the RH values for the entire day.

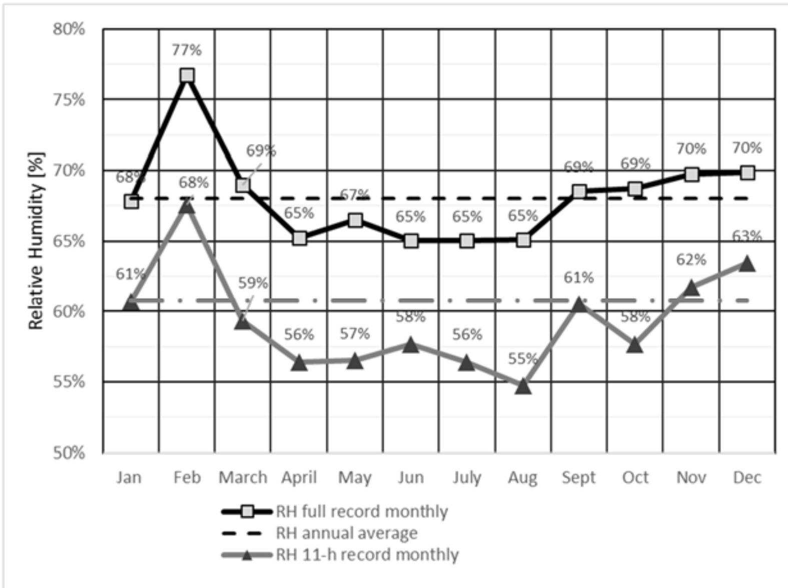


Figure 5.3.13: Comparison Relative humidity Full record and 11-h record

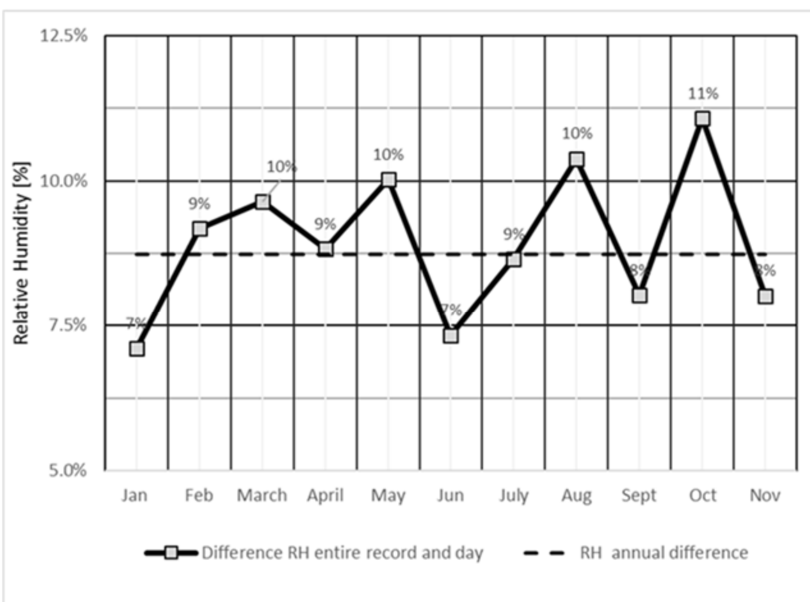
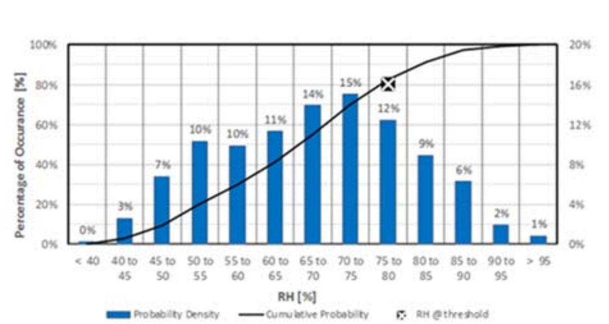


Figure 5.3.14: Comparison Relative humidity Full record and 11-h record; difference between the data of the Full reord and the 11-h record

Positive difference inidcates that the RH of the full record is larger. Also, the RH during the 11-h perios is lower

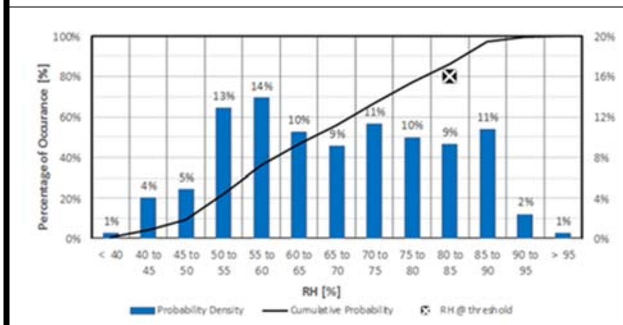
Figure 5.3.15 (a) through (f) shows examples of probability distribution of the RH for the full and 11-h records. A comprehensive record of the monthly and annual probability distributions of RH or the full and 1-h records are presented in **Appendix A.** The probability distributions in Figure 5.3.15 suggest that the distribution in the early year, e.g. January,

Full record



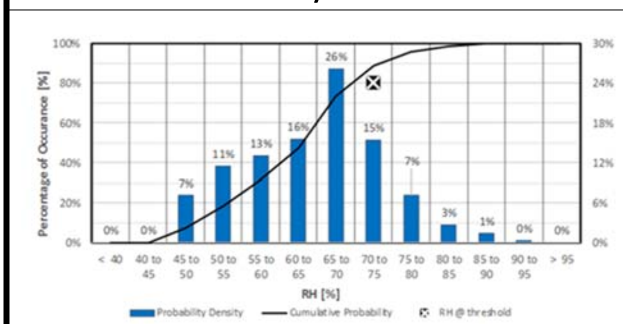
Average 68.1%, 80% threshold 78%; CV = 19%

(a) Relative humidity (RH); **Full record**, Probability of exceedance for the **entire year**



Average 67.8%, 80% threshold 83%; CV = 20%

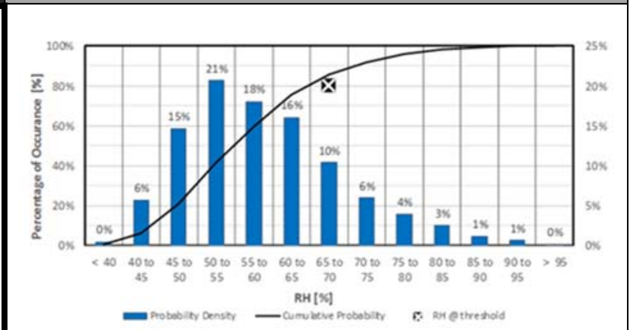
(b) Relative humidity (RH); **Full record**, Probability of exceedance for **January**



Average 65%, 80% threshold 73%; CV = 14 %

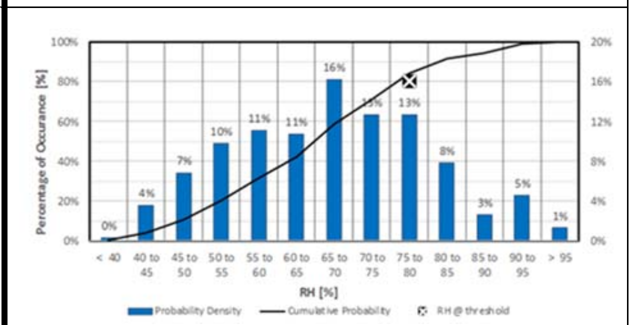
(c) Relative humidity (RH); **Full record**, Probability of exceedance for **June**

11-h record



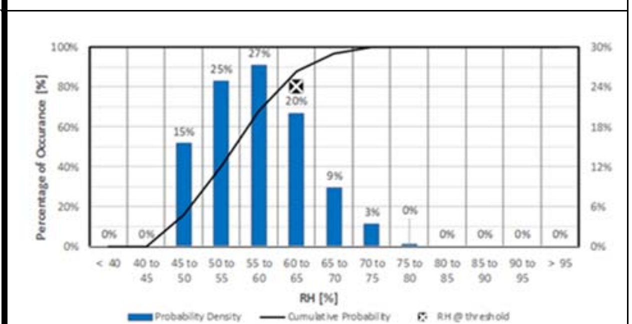
Average 59.3%, 80% threshold 68% ; CV = 18%

(d) Relative humidity (RH); 11-h record, Probability of exceedance for the entire year



Average 67.6%, 80% threshold 78%; CV = 20 %

(e) Relative humidity (RH); **11-h record**, Probability of exceedance for **June**



Average 57.7%, 80% threshold 63%; CV = 11 %

(f) Relative humidity (RH); **11-h record**, Probability of exceedance for **June**

Figure 5.3.15: Probability distributions of RH for the full record and the 11-h record. - The annual distributions and the January and June distributions are shows to characterize the RH conditions.
Note: Coefficient of variation (CV) is ratio of the standard deviation to the mean

Dry bulb temperature: The averages, maximum and minimum of the dry bulb temperature (DBT) for the full record and the 11-h record are presented in Figures 5.3.16 through 5.3.19.

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.9**	**4.4**
Jan	61.0	70.0	48.9	4.9
Feb	65.1	73.9	50.0	6.1
March	62.5	72.0	51.1	4.5
April	61.6	69.1	52.0	4.0
May	64.6	71.1	57.0	3.0
Jun	65.5	69.4	60.1	1.9
July	66.9	75.0	60.1	2.7
Aug	66.9	73.9	61.0	2.3
Sept	68.4	75.9	61.0	2.8
Oct	67.3	73.0	61.0	2.4
Nov	65.7	71.1	55.9	2.8
Dec	64.1	73.0	52.7	5.1

Figure 5.3.16: Drybulb temperature (DBT) for Full record

Annual and monthly averages, min and max values of DBT for the full record

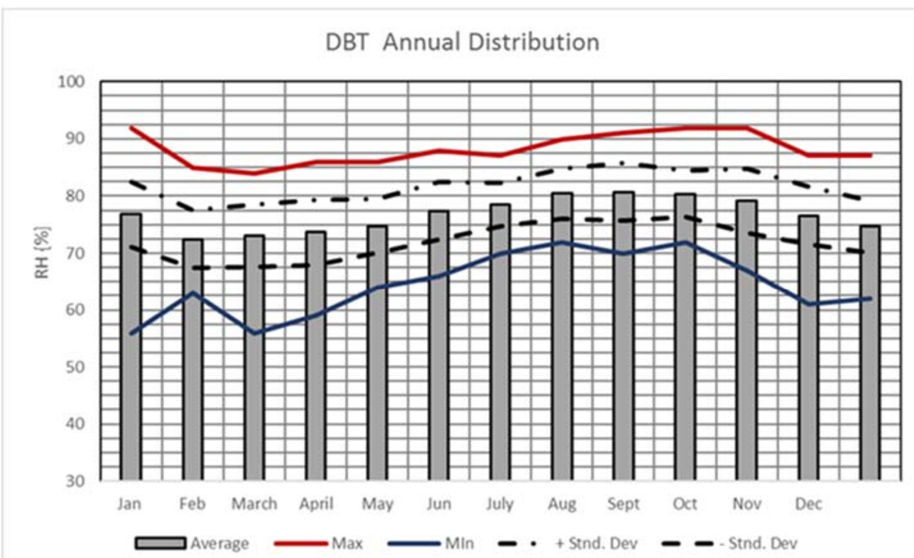


Figure 5.3.17: DBT for Full record;

monthly averages, min and max values of DBT for the full record

	average F	Max F	Min F	stnd. Dev. F
Annual	80.9	91.9	60.1	4.6
Jan	76.1	84.9	64.9	4.4
Feb	77.1	84.0	60.1	4.0
March	78.3	86.0	64.9	3.7
April	78.9	86.0	70.0	3.4
May	82.0	88.0	72.0	3.0
Jun	82.0	87.1	75.0	2.5
July	84.5	90.0	77.0	2.8
Aug	85.4	91.0	75.9	3.1
Sept	83.8	91.9	75.0	3.1
Oct	84.2	91.9	75.0	3.5
Nov	80.4	87.1	64.9	3.8
Dec	77.7	87.1	64.9	3.9

Figure 5.3.18: Drybulb temperature (DBT) for 11-h record

Annual and monthly averages, min and max values of DBT for the 11-h record

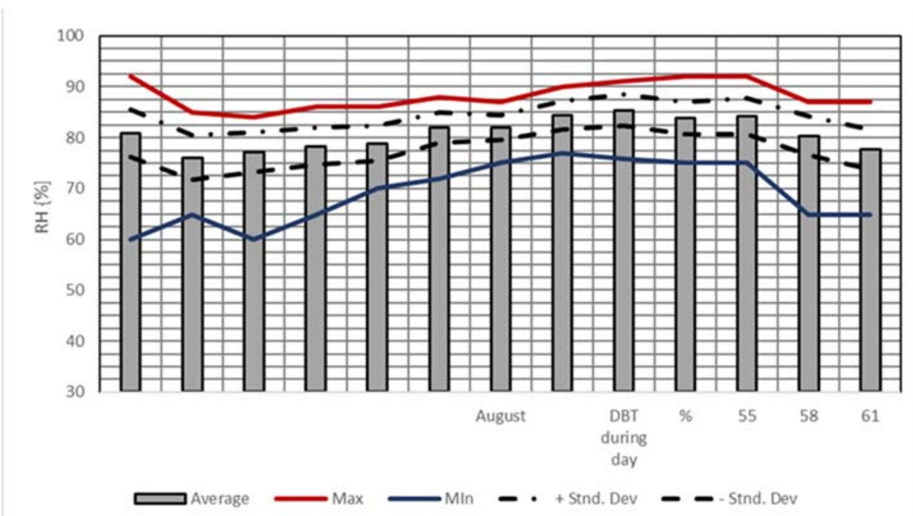


Figure 5.3.19: DBT for 11-h record;

monthly averages, min and max values of DBT for the 11-h record

The monthly average DBT for the full and 11-h record are compared in Figure 5.3.20. The results suggest that the DBT monthly averages are higher for the 11-h record than for the full record. Figure 5.2.21 indicates the monthly values of the DBT values of the 11-h record exceeding those of the full record. This indicates that DBT during daylight hours are significantly above the DBT taken during the entire day.

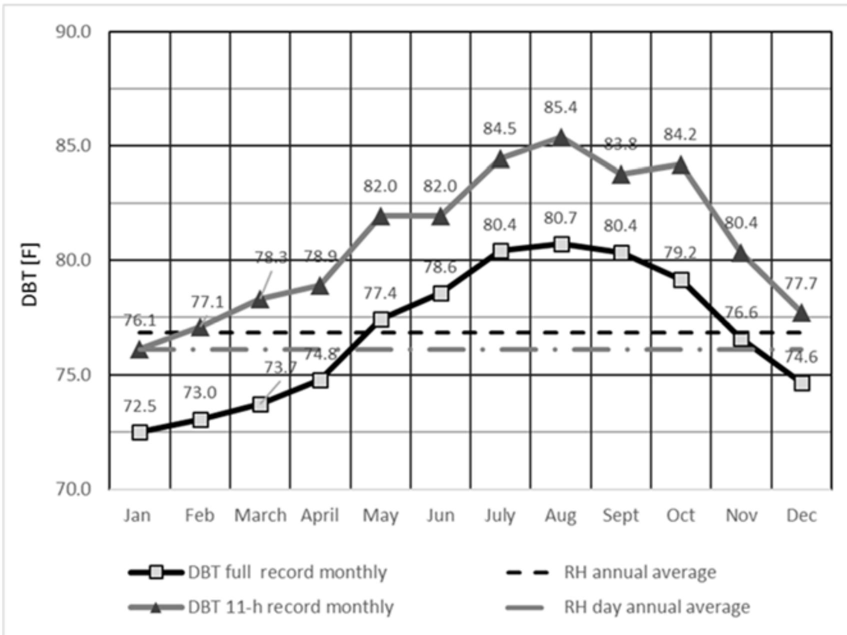


Figure 5.3.20: Comparison of DBT Full record versus 11-h record

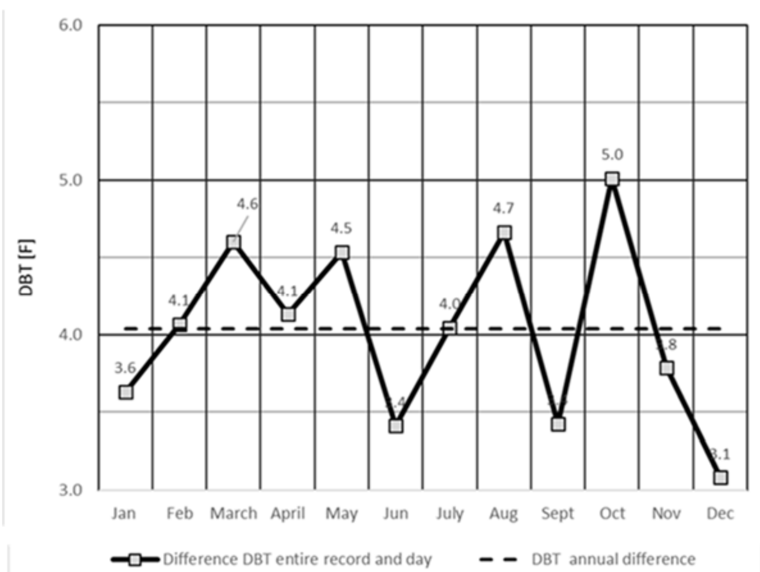
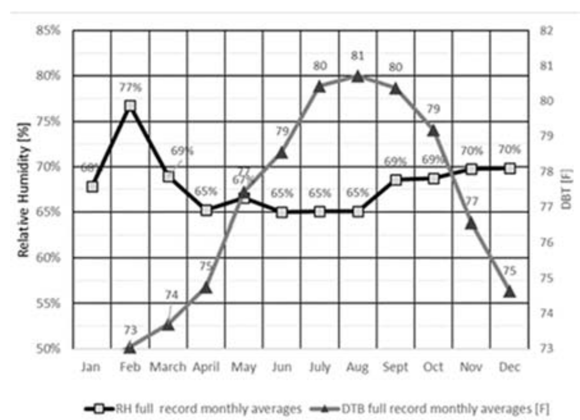


Figure 5.3.21: Comparison DBT Full record and 11-h record; difference between the data of the Full record and the 11-h record

Positive difference inidcates that the DBT of the 11-h record is always higher than the full record.

<u>Mapping dry bulb temperature and relative humidity data:</u> It is important to look at the combination of the dry bulb temperatures and the relative humidity. Figure 5.3.22 (a) and (b) compares the monthly averages for RH and DBT for the full and 11-h records, respectively.

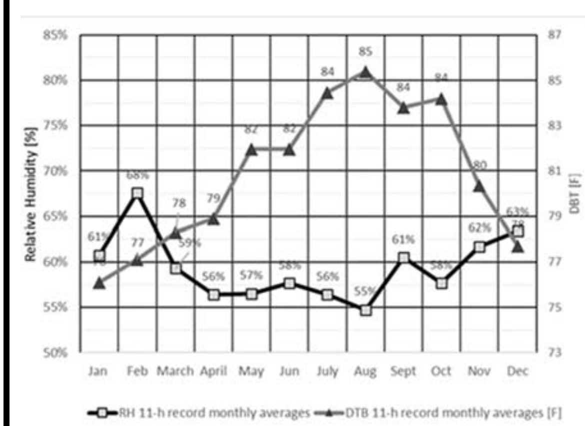
Full record



	RH full record monthly averages	DTB full record monthly averages [F]
Annual	68%	77
Jan	68%	72
Feb	77%	73
March	69%	74
April	65%	75
May	67%	77
Jun	65%	79
July	65%	80
Aug	65%	81
Sept	69%	80
Oct	69%	79
Nov	70%	77
Dec	70%	75

(a) DBT and RH values for Full record

11-h record



	RH 11-h record monthly averages	DTB 11-h record monthly averages [F]
Annual	59%	81
Jan	61%	76
Feb	68%	77
March	59%	78
April	56%	79
May	57%	82
Jun	58%	82
July	56%	84
Aug	55%	85
Sept	61%	84
Oct	58%	84
Nov	62%	80
Dec	63%	78

(b) DBT and RH values for 11-h record

Figure 5.3.22: Dry bulb temperature (DBT) and Relative humidity (RH) annual and monthly averages for the full and 11-h record

Psychrometric charts: Psychrometric data for the full record and the 11-h record is presented in Figure 5.3.23 (a) and (b), respectively. Figure 5.2.24 shows the annual data set of observations on the ASHRAE psychrometric chart (metric). The centroid of all observation of the data set is determined with the 1st moment for the data sets.

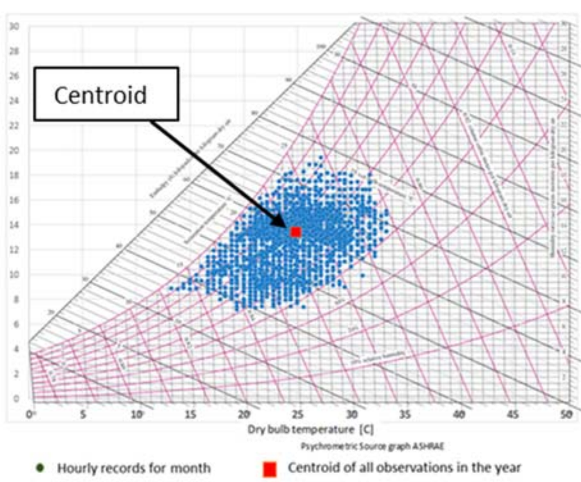
period / month	DBT F	hum. Ratio g/kg	humidity grains / lb	RH %	WBT F	Dew F
Year	76.8	13.329	93.3	67.3	68.8	65.1
1	72.5	11.470	80.3	67.0	64.9	60.9
2	73.1	13.363	93.5	76.3	67.7	65.2
3	73.8	11.470	80.3	64.1	65.3	61.8
4	74.8	11.852	83.0	64.1	66.2	61.8
5	77.5	13.219	92.5	65.2	68.9	64.9
6	78.6	13.488	94.4	64.1	69.6	65.5
7	80.5	14.318	100.2	63.9	71.2	67.1
8	80.8	14.406	100.8	63.6	71.4	67.3
9	80.4	15.110	105.8	67.5	72.1	68.7
10	79.3	14.622	102.4	67.8	71.2	67.7
11	76.6	13.550	94.9	68.8	69.1	65.6
12	74.7	12.790	89.5	69.3	67.5	64.0

(a) Psychrometric variables for the Full record

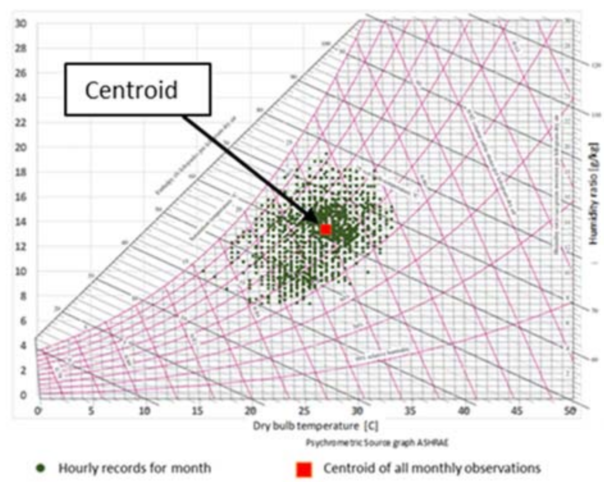
period / month	DBT F	hum. Ratio g/kg	humidity grains / lb	RH %	WBT F	Dew Pnt. F
Year	80.9	13.323	93.3	58.7%	70.0	65.1
Jan	76.1	11.621	81.3	60.2%	66.3	61.3
Feb	77.1	13.504	94.5	67.4%	69.1	65.5
March	78.3	12.233	85.6	58.8%	67.8	62.7
April	78.9	11.819	82.7	55.8%	67.5	61.8
May	82	13.109	91.8	55.8%	70.1	64.7
June	82	13.546	94.8	57.6%	70.7	65.6
July	84.5	14.199	99.4	55.6%	72.2	66.9
Aug.	85.4	14.199	99.4	54.1%	72.4	66.9
Sept	83.8	14.933	104.5	59.8%	72.9	68.3
Oct.	84.2	14.391	100.7	56.9%	72.3	67.3
Nov.	80.4	13.611	95.3	61.0%	70.3	65.7
Dec.	77.7	12.880	90.2	63.1%	68.5	64.2

(b) Psychrometric variables for the 11-h record

Figure 5.3.23: Psychrometric data for the full and 11-h records.



(a) Psychrometric depiction of the annual data set for the Full record



(b) Psychrometric depiction of the annual data set for the 11-h record

Figure 5.3.23: Psychrometric data for the full and 11-h records.

The centroids for the monthly averages of the full and 11-h records are depicted in Figure 5.3.24. The data indicates that the centroid humidity of both data sets are the same, while the centroids of the 11-h record indicate larger dry bulb temperatures.

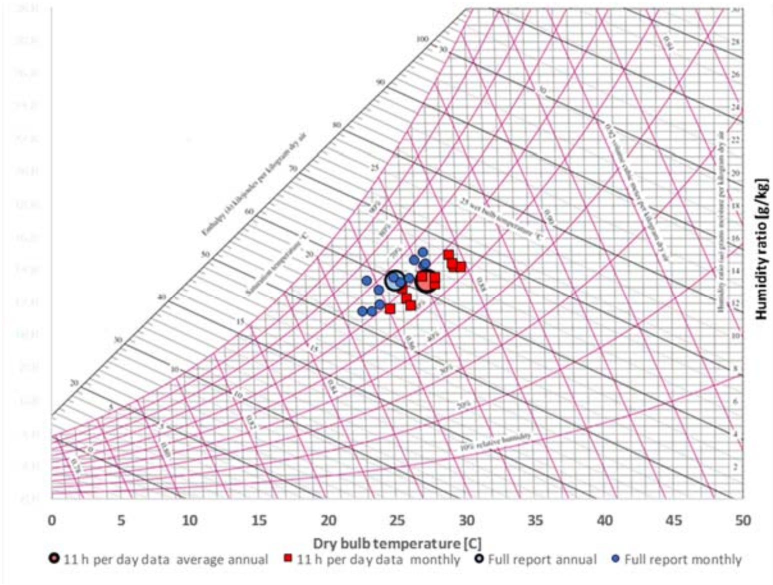


Figure 5.3.24: Psychrometric conditions Centroids for the full record and the 11-h record

The blue data markers depict the monthly average centroids of the full record.

The red data markers depict the monthly average centroids of the full record.

Ventilation Load Indexes (VLI): Harriman (1997) described that the HVAC industry in the 1990s became increasingly concerned with the importance of controlling humidity in buildings. These concerns were based on indoor air quality problems associated with excess moisture in AC systems. Harriman proposed the “ventilation load index” (VLI) as the load generated by one cubic foot per minute of fresh air brought from the weather to space-neutral conditions over the course of one year. The index consists of two numbers, separating the load into dehumidification and cooling components: latent ton-hours per cfm per year + sensible ton-hours per cfm per year. For example, a ventilation air load index of 6.7 + 1.1 indicates that the total annual latent load is 6.7 ton-hours per cfm, and the annual sensible load is 1.1 ton-hours per cfm.

The benefits of using the VLI index are similar to the “degree-day” as shorthand for expressing heating and cooling loads on the envelope of a building, or the “SEER” as a means of expressing the relative efficiency of cooling equipment over time. The VLI allows for quick comparisons between loads in different geographic locations. The author emphasized that the VLI index supports HVAC system design and equipment selection according to climate and amount of outside air.

The VLI index can be calculated with a TMY file to compute hourly latent and sensible loads between the outside weather and the “space-neutral” conditions. The VLI index for a given location is calculated using the differences between outside air temperature and humidity levels and temperature and humidity in the conditioned space. The uniqueness of the method is that calculation are made for every hour of the representative year. The hourly results are summed to form the latent (dehumidification) and the sensible load portion of the index. The calculation does not consider hours when the sensible load and the latent load does not exist, this means when the outside temperature and humidity level are below the reference.

Figure 5.3.25 shows VLI indices for several locations in the US, as presented by the author. The figure indicates where dehumidification or sensible cooling is higher. By separating and quantifying the annual loads for the latent and sensible components of the total load, systems can be configured to control temperature and humidity independently. It must be pointed out though that the VLI index only quantifies the difference in latent and sensible cooling load. The peak load, which is actually used to select the capacity of the cooling is not quantify.

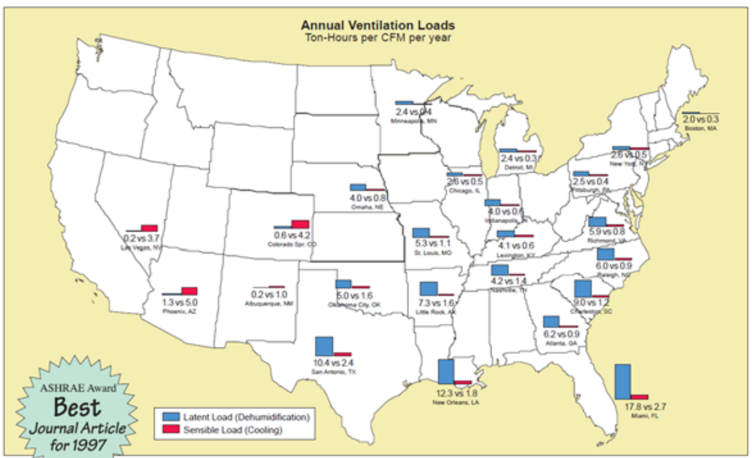
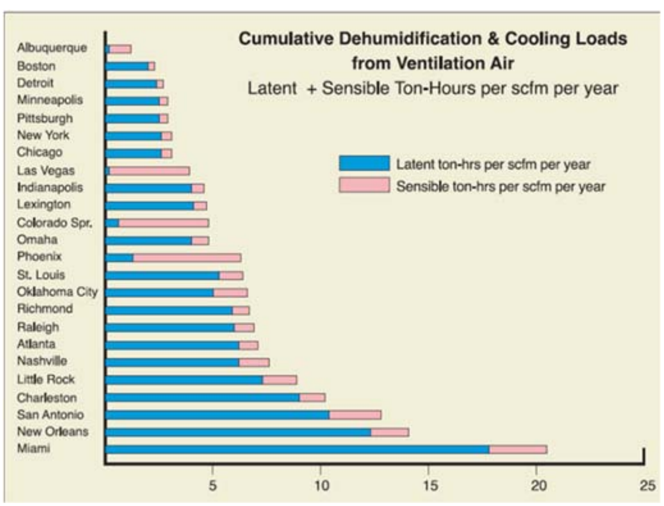


Fig. 1: Map of Ventilation Load Indexes (VLI) for selected continental U.S. locations

Figure 5.3.25: Ventilation Load Indexes (VLI) for selected continental U.S. locations (Harriman et al, 1997)



City	State	Ventilation Load Index (Ton-hrs/scfm/yr) Latent + Sensible	Total	Cumulative Load Ratio Latent:Sensible
Albuquerque	NM	0.2 + 1.0	1.2	0.2:1
Boston	MA	2.0 + 0.3	2.3	6.4:1
Detroit	MI	2.4 + 0.3	2.7	7.4:1
Minneapolis	MN	2.4 + 0.4	2.8	6.2:1
Pittsburgh	PA	2.5 + 0.4	2.9	5.8:1
New York	NY	2.6 + 0.5	3.1	5.1:1
Chicago	IL	2.6 + 0.5	3.1	5.0:1
Las Vegas	NV	0.2 + 3.7	3.9	0.04:1
Indianapolis	IN	4.0 + 0.6	4.6	6.6:1
Lexington	KY	4.1 + 0.6	4.7	7.4:1
Colorado Spr.	CO	0.6 + 4.2	4.8	0.1:1
Omaha	NE	4.0 + 0.8	4.8	5.3:1
Phoenix	AZ	1.3 + 5.0	6.2	0.3:1
St. Louis	MO	5.3 + 1.1	6.4	4.7:1
Oklahoma City	OK	5.0 + 1.6	6.6	3.2:1
Richmond	VA	5.9 + 0.8	6.7	7.2:1
Raleigh	NC	6.0 + 0.9	6.9	6.8:1
Atlanta	GA	6.2 + 0.9	6.9	6.7:1
Nashville	TN	6.2 + 1.4	7.6	4.6:1
Little Rock	AK	7.3 + 1.6	8.8	4.7:1
Charleston	SC	9.0 + 1.2	10.3	7.3:1
San Antonio	TX	10.4 + 2.4	12.8	4.4:1
New Orleans	LA	12.3 + 1.8	14.1	6.8:1
Miami	FL	17.8 + 2.7	20.5	6.7:1

Since Hawaii was not included in Harriman's list of VLI for US locations the VLI for Honolulu was calculated using the same "space-neutral" conditions as 75°F, 50%rh (65 gr/lbs.), which the Harriman used and referred to as "consistent with human comfort". Figure 5.3.26 compares the resulting VLI for Honolulu (14.0 vs. 2.6) with the some of the locations in Harriman's paper.

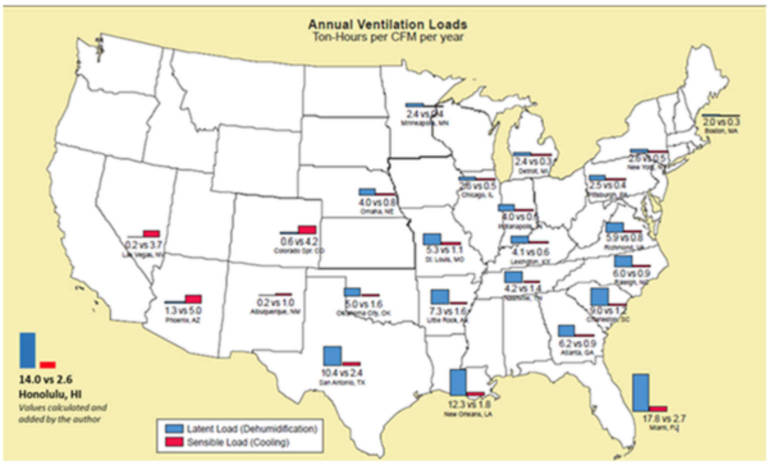
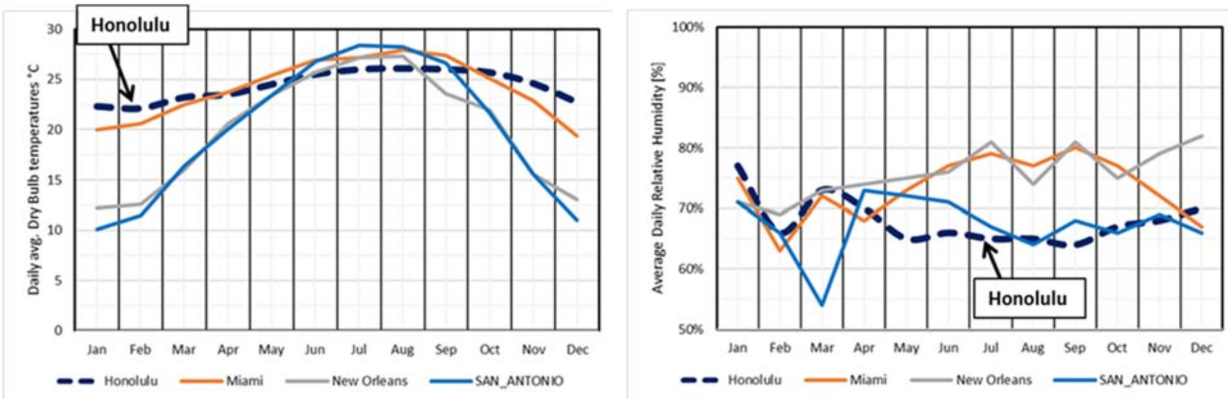


Figure 5.3.26: Ventilation Load Indexes (VLI) calculated for Honolulu and added to data by Harriman (1997)

		ton-hours per cfm per year			Load Ratio		
		Latent	Sensible	Sum	Latent:Sensible		
Phoenix	AZ	1.3	5	6.2	0.3	:	1
St. Louis	MO	5.3	1.1	6.4	4.7	:	1
Oklahoma City	OK	5	1.6	6.6	3.2	:	1
Richmond	VA	5.9	0.8	6.7	7.2	:	1
Raleigh	NC	6	0.9	6.9	6.8	:	1
Atlanta	GA	6.2	0.9	6.9	6.7	:	1
Nashville	TN	6.2	1.4	7.6	4.6	:	1
Little Rock	AK	7.3	1.6	8.8	4.7	:	1
Charleston	SC	9	1.2	10.3	7.3	:	1
San Antonio	TX	10.4	2.4	12.8	4.4	:	1
New Orleans	LA	12.3	1.8	14.1	6.8	:	1
Honolulu,	HI	14.0	2.6	16.6	5.4	:	1
Miami	FL	17.8	2.7	20.5	6.7	:	1

Among the locations listed, the VLI value of 14.0 vs 2.6 identifies Honolulu as second to Miami in regard to combined latent and sensible loads.

Figure 5.3.27 shows a comparison of the dry bulb temperatures a relative humidity levels of Honolulu, San Antonio, New Orleans and Miami, which are the four pertinent locations with the highest VLI values. Figure 5.3.27 indicates that Honolulu does not depict significant seasonal temperature swings, such as in as New Orleans and San Antonio, and has lower annual RH values than Miami.



Annual distribution of monthly average dry bulb temperatures (DBT)

Annual distribution of monthly average relative humidity (RH)

Figure 5.3.27: Annual distribution of monthly average DBT and RH for four locations with the highest VLI

Summary: The typical Hawaii climate features high temperatures and humidity levels throughout the year. The climatic data stresses the need for dehumidification year round. While there are certain seasonal temperature and humidity swings in the Hawaii climate, they are well less pronounced than I all other locations in the US, save Miami. Two data sets were obtained from the TMY file for Honolulu, a “full record” and a “11-hour (h)” data record, which represent all 8670 hourly TMY data points and data points between 8:00 am and 7:00 pm, respectively. The 11-h record represent typical climatic conditions during business hours with space conditioning. The results suggest that the average temperatures and relative humidity levels differ between the two records.

5.4 Solar Heat and other Low Temperature Sources

Solar heat or other cost-effective low temperature heat are of importance to support the economic argument for desiccant cooling. Heat is required for the regeneration of the desiccant material and also for thermally driven solar. Using solar heat or low temperature waste heat reduces the energy costs for desiccant regeneration.

Figure 5.4.1 illustrates that the following discussion of generic key characteristics of desiccant and cooling technologies related to the framework introduced in Section 5.1 is concerned with the availability of solar heat or other low temperature heat source in Hawaii. This section discusses the aspect of adding process heat derived from solar thermal energy conversion and other low temperatures sources to the feasibility assessment of desiccant cooling

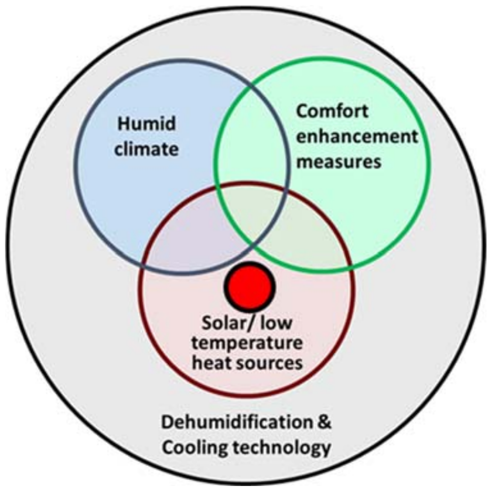


Figure 5.4.1: Key characteristics of desiccant and cooling technologies – Solar or low temperature heat sources

Desiccant cooling technology performance with solar or low temperature heat sources: In this case the economic feasibility of desiccant and sensible cooling technology is supported by using solar derived or other low temperature heat sources. Using alternative heat sources (e.g. alternatives to conventional fuel such as natural gas) for desiccant regeneration presents a significant cost and environmental benefit.

Solar heat can be obtained from various types of solar thermal energy converters, e.g. solar thermal collectors. There are two main types of solar thermal panel, focusing and non-focusing solar thermal collectors. Global irradiance on horizontal surfaces, which is described by global horizontal irradiance [GHI], is the measure of the density of the available solar resource per surface area. Global irradiance could also be defined on “optimum” tilt angle for collectors, i.e. for a receiving surface oriented towards the Equator tilted to maximize the received energy over the year.

The effectiveness of direct irradiance uses the metric of direct normal irradiance (DNI) – the direct beam irradiance received on a surface perpendicular to the sun’s rays. Thus DNI is a useful measure for tracking surface and it can be regarded as the maximum solar resource that can be used. The respective proportions of direct and diffuse irradiance are of primary importance for collecting the energy from the sun and have many practical implications. Non-concentrating technologies take advantage of the global radiation, direct and diffuse (including the reflections from the ground or other surfaces) and do not require tracking.

Values for the two metrics of estimating the solar thermal potential for the focusing and non-focusing solar thermal collectors, e.g. the Direct Normal Irradiance (DNI) and the Global horizontal irradiance (GHI), respectively, values have been calculated with the TMY file for Honolulu. The values for Honolulu DNI and GHI are presented in Tables 5.4.1 and 5.4.2, respectively.

	average [W/m^2]	Max [W/m^2]	Min [W/m^2]	sum [Wh/m^2]	hours [h]	DNI kWh/m2 /day
Jan	394	964	1	142,513	362	4.6
Feb	415	936	1	123,950	299	4.4
March	388	908	1	139,772	360	4.5
April	383	986	2	148,896	389	5.0
May	388	926	1	164,311	424	5.3
June	379	880	1	170,275	449	5.7
July	425	934	1	184,112	433	5.9
Aug	472	944	1	195,868	415	6.3
Sept	420	913	5	156,144	372	5.2
Oct	405	966	1	157,113	388	5.1
Nov	372	954	1	132,866	357	4.4
Dec	409	964	1	139,147	340	4.5
				1,854,967	4,588	5.1
			kWh/m^2>>>	1,855		average

Table 5.4.1: Direct Normal Irradiance (DNI) values for Honolulu

The highlighted column DNI indicates the monthly average of the daily irradiation received.

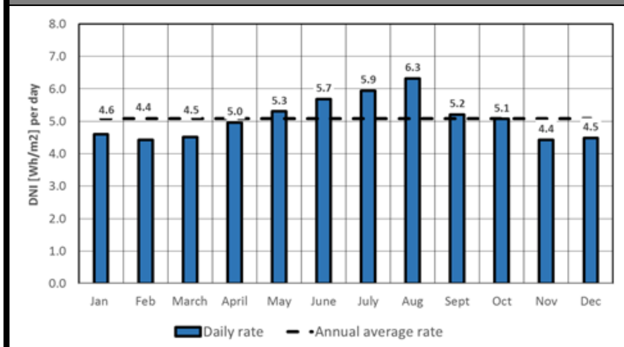
	average [W/m^2]	Max [W/m^2]	Min [W/m^2]	Stnd. Dev [W/m^2]	sum [Wh/m^2]	hours [h]	GHI kWh/m2 /day
Jan	339	816	1	235	123,502	364	4.0
Feb	374	884	2	266	125,602	336	4.5
March	412	1039	1	294	162,564	395	5.2
April	447	1101	13	294	174,458	390	5.8
May	453	1071	1	314	195,549	432	6.3
June	433	1046	1	310	194,737	450	6.5
July	458	1079	1	318	202,251	442	6.5
Aug	489	1036	2	310	203,056	415	6.6
Sept	464	982	1	293	174,566	376	5.8
Oct	399	942	2	273	155,505	390	5.0
Nov	348	822	3	232	124,484	358	4.1
Dec	338	788	1	218	116,737	345	3.8
					1,953,011	4,693	5.3
				kWh/m^2 >>>	1,953		average

Table 5.4.2: Global horizontal irradiance (GHI) values for Honolulu

The highlighted column GHI indicates the monthly average of the daily irradiation received.

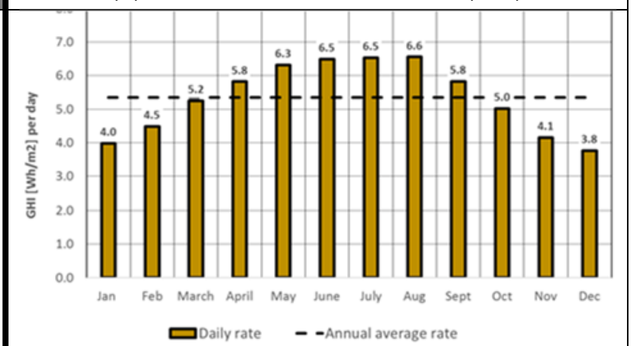
The annual distribution of DNI and GHI for Honolulu is depicted in Figure 5.4.2 (a) and (b), respectively.

(a) Direct Normal Irradiance (DNI)

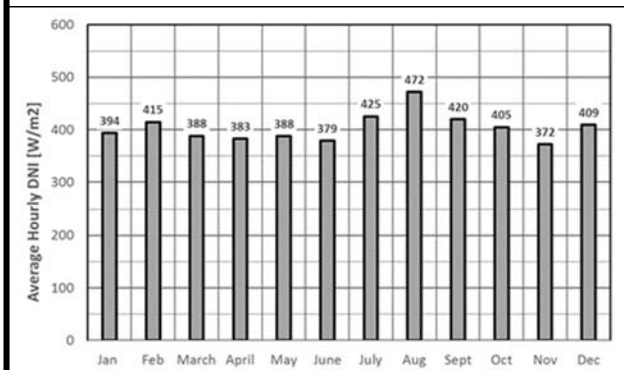


DNI – average monthly value of daily DNI (energy received) in kWh per m^2 per day

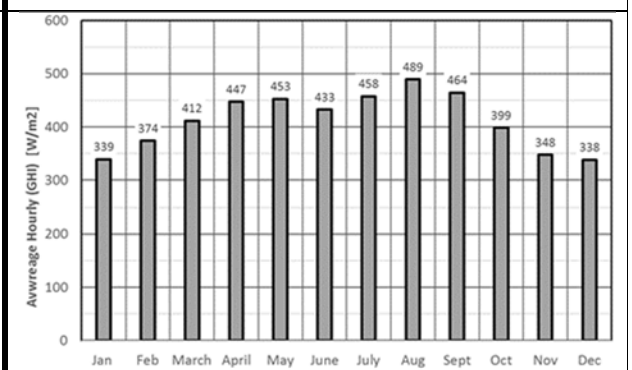
(b) Global horizontal irradiance (GHI)



GHI– average monthly value of daily GHI (energy received) in kWh per m^2 per day



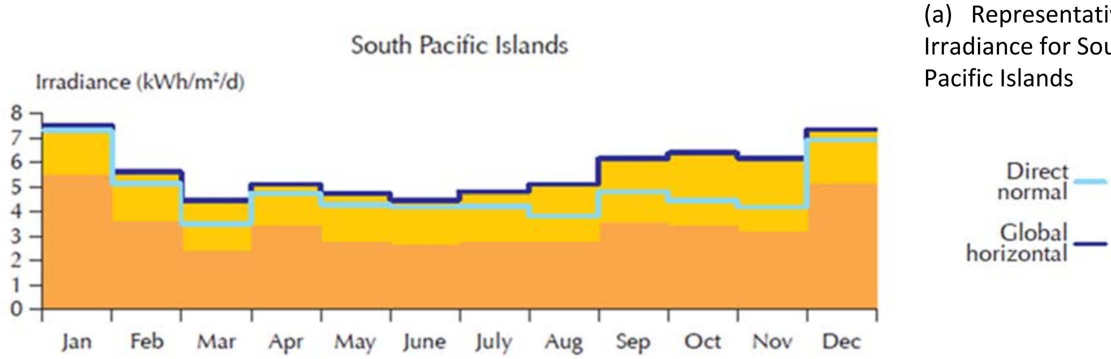
DNI – average monthly value of hourly DNI (energy flux) in W per m^2



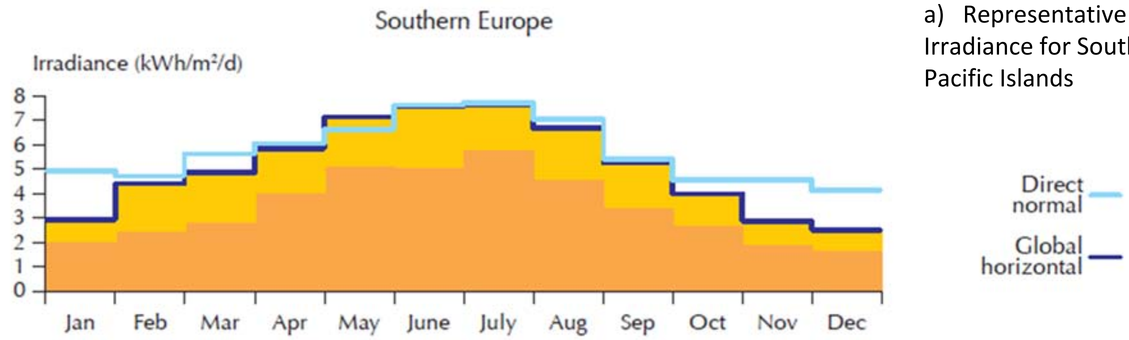
GHI – average monthly value of hourly GHI (energy flux) in W per m^2

Figure 5.4.2: annual distribution of DNI and GHI for Honolulu

The annual distribution of DNI and GHI values were compared to data published by the IEA for South Pacific and South Europe. Both of these regions have been identified as having significant potential for solar thermal energy conversion. Figure 5.4.3 (a) and (b) shows representative values for DNI and GHI for the South Pacific and South Europe regions, respectively. Table 5.4.2 shows the annual distribution of DNI and GHI values for Honolulu and compares them to the repetitive values for South Pacific Islands and South Europe. Figures 5.4.4 shows illustrates the values in Table 5.4.3. The comparison between representative values in is of significance since desiccant cooling technology has been tested or is being evaluated for regions outside Honolulu. The selection of the appropriate solar energy conversion technology will depend on the required solar energy demand but also on the absolute process temperature for desiccant regeneration.



(a) Representative Irradiance for South Pacific Islands

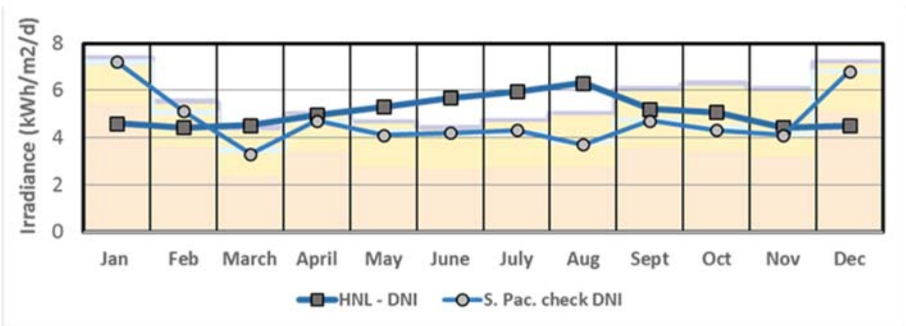


a) Representative Irradiance for South Pacific Islands

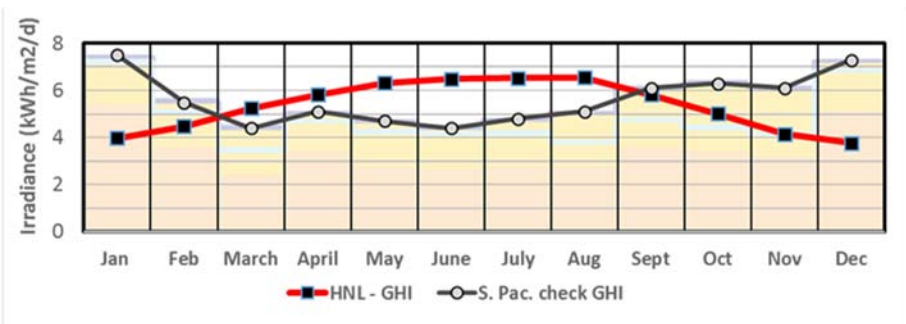
Figure 5.4.3: Representative Irradiance for South Pacific Islands and South Europe (source IEA, 2011)

	Honolulu, Hawaii		Representative values for South Pacific Islands		Representative values for South Europe	
	DNI kWh/m2 /day	GHI kWh/m2 /day	DNI kWh/m2 /day	GHI kWh/m2 /day	DNI kWh/m2 /day	GHI kWh/m2 /day
Jan	4.6	4.0	7.2	7.5	4.8	2.8
Feb	4.4	4.5	5.1	5.5	4.5	4.3
March	4.5	5.2	3.3	4.4	5.3	4.8
April	5.0	5.8	4.7	5.1	5.7	5.8
May	5.3	6.3	4.1	4.7	6.4	7.0
June	5.7	6.5	4.2	4.4	7.4	7.4
July	5.9	6.5	4.3	4.8	7.5	7.5
Aug	6.3	6.6	3.7	5.1	6.8	6.6
Sept	5.2	5.8	4.7	6.1	5.2	5.1
Oct	5.1	5.0	4.3	6.3	4.3	4.0
Nov	4.4	4.1	4.1	6.1	4.4	2.8
Dec	4.5	3.8	6.8	7.3	4.1	2.4
average >>	**5.1**	**5.3**	**4.7**	**5.6**	**5.5**	**5.0**

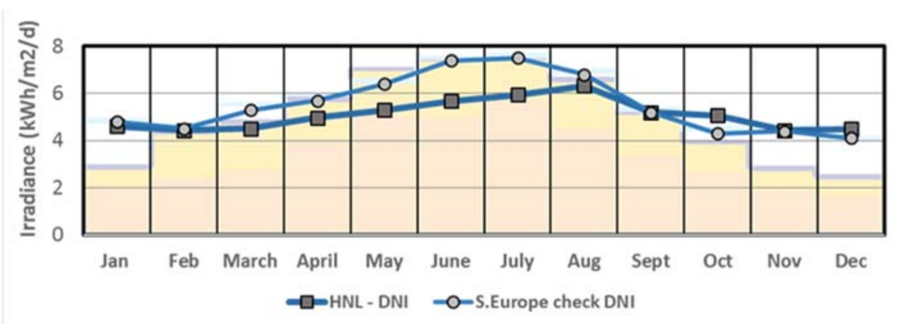
Table 5.4.3: Annually distribution of DNI and GHI values for Honolulu and representative values for South Pacific Island and South Europe



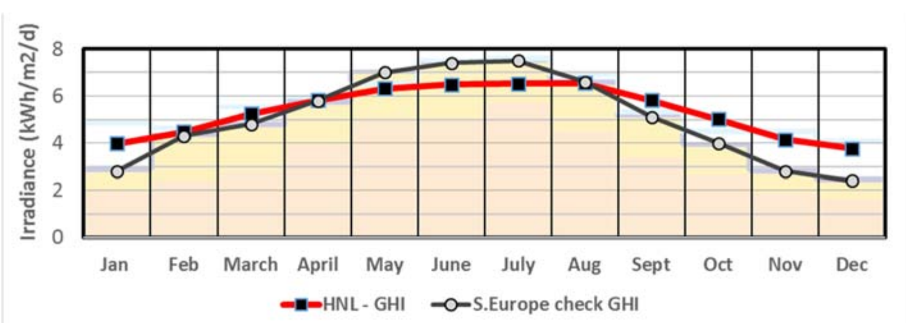
DNI Honolulu vs. representative South Pacific Islands



GHI Honolulu vs. representative South Pacific Islands



DNI Honolulu vs. representative South Europe



GHI Honolulu vs. representative South Europe

Figure 5.4.4: Comparison of annually distribution of DNI and GHI values for Honolulu and representative values for South Pacific Island and South Europe

Other forms of low temperature heat sources: In addition to solar heat, other forms of low temperature heat can be used to drive regeneration of desiccant material and thermally derived sensible cooling. These low temperature sources include especially waste heat from cogeneration. Figure 5.4.5 shows Hawaii's electricity production by source (Source: EIA). As can be seen 84% of the electricity is produced by thermal power plants using petroleum and coal. Another 6% use thermal power plants driven by biofuel and geothermal. Therefore, as a sum about 90% of Hawaii's electricity production uses heat converted to electricity, and therefore ha to reject significant amounts of waste heat.

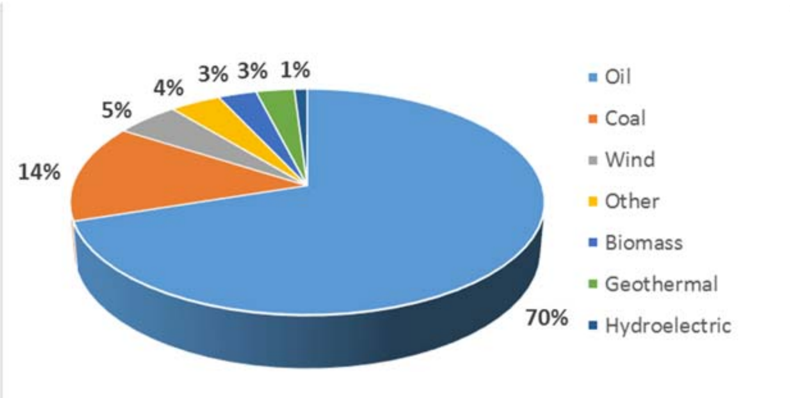


Figure 5.4.4: Hawaii's Electricity Production by Source (Source: EIA)

Using values for heat rates for petroleum and coal based power plants published by the EIA for 2004 through 2014, the average heat rate of Hawaii's thermal power plants is about 10,600 BTU per kWh or electricity produced. Heat rate is one measure of the efficiency of a generator or power plant that converts a fuel into heat and into electricity. Energy Information Administration (EIA) expresses heat rates in British thermal units (Btu) per net kWh generated. Applied to Hawaii this means that about two thirds of the energy source input in form of petroleum and coal is discharged to the environment in the form of waste heat. This reflects an annual amount of waste heat in the order of 5.5 8 10^{12} BTU.

Cogeneration, which means combined heat and power electricity production, can capture waste heat and convert it to useful energy for desiccant regeneration and thermally powered chillers. Installing cogeneration plants close to the demand for process heat would be a "game changer" for Hawaii. Using the presently used fossil fuels petroleum and coal would require oil and coal transported to decentralized cogeneration power plants. This would entail significant logistic of transporting the fuel over roads and storing it close to the cogeneration plants. The use of natural gas would solve, or at least significantly mitigate, the transport logistic challenges associated with petroleum or coal as cogeneration fuel. Gas can be piped to the cogeneration plant and does not require on-site storage. In addition, natural gas is a cleaner burning fuel than petroleum (especially heavy petroleum fractions) and coal and environmental impacts would be significantly smaller.

At the present time, natural gas is not available in Hawaii and synthetic natural gas, produced in Hawaii, takes the place of gaseous energy source. According to future energy plans of Hawaii, natural gas could be shipped to the islands and distributed in the existing pipeline system. Thus natural gas could be a viable energy source for the future and desiccant cooling could take advantage of waste heat from cogeneration plants powered by natural gas. Biofuel is an alternative fuel for cogeneration and would be a good fuel type for Hawaii to reach Clean Energy goals.

Summary: Solar heat and other low temperature heat sources create cost advantages for desiccant cooling applications. The solar resource for Hawaii is assessed by means of the two main metrics Global horizontal irradiance (GHI) and Direct Normal Irradiance (DNI). The solar resource of Honolulu compared favorably to representative values of the South Pacific Islands and South Europe. About 84% of the electricity in Hawaii is used by thermal power plants, which discharge about two thirds of the thermal potential of the petroleum and coal fuel input as waste heat to the ocean. The use of cogeneration, or combined heat and power (CHP), can open use a significant source of virtually free energy for desiccant. Besides conventional types of fuels, such as petroleum and coal, natural gas could be future attractive fuel choice for CHP, as well as biofuels produced as Hawaii indigenous fuel.

5.5 Comfort Related Measures that Support Desiccant Cooling Applications

This section discusses aspects related to occupant comfort which can increase the feasibility of desiccant cooling applications (see Figure 5.5.1). Comfort related aspects do not directly improve the performance of the desiccant cooling systems but improve the environment in which a desiccant cooling systems operates, by lowering cooling needs.

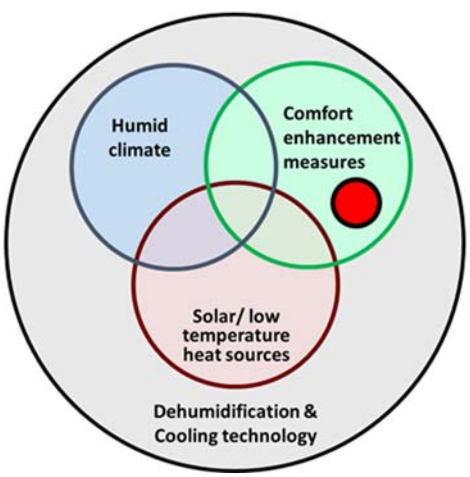


Figure 5.5.1: Key characteristics of desiccant and cooling technologies – Comfort enhancement efforts

<u>Feasibility of desiccant cooling technology is affected by the readiness of the occupants for innovative space conditioning.</u> While comfort related aspects do not necessarily affect the technical performance of desiccant cooling applications, indoor comfort creates a suitable framework to implement innovative and energy saving space conditioning. Recent developments of new comfort standards depart from the conventional understanding of occupant comfort as they take advantage of the subjective comfort experience of occupant, who favor thermal variation and other properties that establish the indoor environment.

Desiccant cooling systems are too often considered merely to replace components of conventional HVAC systems, operating in the same paradigm as conventional HVAC systems. This is a somewhat limiting approach, since desiccant cooling applications are uniquely qualified to bring about significant changes to the HVAC changes and open up new opportunity to align to changing understanding of occupant comfort.

The ultimate goal of space conditioning is to provide a healthy and comfortable indoor air quality and thermal environment. In accordance with ASHRAE "*thermal comfort is the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation*". The conventional HVAC approach focuses on providing thermal conditions which are the same throughout the space and do not change over time. The level of comfort is basically defined as a temperature setting which is a uniformly distributed inside the space.

Comfort, however, is a subjective experience by the occupants, who appreciate spatial variations of the indoor climate conditions, in the same way as humans prefer the outdoor non-steady state environment over an artificially perceived indoor environment. The conventional understanding of comfort does not typically consider properties that are unevenly distributed inside the conditioned space. An example is moving air above a relatively low air speed threshold inside the conditioned space which is considered a nuisance, or draft, in the conventional indoor comfort assessment. Newer developments and understanding of indoor comfort, however, use increased air speed as a cooling effect without the need to lower the air temperature in the space.

Preferred comfort settings: The standard ASHRAE 55 provides metrics to evaluate if thermal comfort conditions are met or not. ASHRAE 55 is a guidance document only and lacks the enforcing nature of a code. Many building operators, however, have decided to follow the ASHRAE 55 standard. Figure 5.5.2 shows the ASHRAE 55-2013 definition of acceptable indoor thermal conditions.

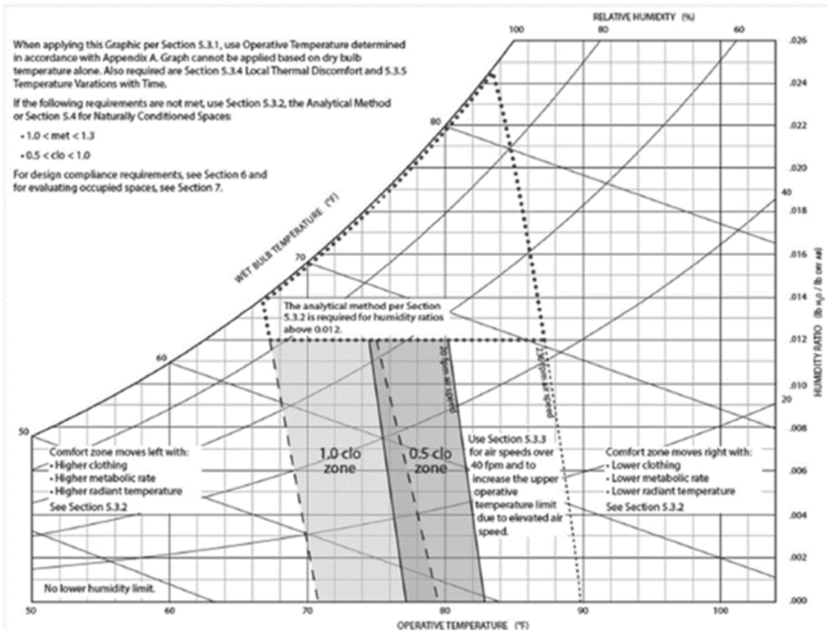


Figure 5.5.2: Defining the boundaries of acceptable indoor comfort in accordance with ASHRAE 55

The shades areas represent acceptable indoor thermal conditions depending on the clothing factor, Clo

Figure 5.5.3 suggests that acceptable indoor thermal conditions can be attain in a wide range of dry bulb temperature and relative humidity conditions, including high humidity values of 80% or even above. In practical applications, however, building operators prefer a more conservative selection of dry bulb temperatures (DBT) and relative humidity (RH) ranges in order to assure not only satisfactory thermal indoor conditions but also healthy indoor air quality.

A suitable and conservative rage is 22 to 27°C of DBT (72 to 81 F) and 40% to 60% of RH. The humidity range, in particular, needs not to be too high and not too low to avoid unwanted and potentially hazardous indoor air quality conditions. Figure 5.5.3 illustrates the recommended range of 40% to 60% RH.

The data sets for the dry bulb temperatures and relative humidity for the full and 11-h record were analyzed to determine the probability of outdoor air being outside the referred range, thus requiring cooling and/or dehumidification of the supply air. Figure 5.5.4 and 5.5.5 show how the hourly psychrometric data points relate to the preferred range for the full and 11-h record, respectively. Table 5.5.1 shows characteristics of the full and 11-h data record as they are depicted in Figure 5.5.4 and 5.5.5, respectively.

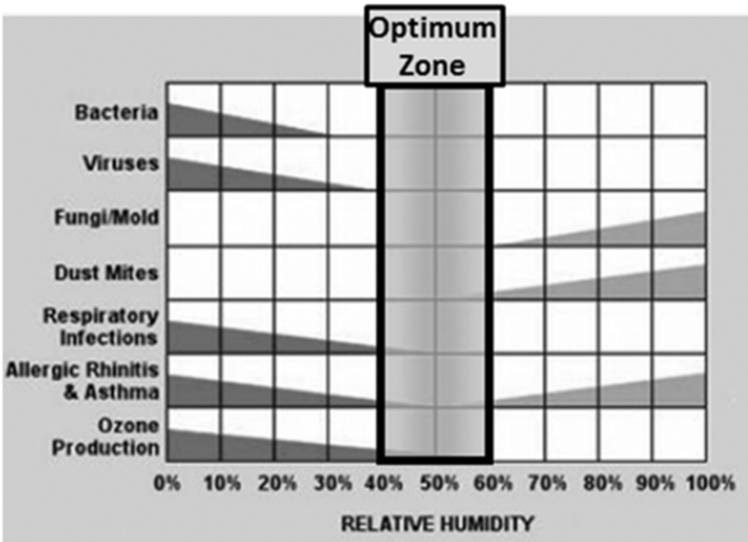


Figure 5.5.3: Recommended indoor rage of relative humidity (source ASHRAE recommendations)

The optimum zone of indoor relative humidity (RH) is between 40% and 60%. At RH values either higher of lower unwanted and potentially hazardous indoor air quality conditions occur and affect occupants and the indoor environmental quality.

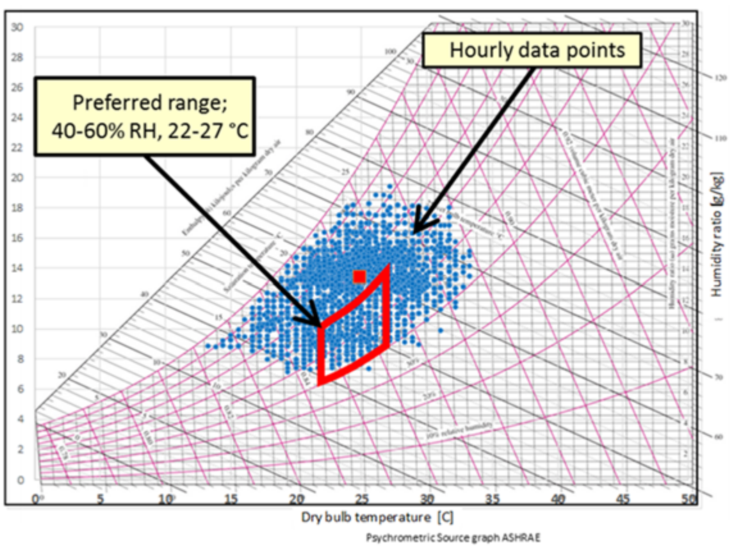


Figure 5.5.4: Hourly data point of DBT an RH, Full record

The red frame indicates the 72F to 81F DBT and 40% to 60% RH comfort range as defined before.

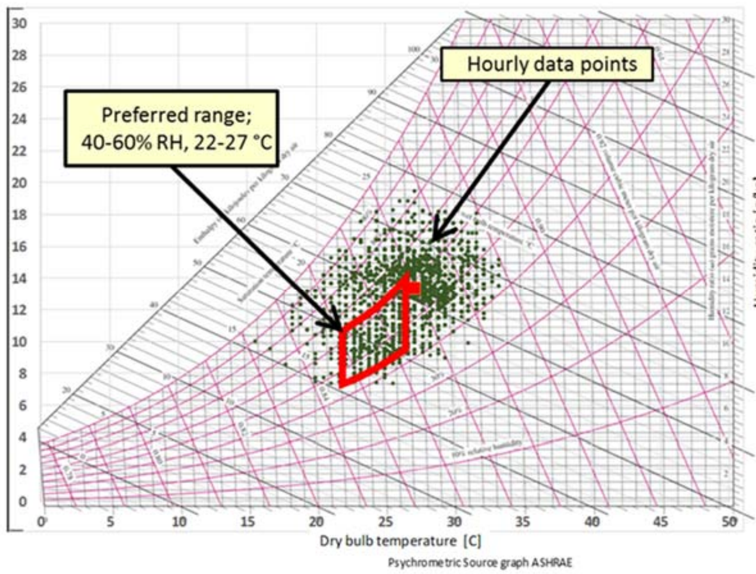


Figure 5.5.5: Hourly data point of DBT an RH, 11-h record

The red frame indicates the 72F to 81F DBT and 40% to 60% RH comfort range as defined before.

Table 5.5.1 and Figure 5.5.6 show and compare the annual distribution of data points that are outside the comfort range for the full and 11-h data record. The data in Table 5.5.1 and Figure 5.5.6 suggest that the data points for both the full and the 11-h data records are more frequently outside the comfort range during the summer months.

Table 5.5.2 present the percentages of data points for both the full and 11-h data record during which either cooling or dehumidification is required in order to stay inside the comfort range (72F to 81F DBT; 40% to 60% RH). Figures 5.5.7 and 5.5.8 shows the cooling and dehumidification needs for the full and 11-h data records, respectively. The data presented in Table 5.5.2 and Figures 5.5.7 and 5.5.8 suggest that the cooling and dehumidification needs are different between the full and 11-h data records. During the full data record, which includes day and night times over the entire year, dehumidification requirements surpass cooling needs. During the daytime hours, e.g. for the 11-h-data record, the cooling needs during the summer months are significantly larger than the dehumidification needs.

It should be noted at this point that the definition of dehumidification and cooling needs consider only the outdoor DBT and RH being larger than 60% and 81F, respectively.

Table 5.5.1: Percentage of hourly data points of outside air outside the comfort range (see Figures 5.5.4 and 5.5.5) for the Full and the 11-h data records

	Full record		11-h record	
	8,760 data points		4,015 data points	
	% of montly record hours	% of yearly record hours	% of montly record hours	% of yearly record hours
year	N/A	92%	N/A	84%
Jan	81%	8%	64%	5%
Feb	89%	7%	77%	6%
March	83%	8%	62%	5%
April	83%	7%	65%	5%
May	93%	9%	88%	7%
June	95%	9%	91%	7%
July	99%	9%	99%	8%
Aug.	99%	9%	99%	8%
Sept	99%	9%	99%	8%
Oct.	99%	9%	99%	8%
Nov.	95%	9%	89%	7%
Dec.	84%	8%	72%	6%

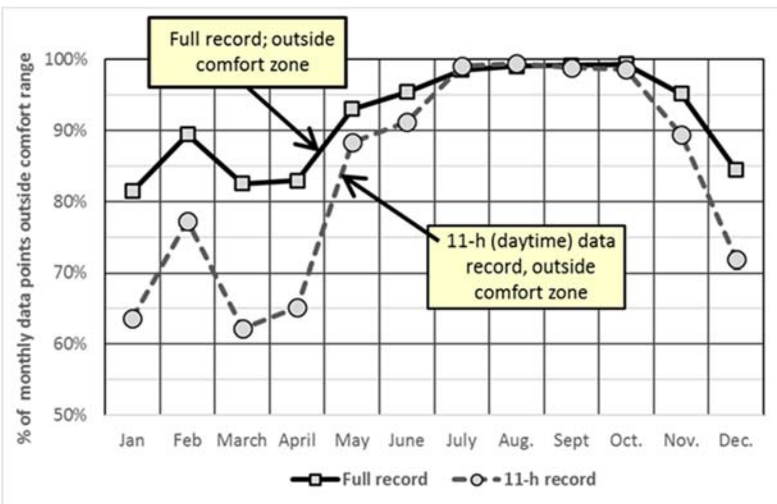


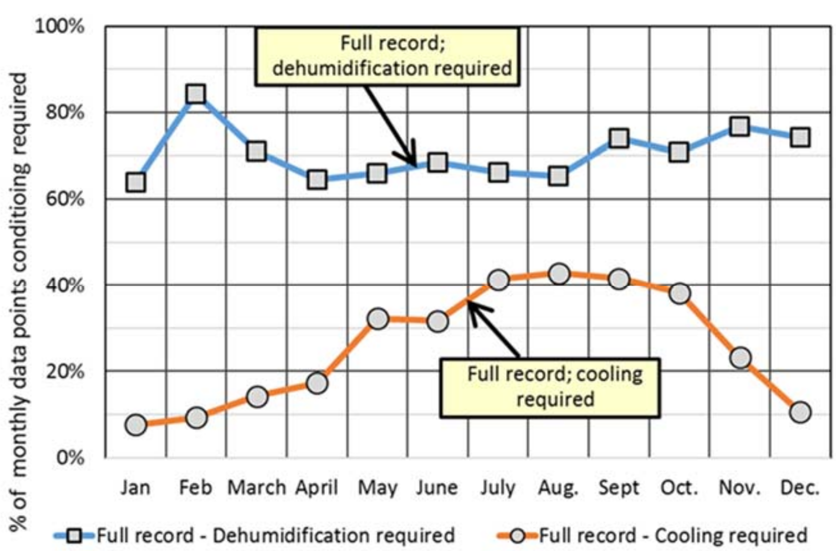
Figure 5.5.6: Data points of outside air outside the comfort range (see Figures 5.5.4 and 5.5.5) for the Full and the 11-h data records

Note: the data depicted refers to the data point, which are the hourly records.

Note: the conditions indicated are for the outdoor air only

Table 5.5.2: Percentage of required dehumidification and cooling during hours of operation of the full and 11-h record

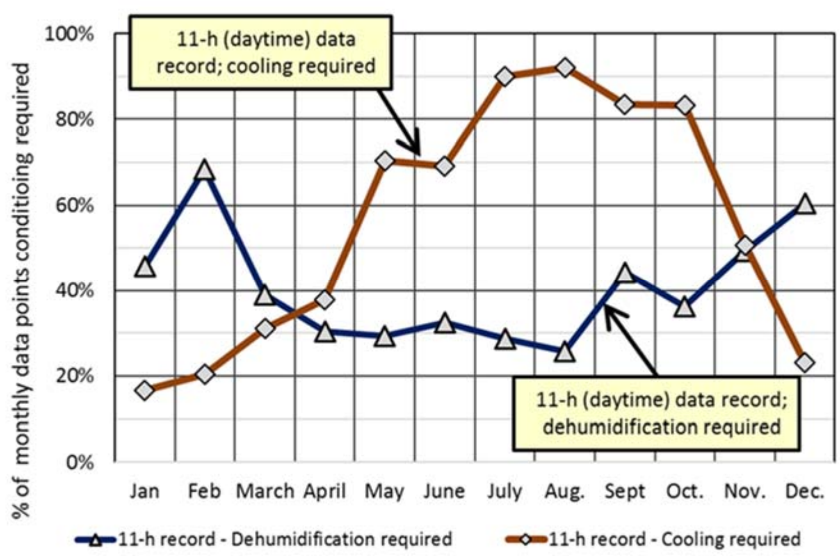
	Full record (8760 data points)				11-h record (4015 data points)			
	Dehumidification required		Cooling required		Dehumidification required		Cooling required	
	% of montly record hours	% of yearly record hours	% of montly hours	% of yearly observations	% of montly hours	% of yearly observations	% of montly hours	% of yearly observations
Year	N/A	70%	N/A	26%	N/A	19%	N/A	26%
Jan	70%	5%	8%	1%	46%	2%	17%	1%
Feb	64%	6%	9%	1%	69%	2%	20%	1%
March	84%	6%	14%	1%	39%	2%	31%	1%
April	71%	5%	17%	1%	30%	1%	38%	1%
May	64%	6%	32%	3%	29%	1%	70%	3%
June	66%	6%	32%	3%	32%	1%	69%	3%
July	68%	6%	41%	4%	29%	1%	90%	4%
Aug.	66%	6%	43%	4%	26%	1%	92%	4%
Sept	65%	6%	42%	3%	44%	2%	83%	3%
Oct.	74%	6%	38%	3%	36%	1%	83%	3%
Nov.	71%	6%	23%	2%	49%	2%	51%	2%
Dec.	77%	6%	11%	1%	60%	2%	23%	1%
	70%		26%		41%		56%	
	average		average		average		average	



Figures 5.5.7: Cooling and dehumidification needs for the FULL data records

This data record includes **all hourly climatic** observation of the TMY file.

Note: The conditions indicated are for the outdoor air only. Indication of “cooling required” indicate that the outdoor air has to be cooled.



Figures 5.5.8: Cooling and dehumidification needs for the 11-h data record

This data record includes the hourly climatic observation for the **daytime hours** only.

Note: the conditions indicated are for the outdoor air only. Indication of “cooling required” indicate that the outdoor air has to be cooled.

Hawaii specific aspects to attain occupant comfort: The conventional approach of determining occupant comfort in conditioned spaces is based on the metrics of Predicted Mean Vote (PMV) and the Predicted Percentage of Dissatisfied (PPD). PMV is assessed on the basis of six parameters, four of which are objective parameters valid for the indoor thermal environment and two are individual parameters of the occupants. The four objective parameters include the dry bulb temperature, the mean radiant temperature, the air speed and the relative humidity. The individual parameters are the metabolic rate and the clothing insulation, which vary between occupants. Therefore, occupants of a conditioned space with the same indoor thermal conditions could have significantly different comfort experiences, because their metabolic rate and clothing insulation are different.

In Hawaii people using lighter clothing year round than nationwide. Therefore, occupants in Hawaii typically can endure higher indoor temperatures, while still experience satisfactory indoor comfort than people from other regions in our nation. This fact is supported by the so-called “Adaptive comfort” standard, which ASHRAE 55 recommends using to assess thermal comfort in a naturally ventilated space. The ASHRAE 55 explains that occupants will attune their expectations for indoor thermal environment in accordance with prevailing outdoor temperatures. In addition, as a general observation people in Hawaii have more tolerance to natural ventilation rather mandatory air conditioning. In fact, a frequently observed complaint is the statement that conditioned spaces are too “cold” to feel comfortable. These particulars will likely facilitate the design of innovative desiccant cooling systems suitable for Hawaii climatic conditions.

Comfort enhancement technologies: Two comfort enhancement measures presented hereafter have positively affect energy demands for space conditioning. These two measures are the use of ceiling fans and radiant cooling.

Ceiling fans have been used for a long time to provide cheap and simple cooling effects by increasing the air flow past the human body. The physical phenomenon of the cooling effect on the human body is the increase of the convective heat transfer and the increase of the evaporative heat loss due to an increase air movement past the outer skin. The degree of heat loss due to increased convection and evaporation is accentuated by lighter clothing or bare skin. It must be stressed that the cooling effect is not due to cold ambient air temperatures but due to the increased rate of heat transfer, or better heat loss, from the human body to the surrounding air. This means that the occupant is at the same heat balance for warmer but moving air than for somewhat colder and still air. The difference between the perceived and actual air temperatures, for the moving and still air, respectively, is dependent on the air speed.

ASHRAE provides a relationship between the air speed and the perceived lower air temperatures, which is illustrated in Figure 5.5.9. The approach presents a simplistic, yet very useful relationship between air speed over the human body and perceived cooling.

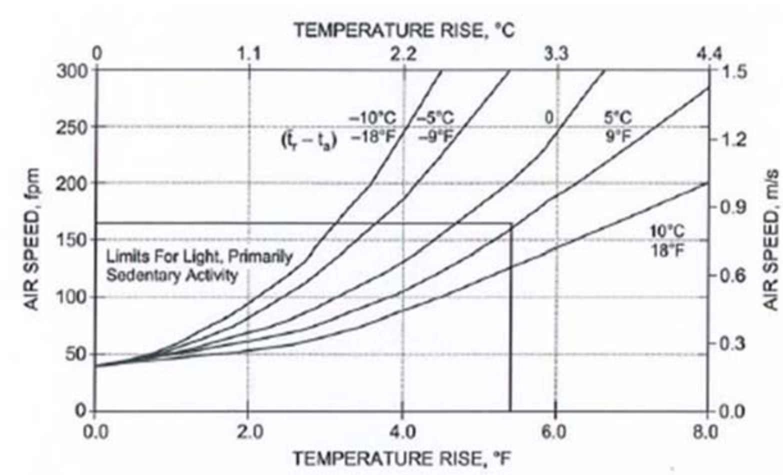


Figure 5.5.9: Air speed required to offset increased air and radiant temperature

(Source ASHRAE 55 -2010)

The "0" graph refers to identical air and mean radiant temperatures, this means absence of radiant temperature asymmetry

Radiant cooling: It was pointed out in Section 4.1 that radiant cooling can save energy by substituting the energy intensive transport of heat energy through chilled air by transporting heat through chilled water. Thus energy intensive fan power to move air is replaced by much more energy effective pumping of chilled water. Besides these advantages, radiant cooling also has the advantage of lowering the operative temperature through actively cooled radiant surfaces. The experience of thermal comfort is closely related to the operative temperature, which is defined as a weighted average of air and mean radiant temperatures. In the simplest, yet most common terms, the operative temperature is calculated without weighing the air or mean radiant temperatures, thus a direct average of air and mean radiant temperature.

With installing a radiant ceiling or better yet a radiant room envelope, the operative temperatures is always below the prevailing air temperature and thus the air temperature can be increased accordingly without a loss in thermal comfort sensation.

Using the phenomenon of perceived temperature reduction through ceiling fans: The fact that ceiling fans reduce the perceived operative temperature reduction is further discussed at this point. It is determined how cooling and dehumidification needs are affected by the presence of ceiling fan and their induced air movement. In this evaluation it is assumed that individually controlled ceiling fans are installed to provide increase air speed for occupants and thereby allow the air temperature in the space to be above the upper limit of the comfort zone, in accordance to the temperature rise due to air speed and using the curve “0” in Figure 5.5.9.

Using the relationship depicted in Figure 5.5.9, the range of 2 to 5 fpm for temperature rise in a conditioned space is used to quantify the level by which the actual air temperature can be higher than the perceived temperature. Therefore, the upper allowable temperature of the comfort range is raised from the upper limit of 81 F, based on the previously used comfort zone, by the temperature rise resulting from the particular air speed. The detailed results for air speeds 4 and 2 are depicted in Tables 5.5.3 and 5.5.4, as well as in Figures 5.5.10 through 5.5.13. The results with and without ceiling fan operation provide a comparison of the level of improvement. Figure 5.5.14 show the improvements of cooling as a function of the increased air speed from the ceiling fan for the data used in the analysis. The figure suggests that the highest gradient of improvement for the outdoor conditions considered happens for air speeds corresponding the allowable temperature rises of 3 and 4 F.

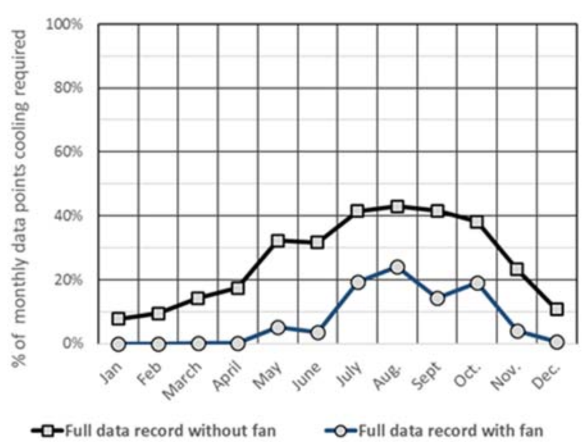


Figure 5.5.10: Cooling requirement with and without ceiling fan for Full data record, **for 4 F temp. rise**

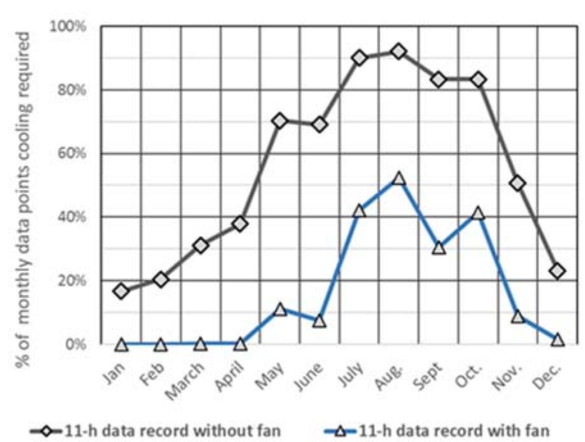


Figure 5.5.11: Cooling requirement with and without ceiling fan for 11-h data record, **for 4 F temp. rise**

	Temperature rise 4 F : air speed ~ 130 fpm or 0.7m/sec			
	Full record		11-h record	
	8,760 data points		4,015 data points	
	Cooling requirements		Cooling requirements	
	Coling when DBT > 81F	Coling when DBT > 85F	Coling when DBT > 81F	Coling when DBT > 85F
	Without ceiling fans	With ceiling fans	Without ceiling fans	With ceiling fans
	% of montly record hours	% of montly record hours	% of montly record hours	% of montly record hours
Jan	8%	0%	17%	0%
Feb	9%	0%	20%	0%
March	14%	0%	31%	0%
April	17%	0%	38%	0%
May	32%	5%	70%	11%
June	32%	3%	69%	8%
July	41%	19%	90%	42%
Aug.	43%	24%	92%	52%
Sept	42%	14%	83%	31%
Oct.	38%	19%	83%	41%
Nov.	23%	4%	51%	9%
Dec.	11%	1%	23%	1%
Average >>	26%	8%	56%	16%
	% of yearly data record hours		% of yearly data record hours	
	23%	7%	23%	8%

Table 5.5.3: Cooling requirement with and without ceiling fan

For temperature rise of 4F; air speed 130 fpm or 0.7 m/sec

130 fpm is a low but comfortable air speed for occupants

	Temperature rise 2 F : air speed ~ 70 fpm or 0.4 m/sec			
	Full record		11-h record	
	8,760 data points		4,015 data points	
	Cooling requirements		Cooling requirements	
	Coling when DBT > 81F	Coling when DBT > 83F	Coling when DBT > 81F	Coling when DBT > 83F
	Without ceiling fans	With ceiling fans	Without ceiling fans	With ceiling fans
	% of montly record hours	% of montly record hours	% of montly record hours	% of montly record hours
Jan	8%	1%	17%	3%
Feb	9%	0%	20%	1%
March	14%	2%	31%	5%
April	17%	3%	38%	5%
May	32%	15%	70%	32%
June	32%	16%	69%	34%
July	41%	30%	90%	65%
Aug.	43%	34%	92%	73%
Sept	42%	26%	83%	56%
Oct.	38%	28%	83%	62%
Nov.	23%	10%	51%	23%
Dec.	11%	3%	23%	7%
Average >>	26%	14%	56%	30%
	% of yearly data record hours		% of yearly data record hours	
	23%	14%	23%	14%

Table 5.5.4: Cooling requirement with and without ceiling fan

For temperature rise of 2F; air speed 70 fpm or 0.4m/sec

70 fpm is a low but comfortable air speed for occupants

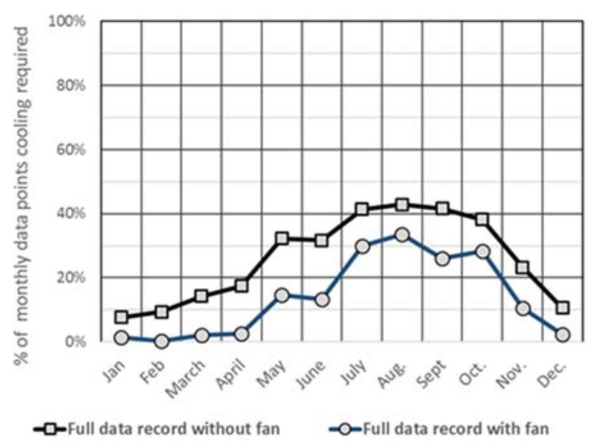


Figure 5.5.12: Cooling requirement with and without ceiling fan for Full data record, **for 2 F temp. rise**

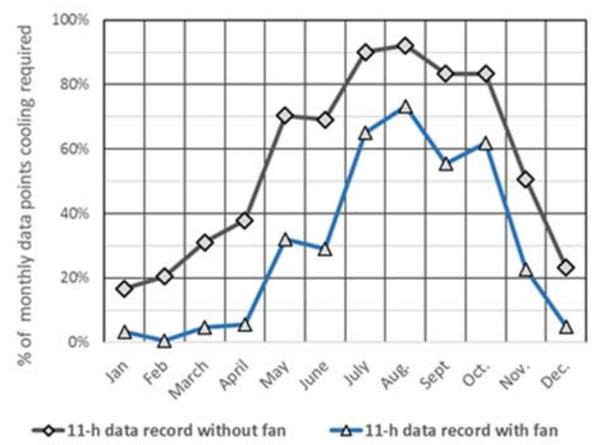


Figure 5.5.13: Cooling requirement with and without ceiling fan for 11-h data record, **for 2 F temp. rise**

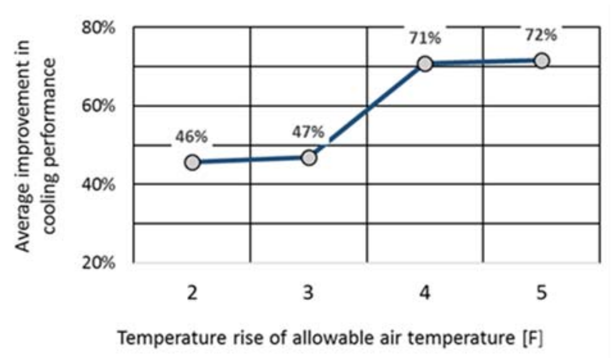


Figure 5.5.14: Averaged improvement of cooling performance due to ceiling fan operation

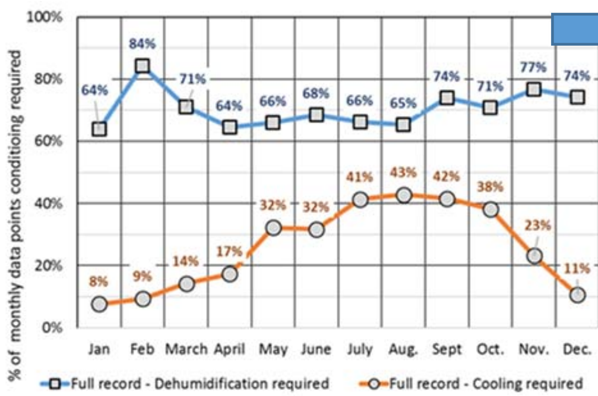
For the averaged improvement the results of the Full and11-h data records were averaged

When considering an effective air speed that corresponds to 4 F allowable temperature rise, the cooling needs are significantly reduced in order to stay within the comfort range. Figure 5.5.17 (a) and (b) shows the expected dehumidification and cooling needs when assuming an allowable air temperature rise of 4 F due to ceiling fan operation. Figure 5.5.15 shows the significant reduction in cooling needs if ceiling fans produce air movement over the body of building occupant. Obviously, the dehumidification needs are not affected by the ceiling fans and stay at the same level with or without air speeds.

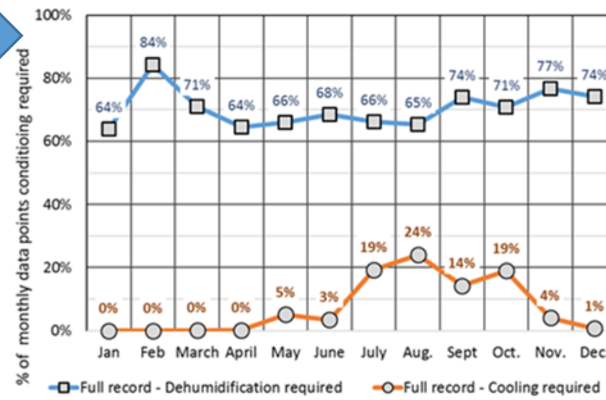
The results indicate that the use of ceiling fans can significantly reduce the demand on cooling of the supply air, by increasing the perceived temperature in the space due to the increased air movement over the human body, generated by ceiling fans. It must be noted that, for the sake of brevity, the discussion about cooling and dehumidification needs are considering ONLY OUTSIDE AIR conditions. Internal gain of sensible and humidity are dependent on the use pattern of the building and can vary significantly.

Without increased air speed

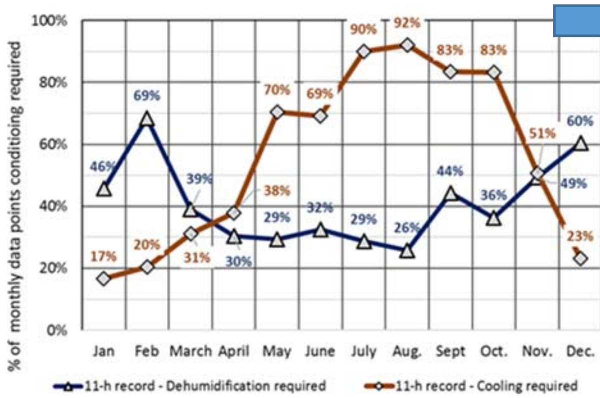
With Temperature rise 4 F: air speed ~ 130 fpm or 0.7m/sec



Dehumidification and cooling needs for **Full** data record; no air speed



Dehumidification and cooling needs for **Full** data record; 4 fpm air speed



Dehumidification and cooling needs for **11-h** data record; no air speed

Dehumidification and cooling needs for **11-h** data record; 4 fpm air speed

Figure 5.5.15: Dehumidification and cooling needs with and without increase air speed

SECTION 6 - CANDIDATE TECHNOLOGY AND SYSTEM INTEGRATIONS

This section presents and evaluates candidate desiccant cooling technology solutions which are assessed for their feasibility in a Hawaiian specific climatic and logistic framework.

6.1 Technologies and System Components Considered

Figure 6.1.1 illustrates the technologies involved and the main functions. The term "desiccant cooling" is used in this feasibility assessment, which follows the technical literature describing air-conditioning applications that are supported by desiccant dehumidification as "*desiccant cooling*".

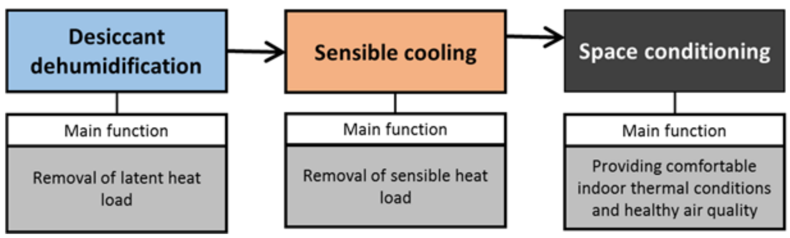


Figure 6.1.1: Categories of desiccant and cooling technology considered in feasibility assessment

Optimum performance of desiccant cooling systems is dependent on effective integration of the different technologies. Figure 6.1.2 presents available components and technologies for the integrated system. These system components and technologies are combined to establish integrated desiccant cooling candidate systems, whose different performance will be quantified in Sections 7 through 9.

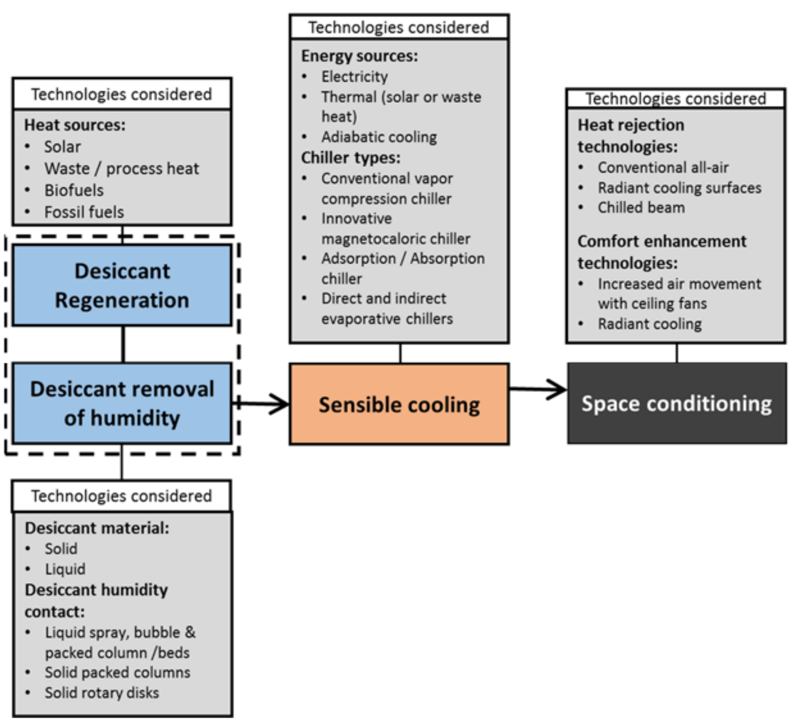


Figure 6.1.2: System components and technologies considered for integrated desiccant cooling candidate systems

6.2 Brief Review of Desiccant Technologies

General Notes:

- Solid desiccant humidity removal technologies have been more widely used in building conditioning than liquid desiccant technologies.
- Liquid desiccant systems, however, have attracted significant interest in the last couple of years, and there have been several very promising desiccant cooling technology applications which are based on liquid desiccant applications.
- One significant advantage of liquid desiccant systems is their better regeneration performance using solar or low temperature heat sources.
- Regeneration (desorption) processes use the same mass transfer mechanisms as the humidity removal (sorption) processes;

The following desiccant technologies are considered for this feasibility review.

Liquid desiccant packed column:

Working principle: The typical system configuration has two process vessels, a larger dehumidification (sorption) process vessel and a smaller regeneration (desorption) vessel. In both vessels, liquid desiccant solution is either sprayed or distributed on the packing inside the vessel (e.g. packing of Rashing rings and other geometries). The liquid desiccant solution runs down on the packing material and shear forces stimulate surface renewal. Moist air is flowing upwards and is brought into contact with the liquid desiccant solution thereby causing gas transfer. The desiccant solution drips down into a pan and from where a portion is recirculated and the remaining portion is pumped to regeneration vessel where the liquid is regenerated by adding heat.

Typical applications: Typically, systems are employed in large, central systems because of the complexity of the liquid spray dehumidifier in comparison to solid desiccant units. Some large systems connect several dehumidification process vessels to a single regenerator reactor.

Advantages: The packed column process has good control of heating and cooling. As an added benefit, independent of the thermal treatment, the liquid desiccant can eliminate particles from the supply air flow.

Disadvantages: Disadvantages of the system include process inertia, which makes adaptation of rapidly changing humidity conditions difficult.

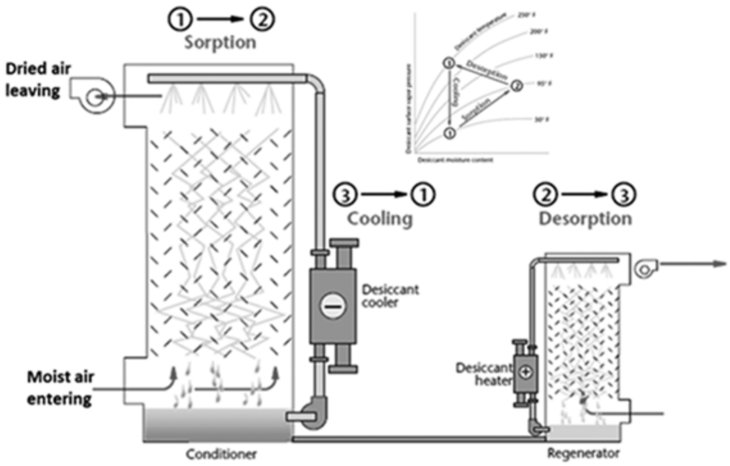


Figure 6.2.1: Liquid desiccant packed column – process vessel
(Literature review, 2016)

Solid desiccant packed tower:

Working principle: The packed tower dehumidifier system consists of pairs of process vessels which are filled with solid granular desiccant material, such as silica gel or molecular sieve. The pairs of vessels operate batch-wise, or in a process swing mode. In this process one vessel operates in dehumidification mode while the other regenerates the desiccant material after it becomes saturated with moisture

Typical applications: Since drying and reactivation take place in separate, sealed compartments, the packed tower dehumidifier is frequently used to dry pressurized process gases.

Advantages: The process can achieve very low dew points, as low as below -40°F.

Disadvantages: Since the system operates batch wise and not continuous, properties of the processed air change during one cycle of the swing operation. At the start of dehumidification, dry and cool desiccant is exposed to the process airstream. Since the vapor pressure is at a maximum, the process air can be deeply dried. As its moisture capacity fills up, the air is not dried as efficiently.

Rotating horizontal bed

Working principle: A series of shallow, perforated trays hold granular desiccant material. The assembly rotates continuously between sections where process and reactivation airstreams flow through the desiccant. During dehumidification trays rotate through a section where moist process air passes vertically through the trays and the desiccant material adsorbs moisture. In regeneration mode trays rotate to a section where reactivation airstream heats the desiccant, raising its vapor pressure and therefore moisture is released into the reactivation air. The hot desiccant material is cooled by cold air flowing through the trays before the tray reaches the section where the desiccant is again exposed to the moist process air.

Typical applications: The technology is used in building applications where it can dry larger quantities of moist to dew points used in space conditioning.

Advantages: The systems have low first costs. The design and operation is simple, expandable and easy to manufacture.

Disadvantages: Operating costs can be higher than with other designs, especially when energy costs are high. Uneven settlement and resulting distribution of the granulated desiccant material can increase the possibility of uneven flow and leakage of air flow through the beds. With regard to energy efficiency, a parallel flow arrangement of process and reactivation air does not perform as well as a counter flow arrangement.

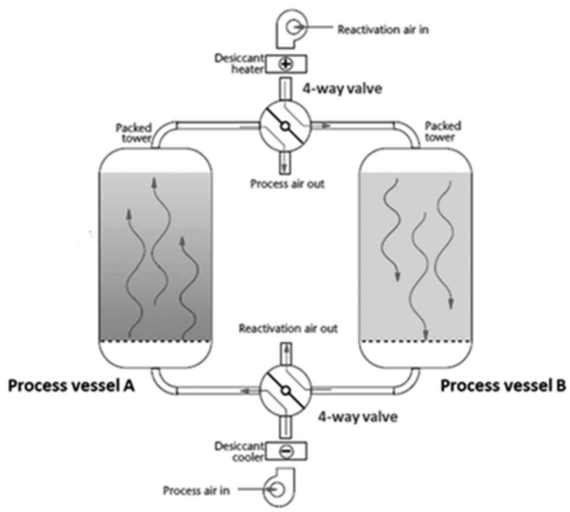


Figure 6.2.2: Solid packed tower dehumidifier
(Literature review, 2016)

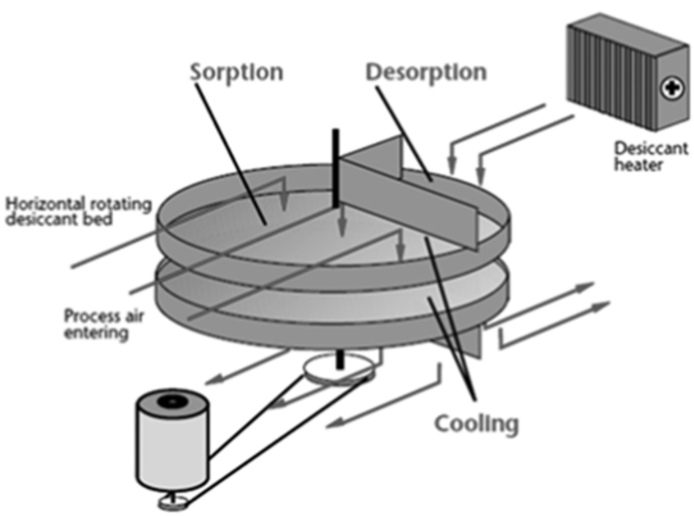


Figure 6.2.3: Rotating horizontal bed dehumidifier
(Literature review, 2016)

Multiple vertical bed:

Working principle: Solid desiccant matrix is installed in a vertical bed configuration on a circular carousel with eight or more towers. This vertical bed configuration combines feature of a packed tower and a rotating horizontal bed design in an arrangement that is well-suited to atmospheric pressure dehumidification applications. Process or activation air streams pass through the relatively high vertical cells. These towers are exposed to alternative process and reactivation air streams.

Typical applications: Systems have been installed and used in building operations.

Advantages: The rotating multiple vertical bed assembly operates more or less continuously and can provide a constant outlet moisture condition on the process air. The systems can achieve low dew points since leakage between process and reactivation air circuits is minimized.

Disadvantages: The multiple vertical bed is mechanically more complex, has a higher first cost and requires more complex operation than the horizontal bed dehumidifier.

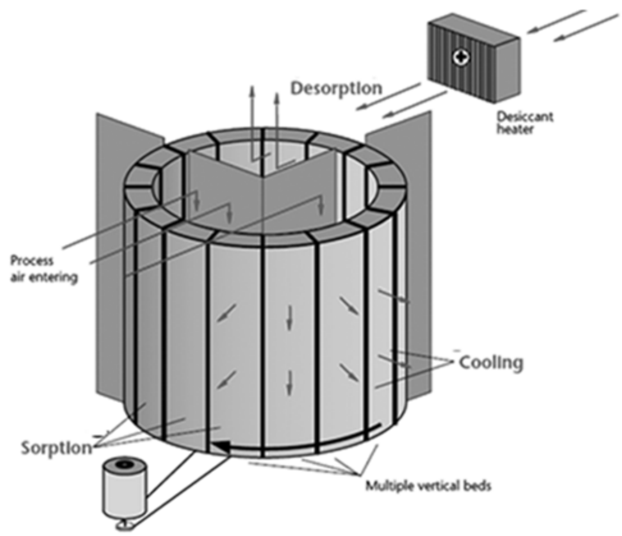


Figure 6.2.4: Multiple vertical bed dehumidifier
(Literature review, 2016)

Rotating Honeycomb ™ or Solid Desiccant Dehumidification Wheel:

Working principle: A rotating wheel assembly with a horizontal axis has a light weight structure with corrugated and desiccant impregnated material. The process air passes through the desiccant which adsorbs moisture until it becomes saturated. As the wheel turns to the reactivation section, the heated reactivation air stream dries out the desiccant material in a counter flow. After rotating through the reactivation section, the desiccants in the corrugated wheel structure are cooled by small portion of the process air, which lowers the surface vapor pressure of the desiccant material.

Typical applications: Systems have been installed and used in many buildings.

Advantages: Air streams passes of the through the slowly rotating wheel, resulting in low pressure losses. The structure is lightweight and offers an effective moisture removal capacity to overall weight ratio, which lowers mechanical power to drive the system. The system has good energy performance and relatively low maintenance efforts and costs.

Disadvantages: The desiccant wheel has fewer disadvantages compared to other solid desiccant systems.

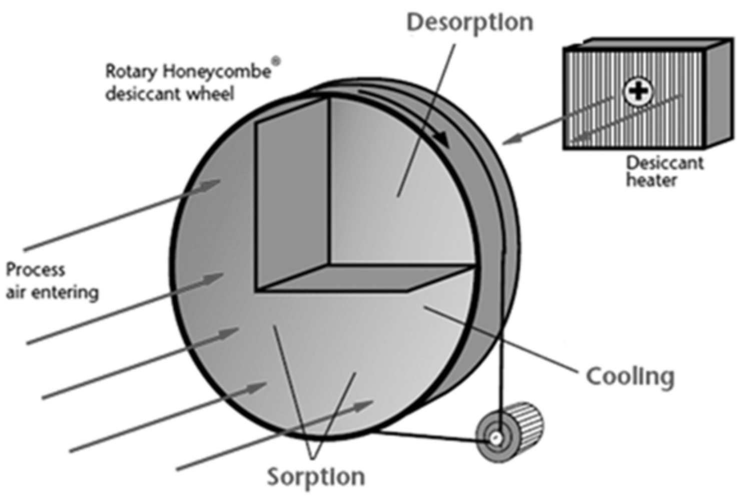


Figure 6.2.4: Rotating Honeycombe ™ or Solid Desiccant Dehumidification Wheel
(Literature review, 2016)

The selection of the desiccant material, both solid and liquid, is very much dependent on the manufacturer of the desiccant systems. Desiccants material is far from standardized and many desiccants are fabricated by vendors to accomplish their specific performance envelope.

6.2 Brief Review of Desiccant Regeneration Technologies

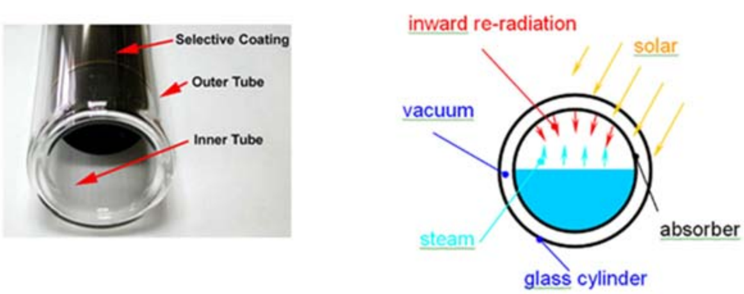
The regeneration process increases the vapor pressure of the desiccant material, thereby leading to the loss of humidity. This process is referred to as desorption and the driving force to achieve desorption is heat or pressure, with heat representing most of the air drying applications and all of the applications in building conditioning.

The heat can be provided most effectively by low cost sources, such as solar heat or waste heat from combined heat and power facilities (CHP).

As was presented in Section 5, Hawaii offers favorable conditions to use solar thermal energy conversion to be used in desiccant regeneration. Either focusing or non-focusing solar thermal collectors can be used. Non-focusing, or non-tracking collectors are of simpler design and operation and the first costs are significantly lower. Non-focusing solar panels, however, produce heated water that is not as hot as from focusing panels. The design and selection of the desiccant solid material or liquid solution determines the required solar heat rate and the temperature.

In private communication with Dr. Lowenstein of AIL Research a suitable solar thermal collector was discussed that has favorable performance for Hawaii. The solar collector has been used in desiccant regeneration systems by AIL Research. Figure 6.2.1 shows the basic design of the working scheme of the Steam-Generating (SG) Solar Collector and images of a completed installation.

Horizontal and angled installation of Steam-Generating Solar Collector



Steam-Generating Solar Collector:

- Lower manufacturing cost for high performance, non-concentrating solar thermal collectors
- Simplify installation for large arrays of solar thermal collectors
- Reduce parasitic power requirements
- Provide passive protection against stagnation

Information by AIL Research, Inc.

Figure 6.2.1: Steam-Generating Solar Collector to provide suitable heat for desiccant regeneration

Waste heat from CHP plants is another cost effective source of low temperature heat for regeneration. The temperature and heat flow rate is dependent on the CHP engine, but typically does not create design barriers and desiccant applications are quite straightforward.

6.3 Sensible Cooling Technology Considered for Candidate Systems

Sensible cooling is a central part of the desiccant cooling system. The desiccant sorption process only removes humidity and no sensible heat; in fact the heat of sorption creates heat that is added to the supply air. The sensible heat removal removes the heat of sorption and also lowers the supply air temperature to the level that is desired. There are several ways how the processes of dehumidification and sensible cooling can be integrated into the overall desiccant cooling system.

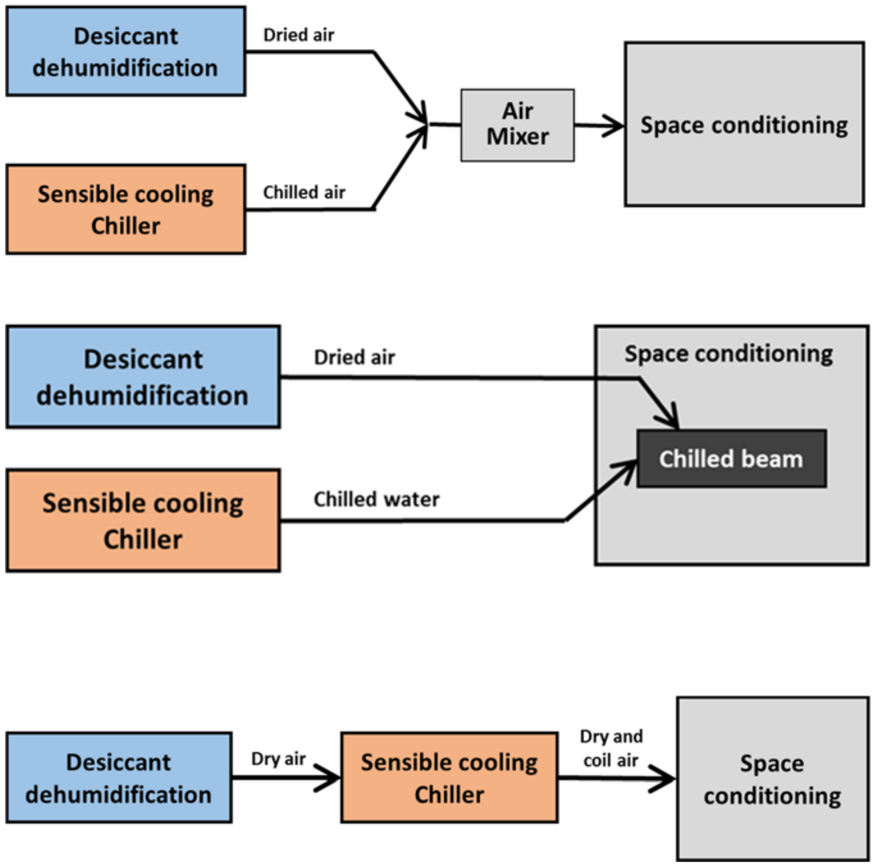
Three of system integration are presented in Figure 6.3.1.

(a) Parallel "all-air" integration of dehumidification and sensible cooling; The air streams from the chiller and the desiccant system are mixed and provide cool and dry air to the conditioned space. The chiller for the sensible cooling is indirectly removing heat from the supply air stream, which means from an air-working fluid heat exchanger. The type of chiller can be driven by electricity or solar / waste heat.

(b) Parallel "air and hydronic" integration of dehumidification and sensible cooling; This proposed integration of desiccant dehumidification and chiller is a special application that works with a chilled beam inside a conditioned space. Here, the desiccant system provides dried air which is conveyed to the chilled beam air intake. A chiller generates chilled water that is piped to the chilled beam and provides sensible cooling inside the conditioned space. The air outlet of the chilled beam provides cool and dry air to the conditioned space

(c) Sequential "all-air" integration of dehumidification and sensible cooling; The dried air stream from the desiccant system provides the chiller with dry air that is cooled further before it enters the conditioned space. This sequential working process is the most frequently presented in the literature since if represents simple construction and controls. While for chillers that indirectly cool the supply air the sequential "all-air" integration is an option, the sequential process is mandatory for conventional evaporative cooling. The Hawaii climate has a high humidity and therefore conventional evaporative would not be efficient, unless the air supply to the evaporative chiller is first dried to a certain extent.

It should be noted that the basic flow diagrams presented in Figure 6.3.1 omit the return air from the conditioned space. The more detailed process diagram will use the return air from the conditioned space for energy recovery and/or as a sink for humidity to precondition the supply air. But the inclusion of the supply air is only relevant to design detail and not relevant for the assessment of feasibility of desiccant cooling systems.



<u>(a) Parallel "all-air" integration of dehumidification and sensible cooling;</u> The air streams from the chiller and the desiccant system are mixed and provide cool and dry air to the conditioned space

<u>(b) Parallel "air and hydronic" integration of dehumidification and sensible cooling;</u> The air stream from the desiccant system is added to air intake of the chilled beam. The chilled beam provides cool and dry air to the conditioned space

<u>(c) Sequential "all-air" integration of dehumidification and sensible cooling;</u> The dried air stream from the desiccant system provided to chiller, which removes sensible heat, this sequential integration is required **for evaporative cooling in the Hawaii climate**.

Figure 6.3.1: Candidate system integration of dehumidification and cooling in the desiccant cooling system

Table 6.3.1 gives an overview of the types of chillers or coolers which are considered for eight candidate desiccant cooling system integrations. These candidate system integrations are described in more detail in Section 6.4 through 6.13.

Table 6.3.1: Eight sensible cooling technologies considered for the candidate desiccant cooling system integrations.

Cooling technology ID.	Type of Chiller/ Cooling	Primary energy source for sensible heat rejection	**Brief Description of cooling technology**
1	Evaporative chiller; direct or indirect; conventional processes	Adiabatic cooling of the supply air (direct) or secondary (indirect)	The evaporative chiller is either of a direct or indirect type. The sensitive cooling is based on an adiabatic process that removes heat from the supply air by evaporation from exposed water surface. • For <u>direct evaporative chillers</u> the water and supply air are brought into direct contact, which increases the humidity of the supply air. • For <u>indirect evaporative chillers</u> the water comes into contact with a secondary air flow, thereby cooling the secondary air flow. An air-air heat exchanger transfers the heat from the supply air to the secondary air flow. This separation of supply and secondary air flows and cooling by means of a heat exchanger does not increase the humidity the supply air.
2	Indirect evaporative chiller; <u>enhanced process</u>	Adiabatic cooling of the secondary air stream	The indirect evaporative chiller is driven by an advanced evaporative process, referred to as dew point reduction process, such as the M-Cycle. While the conventional evaporative process is limited by the wet bulb temperature, the advanced evaporative cooling process can deliver cooling that approaches the dew point in lieu of the web bub temperature.
3	Adsorption chiller	Thermal energy input, either solar heat of waste heat from CHP	The adsorption chiller works in a close cycle process where a solid desiccant induces evaporation of water contained in a low pressure process vessel. The heat of evaporation cools chilled water.
4	Absorption chiller	Thermal energy input, either solar heat of waste heat from CHP	The absorption chiller works in a close cycle process where a liquid desiccant solution induces evaporation of water contained in a low pressure

Table 6.3.1: Eight sensible cooling technologies considered for the candidate desiccant cooling system integrations.

Cooling technology ID.	Type of Chiller/ Cooling	Primary energy source for sensible heat rejection	**Brief Description of cooling technology**
			process vessel. The heat of evaporation cools chilled water.
5	Magnetic chiller	Electricity	The magnetic chiller works on the basis of the magneto caloric effect. Instead of liquid working fluid the magneto caloric effect derives a heat pump effect through the movement a magnetic field relative to solid “working fluid”, which is a special alloy with high magneto caloric effects.
6	Membrane based evaporative chiller	Adiabatic cooling of the supply air stream	This chiller combines the dehumidification process with the sensible cooling. The system presented is referred to as a DEVAP systems. Liquid desiccant is contained within a membrane and exchange from humidity for dehumidification and evaporative cooling is transferred through the membrane. The containment in membrane avoid the possible carry over of desiccant solution into the air stream.
7	Liquid-Desiccant Direct-Expansion air conditioner (LDDX)	Electricity for chiller; solar or waste heat for the desiccant regeneration	The (LDDX) system works with a conventional DX-air chiller (vapor compression) and the so-called Wicking-Fin Technology. This technology embeds a tubular heat exchanger into a bed of porous contact media. The process provides simultaneous sensible cooling and desiccant dehumidification.
8	Conventional vapor compression chillers	Electricity	The chiller is a conventional chiller that is either a DX-chiller or a chilled water generator.

6.4 Candidate Desiccant System Integrations

Eight high level system integrations of desiccant and sensible cooling technologies have been selected for the assessment of the feasibility of desiccant cooling in Hawaii's climate. The <u>main differentiators of the eight candidate systems is how desiccant dehumidification is interfaced with sensible cooling</u>.

Table 6.4.1 summarizes the high level system configurations of candidate system integration for the desiccant cooling. Besides the high level descriptions, a range of second level variations are considered which are discussed in Section 6.5.

Table 6.4.1: Summary of eight high-level system configurations of candidate system integrations for the desiccant cooling; high level system integrations A to H

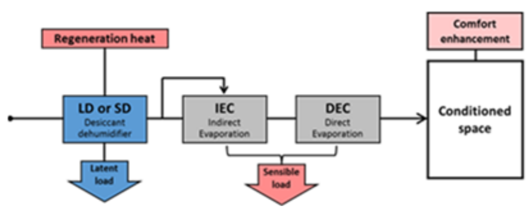
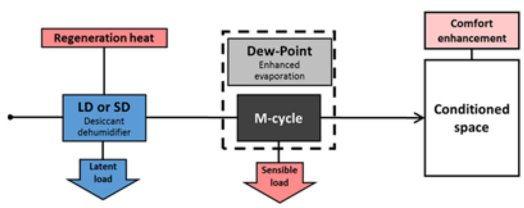
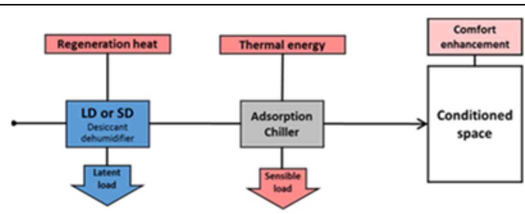
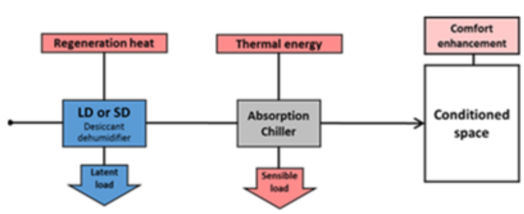
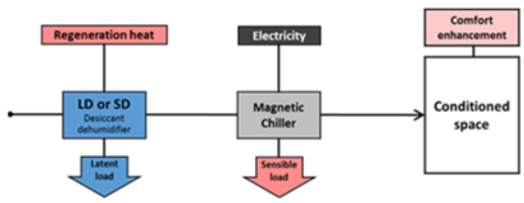
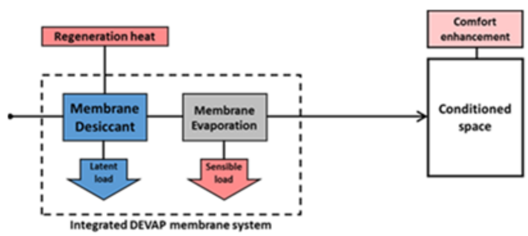
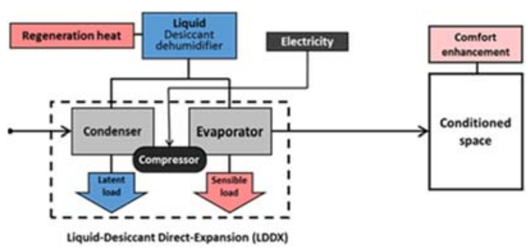
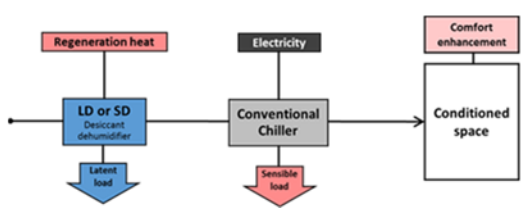
No. of system integration	Image (thumbnail)	Sensible cooling technology for desiccant cooling systems
A		Desiccant and Direct / Indirect Evaporative Cooling downstream of desiccant dehumidification
B		Enhanced evaporative cooling - M-Cycle downstream of desiccant dehumidification
C		Adsorption chiller downstream of desiccant dehumidification
D		Absorption chiller downstream of desiccant dehumidification

Table 6.4.1: Summary of eight high-level system configurations of candidate system integrations for the desiccant cooling; high level system integrations A to H

No. of system integration	Image (thumbnail)	Sensible cooling technology for desiccant cooling systems
E		Magnetic (magneto caloric effect) chiller downstream of desiccant dehumidification
F		Integrated DEVAP membrane system,
G		Liquid-Desiccant Direct-Expansion (LDDX) air-conditioner, integrated dehumidification and sensible cooling
H		Conventional chiller, downstream of desiccant dehumidification

6.5 Variations of High-level System Configurations

The eight high level systems can have variations in terms of desiccant material, source of regeneration heat, and comfort enhancement technologies. These variations are described in Tale 6.5.1.

Table 6.5.1: Variations for high level system integrations of desiccant and sensible cooling technologies

No.	Main component	Category / Variation	Description / comments
1	**Desiccant dehumidifier**		
1.1		Liquid desiccant	A liquid desiccant system accomplishes the dehumidification / sorption
1.2		Solid desiccant	A solid desiccant system accomplishes the dehumidification/ sorption
2	**Heat source for desiccant regeneration**		
2.1		Solar	Solar energy can be used to regenerate the desiccant material
2.2		Waste heat / CHP	Heat energy that is generated, discarded and typically rejected to the environment can be used to regenerate the desiccant material
2.3		Biofuel	Biofuel can be used as the heat source to regenerate the desiccant material
2.4		Fossil fuel	The conventional fuel used for desiccant regeneration is natural gas; In Hawaii no natural gas, but synthetic natural gas from refining crude oil, would be available
3	**Thermal energy for the Absorption / Adsorption chillers**		
3.1		Solar	Solar energy can be used to drive a thermal closed adsorption/ absorption chiller cycle.
3.2		Waste heat / CHP	Heat energy that is generated, discarded and typically rejected to the environment can be used to drive a thermal closed adsorption/ absorption chiller cycle.
4	**Comfort enhancement**		
4.1		Ceiling fans	Ceiling fans create a cooling effect based on increase air movement over the body the creates cooling without lowering the room temperature.
4.2		Radiant cooling	Radiant cooling devices enhance the radiant heat transfer rate and therefore lower the operational temperature without the need to lower the air temperature. The thermal comfort experience of occupant is based on the operative temperature

Table 6.5.1: Variations for high level system integrations of desiccant and sensible cooling technologies

No.	Main component	Category / Variation	Description / comments
			and not on the air temperature or the radiant temperature alone.

6.6 Candidate System Integration–A: Desiccant and Direct / Indirect Evaporative Cooling

The candidate system configuration A is depicted in Figure 6.6.1. The system components and performance characteristics of candidate system configuration A is described in the following.

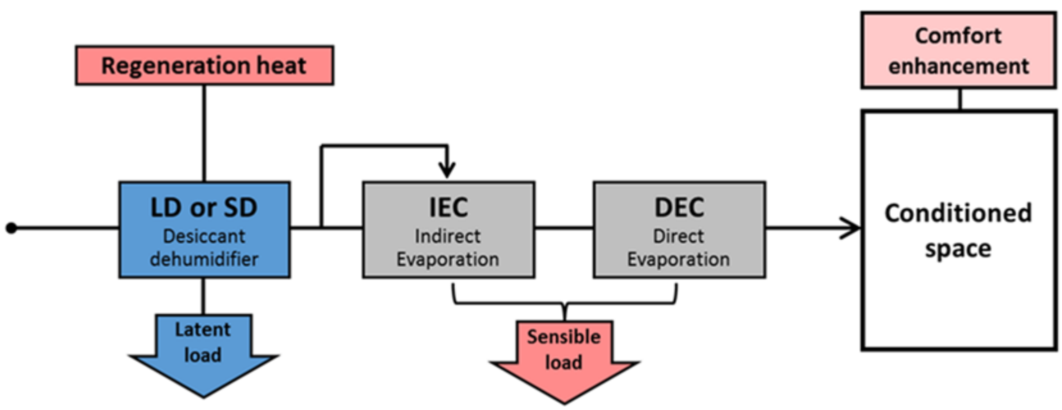


Figure 6.4.1: System integration A with Desiccant and Direct / Indirect Evaporative Cooling

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the desiccant dehumidification unit which removes a certain humidity of contained in the outdoor air supply. The downstream sensible heat removal is accomplished by means of conventional evaporative cooling.
Main objective	Demonstrate that conventional evaporative cooling is effective in the humid and hot climate of Hawaii if the process air is partly dried by an upstream desiccant system. As a result, an efficient desiccant cooling system can be formed.
Desiccant material used	Solid or liquid desiccant material

Aspect of system	Description
Sensible cooling technology	Conventional evaporative cooling: Indirect evaporative cooling in the first stage and direct evaporative cooling in the second stage (the basics of the evaporative cooling process is described at the end of this section)
Working fluid (other than water)	**NONE**
Energy savings projected	The energy savings are projected as **HIGH** (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**
Complexity of sensible cooling system	**MEDIUM**
Integration of desiccant and sensible cooling systems	The integration of desiccant and sensible cooling processes hinges on the supply of partly dehumidified air to the evaporative cooling stage. The humid climate in Hawaii requires that the wet bulb temperature is reduced to make conventional evaporative feasible.
Integration with space conditioning options	Only all-air space conditioning is possible with the direct evaporative cooling process
Fuel used for processes	Desiccant system: Variable heat sources can be used for desiccant regeneration Cooling system: No external fuel source is used for the evaporative cooling process
Ease of using low temperature heat sources	**HIGH**; in effect the competitiveness of the entire desiccant cooling system is based on the use renewable or low temperature heat sources
Maturity technical / market	Technical and market maturity of the entire system is HIGH; there are many conventional evaporative cooling systems in operation
Outlook for use of the system integration	The outlook of the desiccant system is positive

Description of the sensible heat load rejection: **Evaporative Chiller / Coolers:**

For many decades, various evaporative cooling designs have provided energy-efficient space cooling. The conventional evaporative cooling process requires consistently low wet-bulb temperatures to operate, thus limiting the capacity and geographic reach of the system to locations with hot but dry climate. Recent developments in evaporative cooling systems have improved the energy efficiency, lowered the water consumption, and expanded the applicability of evaporative cooling beyond the limits set forth by wet bulb temperature.

The basic process that provides the cooling effect is to lower the temperature and increase the humidity of air by using latent heat of evaporation, changing liquid water to water vapor. In this process, the energy (enthalpy) in the air does not change. The conversion of sensible heat to latent heat causes a decrease in the ambient temperature as water is evaporated providing useful cooling. The temperature of dry air can be dropped significantly through the phase transition of liquid water to water vapor (evaporation), which can cool air using much less energy than refrigeration. In dry climates, evaporative cooling of air has the added benefit of conditioning the air with more moisture for the comfort of building occupants. In moist climates, such as Hawaii, the driving force for evaporation is decreased by high wet bulb and dew point temperatures.

<u>Direct evaporative cooling systems:</u> Direct evaporative coolers pass supply air through a wetted surface. The water on this surface evaporates into the air, thus simultaneously providing cooling and increasing the air's humidity. The heat of the supply air is used to evaporate water.

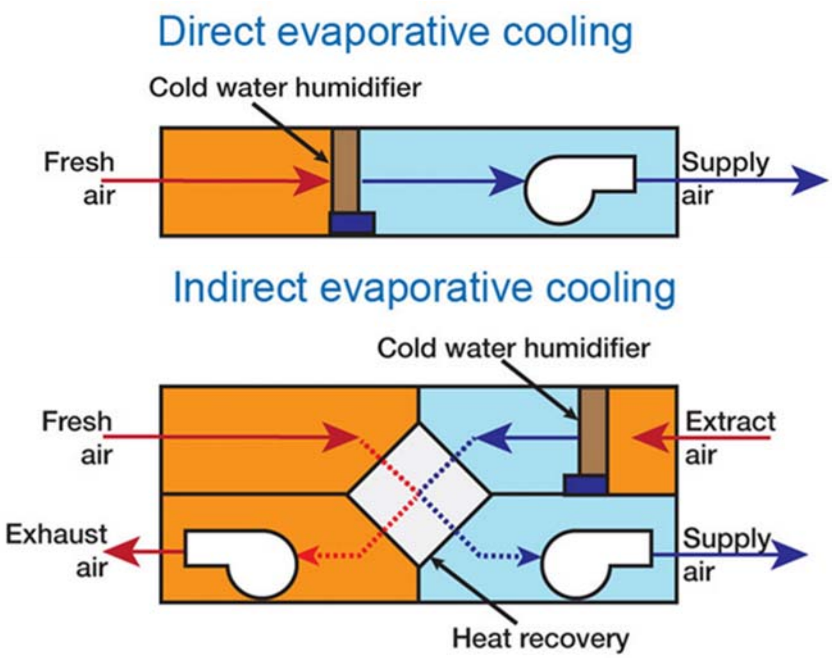
A mechanical direct evaporative cooler unit uses a fan to draw air through a wetted membrane, or pad, which provides a large surface area for the evaporation of water into the air. Water is sprayed at the top of the pad so it can drip down into the membrane and continually keep the membrane saturated. Any excess water that drips out from the bottom of the membrane is collected in a pan and recirculated to the top. Single stage direct evaporative coolers are typically small in size as it only consists of the membrane, water pump, and centrifugal fan.

Passive direct evaporative cooling can occur anywhere that the evaporative cooled water can cool a space without the assist of a fan. The passive direct evaporative cooling process is of no interest to this feasibility study.

<u>Indirect evaporative cooling systems</u>: Unlike direct systems, indirect evaporative coolers pass supply air through a heat exchanger channel that is evaporatively cooled by a secondary airstream. Because the supply air never comes in contact with the water, it does not become more humid, leading to more comfortable space cooling and improved indoor air quality. Due to the added heat transfer step of a heat exchanger the thermal efficiency is lower than for the direct evaporative cooler.

<u>Two-stage evaporative cooling, or indirect-direct</u>: In the first stage of a two-stage cooler, warm air is pre-cooled indirectly without adding humidity (by passing inside a heat exchanger that is cooled by evaporation on the outside). In the direct stage, the pre-cooled air passes through a water-soaked pad and picks up humidity as it cools. Since the air supply is pre-cooled in the first stage, less humidity is transferred in the direct stage, to reach the desired cooling temperatures. The result is cooler supply air with a RH between 50-70%, depending on the climate, compared to a traditional direct evaporative system that produces about 70–80% relative humidity in the conditioned air.

Figure 6.3.2 illustrates the working principle of a direct and indirect cooler, respectively.



Direct evaporative cooling brings evaporative water in contact with the supply air, thereby increasing the humidity of the supply air.

Indirect evaporative cooling transfers the loss of heat from the evaporative cooling of a secondary air flow to the supply air via a heat exchanger. Since direct contact between the supply air and the evaporating water is avoided the humidity level in the supply air does not change.

Figure 6.6.1: Basic process diagrams of direct and indirect evaporative cooling applications (source: http://www.condair.co.uk)

6.7 Candidate System Integration–B: Desiccant and Enhanced Evaporative Cooling - M-Cycle

The candidate system configuration B is depicted in Figure 6.7.1. The system components and performance characteristics of candidate system configuration B is described in the following.

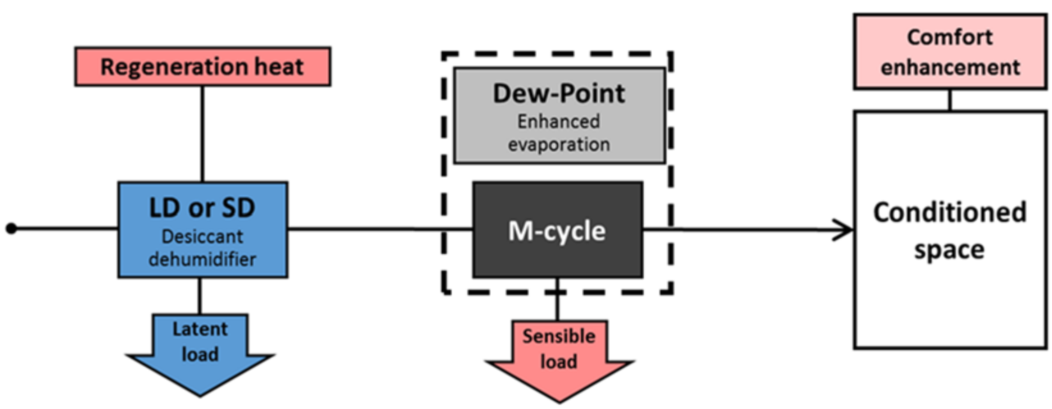


Figure 6.7.1: System integration B - with Desiccant and enhanced evaporative cooling - M-Cycle

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the desiccant dehumidification unit which removes a certain humidity of contained in the outdoor air supply. The downstream sensible heat removal is accomplished by means of an innovative indirect evaporative cooling process. The dew point evaporative process used for this discussion is the so-called M-(Maisotsenko) Cycle. In this process the evaporative cooling is not limited by the wet bulb temperature but by the dew point temperature, which is always lower than the WBT.
Main objective	Demonstrate that innovative “dew point evaporative cooling” works in conjunction with desiccant dehumidification to form an efficient desiccant cooling system.
Desiccant material used	Solid or liquid desiccant material
Sensible cooling technology	Innovate dew point evaporative cooling is used to indirectly cool the supply air (the basics of the dew-point evaporative cooling process is described at the end of this section)
Working fluid (other than water)	**NONE**
Energy savings projected	The energy savings are projected as **HIGH** (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**
Complexity of sensible cooling system	**MEDIUM**
Integration of desiccant and sensible cooling systems	The integration of desiccant and sensible cooling processes is straight forward and includes the cooling of the supply air. Differently from conventional evaporative cooling (see candidate system A in Section 6.6) the supply air to the evaporative cooling process does not have to be dried.
Integration with space conditioning options	Both all-air of hydronic heat rejection for space conditioning are possible.
Fuel used for processes	Variable heat sources can be used for desiccant regeneration No external fuel source is used for the dew-point evaporative cooling process
Ease of using low temperature heat sources	Desiccant system; **HIGH**; in effect the competitiveness of the entire desiccant cooling system is based on the use renewable or low temperature heat sources Cooling system: Is usually not needed, but system can have small performance increase when external heat is applied.

Aspect of system	Description
Maturity technical / market	Technical and market maturity of the entire system is **LOW**; the technology has only a few pilot installations
Outlook for use of the system integration	The outlook of the cooling desiccant system using dew-point evaporative cooling is promising, according to NREL.

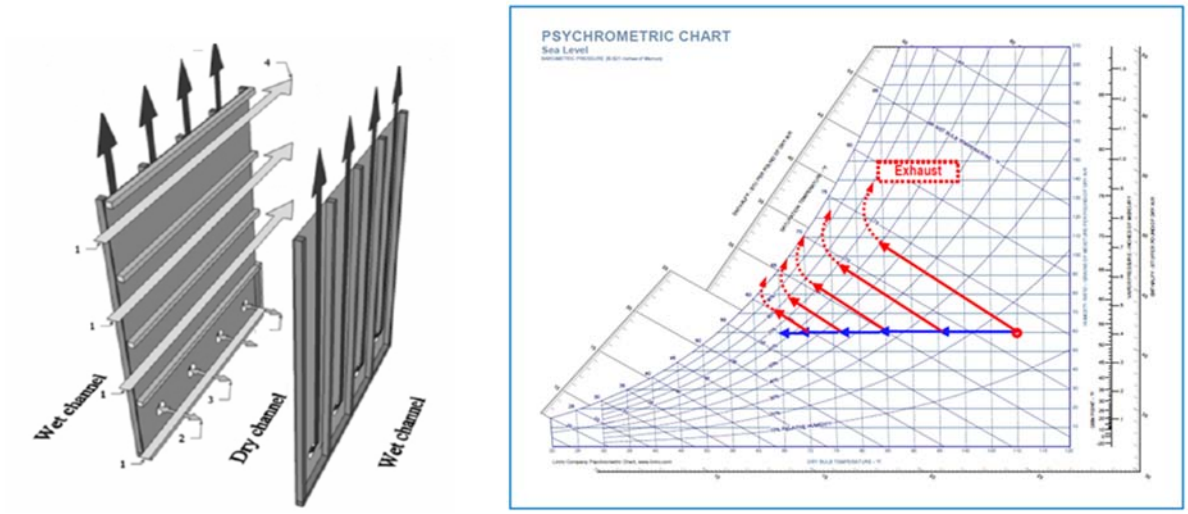
Description of the sensible heat load rejection: **Dew point evaporative cooling (M-cycle):**

Achievable low temperatures of dew-point evaporative processes are not limited by the wet bulb temperature but by the dew-point temperatures. The dew-point evaporative process is demonstrated by the example of the M-cycle, which stands for the Maisotsenko-Cycle named by its developer. These properties make the M - Cycle ideal for many evaporative cooling applications, especially in hot and humid climates, such as Hawaii.

Dew point evaporative cooling processes take advantage of the fact that when a parcel of air is sensibly cooled, the saturated water vapor pressure decreases, reducing its wet bulb temperature, thus increasing its evaporative cooling potential. Consequently, as the working fluid is humidified, the temperature of the evaporative cooling liquid that it is in contact with is also cooled to theoretically as low as the incoming air dew point temperature. This is accomplished through the sensible pre - cooling of the incoming air, lowering its wet bulb temperature. This wet bulb depression is limited to the incoming air dew point temperature.

The operating efficiency of the dew-point evaporative process can be enhanced by reducing the dew point temperature (thus the wet bulb temperature) of the incoming process air through dehumidification with solid and/or liquid desiccants. This is especially true in high absolute humidity climates where the dew point temperature of ambient air is higher than about 60 ° F to 70 ° F (GTI, 2010). Above these dew point temperatures, the effect of changing absolute humidity, with changing temperature, will become even larger.

The M-cycle uses a wet channel and a dry channel to achieve a very effective evaporative cooling effect. Water is sent through the wet channel where it gets evaporated. Figure 6.7.2 (a) and (b) illustrate the separate wet and dry channel of the M-cycle process and the psychrometric performance characteristics. Warm dry air is changed to cool moist air. Heat in the air is used to evaporate water; no heat is added or removed making it an adiabatic process. After getting processed in wet channel, thermodynamically indirect evaporative cooled air passes over the dry side of a plate. The wet side absorbs heat from the dry side by evaporating water and thereby cooling the dry side with the latent heat of vaporizing water into the air. Figure 6.7.2 (b) shows that an amount of cooled air is being discarded, which has an effect on the overall performance efficiency of the system.



(a) Wet and Dry Channels of M-Cycle Cross Flow Heat Exchanger (Wani, 2012)

(b) Psychrometric performance of M-Cycle, illustrating successive exhaust of warmer process air (Dean, 2012)

Figure 6.8.2: The M-cycle typical cross section and psychrometric characteristics

6.8 High Level System Integration – C: Desiccant and Adsorption chiller

The candidate system configuration C is depicted in Figure 6.8.1. The system components and performance characteristics of candidate system configuration C is described in the following.

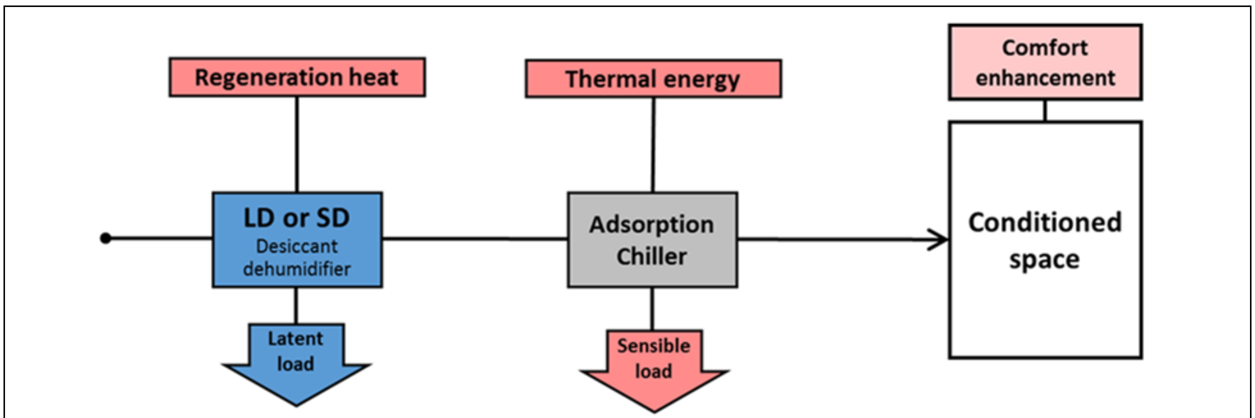


Figure 6.8.1: System integration C with Desiccant and Adsorption chiller

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the desiccant dehumidification unit which removes a certain humidity of contained in the outdoor air supply. The downstream sensible heat removal is accomplished by means of an Adsorption chiller. The chiller generates chilled water which cools the supply air. Chilled water can also be pumped to a radiant cooling ceiling or to a chilled beam.
Main objective	Demonstrate that Adsorption chillers work in conjunction with desiccant dehumidification to form an efficient desiccant cooling system.
Desiccant material used	Solid or liquid desiccant material
Sensible cooling technology	An adsorption used to provide cooling to the supply air through heat exchanger or to supply chilled water to radiant ceiling and chilled beam.
Working fluid (other than water)	Solid desiccant material (such as activated carbon, zeolite, silica gel) is used in the closed cycle (encapsulated) adsorption chiller process vessel
Energy savings projected	The energy savings are projected as **HIGH** (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**
Complexity of sensible cooling system	**LOW**
Integration of desiccant and sensible cooling systems	The integration of desiccant and sensible cooling processes is straight forward and includes the cooling of the supply air.
Integration with space conditioning options	Both all-air or hydronic space rejection for space conditioning are possible. Chilled water is generated
Fuel used for processes	<u>Desiccant system:</u> Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) <u>Cooling:</u> Variable heat sources can be used for thermal energy conversion, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel)
Ease of using low temperature heat sources	**HIGH**; for both the desiccant and the cooling processes. The competitiveness of the entire desiccant cooling system is mainly based on the use renewable or low temperature heat sources
Maturity technical / market	Technical maturity of the entire system is **HIGH**; the technology has only a few pilot installations; market maturity is **MEDIUM** due to limited number of vendors

Outlook for use of the system integration	The outlook of the cooling desiccant system using adsorption chillers is very promising.

Description of the sensible heat load rejection: **Adsorption chiller**

Adsorption chillers use solid desiccants that adsorb water vapor on their surface or within their structure to drive a refrigeration cycle. The adsorption cycle operates batch wise, which is not a continuous process, but is based on cyclical adsorbing of vapor into the solid desiccant material and desorbing the vapor from the bed with by means of a heat source. In essence the process is similar to the conventional vapor compression cycle in regard to evaporating and subsequently condensing of a working fluid. In the case of the adsorption chiller the working fluid is water.

In order to provide continuous cooling, two or more adsorption beds oscillate between the adsorption and desorption state, such that, as one bed reaches saturation and must be regenerated, the other bed is fully regenerated and ready to adsorb vapor. Figure 6.8.3 illustrates the process of an adsorption chiller. In the figure heat exchanger A, (e.g. desiccant vessel) adsorbs moisture from the evaporator, where refrigerant (here water) evaporates reducing temperatures on the air-liquid internal heat exchanger, thereby generating chilled water. At the same time an external heat source warms the heat exchanger B (e.g. desiccant vessel) thus causing the desiccant to release water vapor which flows to the condenser where it transfers latent heat of condensation to the cooling water loop. Figure 6.8.4 shows a commercially available adsorption chiller sold be the company Invensor (Invensor.com). The adsorption chiller depicted has a maximum cooling capacity of 35 $kW_{thermal}$.

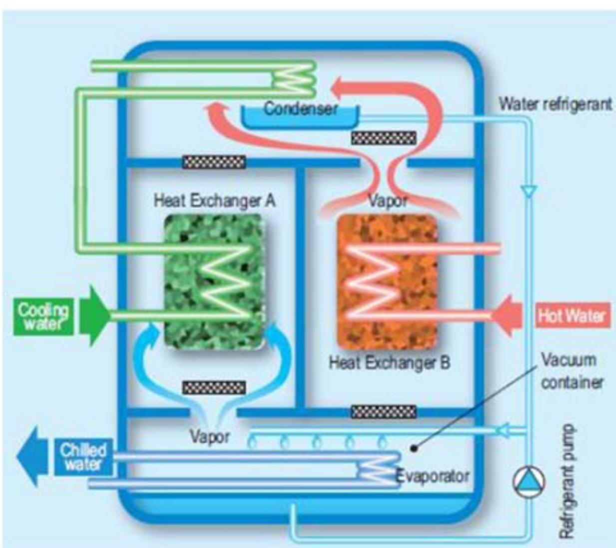


Figure 6.8.3: Process scheme of an adsorption chiller (DOE, 2014)

Figure 6.8.3: Commercially available Adsorption chiller (source: Invensor.com)

6.9 High Level System Integration – D: Desiccant and Absorption chiller

The candidate system configuration D is depicted in Figure 6.9.1. The system components and performance characteristics of candidate system configuration D is described in the following.

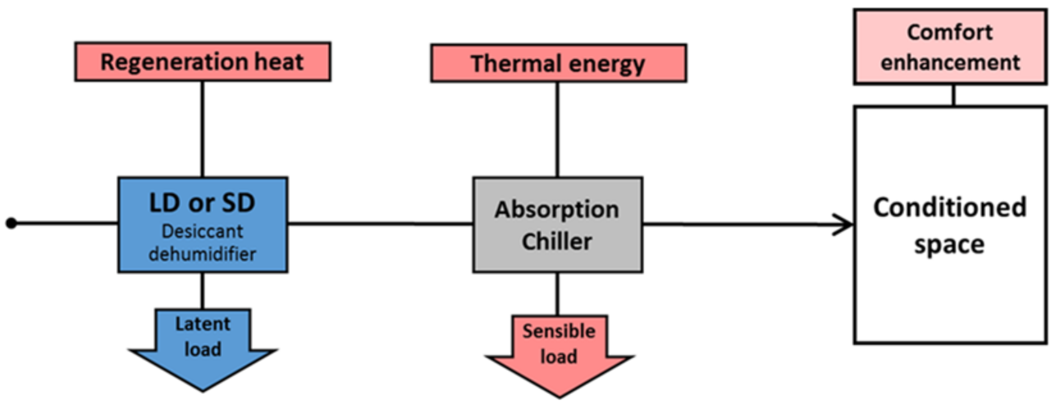


Figure 6.9.1: System integration D with Desiccant and Absorption chiller

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the desiccant dehumidification unit which removes a certain humidity of contained in the outdoor air supply. The downstream sensible heat removal is accomplished by means of an Absorption chiller. The chiller generates chilled water which cools the supply air. Chilled water can also be pumped to a radiant cooling ceiling or to a chilled beam.
Main objective	Demonstrate that absorption chillers work in conjunction with desiccant dehumidification to form an efficient desiccant cooling system.
Desiccant material used	Solid or liquid desiccant material; in many applications lithium-bromide (LiBr) salt solutions is used as the absorbent
Sensible cooling technology	An absorption chiller used to provide cooling to the supply air through heat exchanger or to supply chilled water to radiant ceiling and chilled beam.
Working fluid (other than water)	Liquid desiccant material (lithium bromide solution) is used in the closed cycle (encapsulated) adsorption chiller process vessel
Energy savings projected	The energy savings are projected as **MODERATE** (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**

Aspect of system	Description
Complexity of sensible cooling system	**HIGH**
Integration of desiccant and sensible cooling systems	The integration of desiccant and sensible cooling processes is straight forward and includes the drying and subsequent cooling of the supply air.
Integration with space conditioning options	Both all-air or hydronic space rejection for space conditioning are possible. Chilled water is generated in the absorption chiller and can be used in air-handling units or in hydronic systems.
Fuel used for processes	<u>Desiccant system:</u> Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) <u>Cooling:</u> Variable heat sources can be used for thermal energy conversion, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel)
Ease of using low temperature heat sources	**HIGH**; for both the desiccant and the cooling processes. The competitiveness of the entire desiccant cooling system is mainly based on the use renewable or low temperature heat sources
Maturity technical / market	Technical maturity of the entire system is **HIGH**; Absorption chillers have been installed at many locations and have a track record, therefore market maturity is **HIGH**
Outlook for use of the system integration	The outlook of the cooling desiccant system using absorption chillers is promising, since more recent design have solved some operational difficulties of older designs.

Description of the sensible heat load rejection: **Absorption chiller**

Absorption cooling process uses solutions of lithium bromide, and water, or other solutions, which have a high affinity to water. The absorption of water vapor into the solutions drives the process. The driving force is similar to the adsorption chiller, save the difference between the use of liquid in lieu of solid desiccants, and no need for batch wise operation, which the absorption does not have.

Figure 6.9.2 illustrates the absorption process. Water in tank A evaporates and water vapor flows towards the absorber (tank B) which contains the absorbing solution, e.g. the solid lithium bromide and water solution. Evaporation of the water in the evaporator (tank A) causes the temperature to decrease. The achievable temperature in the evaporator depends on the evaporator pressure and the effectiveness of the evaporative process. Since mass enters the absorber (tank B) and dilutes the solution, fluid from the absorber is continuously pumped from tank B to the generator (tank C). In the generator, the solution is heated either directly by a natural gas burner, or indirectly by means of a steam coil. While the solution is heated, water evaporates and passes into the condenser (tank D). The regenerated and concentrated solution accumulates in the sump of tank C, and is then returned to the absorber (tank B). Here, it again absorbs water vapor that comes from the evaporator. Water vapor in the condenser (tank D) is cooled by a separate coil of pipe through which cooling water passes. The condensed water is returned to the evaporator (A).

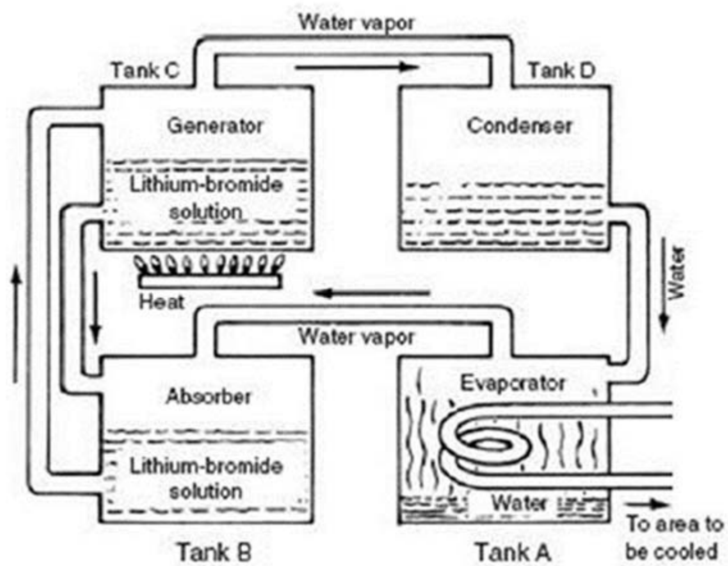
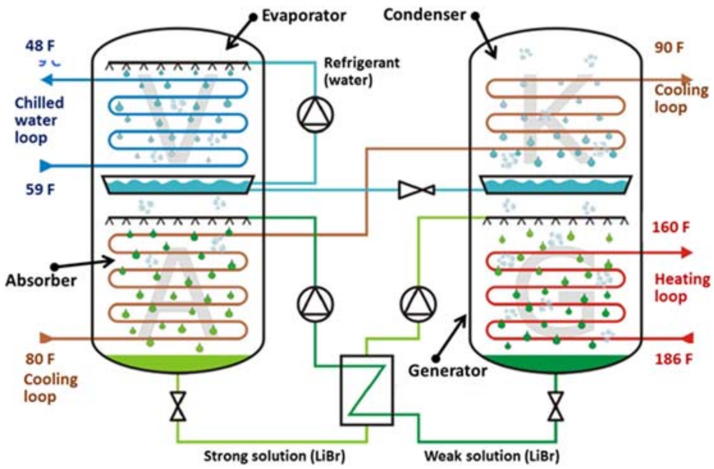


Figure 6.9.2.: Basic system diagram of a single effect absorption chiller

The diagram indicates the mass flow of the absorption and regeneration process. A Lithium-bromide salt solution is the absorbent of choice in many applications. Pressures and energy flows (e.g. heat rejection) are not all indicated. (EERE, 2016)

Absorption chillers have typically been used in large chilled-water cooling applications due to their operational complexity and relative low COP. While the COP is low, absorption chillers are attractive for the reduction of source energy (e.g. use of natural gas) and their ability to provide peak demands without the need for high electric loads. Where higher temperature heating sources are available, absorption chillers can utilize multiple generator effects for improved thermal efficiency.

Smaller absorption chillers for residential or light commercial use have become available lately. The more recent absorption technology has better maintenance and longer useful life-time of the equipment than older technology, which often bedeviled by difficult maintenance, due to the corrosiveness of the liquid absorbent used in the process. Figure 6.9.3 depicts a small scale absorption chiller for light commercial use. The one depicted has a thermal rating of 50 kW. The range of available absorption chillers offered by the manufacturer EWA is 15 to 200 kW (thermal).



Process diagram of single effect absorption chiller — Absorption chiller (50 kW)

Figure 6.9.3: Example of a small absorption chiller for light commercial use (e.g. 50 kW), (source http://www.eaw-energieanlagenbau.de/ modified)

6.10 High Level System Integration – E: Desiccant and Magnetic Chiller

The candidate system configuration E is depicted in Figure 6.10.1. The system components and performance characteristics of candidate system configuration E is described in the following.

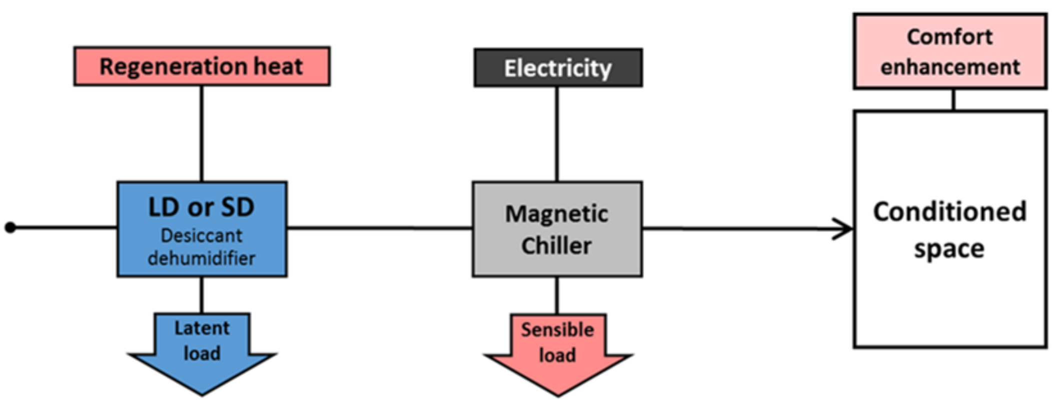


Figure 6.10.1: System integration E with Desiccant and magnetic chiller

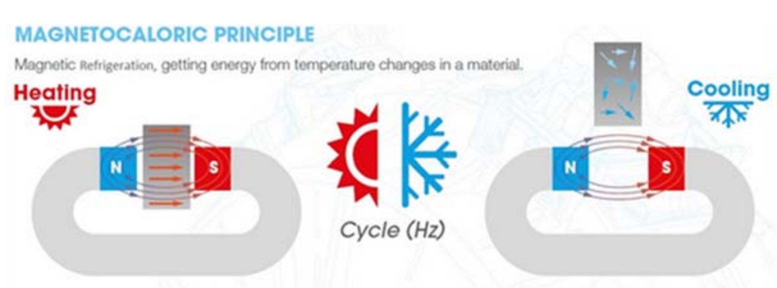
Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the desiccant dehumidification unit which removes a certain humidity of contained in the outdoor air supply. The downstream sensible heat removal is carried out by means of an electrically driven chiller, which work on the basis of the magneto caloric effect, called the "magnetic refrigeration. The chiller generates chilled water which cools the supply air. Chilled water can also be pumped to a radiant cooling ceiling or to a chilled beam.
Main objective	Present an emerging Solid-State chiller technology which is an alternative to conventional chiller technology. There are several promising alternative chiller technologies which are presently developed. The magnetic refrigeration is one of the more promising technology which is close to being commercially viable and available.
Desiccant material used	Solid or liquid desiccant material; in many applications lithium-bromide (LiBr) salt solutions is used as the absorbent
Sensible cooling technology	Chiller that converts magnet field changes to heat flow; so-called magnetic chiller technology
Working fluid (other than water)	Magnetocaloric Materials (MCM), which heat up when immersed in a magnetic field and cool down when removed from it; process is based on the Magnetocaloric Effect (MCE)
Energy savings projected	The energy savings are projected as **MEDIUM** (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**
Complexity of sensible cooling system	**HIGH** based on emerging technology; however, the manufacturers claim the maintenance will be extremely low
Integration of desiccant and sensible cooling systems	The integration of desiccant and sensible cooling processes is straight forward and includes the drying and subsequent cooling of the supply air.
Integration with space conditioning options	Both all-air or hydronic heat rejection for space conditioning are possible. Chilled water is generated in the magnetic chiller and can be used in air-handling units or in hydronic systems.
Fuel used for processes	Desiccant system: Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) Cooling: Electricity
Ease of using low temperature heat sources	**HIGH**; for the desiccant and **NON** existing for the magnetic cooler

Aspect of system	Description
Maturity technical / market	<u>Technical maturity</u> of the entire system is **MODERATE**; magnetic chiller represents an emerging refrigeration and cooling technology; the first commercial installations have been tested by the main vendor "CoolTech Applications"; therefore <u>market maturity</u> is **LOW**
Outlook for use of the system integration	The outlook of the cooling desiccant system using absorption chillers is very promising, due to their energy saving potential, very quiet operation and small sizes.

Description of the sensible heat load rejection: **Magnetic (magneto caloric effect) cooling**

Magnetic cooling operates on the magneto caloric effect, a phenomenon in which a magneto caloric material exhibits reversible temperature change when exposed to a changing magnetic field.

The magneto caloric effect causes the temperature of the magneto caloric material to increase when it is exposed to a magnetic field and decreases when it is removed from it. Thus the temperature changes are reversible, almost instantaneous, allowing a flow of heat, and therefore enabling the removal of heat energy. The controlled magnetic field applies a series of magnetization-demagnetization cycles to the magneto-caloric alloys. Each of these cycles creates a temperature gradient in the material. A rapid succession of these cycles produces the final and stabilized hot and cold temperatures in the refrigerated system. The temperature with the strongest effect (e.g. Curie temperature) depends on the properties of the magneto caloric material. Figure 6.10.2 illustrates the basic working principle of magnetic cooling and presents the world's first commercially available magnetic cooling device developed and sold by French company "Cooltech Applications".



Basic process diagram of magnetic cooling

The Magnetic Refrigeration System; (Cooltech, 2016)

Figure 6.10.2: Working principle of magnetic cooling and presents the world's first commercially available magnetic cooling device (source, http://www.cooltech-applications.com)

The benefits of magnetic cooling are described by Cooltech Applications as follows:

- Energy savings up to 50%: Energy intense mechanical work, such as vapor compression is avoided. Magnets are permanent and do not require an energy source to operate.
- No Refrigerant Gas: Total Avoidance of refrigerant gas and the related drawbacks thus producing cold energy without any polluting gas.
- Cost Reduction: Magnetic Cooling requires a low level of maintenance (low rotational speed, low pressure, no leaks and no hazardous chemicals…) which lowers the operational cost for the end user.
- Reduced noise and vibrations: The Magnetic Refrigeration System operates with a low noise and reduced vibrations.
- Safety: The absence of refrigerant gas avoids impacts from hazardous leaks. Measured magnetic emissions that surrounded the devices were far lower than the ones from a small refrigerator magnet.

6.11 High Level System Integration – F: Integrated DEVAP Membrane System

The candidate system configuration F is depicted in Figure 6.11.1. The system components and performance characteristics of candidate system configuration F is described in the following. The acronym DEVAP system stands for Desiccant Enhanced Evaporative Air Conditioning.

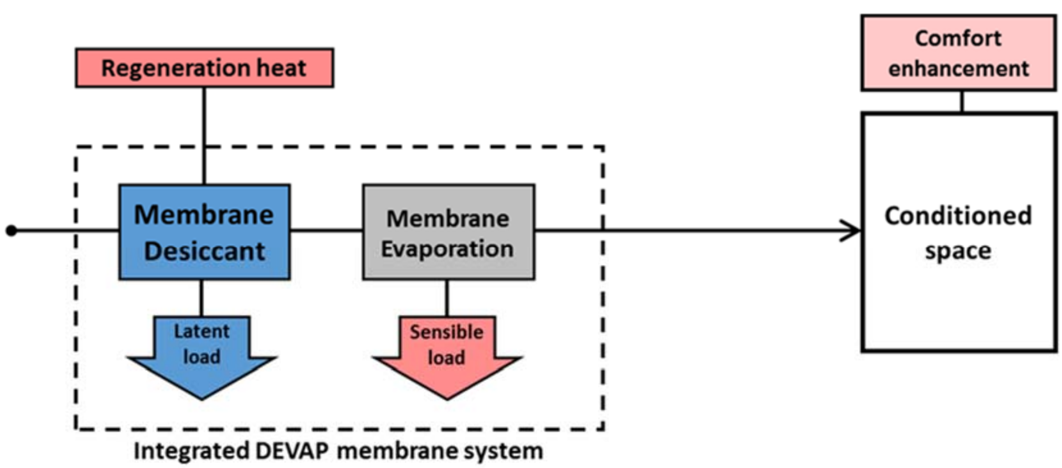


Figure 6.11.1: System integration F with Integrated DEVAP membrane system

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the integrated DEVAP system where desiccant dehumidification removes a certain level of humidity contained in the outdoor air supply. Parallel to dehumidification, sensible cooling is performed inside the same unit.
Main objective	Demonstrate an emerging membrane chiller technology as an energy efficient way to dry and cool supply air with a highly integrated and compact device.
Desiccant material used	Liquid desiccants, lithium chloride (LiCl) or calcium chloride ($CaCl_2$); desiccant is encapsulated in membranes
Sensible cooling technology	Indirect evaporative cooling process
Working fluid (other than water)	NONE
Energy savings projected	The energy savings are projected as HIGH (please refer to Section 7 for details of the energy savings)
Complexity of the integrated desiccant and sensible cooling system	**MEDIUM,** the main objective of the DEVAP system is to provide an integrated and compact cooling system
Complexity of sensible cooling system	**HIGH** based on emerging technology; the manufacturers claim the maintenance requirements will be extremely low
Integration of desiccant and sensible cooling systems	Desiccant and sensible cooling processes are performed in parallel inside the same device.
Integration with space conditioning options	Only all-air space conditioning is possible. The system works with return air.
Fuel used for processes	Desiccant system: Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) Cooling: Adiabatic cooling
Ease of using low temperature heat sources	**HIGH**
Maturity technical / market	Technical maturity of the entire system is **MODERATE**; the DEVAP process represents an emerging integrated dehumidification and cooling technology; the first commercial installations have been installed at selected locations and applications, market maturity is **LOW**
Outlook for use of the system integration	The outlook of the DEVAP technology is very promising, due to their energy saving potential and compact and small size.

Description of the integrated dehumidification and sensible heat load rejection: **DEVAP technology**

The DEVAP technology is a recent innovative chiller technology development, which has been supported by the U.S. Department of Energy. The DOE suggested that energy savings of the DEVAP could be as high as “80 percent less total energy than traditional air conditioning”. DEVAP technology combines hydrophobic membranes, liquid desiccants and evaporative cooling together in a way that has not been done previously. The DEVAP system uses salt solutions instead of conventional CFCs or HCFC refrigerants, contributes to global warming.

The process allows for greater control over the liquid desiccant flow so the water and desiccant remains separated from the air flow and no liquid desiccant is entrained in the supply air stream. Possible entrainment is a significant concern since the system uses up to 70% of return air and the DEVAP membranes encapsulate (e.g. contain) the liquid desiccant solution, thereby avoiding entrainment. The desiccant creates dry air using heat, while the evaporative cooler turns dry air into cold air; the desiccant and evaporative cooler work together to create cold-dry air. The DEVAP process avoids buildings over cooling and the technology is expected to reduce electric power demand and grid strain.

The U.S. Department of Energy plans to roll out this technology in commercial markets first. The energy payback for efficient cooling is higher in commercial applications. Once the DEVAP technology is established, residential use is expected to start as well. The DOE reports disadvantages of the system as high upfront costs and maintenance staff need to undergo training on the technology. Prototypes suggests that the risks are unknown with reliability and longevity being the greatest risks, especially as far as the innovative membrane desiccant technology is concerned. Figure 6.11.2 illustrates the basic working process of the DEVAP process. As illustrated the system works with a significant portion of return air as is required for the thermodynamic process. This is unusual for evaporative cooling, which usually works with 100% outside air. Figure 6.11.3 shows the dehumidification and sensible cooling stages of the device.

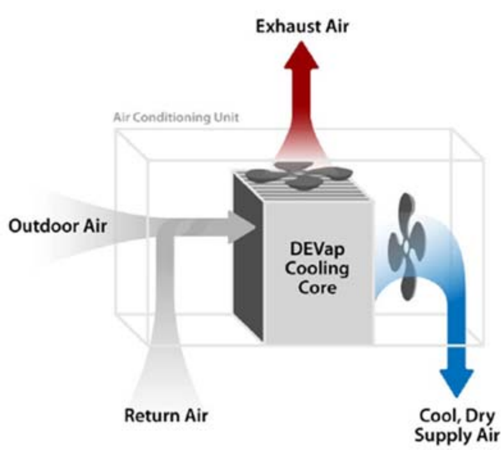


Figure 6.11.2: Basic work principle of the DEVAP process

The figure illustrates that the system is an integrated and compact device. Typically, the combination of

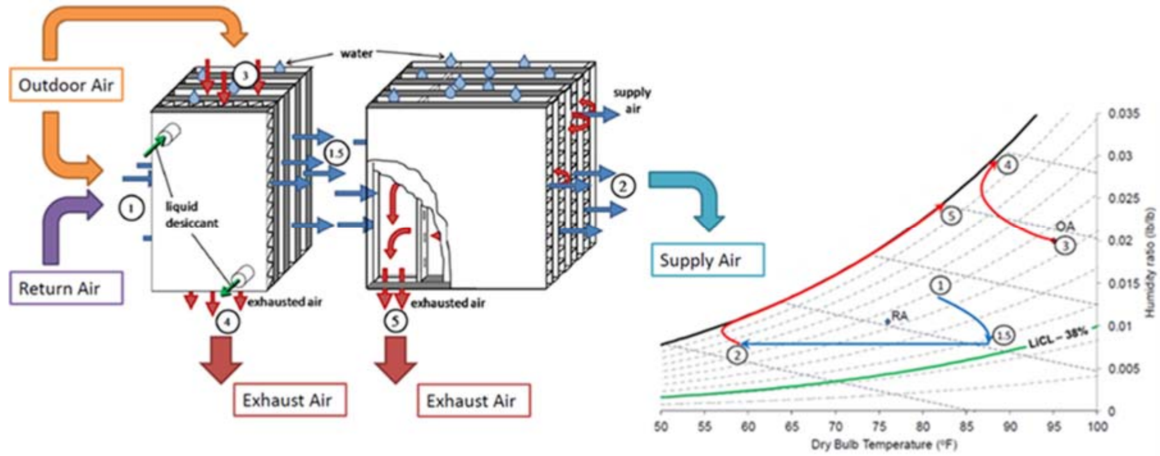


Figure 6.11.3: The DEVAP technology, process diagram and psychometrics (NREL, 2013)

6.12 High Level System Integration – G: Liquid-Desiccant Direct-Expansion (LDDX) AC

The candidate system configuration G is depicted in Figure 6.12.1. The system components and performance characteristics of candidate system configuration G is described in the following.

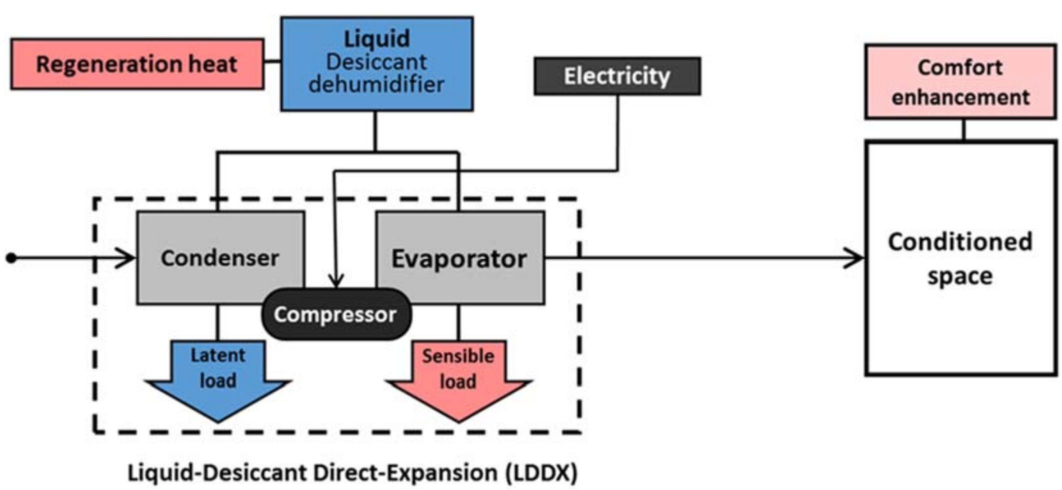


Figure 6.12.1: System integration G with Liquid-Desiccant Direct-Expansion (LDDX) air-conditioner

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the integrated LDDX system where desiccant dehumidification removes a certain level of humidity contained in the outdoor air supply. Parallel to dehumidification sensible cooling is performed inside the same unit by the evaporator of a conventional direct-expansion unit powered by vapor compression cycle.
Main objective	Demonstrate capabilities of the liquid-desiccant direct-expansion air conditioner (LDDX) to provide an efficient process to control indoor humidity. The emerging LDDX technology is an integrated liquids desiccant and sensible heat removal system using a conventional vapor compression cycle and an innovative wicking-fin technology for effective air drying.
Desiccant material used	Liquid desiccants, such as lithium chloride (LiCl), included in the wicking-fin enhanced condenser and evaporator
Sensible cooling technology	Conventional vapor compression
Working fluid (other than water)	conventional CFCs or HCFC refrigerants
Energy savings projected	The energy savings are projected as MEDUIM (please refer to Section 7 for details of the energy savings)
Complexity of the integrated desiccant and sensible cooling system	**LOW,** the main objective of the LDDX system is to provide an integrated and compact desiccant cooling system
Complexity of sensible cooling system	**LOW**; the vapor compression cycle is a well proven cooling technology
Integration with space conditioning options	Only all-air space conditioning is possible, similar to conventional air handling units in buildings.
Fuel used for processes	Desiccant system: Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) Cooling: Electricity
Ease of using low temperature heat sources	**MEDIUM,** only desiccant regeneration can use low temperature sources
Maturity technical / market	Technical maturity of the entire system is **MODERATE**; the LDDX process represents an emerging integrated dehumidification and cooling technology; the first commercial installations have been installed at selected locations and applications, market maturity is **LOW**

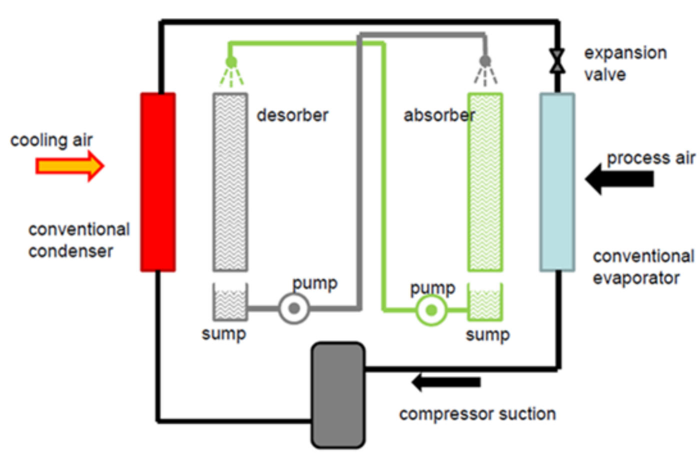
Aspect of system	Description
Outlook for use of the system integration	The outlook of the LDDX technology is very promising, due to their energy saving potential and compact and small size. The possibility of retrofitting existing AC systems increases the positive outlook.

Description of the integrated dehumidification and sensible heat load rejection: **LDDX technology**

The LDDX technology is a small-tonnage, packaged air conditioner that has more than twice the dehumidification capacity (latent cooling) of a conventional air conditioner. High latent cooling is achieved by integrating a liquid desiccant with the evaporator and condenser of a vapor-compression refrigeration circuit. This effective integration is accomplished by replacing the traditional aluminum-finned/copper-tube heat exchangers found in a conventional direct-expansion (DX) air conditioner with a wicking-fin heat exchanger that allows thin films of the liquid desiccant to cover the surfaces of the heat exchanger. On the cooled evaporator surfaces of the LDDX, the cool desiccant absorbs more water than would be condensed in a conventional evaporator. On the heated condenser surfaces, the water that was absorbed in the evaporator is released by the warm desiccant and carried by the condenser's cooling air flow. Since air does not have to be cooled below its dew point to condense the water vapor, the LDDX's evaporator can run at a higher temperature than that of a conventional DX air conditioner. This relatively high evaporator temperature leads to an efficient, high latent cooling system

Figure 6.12.2 shows the basic working principle of the LDDX technology. The main steps in the LDDX process are as follows:

- LDDX uses a conventional DX refrigeration circuit
- Desiccant-wetted absorber dries the saturated air leaving the evaporator
- Desiccant-wetted desorber rejects water to warm, low-RH air leaving condenser
- Reverts to DX AC when desiccant is turned off



<<<< Figure 6.12.2: basic working principle of the LDDX technology

5 5-Ton LDDX system installed by AIL Research (ailr.com)

The LDDX technology provides an energy-efficient means of controlling indoor humidity. In humid climates, the conventional approach to humidity control--overcooling the air to condense the water vapor followed by reheating to maintain comfort--increases cooling loads by about 30%. The LDDX eliminates overcooling and reheating, leading to significant savings.

6.13 High Level System Integration – H: Desiccant and Conventional Chiller

The candidate system configuration H is depicted in Figure 6.13.1. The system components and performance characteristics of candidate system configuration G is described in the following. This system integration uses a conventional vapor compression cycle as the chiller. This candidate system provides the benchmark for assessing the efficiency of the other candidate systems.

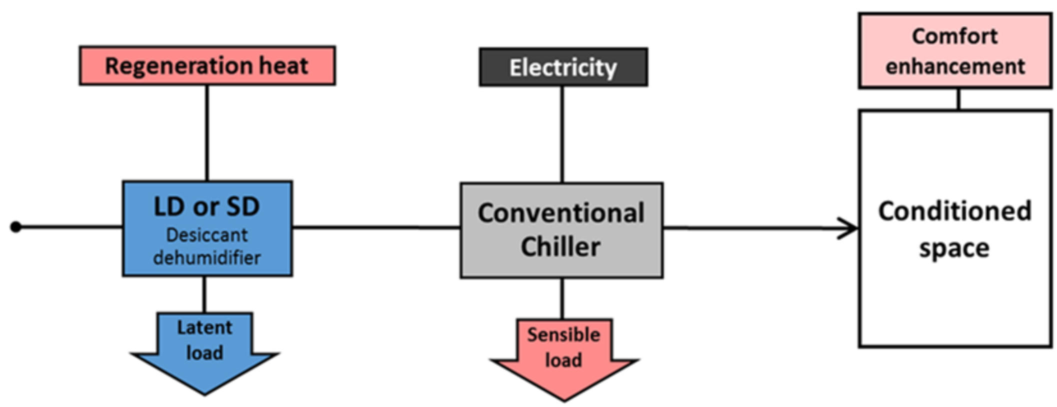


Figure 6.4.1: System integration H with Desiccant and conventional chiller

Aspect of system	Description
Overall working scheme of the system integration:	Outdoor air is passed through the integrated LDDX system where desiccant dehumidification removes a certain level of humidity contained in the outdoor air supply. A conventional chiller based on vapor compression cycle removes sensible heat.
Main objective	Assess the performance improvement by combining a conventional chiller with desiccant dehumidification, so to separate sensible and latent heat removal.
Desiccant material used	Solid or liquid desiccant material can be used
Sensible cooling technology	Conventional vapor compression

Aspect of system	Description
Working fluid (other than water)	conventional CFCs or HCFC refrigerants
Energy savings projected	The energy savings are projected as MEDIUM (please refer to Section 7 for details of the energy savings)
Complexity of desiccant system	**MEDIUM**
Complexity of sensible cooling system	**LOW**
Integration with space conditioning options	Both all-air and hydronic space conditioning is possible
Fuel used for processes	Desiccant system: Variable heat sources can be used for desiccant regeneration, including renewable, low temperature (solar and waste heat), and high temperature (natural gas of biofuel) Cooling: Electricity
Ease of using low temperature heat sources	**MEDIUM,** only desiccant regeneration can use low temperature sources
Maturity technical / market	Technical maturity of the entire system is **HIGH /** market maturity **MEDIUM** based on the limited use of desiccant systems combined with conventional vapor compression chiller technology
Outlook for use of the system integration	The outlook of this system integration is promising, since it opens the possibility of energy efficient retrofits, with many conventional chillers already in place.

Since the conventional vapor compression chiller technology is well known it is not described further.

SECTION 7- PROJECTED ENERGY SAVINGS

This section presents the results of energy savings assessments of the eight candidate systems. The energy assessments use generic, yet realistic assumptions of unit energy use, coefficients of performance, energy conversion effectiveness and manufacturer provided energy efficiency information. It is understood that the energy analysis is not as detailed and therefore precise as can be achieved with more detained energy modelling. A more precise energy analysis is beyond the scope of the present feasibility assessment.

The assumptions used in the assessment of energy savings are as follows:

- The latent and sensible heat load of the supply air is assessed with average annual climatic conditions. This energy analysis uses the ventilation load index (VLI), which was introduced in Section 6. The VLI for the entire data record and the abbreviated data record for the daytime hours (e.g. 11-hour record) were averaged to provide a representative VLI expression. This VLI provides the representative unit sensible and latent heat of the air that is provided to the conditioned space. These representative numbers for the annual latent and sensible unit heat removal rates for the outside air are 10.2 and 2.4 ton-hours per cfm, respectively.
- The latent and sensible heat load of the outside air were augmented by internal latent and sensible heat gain to obtain the overall unit rate of heat removal from the conditioned space that corresponded with the sensible heat rate (SHR) of 50%, which is a conservative estimate for Hawaii. The resulting total representative latent and sensible heat load rates that have to be removed from the conditioned space are 12.2 and 11.7 ton-hours per cfm, respectively.

In order to describe latent and sensible energy in absolute terms, an example building was assumed with the required ventilation flow rate as follows:

Description	Unit	Amount
Type building – type of use	School buildings (students older than 9 years)	
Ventilation rate for occupants (*)	cfm / person	10
Ventilation rate for space (*)	cfm / sqft	0.12
Occupant density (*)	person/ft2	35
Size of the conditioned space	ft2	4,000
Number of students considered		115
Ventilation flow for occupants	cfm	1,150
Ventilation flow for space	cfm	480
Total ventilation flow	cfm	1,630

Note (*): Values for ventilation rates and occupant density are from ASHRAE 62.1.

- The total ventilation rate was applied to various parameters, which included site-to-source energy factor, representative coefficient of performance (COP) for a conventional vapor compression and the efficiency to the dehumidification. The assumptions and input parameters of the calculations of effective energy savings is summarized below:

Description	Unit	Amount
Latent heat load to be removed with AC	ton hours	19,900
	kWh $_{therm}$	70,000
Sensible heat load to be removed with AC	ton hours	19,000
	kWh $_{therm}$	66,800
Total heat load to be removed with AC	ton hours	38,900
	kWh $_{therm}$	136,800
COP for conventional chiller (vapor compression)	[-]	3.2
Total site energy to remove total heat load	kWh $_{electr.}$	42,750
Site-to-source energy factor for electricity	[-]	3.14
Total source energy to remove total heat load	kWh $_{electr.}$	134,200
BENCHMARK Required electric **site** energy to remove sensible heat	kWh $_{electr.}$	20,875
BENCHMARK Required electric **source** energy to remove sensible heat load	kWh $_{electr.}$	65,500
Efficiency of desiccant system to remove latent load	[-]	90%
Required electric **site** energy to remove latent heat load	kWh $_{electr.}$	7,000

Expected values of energy savings for the sensible heat removal was derived from the manufactures and/or published specifications. The energy saving rates were applied to the benchmark sensible heat load. For the system H, the conventional chiller, the energy savings were derived by the increased COP due to increased chilled water temperatures. The resulting energy savings are presented in the Table 7.1. The energy savings for the eight candidate systems are presented in Figure 7.1.

As reported in Section 6 the use of ceiling fans provides a cooling effect due to increased air flow over the human body. For the assessment of the effect of ceiling fans on the energy savings a conservative 45% reduction of sensible cooling due to the ceiling fan is considered; no savings for the latent load occur, obviously. Figure 7.2 shows a comparison of the energy savings for the eight candidate systems with and without the ceiling fans operating. The resulting savings suggest the saving potential and not the actual savings.

Table 7.1: Summary of energy savings calculations for eight candidate systems A through H

Sensible cooling technology used	Candidate system	site energy	source energy	sensible cooling base		savings for sensible cooling		resulting total energy demand		savings		
		benchmark			dehum	savings	resulting	site	source	site energy	source energy	savings
		kWh electr	kWh electr	kWh electr	kWh electr	[%]	kWh electr	kWh electr	kWh	kWh electr	kWh	%
Convent. evapor. cooling	A	42,750	134,200	20,900	7,000	75%	5,200	12,200	38,300	30,550	95,900	**71%**
Advncd. .evapor. Cooling M-cycle	B	42,750	134,200	20,900	7,000	55%	9,400	16,400	51,500	26,350	82,700	**62%**
Adsorption chiller	C	42,750	134,200	20,900	7,000	80%	4,200	11,200	35,200	31,550	99,000	**74%**
Absorption chiller	D	42,750	134,200	20,900	7,000	70%	6,300	13,300	41,800	29,450	92,400	**69%**
Magnetic chiller	E	42,750	134,200	20,900	7,000	50%	10,500	17,500	55,000	25,250	79,200	**59%**
DEVAP system	F	42,750	134,200	20,900	7,000	78%	4,600	11,600	36,400	31,150	97,800	**73%**
LDDX system	G	42,750	134,200	20,900	7,000	30%	14,600	21,600	67,800	21,150	66,400	**49%**
Convetional Chiller	H	42,750	134,200	20,900	7,000	10%	18,800	25,800	81,000	16,950	53,200	**40%**

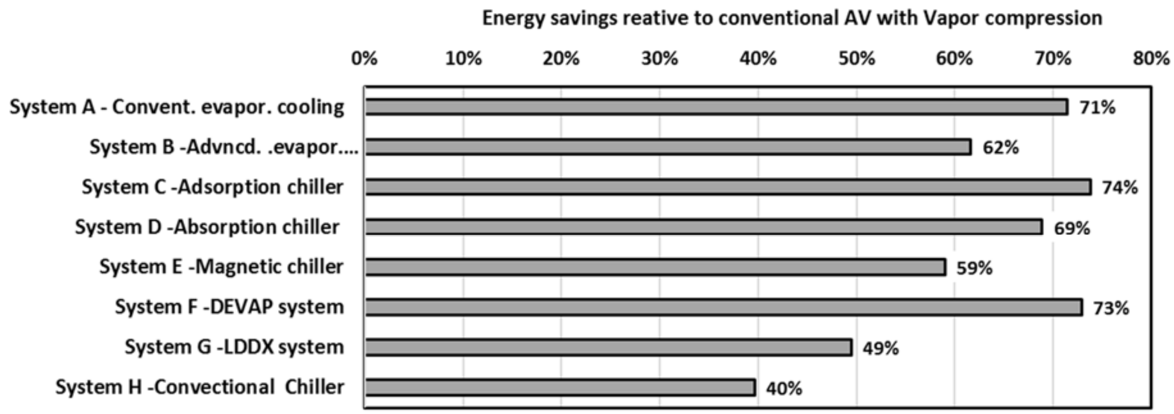


Figure 7.1: Projected average energy savings of eight candidate systems relative to conventional AC with vapor compression cycle a cooling based dehumidification

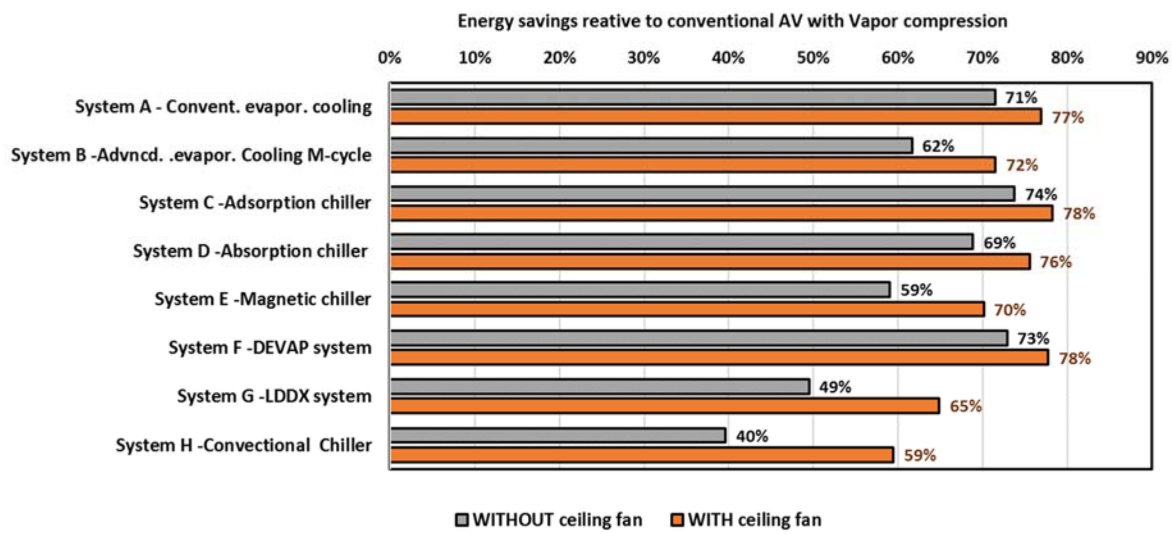


Figure 7.2: Projected average energy savings of eight candidate systems relative to conventional AC with vapor compression cycle a cooling based dehumidification; comparison with and without cooling effect from ceiling fan

SECTION 8 – BENEFIT VERSUS RISK ASSESSMENT

This section presents an assessment of benefits and risks associated with the eight candidate systems and determines a quantitative description, the benefit risk ratio (BRR). The determination of the BRR is intended to gauge the perceived risks versus benefits of using desiccant cooling systems. In order to establish the BRR a two level ranking approach was used. Ranking criteria that described aspects of benefits and risks were assigned overall weights and the eight candidate systems were ranked to their extent of matching the criteria definition.

In order to facilitate assigning of overall weights categories were established, where first categories and then ranking criteria in these categories were assigned overall weights. Table 8.1 illustrates the assignment of overall weights. For the sake of illustration, the table only presents benefit criteria.

Table 8.1: Description of calculating overall weight

(A)	(B)	(C)	(D)	(E)
Criterion No.		1 st level weights	2 nd level weights	overall weight
	Benefits			
B.1	**Financial benefits**	55%	100%	
B.1.A	Energy savings		60%	33%
B.1.B	First costs		40%	22%
B.2	**Technology benefits**	25%	100%	
B.2.A	Potential of peak reductions		30%	8%
B.2.B	Increase comfort (hydronic)		25%	6%
B.2.C	Compactness of unit		45%	11%
B.3	**Other benefits**	20%	100%	
B.3.A	Easy maintenance		60%	12%
B.3.C	Adopting to technology		40%	8%
	sums >>>	100%		100%

Table 8.1: Assignment of overall weights for overall criteria

Columns (A) through (E) have the following purpose:

(A) ID of the criterion
(B) Description of criterion
(C) First level weight for category, sum has to be 100%
(D) Second level weights for criteria in the category, sum has to be 100% for each category
(E) Overall weight is the product of first and second level weight, sum has to be 100%

In a second step a discrete ranking weights with a scale of 1 to 3 were assigned to each criteria and the discrete weight multiplied with the overall weight of the criterion. The sum of all calculated numbers is the representative benefit or risk points factor.

The criteria used are briefly described in Table 8.2. The rationale of assigning a discrete weight of "1" through "3" is presented. For the sake of brevity only the rationale for the high and low marks, which means assigning a 3 or 1, is described. Assigning a "2" refers to using a medium mark. It should also be noted that all ranking criteria are defined to obtain high values that refer to either benefits or risks. This means a case that lower risks is assigned a low value for the risk ranking but a high value for the benefit based ranking.

Table 8.2: Definition of criteria to determine the Benefit to Risk Ratio

Criterion No.	Criterion description	High Mark 3 given	Low Mark 1 given
B.	**Benefits**	**High benefits**	**Low benefits**
B.1.A	Energy savings: The predicted magnitude of energy savings of the desiccant cooling system	3= High energy savings	1 = low energy savings
B.1.B	First costs: The predicted first costs for the installation of the desiccant cooling systems	3= Low first costs	1 = High first costs
B.2.A	Potential of peak reductions: The capability of the desiccant cooling system to shave of peak loads. The capacity of the overall system is the combination of the desiccant dehumidification and sensible cooling systems.	3 = High capacity of peak reduction	1 = Low capacity of peak reduction
B.2.B	Increase comfort (hydronic): The capacity of the cooling system to integrate into a hydronic cooling system	3 = High capacity of hydronic cooling integration	1 = Low capacity of hydronic cooling integration
B.2.C	Compactness of unit: The combination of desiccant and sensible cooling system components can create sizable units. Compactness of the unit is a significant benefit, since it reduces maintenance complication.	3 = Overall system is compact	1 = Overall system is not compact
B.3.A	Easy maintenance: The level of complication is an important factor of building operators to accept a new building system. Easy maintenance procedures are preferred.	3 = Overall system is easy to maintain	3 = Overall system is NOT easy to maintain
B.3.C	Adopting to technology: High performance buildings rely on technology and design features that result in energy savings and a better indoor environment of occupants. Well trained building operators can effectively integrate the desiccant cooling system into the building operation.	3 = building operators do not need much training to efficiently use the desiccant cooling system	3 = building operators do need significant training to efficiently use the desiccant cooling system
R.	**Risks:**	**High risks**	**Low risks**
R.1.A	Backup systems required: Due to the sometimes early stage of development of the desiccant cooling systems, components might require a backup by less sophisticated technology	3 = Backup components are required	1 = Backup components are NOT required
R.1.B	Safety concerns: The operation of the desiccant cooling system might bring about different type of safety risks.	3 = significant safety concerns	1 = few safety concerns
R.2.A	Unknown technology: Building operators might resist since the desiccant cooling system presents unknown technology.	3 = technology is likely perceived as new and complicated	1 = technology is likely NOT perceived as too new and complicated
R.2.B	Support from vendors: There is a risk that not enough support from vendors will be available	3 = There will likely be limited vendor support	1= There will likely be sufficient vendor support

Table 8.2: Definition of criteria to determine the Benefit to Risk Ratio

Criterion No.	Criterion description	High Mark 3 given	Low Mark 1 given
	since the technology does not have sufficient market maturity		
R.3.A	About financial risks: Decision makers might be unsure about investing in a technology that has a certain track record or promises a certain energy savings performance.	3 = feeling unsure about savings and financial performance	1 = feeling convinced about savings and financial performance
R.3.B	About hidden things: Decision makers might be envisioning hidden problems associated with a new technology that are not apparent at the present time. Hidden complications might be more prevalent with technologies of low market maturity.	3 = anticipating hidden complication	1 = NOT anticipating hidden complication

Table 8.3. shows the results of determining the Benefit and Risk points factors.

Table 8.3: Determining the benefits and risk points factors

					Candidate systems A through H; desiccant dehumidification with sensible cooling							
					A	B	C	D	E	F	G	H
Criterion No.		1st level weights	2nd level weights	overall weight	convent. evapor. Cooling	M-cycle evapor. Cooling	adsorption chiller	absorption chiller	magnetic chiller	DEVAP system	LDDX system	convent. Chiller VCAC
	Benefits											
B.1	**Financial benefits**	55%	100%									
B.1.A	Energy savings		60%	33%	3	3	3	3	2	3	2	1
B.1.B	First costs		40%	22%	2	1	1	2	1	1	2	2
B.2	**Technology benefits**	25%	100%									
B.2.A	Potential of peak reductions		30%	8%	3	3	3	3	1	2	2	1
B.2.B	Increase comfort (hydronic)		25%	6%	1	1	3	3	3	1	2	3
B.2.C	Compactness of unit		45%	11%	1	2	3	1	3	3	3	2
B.3	**Other benefits**	20%	100%									
B.3.A	Easy maintenance		60%	12%	2	2	2	1	3	3	2	2
B.3.C	Adopting to technology		40%	8%	2	1	2	2	2	2	2	3
	sums >>>	100%		100%								
		Befefit points score >>>			2.2	2.0	2.4	2.2	2.0	2.3	2.1	1.7
	Risks				A	B	C	D	E	F	G	H
R.1	**Technology related**	25%	100%									
R.1.A	Backup systems required		40%	10%	1	2	2	2	1	2	1	1
R.1.B	Safety concerns		60%	15%	2	1	1	3	2	2	2	1
R.2	**Operator concerns/resistance**	40%	100%									
R.2.A	Unknown technology		40%	16%	1	3	2	1	3	3	2	1
R.2.B	Support from vendors		60%	24%	1	3	2	2	3	3	2	1
R.3	**Management concerns**	35%	100%									
R.3.A	About financial risks		70%	25%	2	3	1	2	2	3	2	1
R.3.B	About hidden things		30%	11%	1	3	1	1	2	3	2	2
		100%		100%								
		Risk points score >>>			1.4	2.6	1.5	1.9	2.3	2.8	1.9	1.1

Table 8.4 presents the benefit versus risks ratio (BRR) for the eight candidate systems. Figure 8.1 illustrates the use of the BRR matrix. As shown in Figure 8.1, a line of BRR = 1 indicates equal benefits and risks. The combination of benefits and risk point factors indicates whether the candidate systems are located above or below the line of BRR=1 line, which indicates that either more benefits or risks can be identified, respectively. Figure 8.2 shows the resulting BRRs for all eight candidate systems inserted into the BRR matrix. Figure 8.3 shows the deviation of the eight candidate from a the neutral BRR = 1 datum.

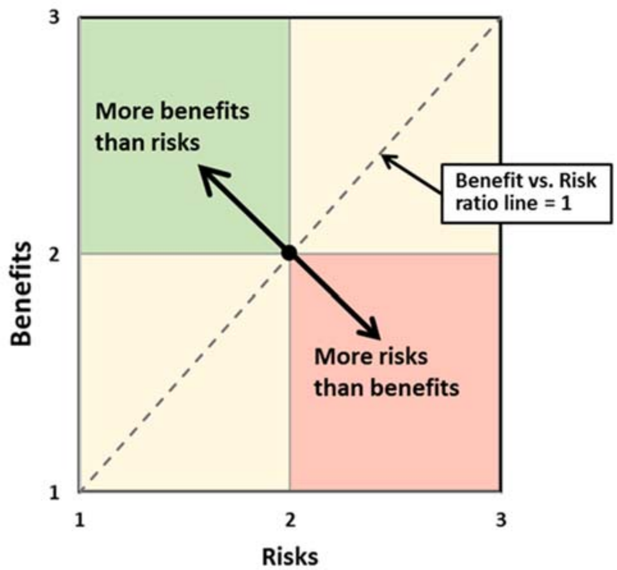


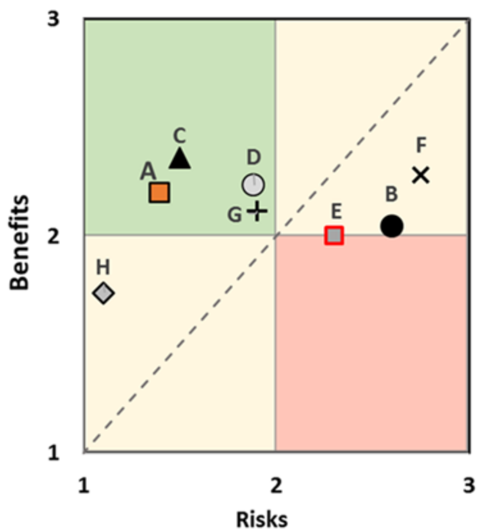
Figure 8.1: BRR matrix

The figure indicates the use of the BRR matrix. The neutral dashed line indicates a BRR value of 1, which mean that benefits and risks outweigh each other. When a resulting combination of benefit and risk point factors is located above the neutral line benefits are larger than risks, and vice versa.

Table 8.4: Results of benefits and risk points factors and Benefits Risk Ratio (BRR)

		benefits score points	Risks score points	benefits vs. risk factor	B vs. R factor deviation from 1.0
System A - Convent. evapor. cooling	A	2.20	1.40	1.58	0.58
System B - Advncd. .evapor. Cooling M-cycle	B	2.04	2.60	0.79	-0.21
System C -Adsorption chiller	C	2.36	1.50	1.57	0.57
System D -Absorption chiller	D	2.24	1.89	1.19	0.19
System E -Magnetic chiller	E	2.00	2.30	0.87	-0.13
System F -DEVAP system	F	2.28	2.75	0.83	-0.17
System G -LDDX system	G	2.11	1.90	1.11	0.11
System H -Convectional Chiller	H	1.74	1.11	1.57	0.57

Figure 8.2: BRR Matrix with Results



A	Convent. evapor. cooling
B	Advncd. .evapor. Cooling M-cycle
C	Adsorption chiller
D	Absorption chiller
E	Magnetic chiller
F	DEVAP system
G	LDDX system
H	Convectional Chiller

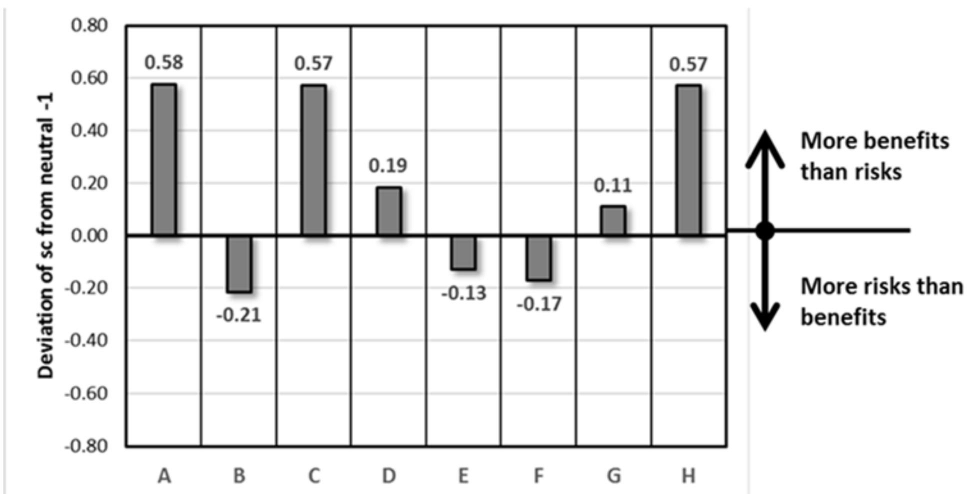


Figure 8.3: Deviation from neutral BRR for eight candidate systems

The results of balancing the benefits and risks suggest that three candidate system have a good benefit to risk ratio. These are the candidate systems A, C and H. Systems A and C represent innovative desiccant cooling systems which use conventional evaporative cooling and adsorption chiller as the sensible cooling technology. System H is the system with a conventional vapor compression chiller, which obviously has low risks because of the known and proven vapor compression chiller technology.

SECTION 9 - RANKING OF CANDIDATE SYSTEMS

This section presents the ranking of the eight candidate. The objective of the ranking is determining the most suitable desiccant cooling candidate system. The ranking will be done with a two-step ranking process, similar to that presented in Section 8 and used to determine the benefit versus risk ratio. The procedure to determine the overall ranking does, however, use both discrete and continuous scores. Before the final ranking is carried out in Section 9.2 the numeric results of the energy savings and benefit to risk ration have to be converted to a continuous score scale. This is being done in Section 9.1.

9.1 Conversion of Numerical Energy Savings and BRR Results to a Continuous Score

Energy savings: The energy savings for the eight candidate systems are converted to scoring values. The conversion uses two representative energy savings numbers and assign them a maximum and minimum score. The individual energy savings are then distributed linearly between these two values. Table 9.1.1 presents the conversion of the energy savings which were calculated for the "**Without** ceiling fan" case to the matching score. Figure 9.1.1 illustrates the distribution of the scores presented in Table 9.1.1 along a line connecting the minimum and maximum scores. Table 9.1.2 presents the conversion of the energy savings which were calculated for the "**With** ceiling fan" case to the matching score. Figure 9.1.2 illustrates the distribution of the scores presented in Table 9.1.2 along a line connecting the minimum and maximum scores.

Without ceiling fans

candidate system No.	Energy savings	Corresponding score
System A	71%	91%
System B	62%	71%
System C	74%	95%
System D	69%	85%
System E	59%	66%
System F	73%	93%
System G	49%	48%
System H	40%	29%

max	74%	95%
min	30%	10%

Table 9.1.1: Conversion of energy savings to scores used in the overall ranking

"Without Ceiling fan" case

A	Convent. evapor. cooling
B	Advncd. .evapor. Cooling M-cycle
C	Adsorption chiller
D	Absorption chiller
E	Magnetic chiller
F	DEVAP system
G	LDDX system
H	Convectional Chiller

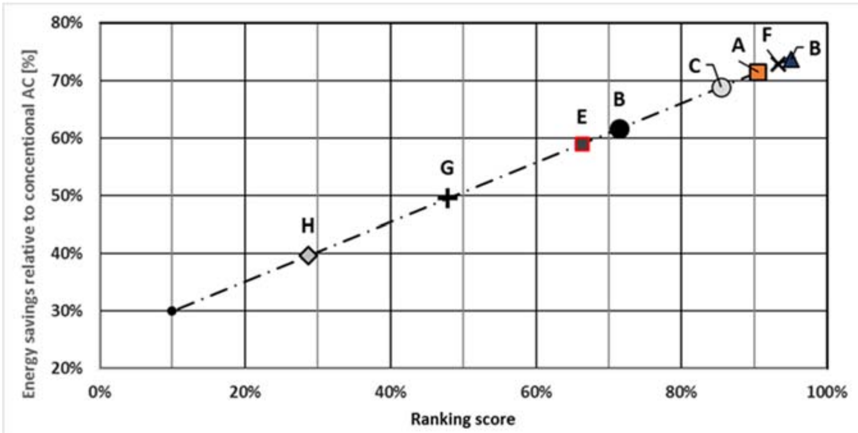


Figure 9.1.1: Illustration of converting energy savings to scores used in the overall ranking

"Without Ceiling fan" case

With ceiling fans

candidate system No.	Energy savings	Corresponding score
System A	77%	101%
System B	72%	91%
System C	78%	104%
System D	76%	98%
System E	70%	88%
System F	78%	103%
System G	65%	78%
System H	59%	67%
max **	74%	95%
min **	30%	10%

note**: max and min values are taken from the case "without ceiling fans"

Table 9.1.2: Conversion of energy savings to scores used in the overall ranking

"**With** Ceiling fan" case

A	Convent. evapor. cooling
B	Advncd. .evapor. Cooling M-cycle
C	Adsorption chiller
D	Absorption chiller
E	Magnetic chiller
F	DEVAP system
G	LDDX system
H	Convectional Chiller

<ins>Benefit to Risk Ratio (BRR):</ins> The values for BRR for the eight candidate systems are converted to scoring values. Table 9.1.3 presents the conversion of the BRR values. Figure 9.1.3 illustrates the distribution of the scores presented in Table 9.1.3 along a line connecting the minimum and maximum scores.

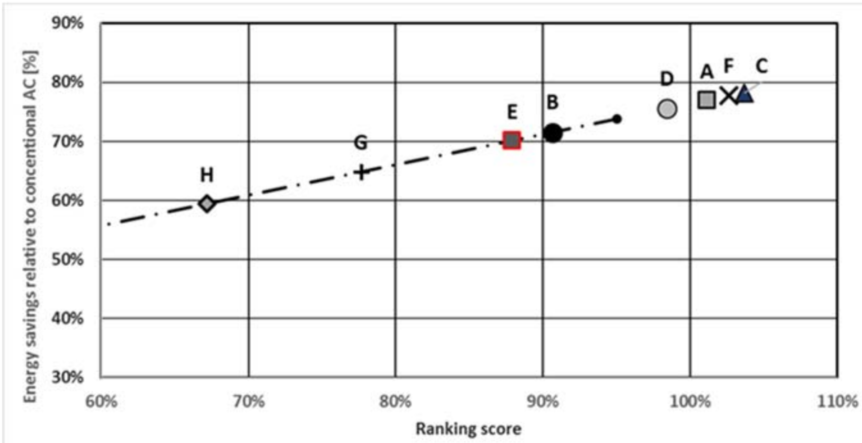


Figure 9.1.2: Illustration of converting energy savings to scores used in the overall ranking

"**With** ceiling fan" case

Table 9.1.3: Conversion of BRR values to scores used in the overall ranking

candidate system No.	BRR values	Corresponding score
System A	158%	95%
System B	79%	10%
System C	157%	95%
System D	119%	53%
System E	87%	19%
System F	83%	15%
System G	111%	45%
System H	157%	94%

max	158%	95%
min	79%	10%

A	Convent. evapor. cooling
B	Advncd. .evapor. Cooling M-cycle
C	Adsorption chiller
D	Absorption chiller
E	Magnetic chiller
F	DEVAP system
G	LDDX system
H	Convectional Chiller

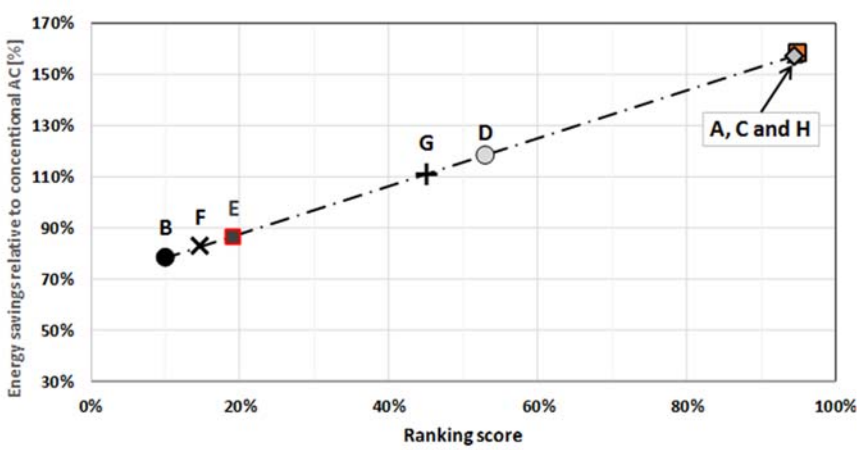


Figure 9.1.3: Illustration of converting BRR values to scores used in the overall ranking

9.2 Determining the Overall Ranking of Candidate Systems

The 0verall ranking is determined in a two-step procedure, as pointed out above. In this particular ranking continuous and discrete ranks are both used. A 1 to 3 range of discrete ranks is used and percentages are assigned to the discrete ranks as follows; Discrete ranks 1, 2 and 3 are assigned the values 10%, 50% and 95%. Different from the benefit versus risk ranking, where positive and negative criteria were used, only positive criteria statements are used in the overall ranking. Therefore, it is self-evident that the criteria ranking by discrete values of 1 and 3 indicate low and high applicability, respectively. The detailed calculations of the overall ranking of the eight candidate systems is presented in Table 9.2.1.

	Overall weights			A			B		
	Sensible cooling technology >>>>			Conventional Evaporative cooling			Advanced Evaporative cooling (M-cycle)		
Ranking	1st level	2nd level	overall	input	resulting	overall	input	resulting	overall
Financial performance	35%	100%							
Predicted energy savings (converted values)		75%	26%	*71%*	*91%*	*24%*	*62%*	*71%*	*19%*
Anticipated first costs		25%	9%	2	50%	4%	2	50%	4%
Benefit to risk ratio (conveted values)	15%		15%	*158%*	*95%*	*14%*	*158%*	*95%*	*14%*
Technology	5%	100%							
Complexity of desiccant technology		40%	2%	2	50%	1%	2	50%	1%
Complexity of sensible cooling		30%	2%	2	50%	1%	2	50%	1%
Ease of efficient space conditioning integration		30%	2%	1	10%	0%	3	95%	1%
Operation of systems	25%	100%							
Complexity of desiccant operation		15%	4%	1	10%	0%	1	10%	0%
Complexity of sensible cooling operation		15%	4%	2	50%	2%	2	50%	2%
Ease of desiccant and sensible cooling integration		70%	18%	1	10%	2%	1	10%	2%
Market status	10%								
Market maturity		10%	10%	3	95%	10%	2	50%	5%
Impact mitigation	10%								
Use of low temperature energy sources		10%	10%	3	95%	10%	3	95%	10%
sum >>	100%	sum >>	100%	ranking score >>>		67%	ranking score >>>		59%

Table 9.2.1: Calculations of the overall ranking , New Installation (Part1); the full size table is presented in Table 9.2.2

	C			D			E		
	Adsoprtion chiller			Absoprtion chiller			Magnetic chiller		
Ranking	input	resulting	overall	input	resulting	overall	input	resulting	overall
Financial performance									
Predicted energy savings (converted values)	*74%*	*95%*	*25%*	*69%*	*85%*	*22%*	*59%*	*66%*	*17%*
Anticipated first costs	1	10%	1%	1	10%	1%	2	50%	4%
Benefit to risk ratio (conveted values)	*157%*	*95%*	*14%*	*119%*	*53%*	*8%*	*87%*	*19%*	*3%*
Technology									
Complexity of desiccant technology	2	50%	1%	2	50%	1%	2	50%	1%
Complexity of sensible cooling	2	50%	1%	2	50%	1%	2	50%	1%
Ease of efficient space conditioning integration	3	95%	1%	3	95%	1%	3	95%	1%
Operation of systems									
Complexity of desiccant operation	1	10%	0%	1	10%	0%	1	10%	0%
Complexity of sensible cooling operation	2	50%	2%	1	10%	0%	2	50%	2%
Ease of desiccant and sensible cooling integration	2	50%	9%	2	50%	9%	2	50%	9%
Market status									
Market maturity	3	95%	10%	3	95%	10%	1	10%	1%
Impact mitigation									
Use of low temperature energy sources	3	95%	10%	3	95%	10%	1	10%	1%
		ranking score >>>	73%		ranking score >>>	63%		ranking score >>>	41%

Table 9.2.1: Calculations of the overall ranking New Installation (Part2); the full size table is presented in Table 9.2.2

	F			G			H		
	DEVAP system			LDDX system			Conventional chiller		
Ranking	input	resulting	overall	input	resulting	overall	input	resulting	overall
Financial performance									
Predicted energy savings (converted values)	*73%*	*93%*	*24%*	*49%*	*48%*	*13%*	*40%*	*29%*	*8%*
Anticipated first costs	1	10%	1%	3	95%	8%	3	95%	8%
Benefit to risk ratio (conveted values)	*83%*	*15%*	*2%*	*111%*	*45%*	*7%*	*157%*	*94%*	*14%*
Technology									
Complexity of desiccant technology	2	50%	1%	3	95%	2%	3	95%	2%
Complexity of sensible cooling	1	10%	0%	3	95%	1%	3	95%	1%
Ease of efficient space conditioning integration	1	10%	0%	1	10%	0%	3	95%	1%
Operation of systems									
Complexity of desiccant operation	1	10%	0%	3	95%	4%	3	95%	4%
Complexity of sensible cooling operation	2	50%	2%	3	95%	4%	3	95%	4%
Ease of desiccant and sensible cooling integration	3	95%	17%	3	95%	17%	1	10%	2%
Market status									
Market maturity	1	10%	1%	3	95%	10%	3	95%	10%
Impact mitigation									
Use of low temperature energy sources	3	95%	10%	2	50%	5%	1	10%	1%
		ranking score >>>	58%		ranking score >>>	69%		ranking score >>>	54%

Table 9.2.1: Calculations of the overall ranking New Installation (Part3); the full size table is presented in Table 9.2.2

	Overall weights			A			B			C			D		
	Sensible cooling technology >>>>			Conventional Evaporative cooling			Advanced Evaporative cooling (M-cycle)			Adsoprtion chiller			Absoprtion chiller		
Ranking	1st level	2nd level	overall	input	resulting	overall	input	resulting	overall	input	resulting	overall	input	resulting	overall
Financial performance	35%	100%													
Predicted energy savings (converted values)		75%	26%	*71%*	*91%*	*24%*	*62%*	*71%*	*19%*	*74%*	*95%*	*25%*	*69%*	*85%*	*22%*
Anticipated first costs		25%	9%	2	50%	4%	2	50%	4%	1	10%	1%	1	10%	1%
Benefit to risk ratio (conveted values)	15%		15%	*158%*	*95%*	*14%*	*158%*	*95%*	*14%*	*157%*	*95%*	*14%*	*119%*	*53%*	*8%*
Technology	5%	100%													
Complexity of desiccant technology		40%	2%	2	50%	1%	2	50%	1%	2	50%	1%	2	50%	1%
Complexity of sensible cooling		30%	2%	2	50%	1%	2	50%	1%	2	50%	1%	2	50%	1%
Ease of efficient space conditioning integration		30%	2%	1	10%	0%	3	95%	1%	3	95%	1%	3	95%	1%
Operation of systems	25%	100%													
Complexity of desiccant operation		15%	4%	1	10%	0%	1	10%	0%	1	10%	0%	1	10%	0%
Complexity of sensible cooling operation		15%	4%	2	50%	2%	2	50%	2%	2	50%	2%	1	10%	0%
Ease of desiccant and sensible cooling integration		70%	18%	1	10%	2%	1	10%	2%	2	50%	9%	2	50%	9%
Market status	10%														
Market maturity		10%	10%	3	95%	10%	2	50%	5%	3	95%	10%	3	95%	10%
Impact mitigation	10%														
Use of low temperature energy sources		10%	10%	3	95%	10%	3	95%	10%	3	95%	10%	3	95%	10%
sum >>	100%	sum >>	100%	ranking score >>>		67%	ranking score >>>		59%	ranking score >>>		73%	ranking score >>>		63%

Candidate system which use only electrcity for sensible cooling

Candidate system which use thermal energy for sensible cooling

Table 9.2.2: Calculations of the overall ranking, full size ; New Installation (Part 1)

	Overall weights			E			F			G			H		
	Sensible cooling technology >>>>			Magnetic chiller			DEVAP system			LDDX system			Conventional chiller		
Ranking	1st level	2nd level	overall	input	resulting	overall	input	resulting	overall	input	resulting	overall	input	resulting	overall
Financial performance	35%	100%													
Predicted energy savings (converted values)		75%	26%	*59%*	*66%*	*17%*	*73%*	*93%*	*24%*	*49%*	*48%*	*13%*	*40%*	*29%*	*8%*
Anticipated first costs		25%	9%	2	50%	4%	1	10%	1%	3	95%	8%	3	95%	8%
Benefit to risk ratio (conveted values)	15%		15%	*87%*	*19%*	*3%*	*83%*	*15%*	*2%*	*111%*	*45%*	*7%*	*157%*	*94%*	*14%*
Technology	5%	100%													
Complexity of desiccant technology		40%	2%	2	50%	1%	2	50%	1%	3	95%	2%	3	95%	2%
Complexity of sensible cooling		30%	2%	2	50%	1%	1	10%	0%	3	95%	1%	3	95%	1%
Ease of efficient space conditioning integration		30%	2%	3	95%	1%	1	10%	0%	1	10%	0%	3	95%	1%
Operation of systems	25%	100%													
Complexity of desiccant operation		15%	4%	1	10%	0%	1	10%	0%	3	95%	4%	3	95%	4%
Complexity of sensible cooling operation		15%	4%	2	50%	2%	2	50%	2%	3	95%	4%	3	95%	4%
Ease of desiccant and sensible cooling integration		70%	18%	2	50%	9%	3	95%	17%	3	95%	17%	1	10%	2%
Market status	10%														
Market maturity		10%	10%	1	10%	1%	1	10%	1%	3	95%	10%	3	95%	10%
Impact mitigation	10%														
Use of low temperature energy sources		10%	10%	1	10%	1%	3	95%	10%	2	50%	5%	1	10%	1%
	100%	sum >>	100%	ranking score >>>		41%	ranking score >>>		58%	ranking score >>>		69%	ranking score >>>		54%

Candidate system which use only electrcity for sensible cooling

Candidate system which use thermal energy for sensible cooling

Table 9.2.2: Calculations of the overall ranking, full size; New Installation (Part 2)

9.3 Results of Overall Ranking – New Installations in Buildings

The overall ranking results of the eight candidate systems are presented in Table 9.3.1 and Figure 9.3.1. The overall ranks reflect energy savings for the case "Without ceiling fans", which reflects the more conservative energy savings. The ranking also reflects the situations where the desiccant cooling systems are installed regardless of the existing building infrastructure; e.g. for New installations.

Table 9.3.1: Results of overall Ranking

For new installations in buildings

System	Description	Overall ranking score	Overall Rank
A	Conventional Evaporative cooling	67%	3
B	Advanced Evaporative cooling (M-cycle)	59%	5
C	Adsoprtion chiller	73%	1
D	Absoprtion chiller	63%	4
E	Magnetic chiller	41%	8
F	DEVAP system	58%	6
G	LDDX system	69%	2
H	Conventional chiller	54%	7

Candidate systems which use only electricity for sensible cooling

Candidate systems which use thermal energy for sensible cooling

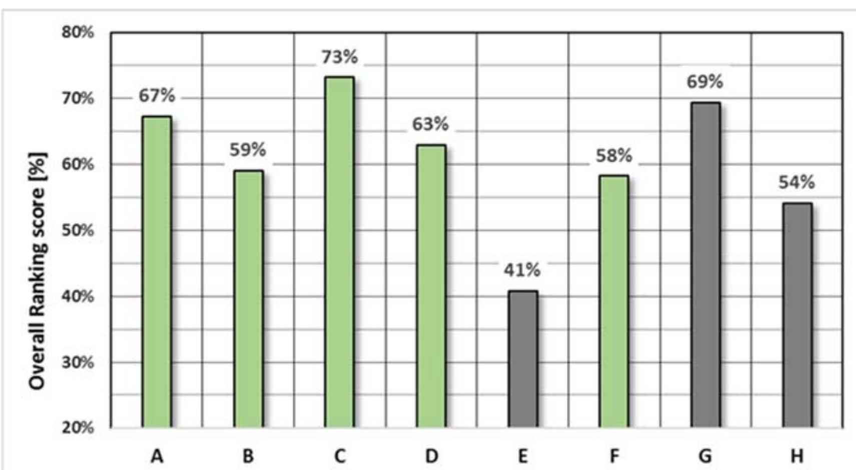


Figure 9.3.1: Results of overall Ranking scores

For new installations in buildings

Candidate systems which use only electricity for sensible cooling

Candidate systems which use thermal energy for sensible cooling

Table 9.3.2 shows the final ranking of candidate systems for NEW Installations in right order. Figure 9.3.2 shows the percentage “lead” of candidate system over the system with the lowest score.

Table 9.3.1: Results of overall Ranking; arranged in right order

For new installations in buildings

System	Description	Overall ranking score	Overall Rank	Lead over lowest rank
C	Adsorption chiller	73%	1	32%
G	LDDX system	69%	2	29%
A	Conventional Evaporative cooling	67%	3	26%
D	Absorption chiller	63%	4	22%
B	Advanced Evaporative cooling (M-cycle)	59%	5	18%
F	DEVAP system	58%	6	17%
H	Conventional chiller	54%	7	13%
E	Magnetic chiller	41%	8	0%

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

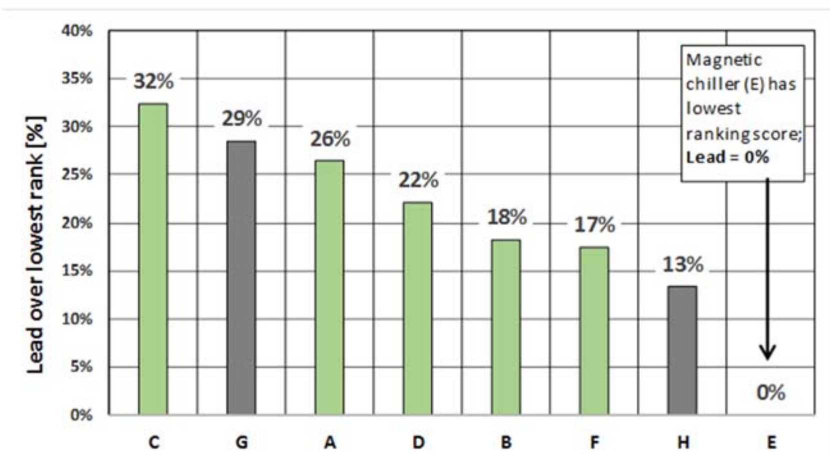


Figure 9.3.2: Results of overall Ranking scores, arranged in right order

For new installations in buildings

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

9.4 Results of Overall Ranking – Building Retrofits

The ability to add desiccant cooling systems to existing buildings present is an important consideration to facilitate implementation to more energy efficient AC based on separate latent and sensible heat removal. The ranking procedure presented in Table 9.2.1 was augmented and decision criteria that quantify the effects of retrofitting buildings were added.

The overall ranking results <u>for retrofitting</u> of the eight candidate systems are presented in Table 9.4.1 and Figure 9.4.1.

System	Description	Overall score	Overall Rank
A	Conventional Evaporative cooling	47%	3
B	Advanced Evaporative cooling (M-cycle)	42%	5
C	Adsoprtion chiller	50%	2
D	Absoprtion chiller	41%	6
E	Magnetic chiller	40%	7
F	DEVAP system	33%	8
G	LDDX system	75%	1
H	Conventional chiller	46%	4

Table 9.4.1: Results of overall Ranking scores
<u>For retrofitting buildings with desiccant cooling systems</u>

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

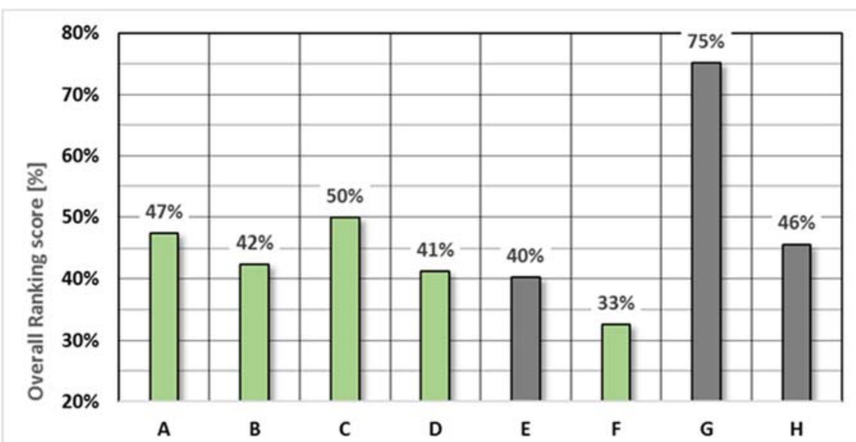


Figure 9.3.1: Results of overall Ranking scores
<u>For retrofitting buildings with desiccant cooling systems</u>

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

Table 9.4.2 shows the final ranking of candidate systems for Building retrofits in right order. Figure 9.4.2 shows the percentage "lead" of candidate system over the system with the lowest score.

Table 9.3.1: Results of overall Ranking; arranged in right order

For Building retrofits:

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

System	Description	Overall score	Overall Rank	Lead over lowest rank
G	LDDX system	75%	1	43%
C	Adsoprtion chiller	50%	2	17%
A	Conventional Evaporative cooling	47%	3	15%
H	Conventional chiller	46%	4	13%
B	Advanced Evaporative cooling (M-cycle)	42%	5	10%
D	Absoprtion chiller	41%	6	9%
E	Magnetic chiller	40%	7	8%
F	DEVAP system	33%	8	0%

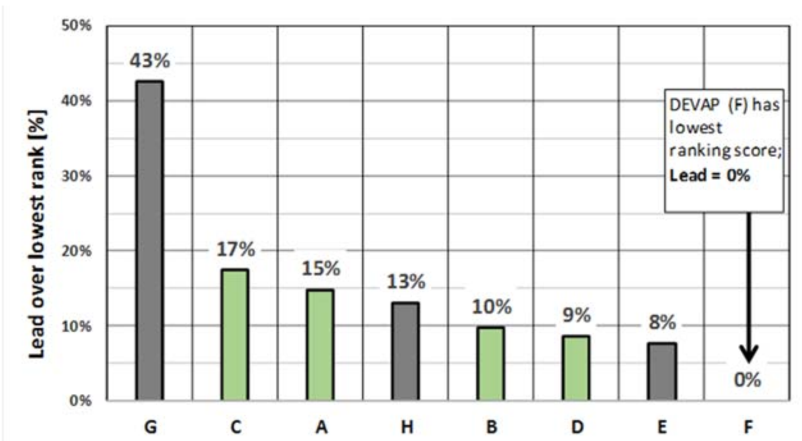


Figure 94.2: Results of overall Ranking scores, arranged in right order

For Building retrofits:

Candidate systems which use only electricity for sensible cooling

Candidate system which use thermal energy for sensible cooling

Concluding remarks for Section 9 and the overall ranking.

<u>Summarizing:</u> Ranking of the eight candidate system included a wide range of ranking criteria. For the ranking continuous and discrete ranking procedures were used. Using continuous ranking system performance metrics were converted to continuous percentage ranking scores. Using discrete ranking scores the range of 1 through 3 discrete scores were given percentage ranking scores.

Ranking was performed for the cases of new installations in buildings and building retrofits. For the two ranking cases different candidate system obtained the highest score.

REFERENCES

ASHRAE (2009) “ASHRAE Handbook—Fundamentals”, American Society of Heating, Refrigerating and Air-Conditioning Engineers

Armstrong, P.R. and Brusewitz, G. H. (1984) "Design and Testing of a Liquid Desiccant Dehumidifier", TRANSACTIONS of the American Society of Agricultural Engineers (ASAE_ —1984

Bargman, R. (2016) "Adsorption vs. Absorption Chillers: Applications and Use Overview", http://hubpages.com/education/Adsorption-vs-Absorption-Chillers-Applications-Use-Overview, accessed April 2016

Brunauer, S.,E et al (1938) "Adsorption of gases in multimolecular layers", J. Am. Chem. Soc. 60 (1938) 309–316

Collier, R.K.(1989) "Desiccant Properties and Their Effect on Cooling System Performance," ASHRAE Transactions 1989, Vol. 95, Pt.1.

Collier, R. et al (1981) "An Overview of Open-cycle Desiccant cooling systems and Materials", JOURNAL OF SOLAR ENERGY ENGINEERING

CoolTech (2016) "Magnetic Refrigeration System", http://www.cooltech-applications.com/, accessed April 2016

Daou, K., et al (2005) "Desiccant cooling air conditioning: a review", Renewable and Sustainable Energy Reviews, 10 (2006) 55–77

Das, R.S., et al (2013) "Investigations on solar energy driven liquid desiccant cooling systems for tropical climates”, Proceedings of the 50th Annual Conference, Australian Solar Energy Society, Melbourne, 2012

Dean, J. (2012) "Dew-Point Evaporative Comfort Cooling", Rocky Mountain Chapter ASHRAE Technical Conference, NREL Publication PR-7A40-55837

DOE (2104) "Energy Savings Potential and RD&D Opportunities for Non-Vapor-Compression HVAC Technologies", Prepared for: U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Building Technologies Office

Energy Center of Wisconsin (ECW) (2000) "Desiccant Dehumidification Systems", https://www.seventhwave.org/sites/default/files/430-1.pdf, accessed March 2016

EERE (2016) "Absorption Cooling Basics", (Energy.gov, Office of Energy Efficiency & Renewable Energy (EERE), http://energy.gov/eere/energybasics/articles/absorption-cooling-basics, accessed April 2016

The Engineering Tool Box (2016), various definitions and parameters, http://www.engineeringtoolbox.com, accessed February to March 2016

Fischer, J.C., J.R. Sand, B. Elkin, and K. Mescher (2002) "Active desiccant, total energy recovery hybrid system optimizes humidity control, IAQ, and energy efficiency in an existing dormitory facility". ASHRAE Transactions, 108 (Part 2), pp. 537-545.

Foo, K.Y.; Hameed, B.H. (2010). "Insights into the modeling of adsorption isotherm systems". Chemical Engineering Journal 156

GAS TECHNOLOGY INSTITUTE (2010) "Maisotsenko Cycle Enhanced Cooling Towers", http://www.gastechnology.org/Expertise/Documents/EndUse-Cooling-Towers-Brochure-Sep2010.pdf, accessed May 2016

Gidwani, B.N. (1984) "OPTIMIZATION OF CHILLED WATER SYSTEMS", Proceedings of the First

Greenchiller (2016) "green Adsorption chillers", http://www.greenchiller.biz/ , accessed April 2016

Green Cooling Initiative, GCI (2016) website http://www.green-cooling-initiative.org/, accessed March 2016

Harriman, L.G. (2002)"The Dehumidification Handbook, Second Edition", Munters Corporation, AMESBURY, MA

Harriman, L.G. et al (1997) "Dehumidification and Cooling Loads From Ventilation Air", ASHRAE Journal, November, 1997

Henderson, H. (2005) "Understanding Dehumidification Performance at Part Load in Commercial Applications", Florida Solar Energy Center, DOE/NETL Project #DE-FC26-01NT41253

InvenSor (2016) "Adsorption Chillers, Product description", http://www.invensor.com/en/products/ltc30e-hi-efficiency-chillers.htm , InvenSor GmbH, Berlin, Germany

Lowenstein, A. et al (1998) "Advanced Commercial Liquid-Desiccant Technology Development Study", National Renewable Energy Laboratory (NREL)

O'Handley, R. (2004) "Magnetic Refrigeration, A Chilling Attraction: An Evaluation of Recent Materials and Technological Developments", Department of Materials Science & Engineering, Massachusetts Institute of Technology

Patel, P. (2015) "Performance Analysis of An Indirect evaporative Cooling System using M–Cycle, IJSRD - International Journal for Scientific Research & Development| Vol. 3,, Issue 05, 2015

Ruivo, C. et al (2011) "Heat and Mass Transfer in Desiccant Wheels", in Advanced Topics in Mass Transfer, Mohamed El-Amin (editor)

San et al. (2002) "Validity of solid-side mass diffusivity in simulation of water vapor adsorbed by silica gel in packed beds", International Journal of Thermal Sciences 41 (2002).

Saudagar, R. T. (2013) "A Review of Emerging Technologies for Solar Air Conditioners", International Journal of Innovative Research in Science, Engineering and Technology, Vol.2, Issue 6, June 2013

Sahlot, M. and Riffat, S. (2016) "Desiccant cooling systems: a review", International Journal of Low-Carbon Technologies 2016, 0, 1–17

SOLAIR (2009) "Absorption Chillers", http://www.solair-project.eu/143.0.html, accessed May 2016

Staton, J. (1998) "Heat and Mass Transfer Characteristics of Desiccant Polymers", Virginia Polytechnic Institute and State University

Sultan, M. et al (2015) "An overview of solid desiccant dehumidification and air conditioning systems", Renewable and Sustainable Energy Reviews 46 (2015) 16–29

TRANE (2016) "Absorption Liquid Chillers", http://www.trane.com/commercial/north-america/us/en/products-systems/equipment/chillers/absorption-liquid-chillers.html, accessed April 2016

Wani, C. et al (2012) "A Review on Potential of Maisotsenko Cycle in Energy Saving Applications Using Evaporative Cooling", International Journal of Advance Research in Science, Engineering and Technology, Vol.01, Issue 01

Wulfinghoff, D.R. (1999) "Energy Efficiency Manual", Energy Institute Press, Symposium on Improving Building Systems in Hot and Humid Climates, August 1984

APPENDICES

There are three appendices attached to this report

APPENDIX A: TYPICAL WEATHER DATA FOR HONOLULU: Dew point, dry bulb temperature, wet bulb temperature and relative humidity

APPENDIX B: TYPICAL WEATHER DATA FOR HONOLULU: Psychrometric charts

APPENDIX C: TYPICAL WEATHER DATA FOR HONOLULU: Direct Normal Irradiance (DNI)

APPENDIX A

TYPICAL WEATHER DATA FOR HONOLULU:

Dew point, dry bulb temperature, wet bulb temperature and relative humidity

Appendix A

TYPICAL WEATHER DATA FOR HONOLULU

This appendix contains:

Time series and probability functions were determined for

1. Dew point
2. Dry bulb temperature
3. Wet bulb temperature
4. Relative humidity with the TMY3 file for Honolulu.

Hourly weather data was determined for two conditions, (1) a “full record”, for all 8760 hours per year, and a “11-h” record, for the time from 8:00 am (start) to 6:00 pm (end).

Data is provided both in SI and Imperial units.

Dew point data (Full record, 8760 hours per year)

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.0**	**4.3**
Jan	60.6	70.0	48.0	4.9
Feb	64.8	73.9	50.0	5.8
March	62.3	72.0	51.1	4.3
April	61.7	70.0	50.0	3.7
May	64.8	72.0	57.0	2.9
Jun	65.7	69.4	60.1	1.7
July	67.1	75.0	60.1	2.5
Aug	67.3	73.9	61.0	2.1
Sept	68.7	75.9	61.0	2.5
Oct	67.4	73.0	61.0	2.1
Nov	65.5	71.1	55.9	2.9
Dec	63.8	73.0	51.1	4.9

n

Figure A-1:
Dew point for Full record

Annual and monthly averages, min and maxvalues of Dew Point for the full record

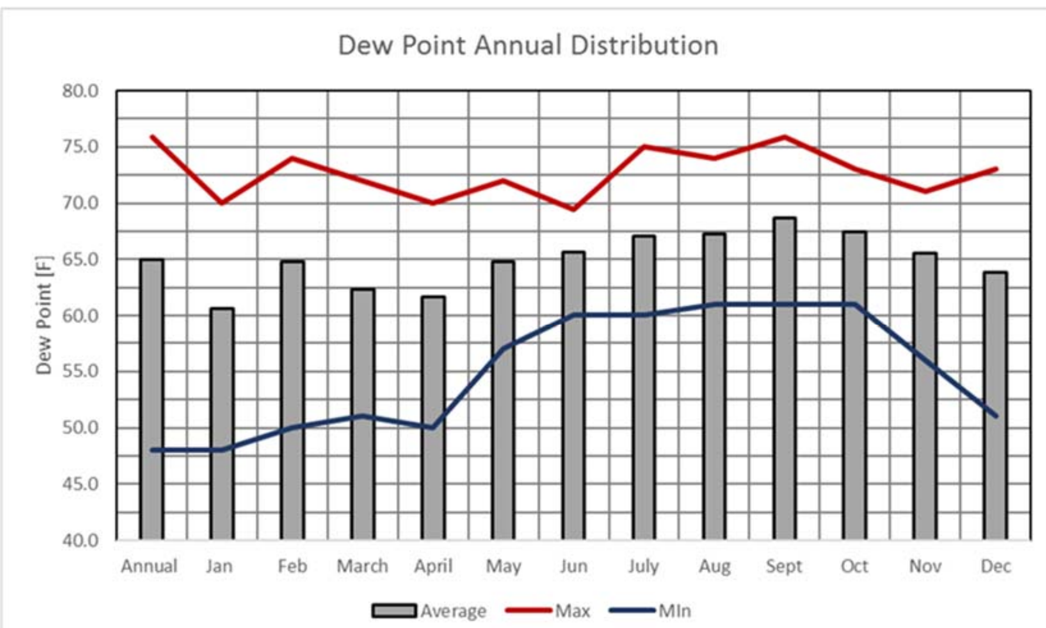


Figure A-2: Dew point for Full record; Annual and monthly averages, min and maxvalues of Dew Point for the full record

Annual
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.0	75.9	48.0	4.3

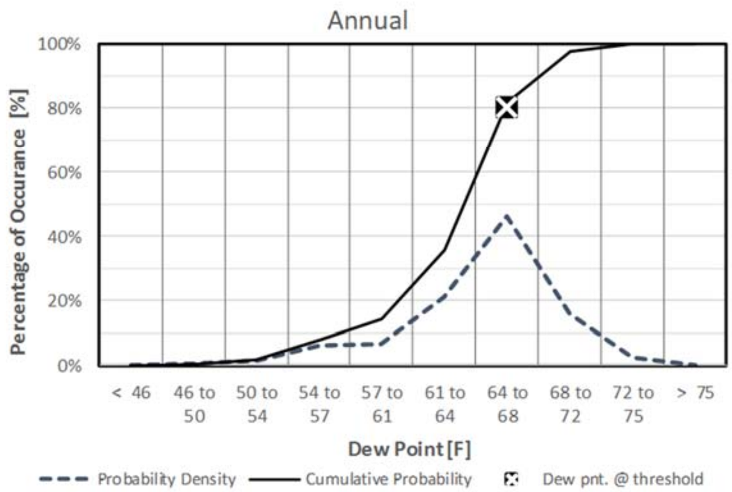


Figure A-3: Dew point for Full record; Probability of exceedance for the the entire year

January
Dew Point [F] Cum. Probability 80% = Dew Pnt. 63 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
60.6	70.0	48.0	4.9

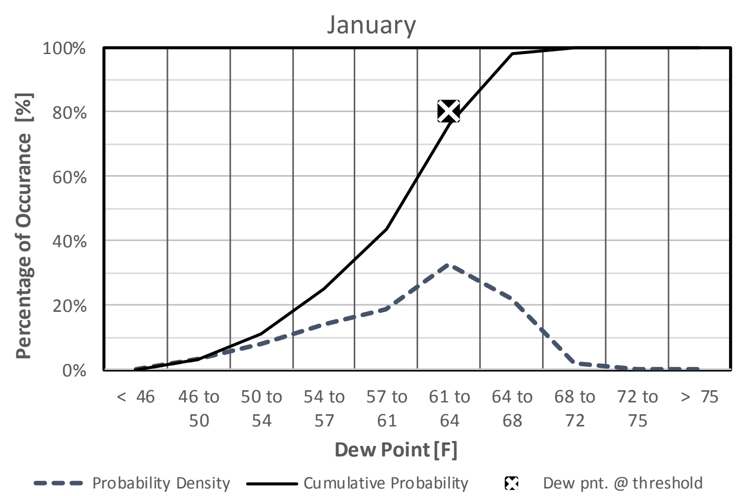


Figure A-5: Dew point for Full record; Probability of exceedance for the month of January

February
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
64.8	73.9	50.0	5.8

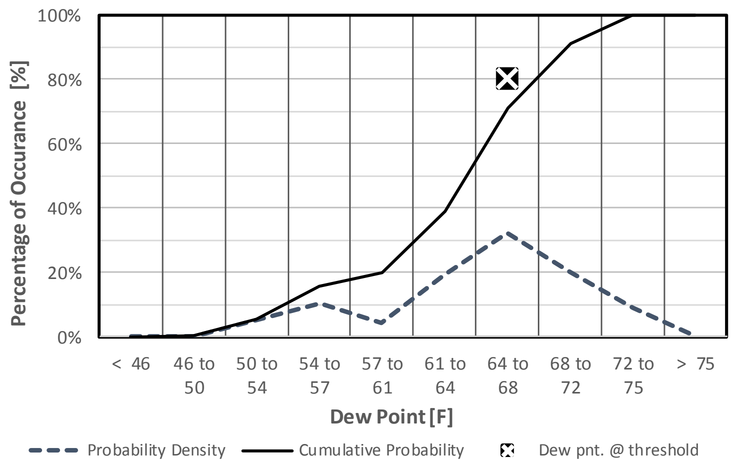


Figure A-6: Dew point for Full record; Probability of exceedance for the month of February

March
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
62.3	72.0	51.1	4.3

Figure A-7: Dew point for Full record; Probability of exceedance for the month of March

April
Dew Point [F] Cum. Probability 80% = Dew Pnt. 63 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
61.7	70.0	50.0	3.7

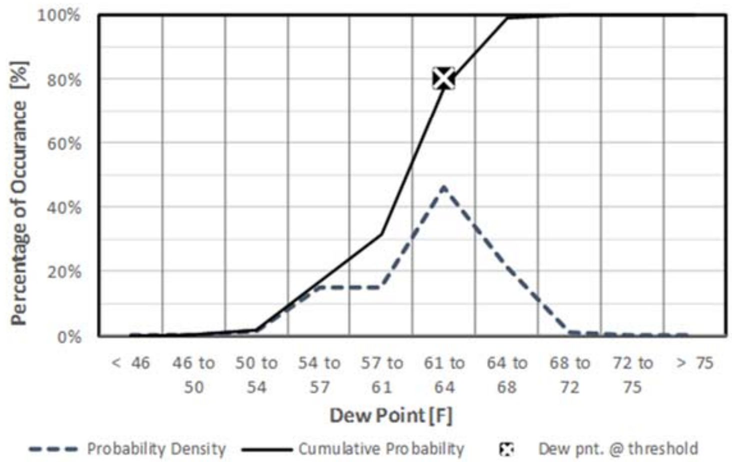


Figure A-8: Dew point for Full record; Probability of exceedance for the month of April

May
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
64.8	72.0	57.0	2.9

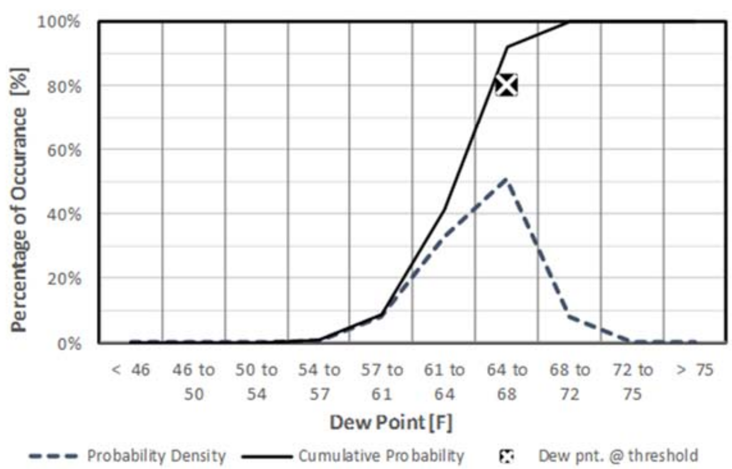


Figure A-9: Dew point for Full record; Probability of exceedance for the month of May

June
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.7	69.4	60.1	1.7

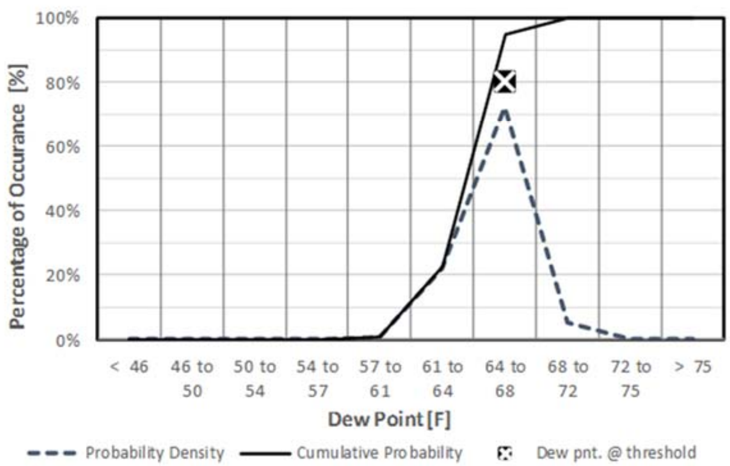


Figure A-10: Dew point for Full record; Probability of exceedance for the month of June

July
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
67.1	75.0	60.1	2.5

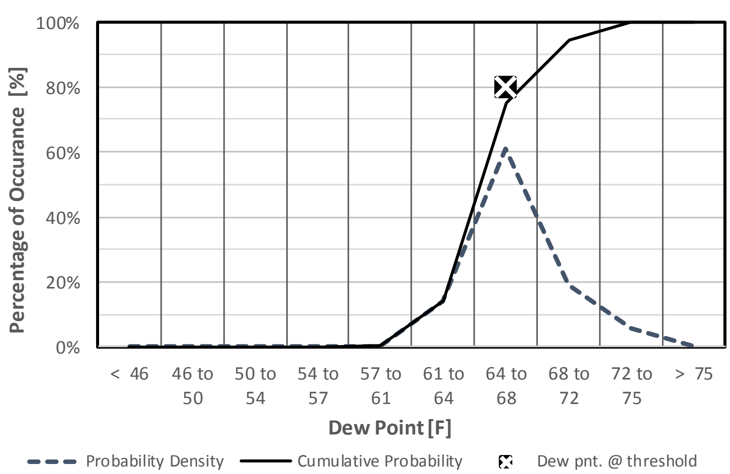


Figure A-11: Dew point for Full record; Probability of exceedance for the month of July

Figure A-12: Dew point for Full record; Probability of exceedance for the month of August

August
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	67.3	73.9	61.0	2.1

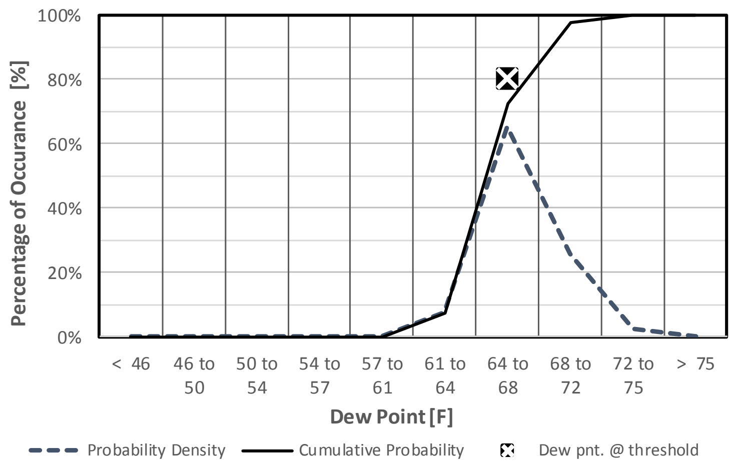
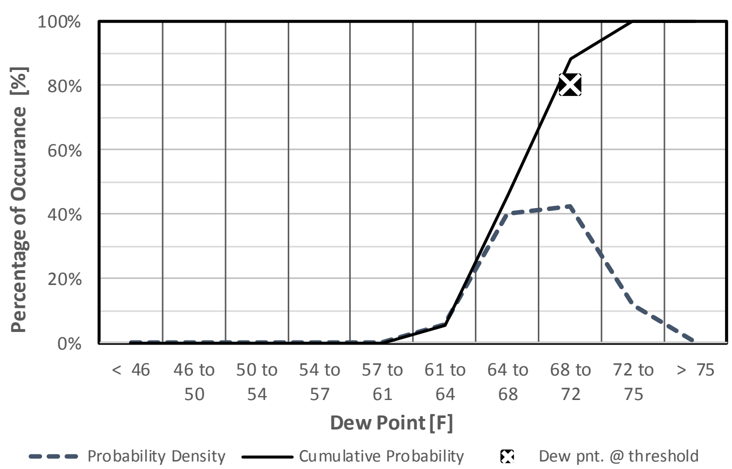


Figure A-13: Dew point for Full record; Probability of exceedance for the month of September

September
Dew Point [F] Cum. Probability 80% = Dew Pnt. 70 F
Entire annual record

	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	68.7	75.9	61.0	2.5



October
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
67.4	73.0	61.0	2.1

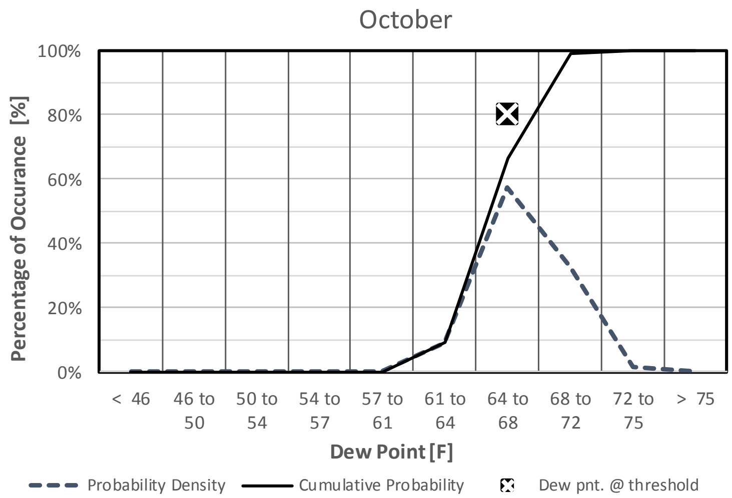


Figure A-14: Dew point for Full record; Probability of exceedance for the month of October

November
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entire annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.5	71.1	55.9	2.9

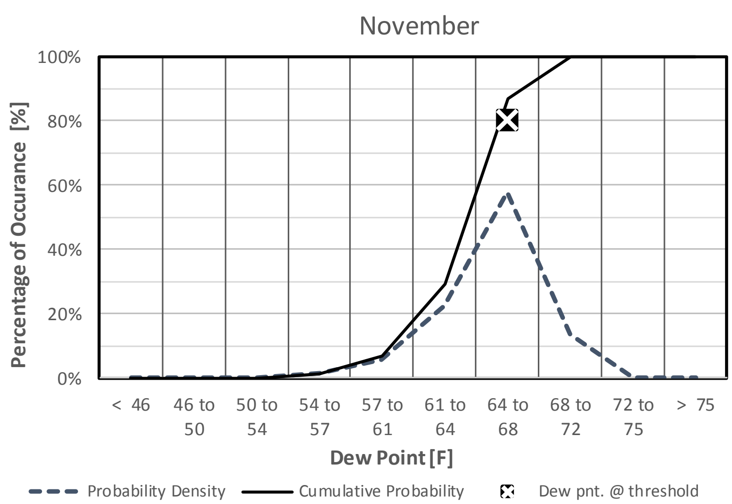


Figure A-15: Dew point for Full record; Probability of exceedance for the month of November

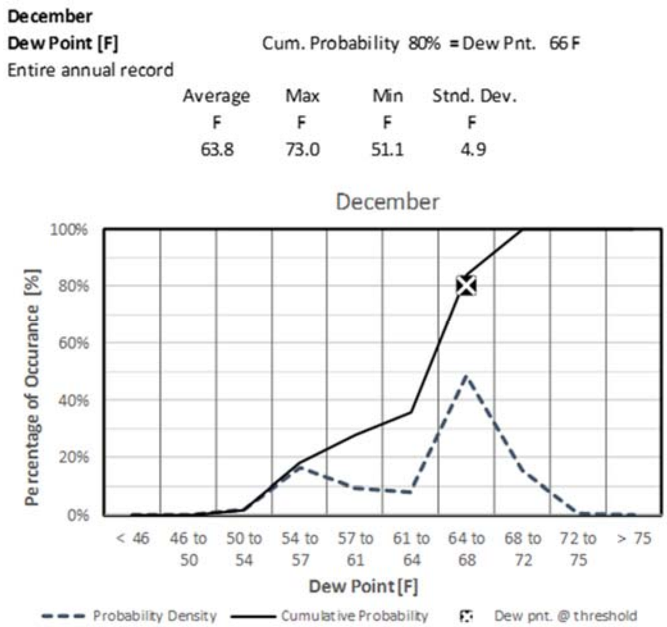


Figure A-16: Dew point for Full record; Probability of exceedance for the month of December

Dew point data (11-h record, 8:00 am to 6:00 pm)

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.9**	**4.4**
Jan	61.0	70.0	48.9	4.9
Feb	65.1	73.9	50.0	6.1
March	62.5	72.0	51.1	4.5
April	61.6	69.1	52.0	4.0
May	64.6	71.1	57.0	3.0
Jun	65.5	69.4	60.1	1.9
July	66.9	75.0	60.1	2.7
Aug	66.9	73.9	61.0	2.3
Sept	68.4	75.9	61.0	2.8
Oct	67.3	73.0	61.0	2.4
Nov	65.7	71.1	55.9	2.8
Dec	64.1	73.0	52.7	5.1

Figure A-17:
Dew point for 11-h record

Annual and monthly averages, min and maxvalues of Dew Point for the11-h record

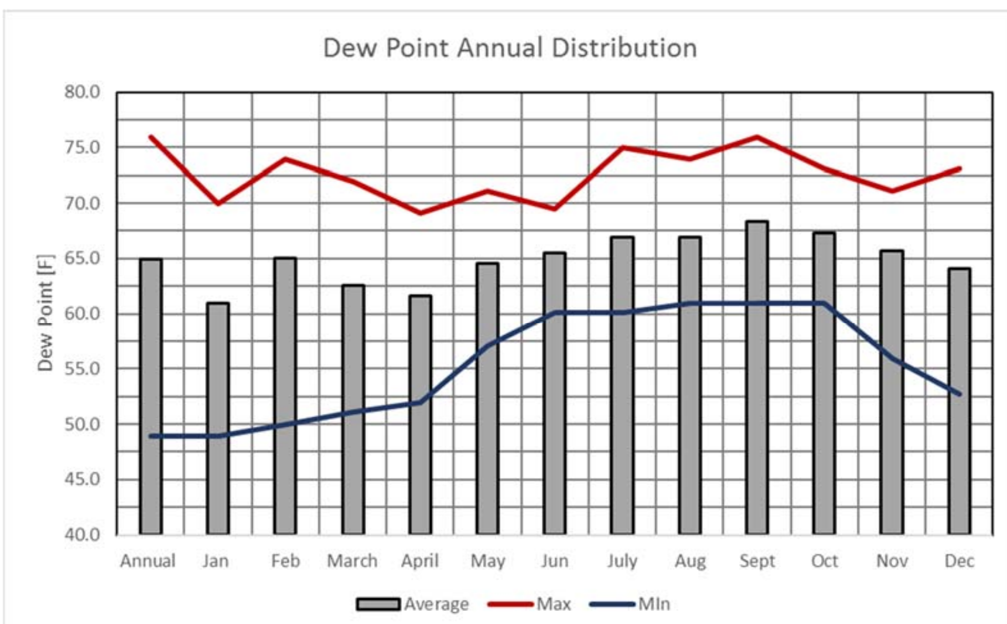


Figure A-18: Dew point for 11-6 record; Annual and monthly averages, min and maxvalues of Dew Point for the full record

Annual
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.0	75.9	48.9	4.4

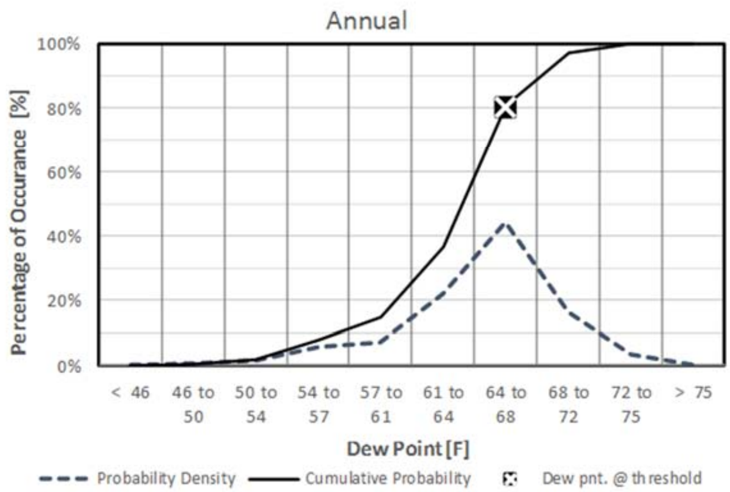


Figure A-19: Dew point for 11-6 record; Probability of exceedance for the the entire year

January
Dew Point [F] Cum. Probability 80% = Dew Pnt. 63 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
61.0	70.0	48.9	4.9

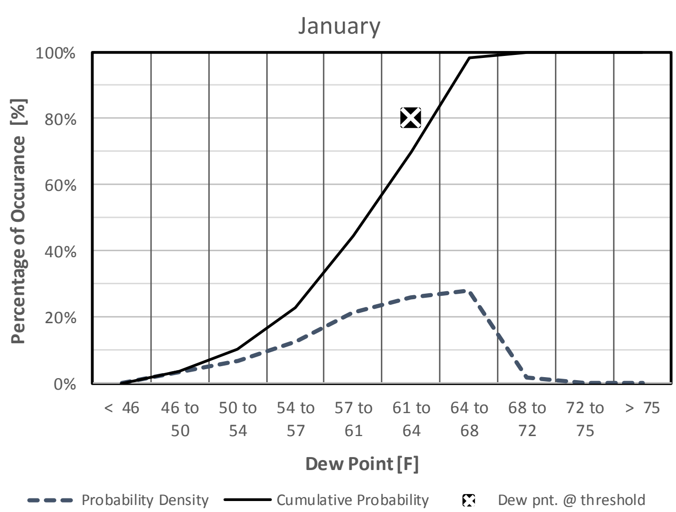


Figure A-20: Dew point for 11-6 record; Probability of exceedance for the month of January

February
Dew Point [F] Cum. Probability 80% = Dew Pnt. 70 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.1	73.9	50.0	6.1

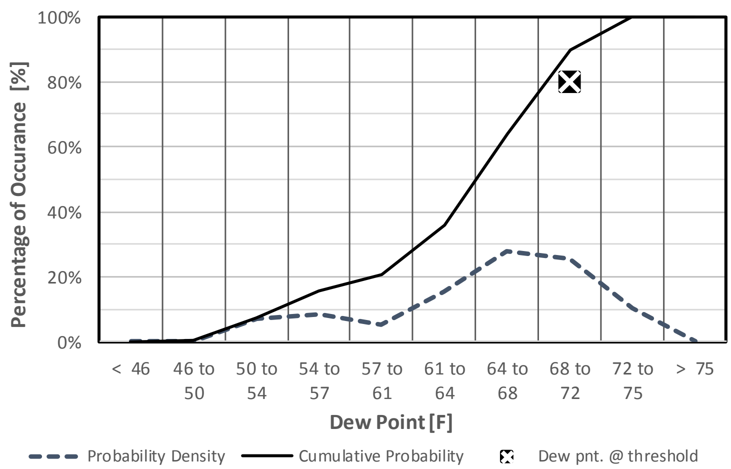


Figure A-21: Dew point for 11-6 record; Probability of exceedance for the month of February

March
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
62.5	72.0	51.1	4.5

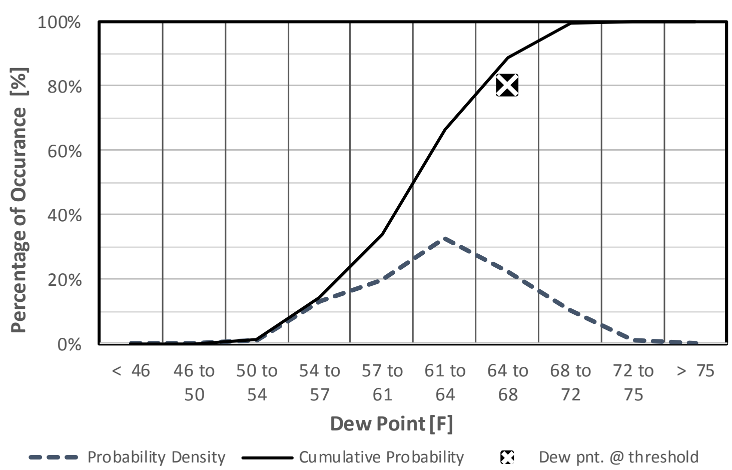


Figure A-22: Dew point for 11-6 record; Probability of exceedance for the month of March

April
Dew Point [F] Cum. Probability 80% = Dew Pnt. 63 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
61.6	69.1	52.0	4.0

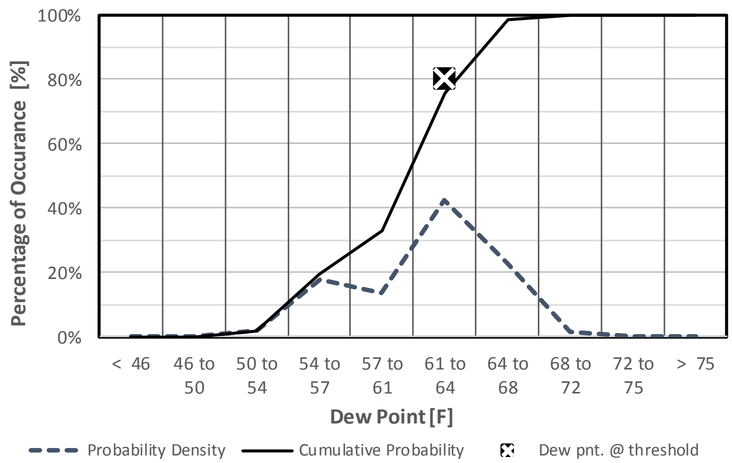


Figure A-23: Dew point for 11-6 record; Probability of exceedance for the month of April

May
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
64.6	71.1	57.0	3.0

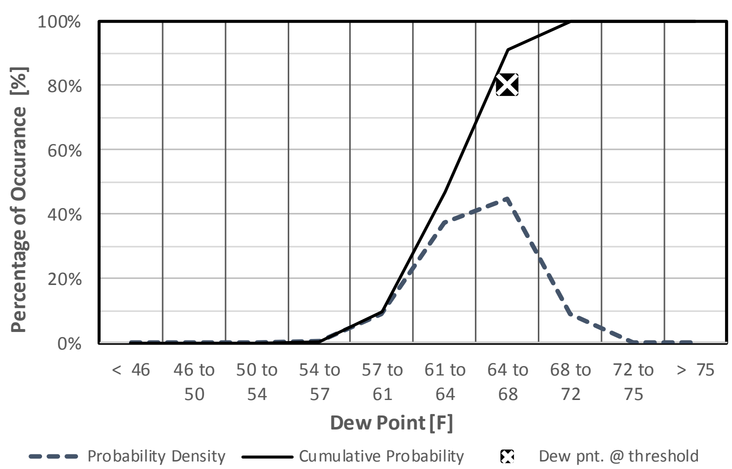


Figure A-24: Dew point for 11-6 record; Probability of exceedance for the month of May

June
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	65.5	69.4	60.1	1.9

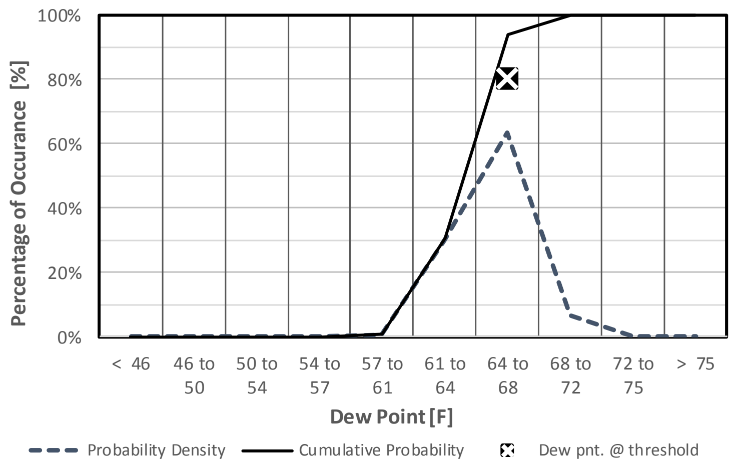


Figure A-25: Dew point for 11-6 record; Probability of exceedance for the month of June

July
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	66.9	75.0	60.1	2.7

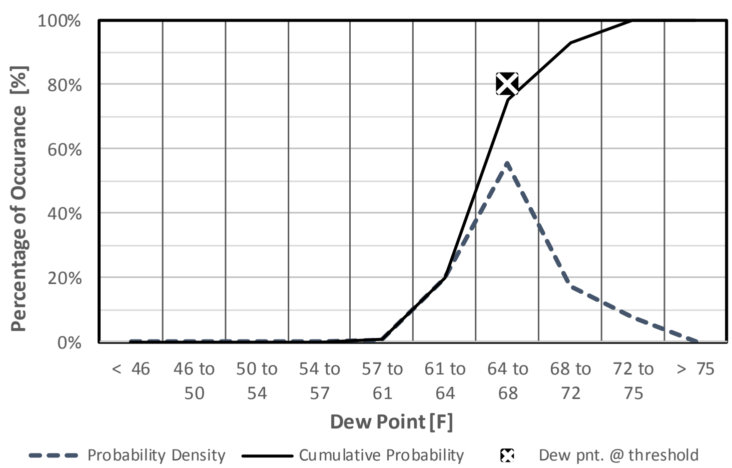


Figure A-26: Dew point for 11-6 record; Probability of exceedance for the month of July

August
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
66.9	73.9	61.0	2.3

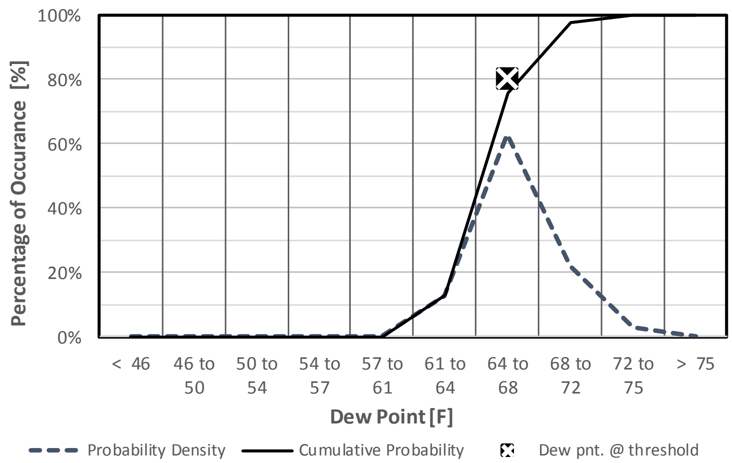


Figure A-27: Dew point for 11-6 record; Probability of exceedance for the month of August

September
Dew Point [F] Cum. Probability 80% = Dew Pnt. 70 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
68.4	75.9	61.0	2.8

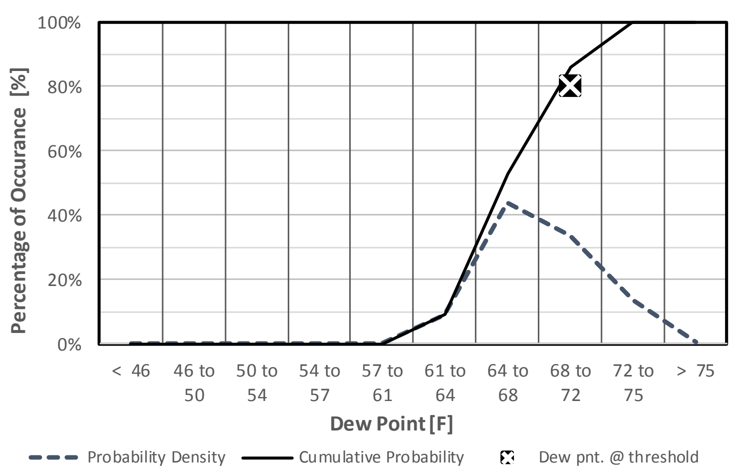


Figure A-28: Dew point for 11-6 record; Probability of exceedance for the month of September

October
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
67.3	73.0	61.0	2.4

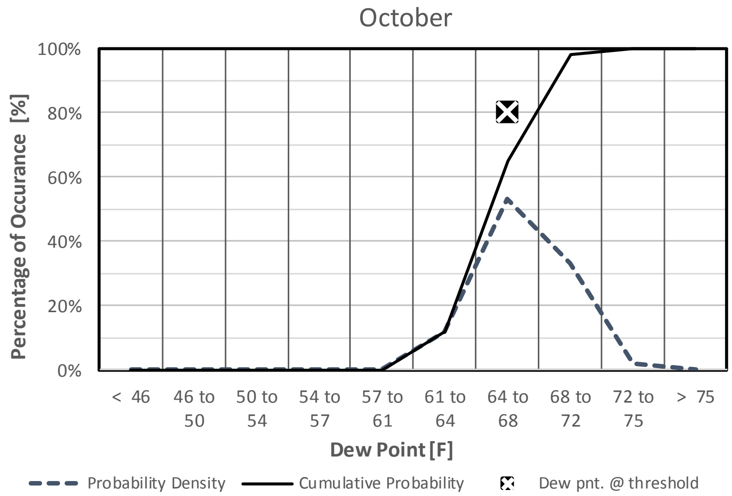


Figure A-29: Dew point for 11-6 record; Probability of exceedance for the month of October

November
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
65.7	71.1	55.9	2.8

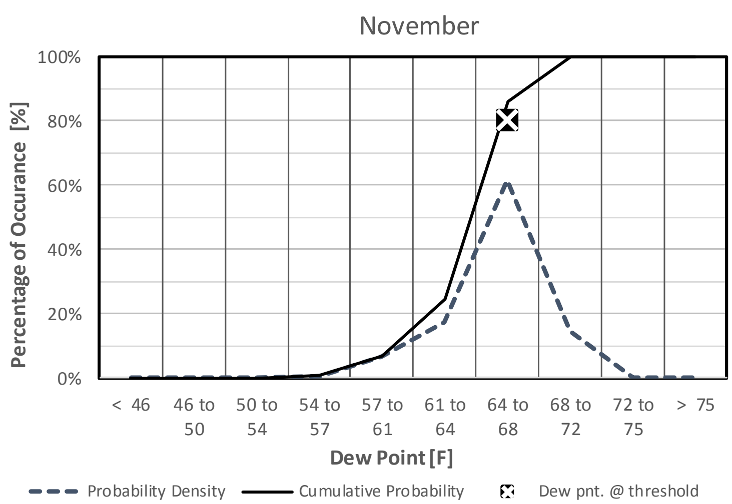


Figure A-30: Dew point for 11-6 record; Probability of exceedance for the month of November

December
Dew Point [F] Cum. Probability 80% = Dew Pnt. 66 F
Entrie annual record

Average	Max	Min	Stnd. Dev.
F	F	F	F
64.1	73.0	52.7	5.1

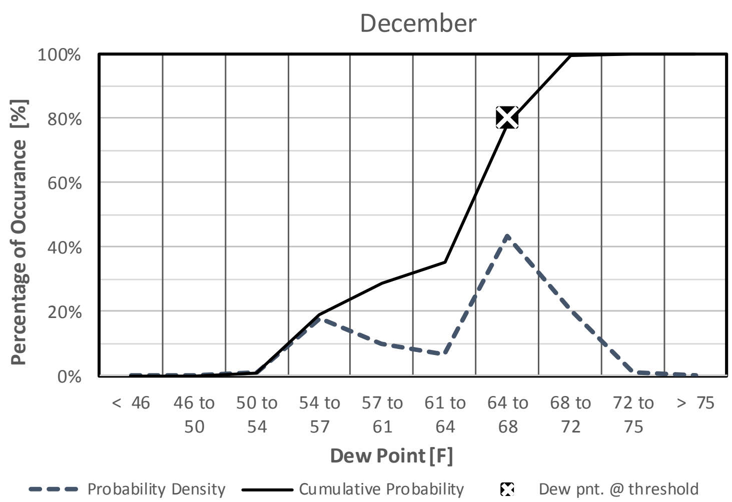


Figure A-31: Dew point for 11-6 record; Probability of exceedance for the month of December

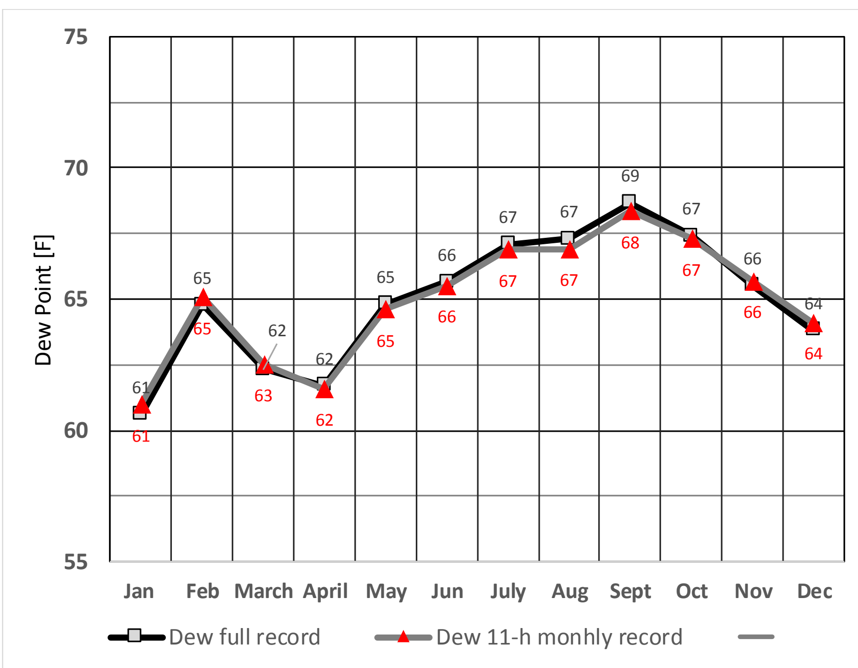


Figure A-32: Comparison full dew point record – 11-h record

Relative humidity Data (Full record, 8760 hours per year)

	average F	Max F	Min F	stnd. Dev. F
Annual	**65.0**	**75.9**	**48.9**	**4.4**
Jan	61.0	70.0	48.9	4.9
Feb	65.1	73.9	50.0	6.1
March	62.5	72.0	51.1	4.5
April	61.6	69.1	52.0	4.0
May	64.6	71.1	57.0	3.0
Jun	65.5	69.4	60.1	1.9
July	66.9	75.0	60.1	2.7
Aug	66.9	73.9	61.0	2.3
Sept	68.4	75.9	61.0	2.8
Oct	67.3	73.0	61.0	2.4
Nov	65.7	71.1	55.9	2.8
Dec	64.1	73.0	52.7	5.1

Figure A-33:
Relative humidity (RH); Full record

Annual and monthly averages, min and maxvalues of RH for the full record

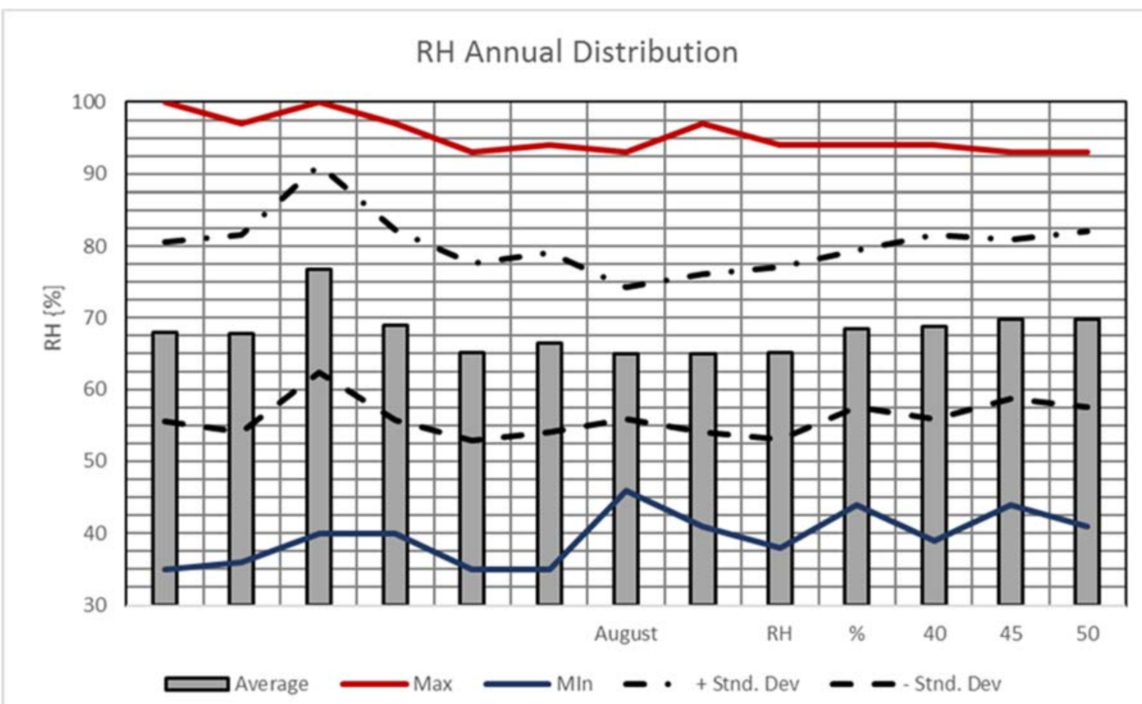


Figure A-34: Relative humidity (RH); Full record; Annual and monthly averages, min and maxvalues of RH for the full record

Cum. Probability 80% = RH 78%

Annual

RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	68.1	100.0	35.0	12.6

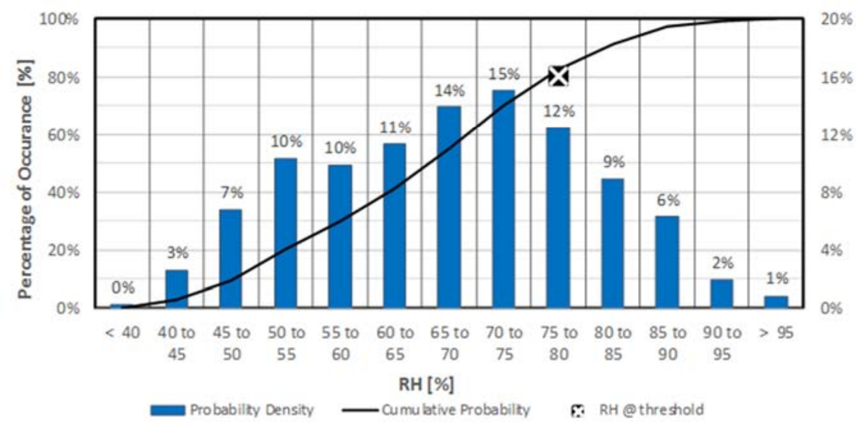


Figure A-35:
Relative humidity (RH); Full record

Probability of exceedance for the the entire year

Cum. Probability 80% = RH 83%

January

RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	67.8	97.0	36.0	13.8

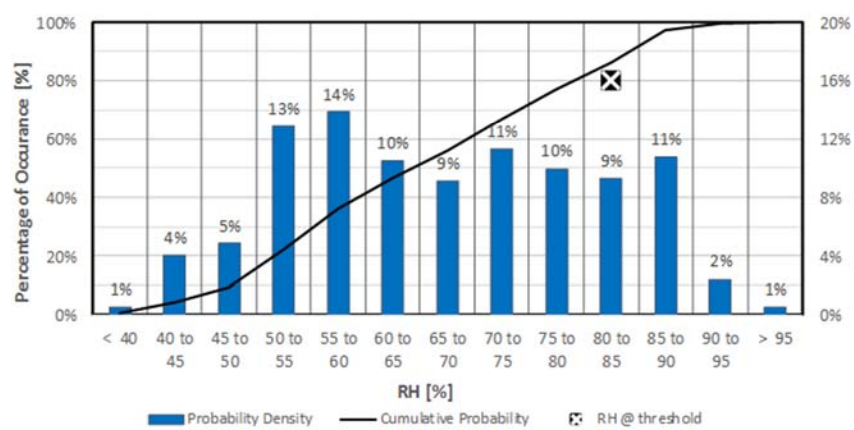


Figure A-36:
Relative humidity (RH); Full record

Probability of exceedance for the month of January

Figure A-37:
Relative humidity (RH); Full record

Probability of exceedance for the month of February

Cum. Probability 80% = RH 93 %

February				
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	76.7	100.0	40.0	14.4

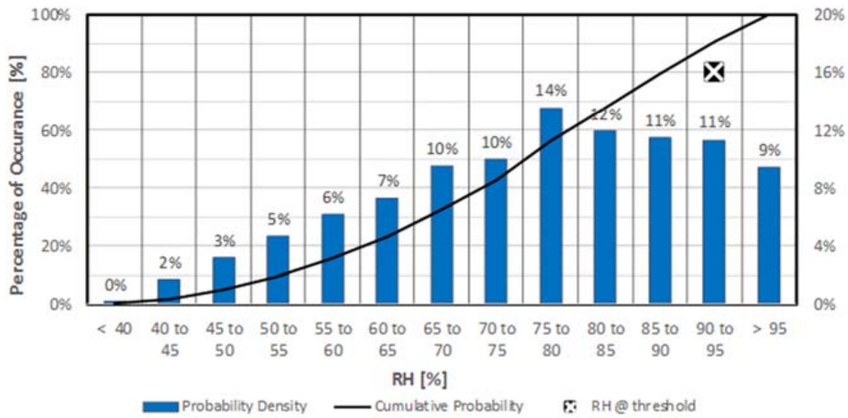


Figure A-38:
Relative humidity (RH); Full record

Probability of exceedance for the month of March

March	Cum. Probability 80% = RH 83 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	69.0	97.0	40.0	13.2

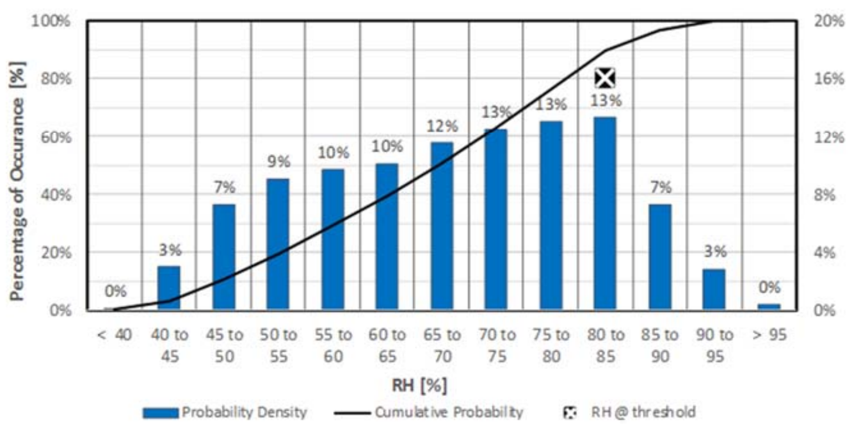


Figure A-39: Relative humidity (RH); Full record; Probability of exceedance for the month of April

April
RH [%]

Cum. Probability 80% = RH 78 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
65.2	93.0	35.0	12.3

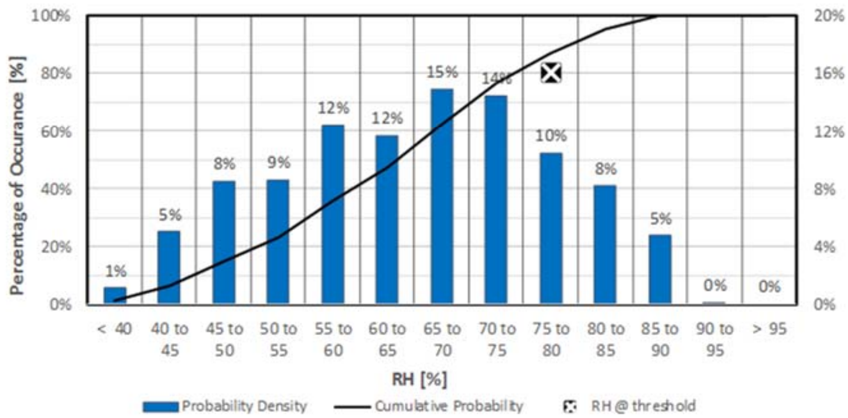


Figure A-40: Relative humidity (RH); Full record ; Probability of exceedance for the month of May

May
RH [%]

Cum. Probability 80% = RH 78 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
66.5	94.0	35.0	12.5

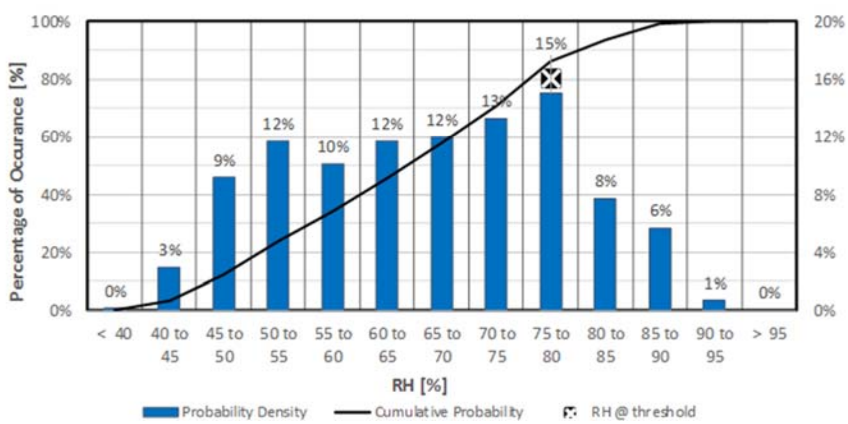


Figure A-41: Relative humidity (RH); Full record; Probability of exceedance for the month of June

June
RH [%]

Cum. Probability 80% = RH 73 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
65.0	93.0	46.0	9.2

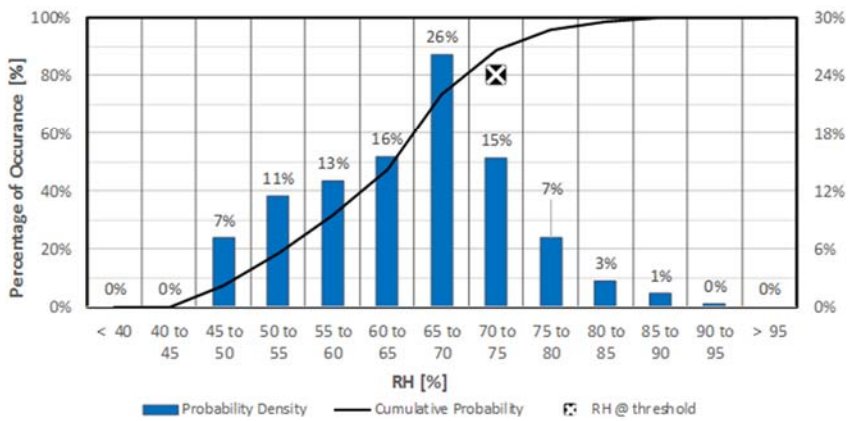


Figure A-42: Relative humidity (RH); Full record; Probability of exceedance for the month of July

July
RH [%]

Cum. Probability 80% = RH 73 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
65.0	97.0	41.0	11.0

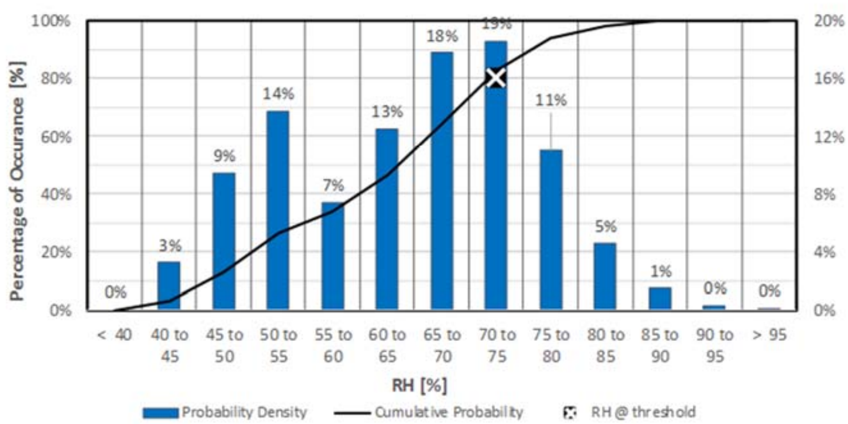


Figure A-43: Relative humidity (RH); Full record; Probability of exceedance for the month of August

August
RH [%]

Cum. Probability 80% = RH 78 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
65.1	94.0	38.0	12.0

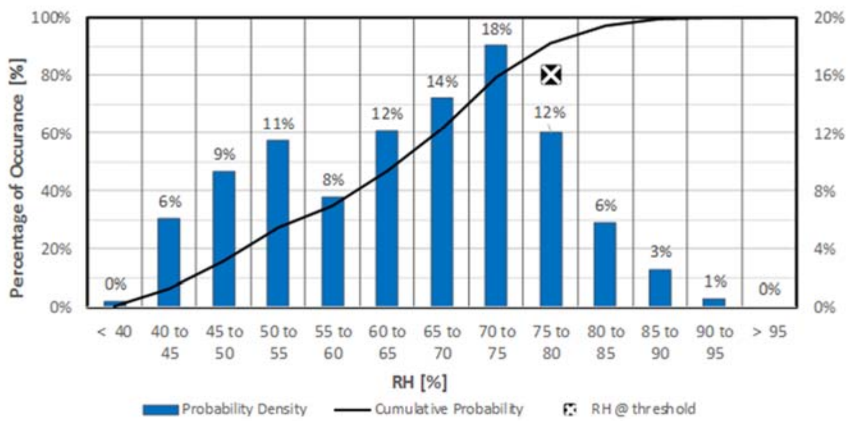
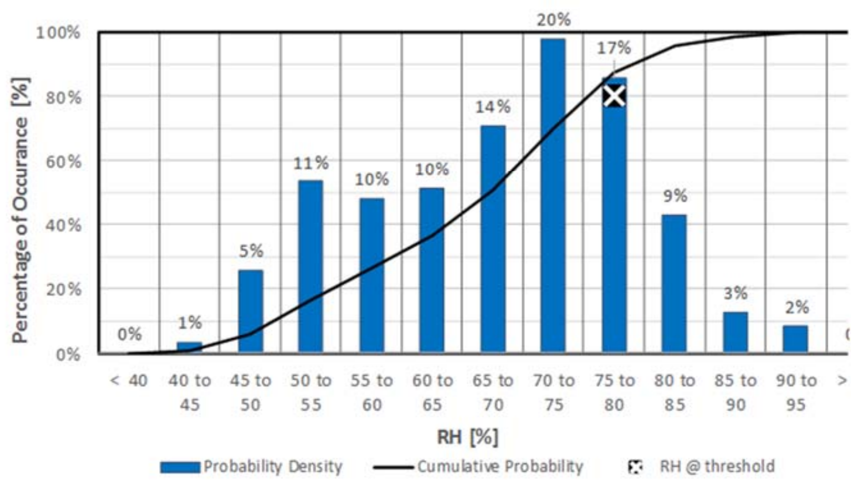


Figure A-44: Relative humidity (RH); Full record; Probability of exceedance for the month of September

September
RH [%]

Cum. Probability 80% = RH 78%

Average	Max	Min	Stnd. Dev.
%	%	%	%
68.5	94.0	44.0	11.0



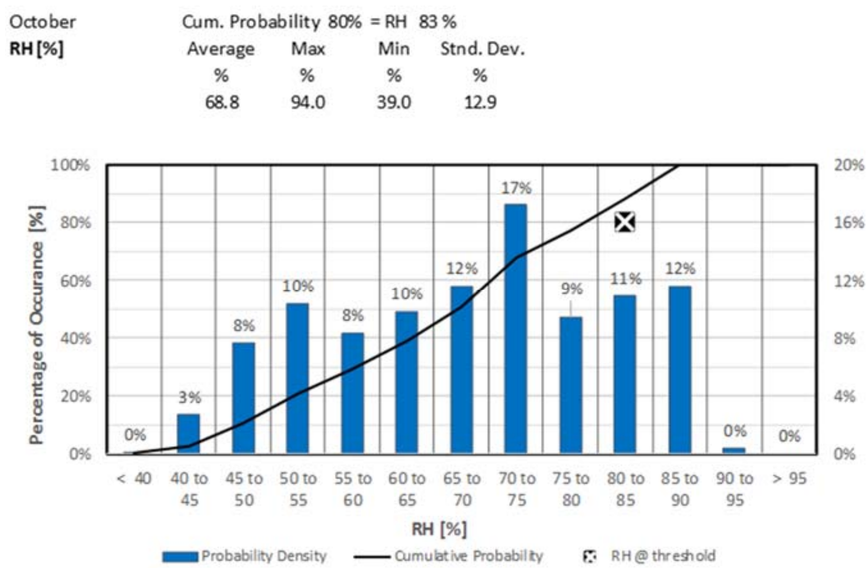


Figure A-45: Relative humidity (RH); Full record; Probability of exceedance for the month of October

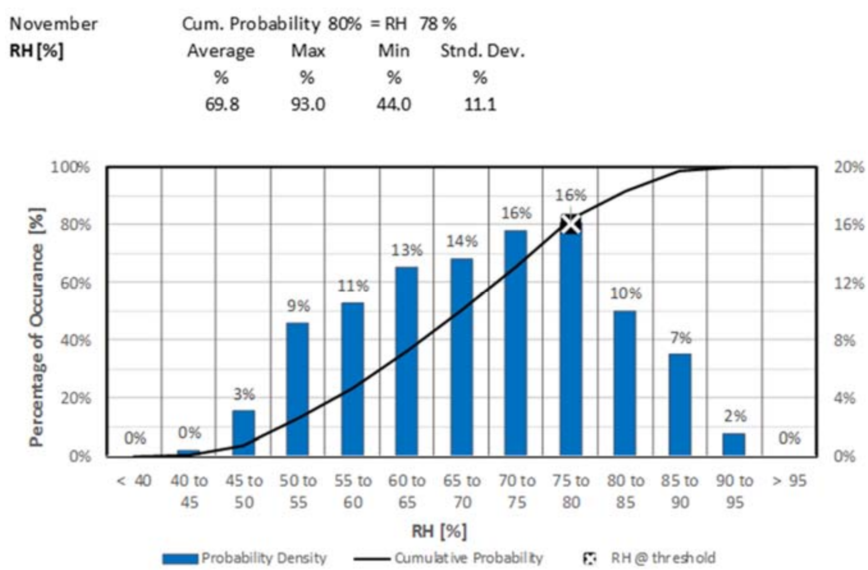
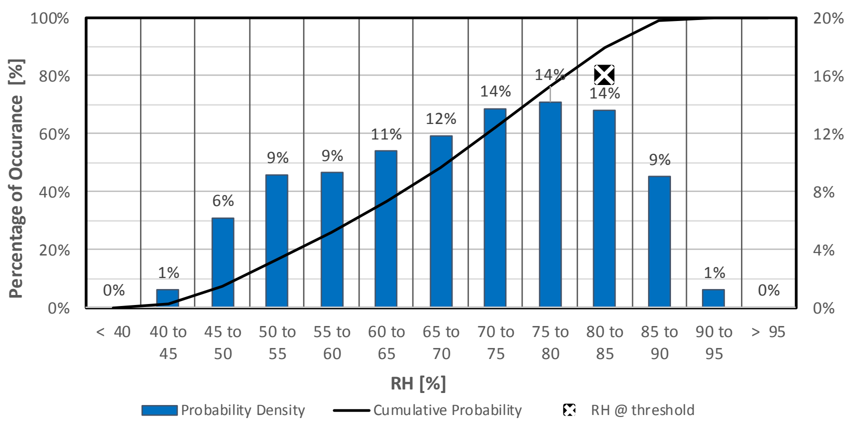


Figure A-46: Relative humidity (RH); Full record; Probability of exceedance for the month of November

December	Cum. Probability 80% = RH 83 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	69.9	93.0	41.0	12.3

Figure A-47: Relative humidity (RH); Full record; Probability of exceedance for the month of December



Relative humidity Data (11-h record, 8:00 am to 6:00 pm)

	average %	Max %	Min %	stnd. Dev. %
Annual	59.3	97.0	35.0	10.4
Jan	60.7	93.0	36.0	11.9
Feb	67.6	97.0	40.0	13.0
March	59.3	93.0	40.0	11.0
April	56.4	87.0	35.0	10.2
May	56.5	90.0	35.0	9.0
Jun	57.7	76.0	46.0	6.5
July	56.4	88.0	41.0	8.9
Aug	54.7	82.0	38.0	7.9
Sept	60.5	91.0	44.0	9.3
Oct	57.7	88.0	39.0	8.6
Nov	61.7	90.0	44.0	8.9
Dec	63.4	93.0	41.0	11.2

Figure A-48:
Relative humidity (RH); 11-h record

Annual and monthly averages, min and maxvalues of RH for the full record

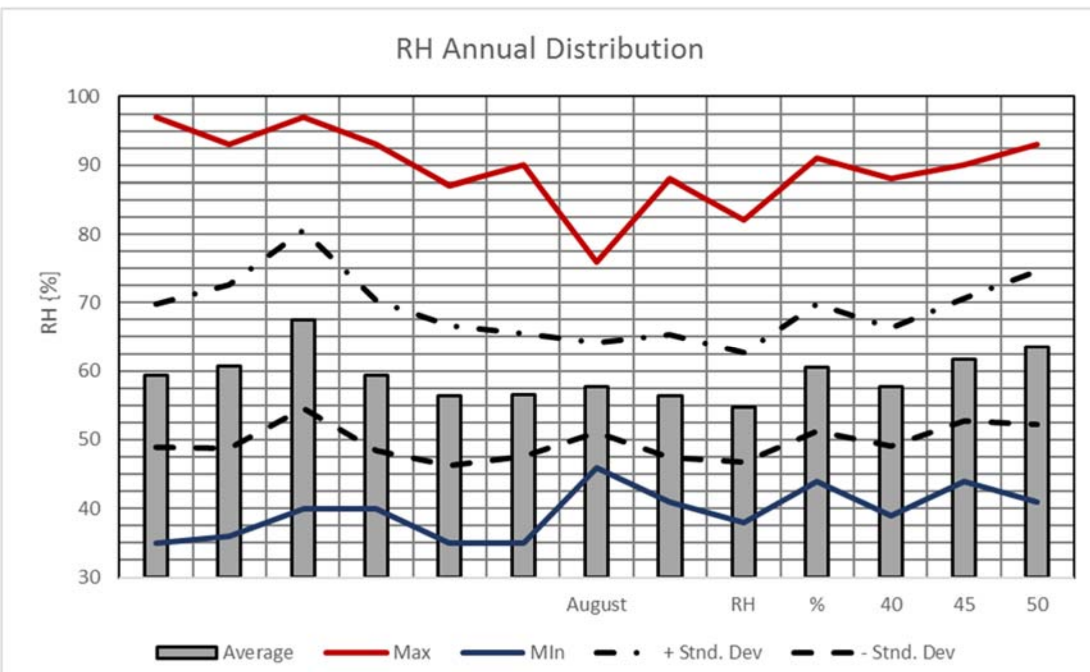


Figure A-49: Relative humidity (RH); 11-h record; Annual and monthly averages, min and maxvalues of RH for the full record

Cum. Probability 80% = RH 68%

Annual
RH [%]

	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	59.3	97.0	35.0	10.4

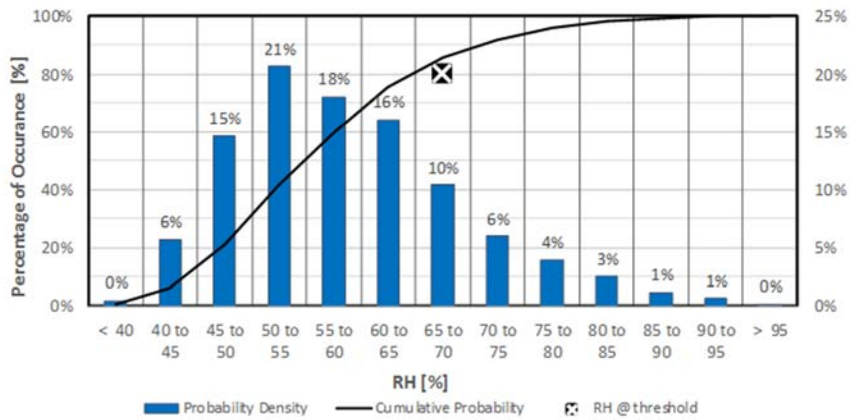


Figure A-50:
Relative humidity (RH); 11-h record

Probability of exceedance for the the entire year

Cum. Probability 80% = RH 68%

January
RH [%]

	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	60.7	93.0	36.0	11.9

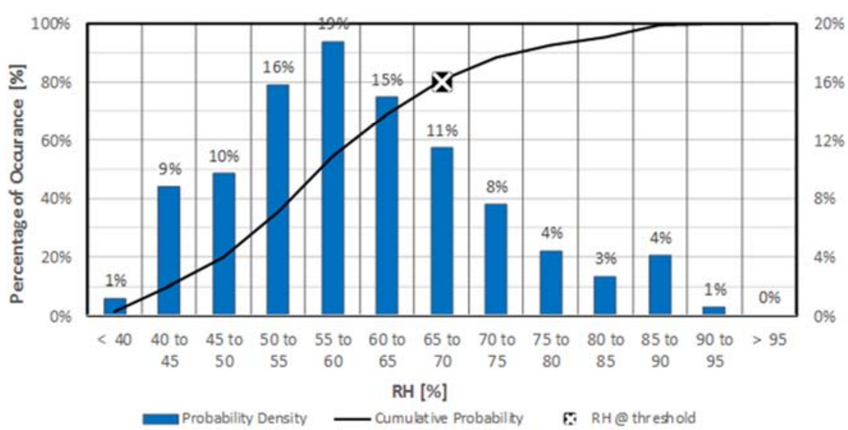


Figure A-51:
Relative humidity (RH); 11-h record
Probability of exceedance for the month of January

Cum. Probability 80% = RH 78 %

February **RH [%]**	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	67.6	97.0	40.0	13.0

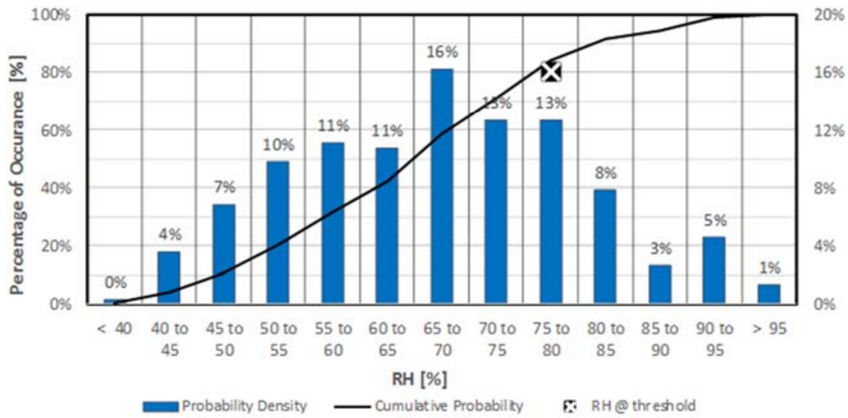


Figure A-52:
Relative humidity (RH); 11-h record

Probability of exceedance for the month of February

March **RH [%]**	Cum. Probability 80% = RH 68 % Average	Max	Min	Stnd. Dev.
	%	%	%	%
	59.3	93.0	40.0	11.0

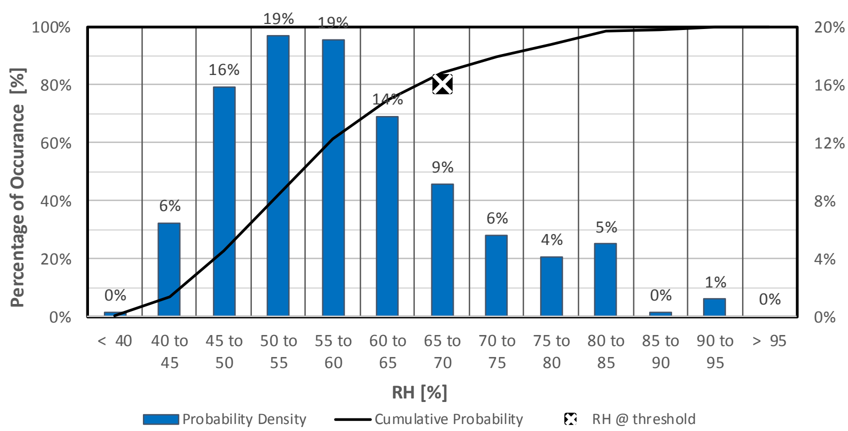


Figure A-53:
Relative humidity (RH); 11-h record
Probability of exceedance for the month of March

Figure A-54: Relative humidity (RH); 11-h record; Probability of exceedance for the month of April

April
RH [%]

Cum. Probability 80% = RH 63 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
56.4	87.0	35.0	10.2

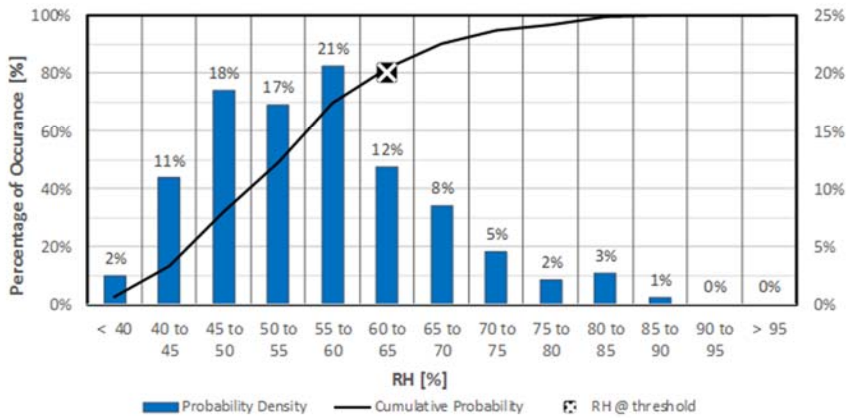


Figure A-55: Relative humidity (RH); 11-h record; Probability of exceedance for the month of May

May
RH [%]

Cum. Probability 80% = RH 63 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
56.5	90.0	35.0	9.0

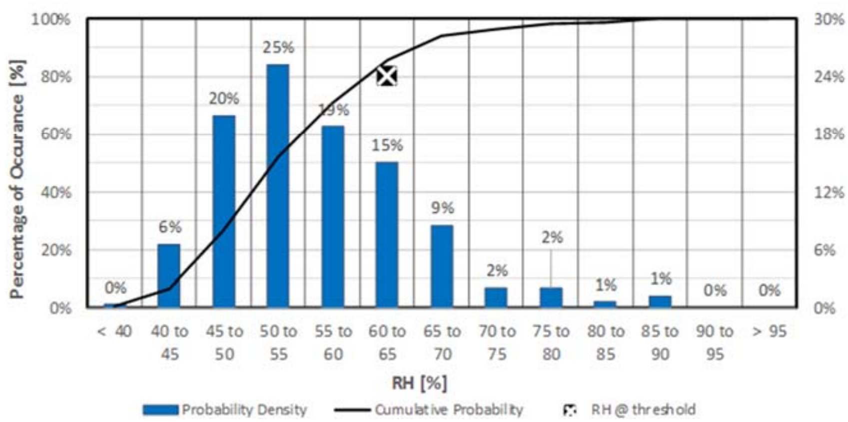


Figure A-56: Relative humidity (RH); 11-h record; Probability of exceedance for the month of June

June
RH [%]

Cum. Probability 80% = RH 63 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
57.7	76.0	46.0	6.5

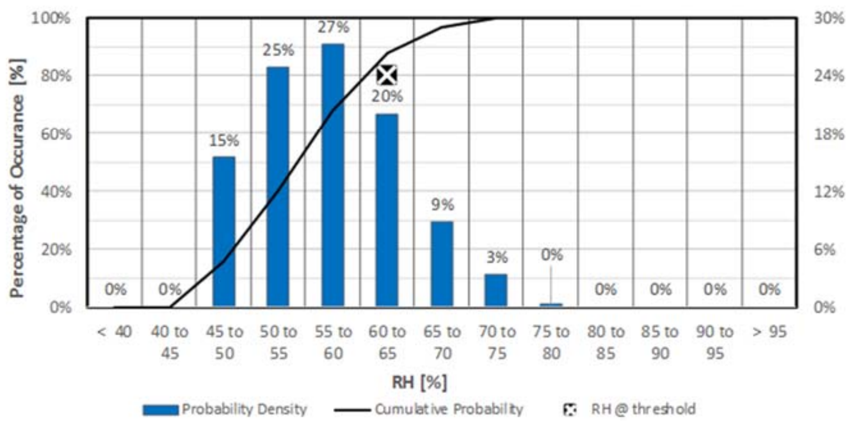
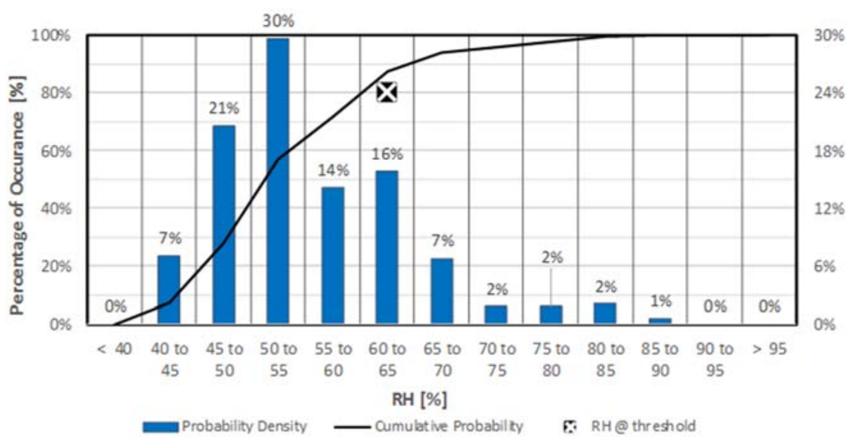


Figure A-57: Relative humidity (RH); 11-h record; Probability of exceedance for the month of July

July
RH [%]

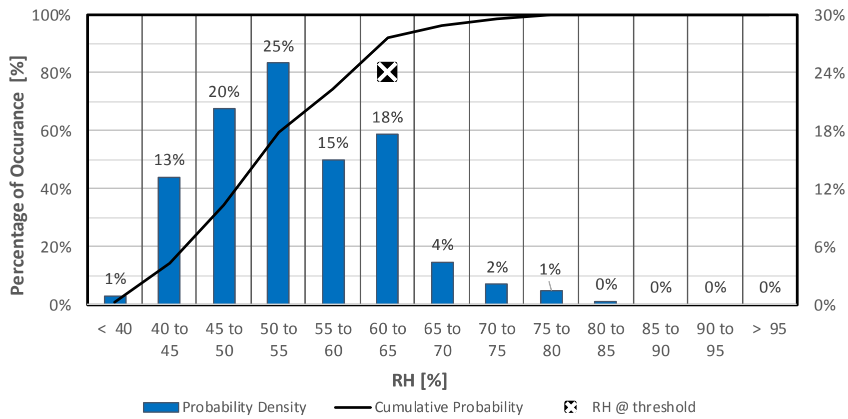
Cum. Probability 80% = RH 63 %

Average	Max	Min	Stnd. Dev.
%	%	%	%
56.4	88.0	41.0	8.9



August	Cum. Probability 80% = RH 63 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	54.7	82.0	38.0	7.9

Figure A-58: Relative humidity (RH); 11-h record; Probability of exceedance for the month of August



Septermber	Cum. Probability 80% = RH 68 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	60.5	91.0	44.0	9.3

Figure A-59: Relative humidity (RH); 11-h record; Probability of exceedance for the month of September

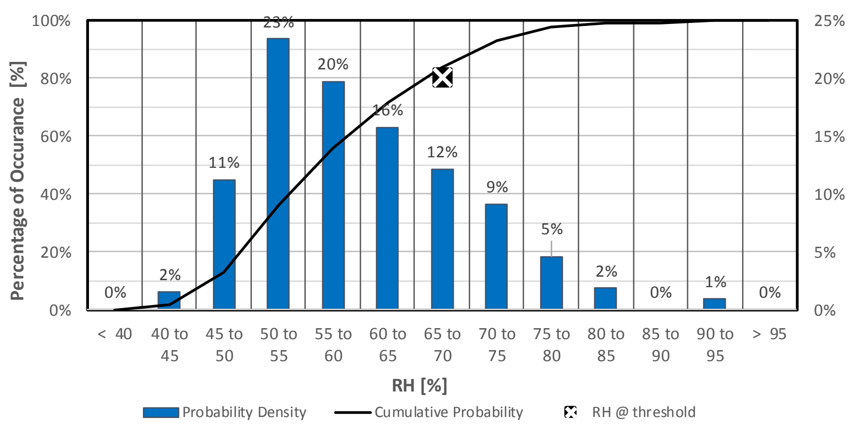


Figure A-60: Relative humidity (RH); 11-h record; Probability of exceedance for the month of October

October	Cum. Probability 80% = RH 63 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	57.7	88.0	39.0	8.6

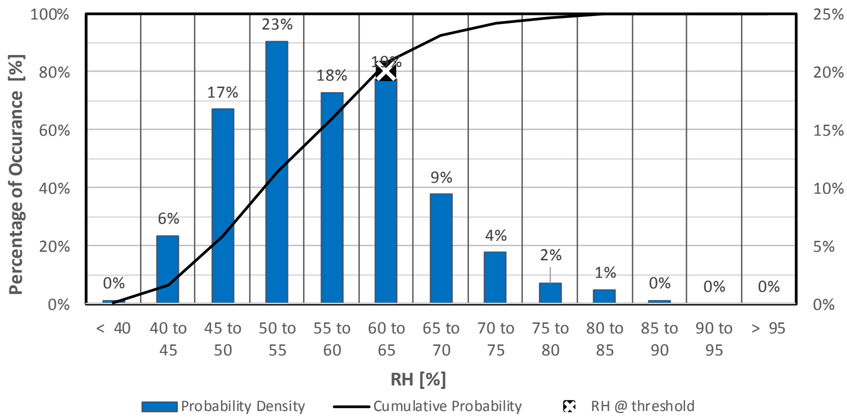
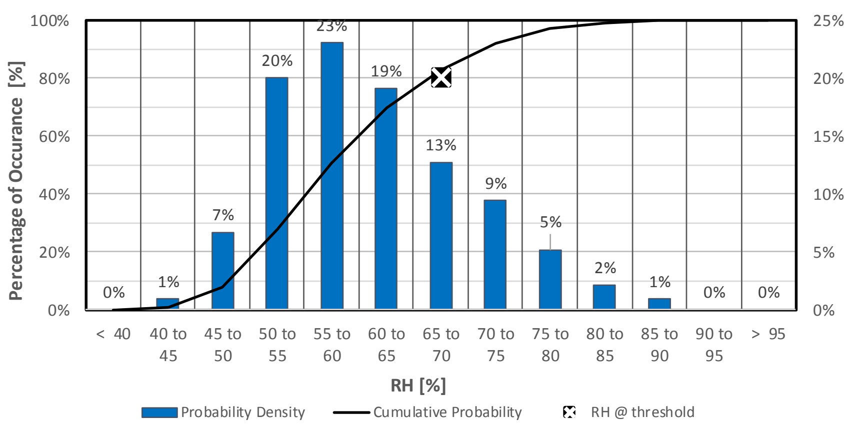


Figure A-61: Relative humidity (RH); 11-h record; Probability of exceedance for the month of November

November	Cum. Probability 80% = RH 68 %			
RH [%]	Average	Max	Min	Stnd. Dev.
	%	%	%	%
	61.7	90.0	44.0	8.9



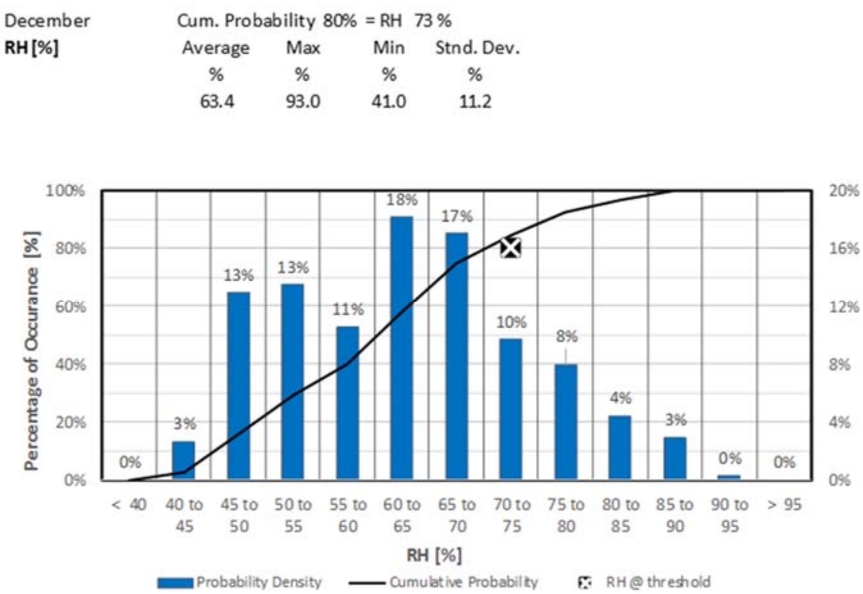


Figure A-62: Relative humidity (RH); 11-h record; Probability of exceedance for the month of December

	RH full record monthly	RH 11-h record monthly
Annual	68%	59%
Jan	68%	61%
Feb	77%	68%
March	69%	59%
April	65%	56%
May	67%	57%
Jun	65%	58%
July	65%	56%
Aug	65%	55%
Sept	69%	61%
Oct	69%	58%
Nov	70%	62%
Dec	70%	63%

Figure A-62: Comparison Relative humidity Full record and 11-h record

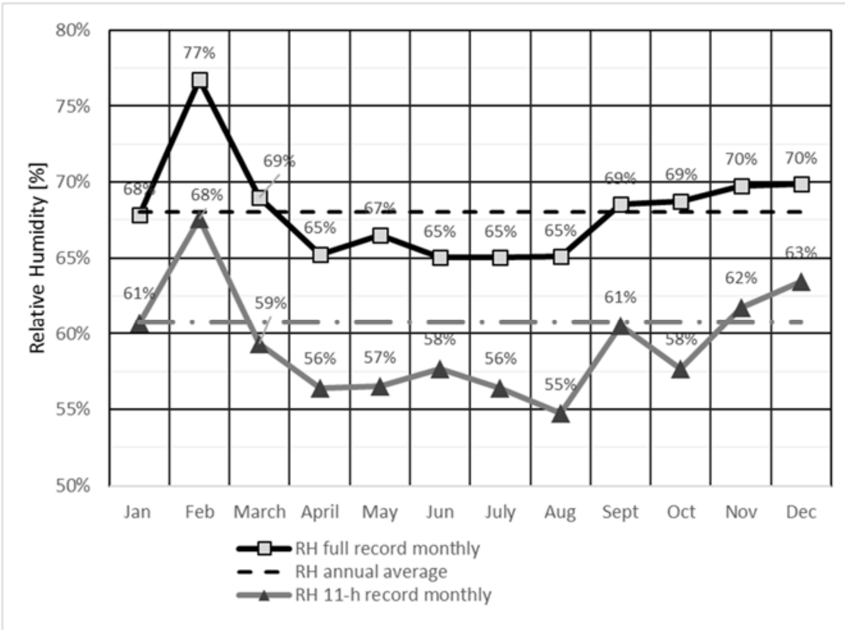


Figure A-62: Comparison Relative humidity Full record and 11-h record

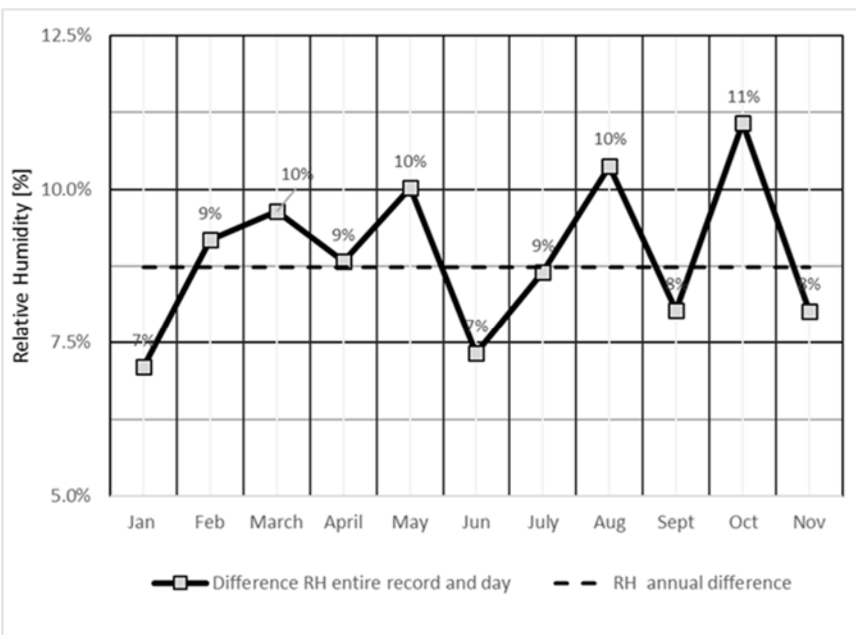


Figure A-62: Comparison Relative humidity Full record and 11-h record; difference between the data of the Full reord and the 11-h record

Positive difference inidcates that the RH of the full record is larger. Also, the RH during the 11-h perios is lower

Dry bulb temperature (Full record, 8760 hours per year)

	average F	Max F	Min F	stnd. Dev. F	+ Stnd. Dev F	- Stnd. Dev F
Annual	77	92	56	6	83	71
Jan	72	85	63	5	77	68
Feb	73	84	56	5	79	68
March	74	86	59	6	79	68
April	75	86	64	5	79	70
May	77	88	66	5	82	72
Jun	79	87	70	4	82	75
July	80	90	72	4	85	76
Aug	81	91	70	5	86	76
Sept	80	92	72	4	84	76
Oct	79	92	67	6	85	74
Nov	77	87	61	5	82	71
Dec	75	87	62	5	79	70

Figure A-63:
Dry bulb temperature; Full record

Annual and monthly averages, min and maxvalues of RH for the full record

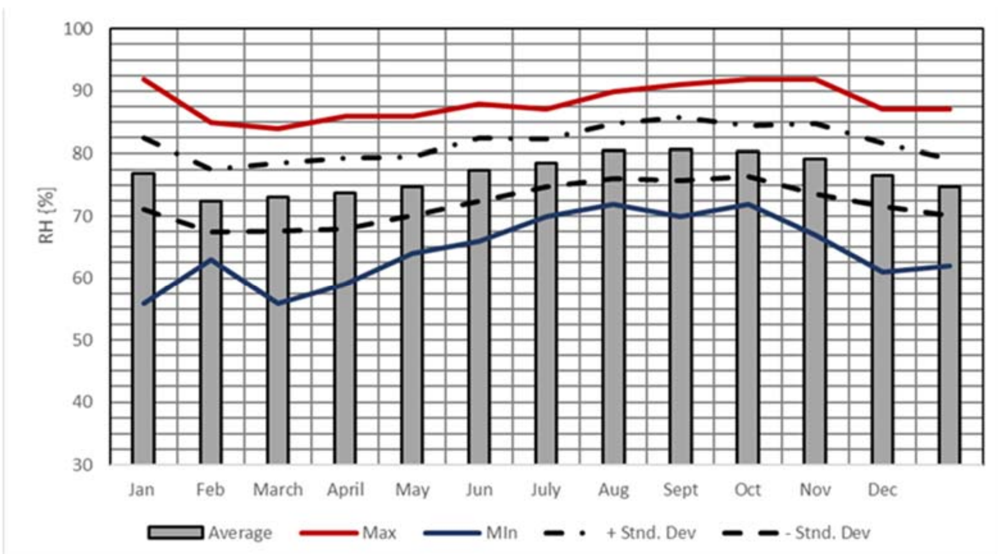


Figure A-64: Dry bulb temperature; Full record , Annual and monthly averages, min and maxvalues of RH for the full record

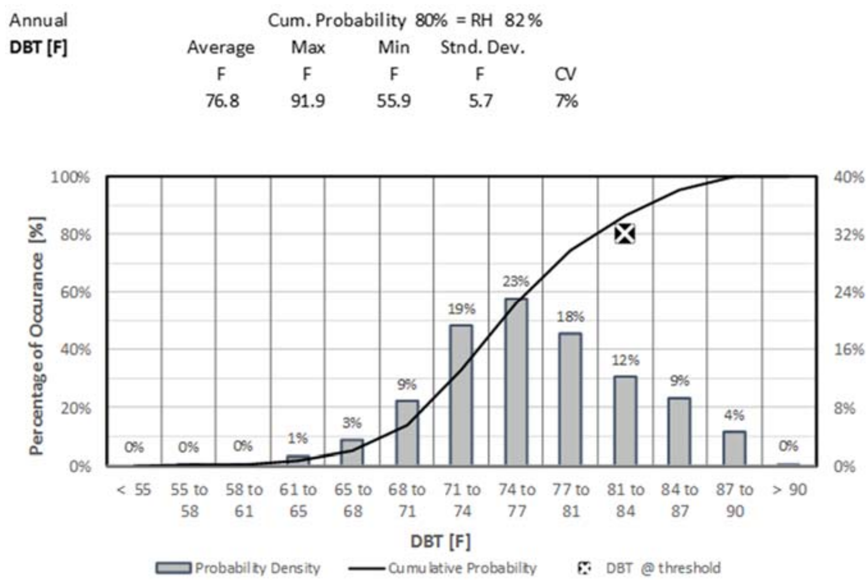


Figure A-65:
Dry bulb temperature; Full record; Probability of exceedance for the the entire year

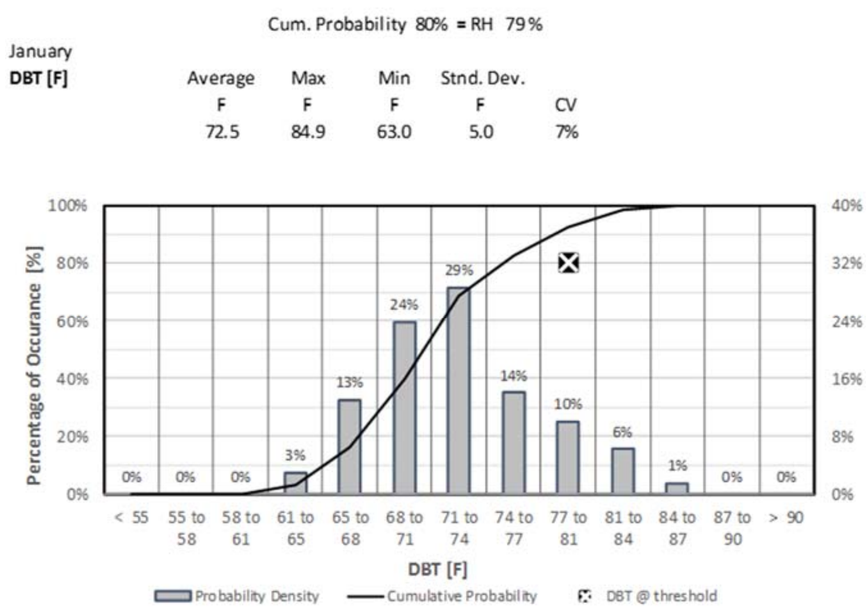


Figure A-66:
Dry bulb temperature; Full record; Probability of exceedance for the month of January

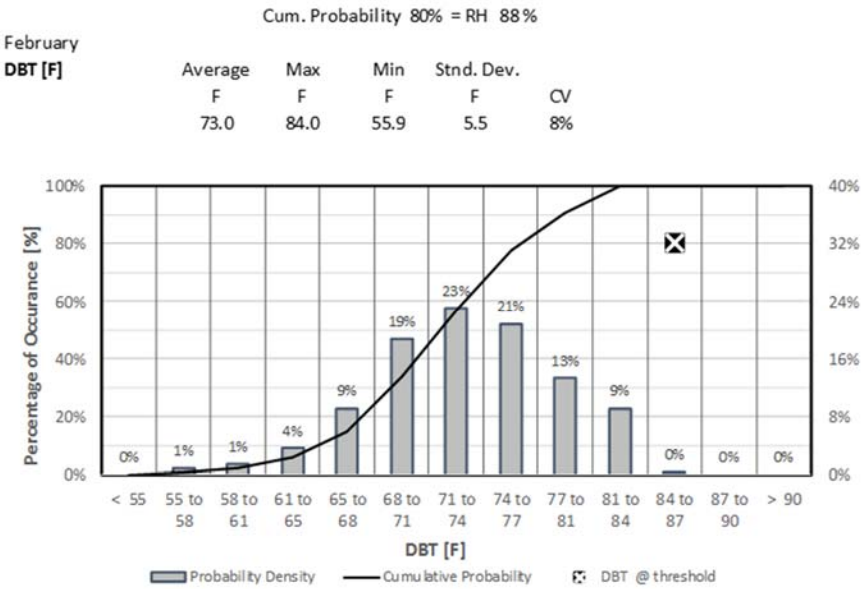


Figure A-67:
Dry bulb temperature; Full record; Probability of exceedance for the month of February

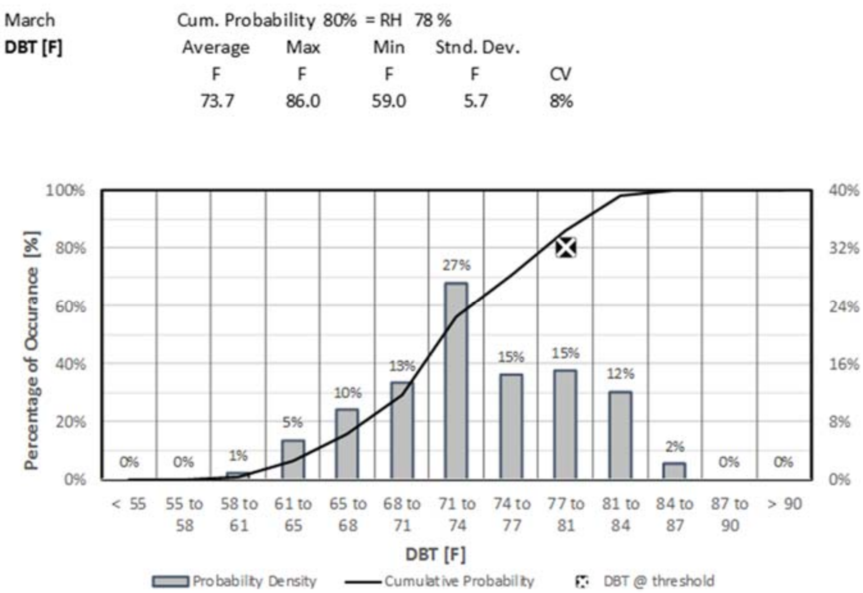


Figure A-68:
Dry bulb temperature; Full record; Probability of exceedance for the month of March

April	Cum. Probability 80% = RH 79 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	74.8	86.0	64.0	4.7	6%

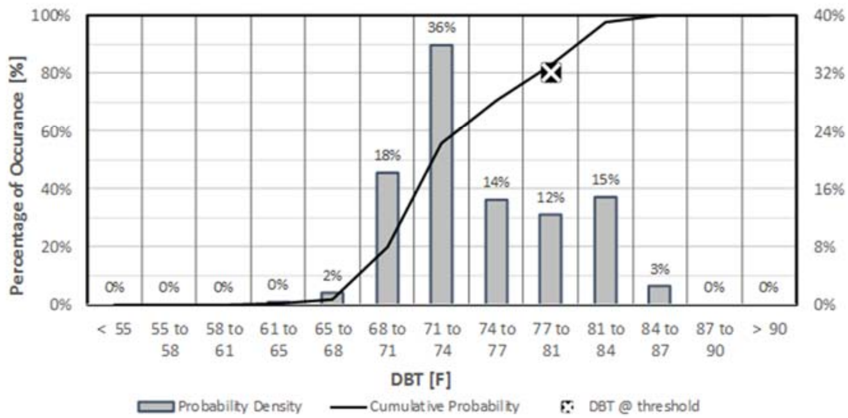


Figure A-69:
Dry bulb temperature; Full record; Probability of exceedance for the month of April

May	Cum. Probability 80% = RH 82 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	77.4	88.0	66.0	5.0	6%

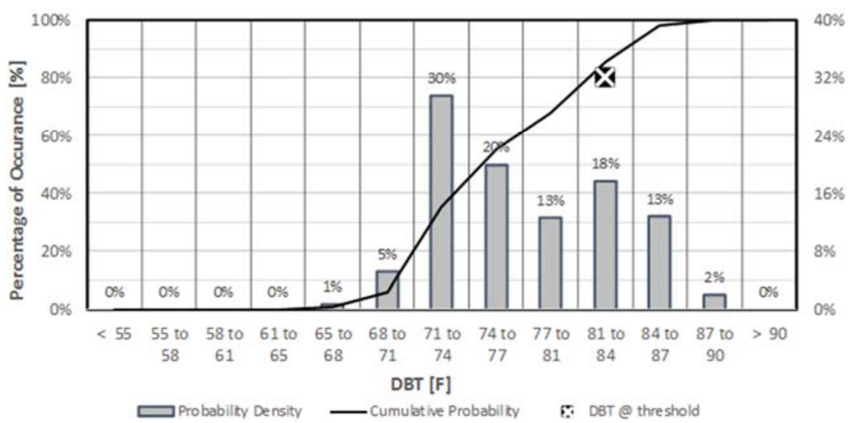
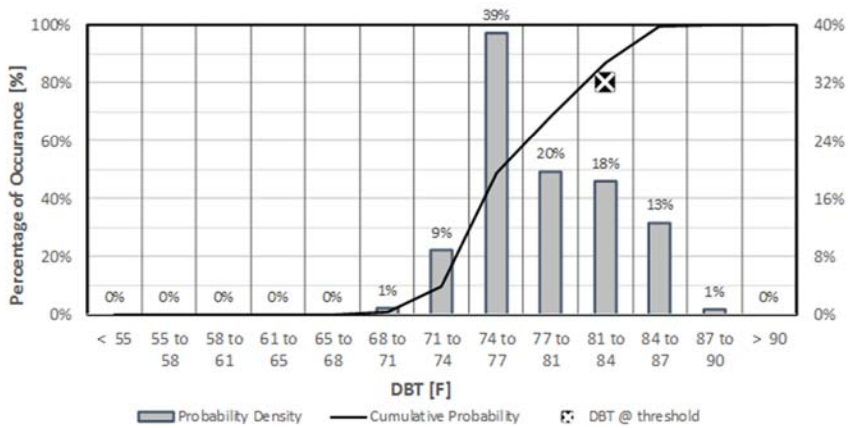


Figure A-70:
Dry bulb temperature; Full record; Probability of exceedance for the month of May

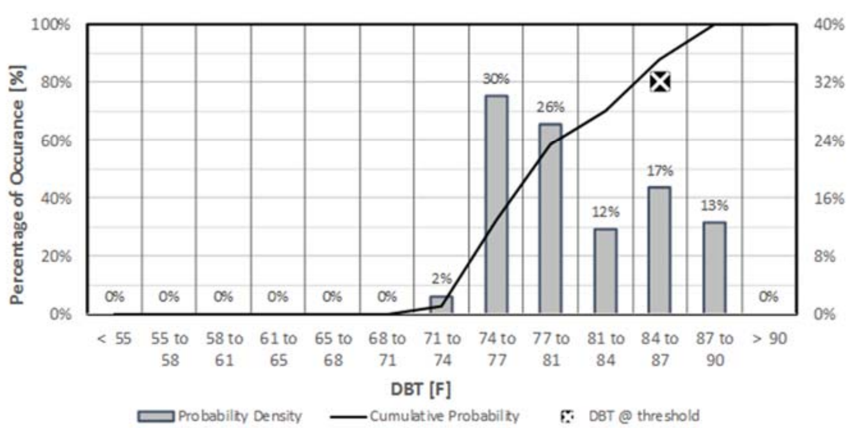
June	Cum. Probability 80% = RH 82 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	78.6	87.1	70.0	3.8	5%

Figure A-71: Dry bulb temperature; Full record; Probability of exceedance for the month of June



July	Cum. Probability 80% = RH 85 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	%	F	F	F	CV
	80.4	90.0	72.0	4.3	5%

Figure A-72: Dry bulb temperature; Full record ; Probability of exceedance for the month of July



August	Cum. Probability 80% = RH 85 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	80.7	91.0	70.0	5.0	6%

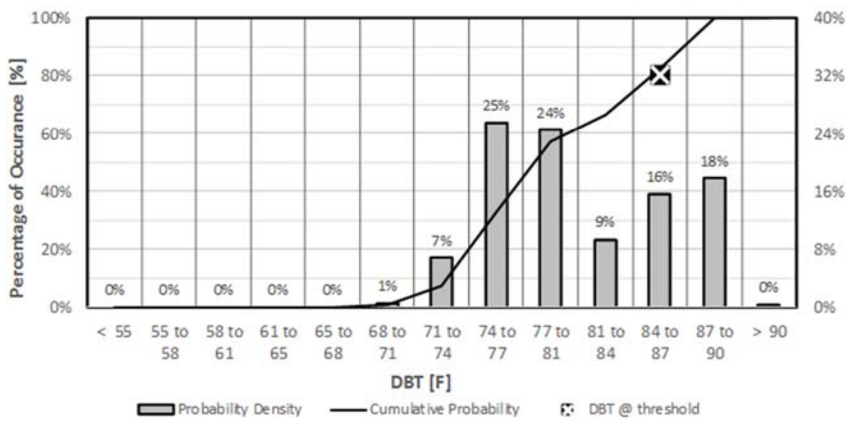


Figure A-73: Dry bulb temperature; Full record; Probability of exceedance for the month of August

September	Cum. Probability 80% = RH 85 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	%	F	F	F	CV
	80.4	91.9	72.0	4.1	5%

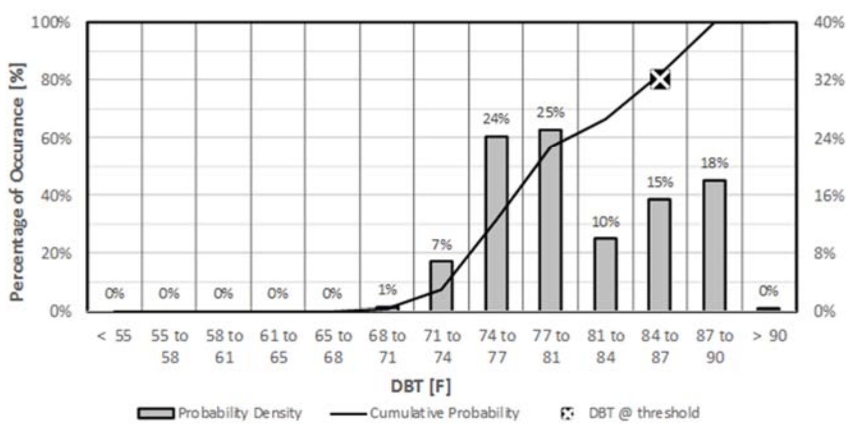
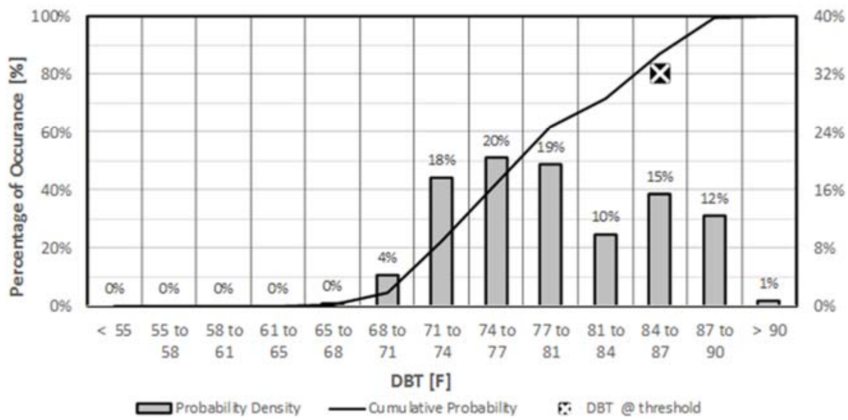


Figure A-74: Dry bulb temperature; Full record ; Probability of exceedance for the month of September

October
DBT [F]

	Cum. Probability 80% = RH 85 %				
	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	79.2	91.9	66.9	5.7	7%

Figure A-75: Dry bulb temperature; Full record; Probability of exceedance for the month of October



November
DBT [F]

	Cum. Probability 80% = RH 82 %				
	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	76.6	87.1	61.0	5.1	7%

Figure A-76: Dry bulb temperature; Full record; Probability of exceedance for the month of November

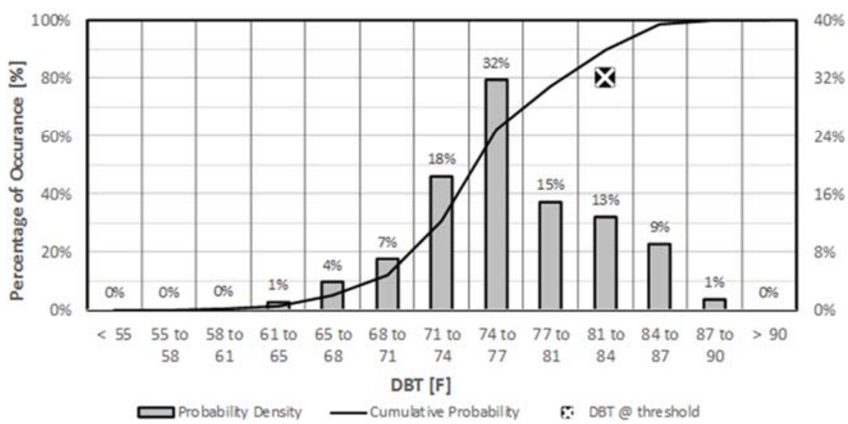
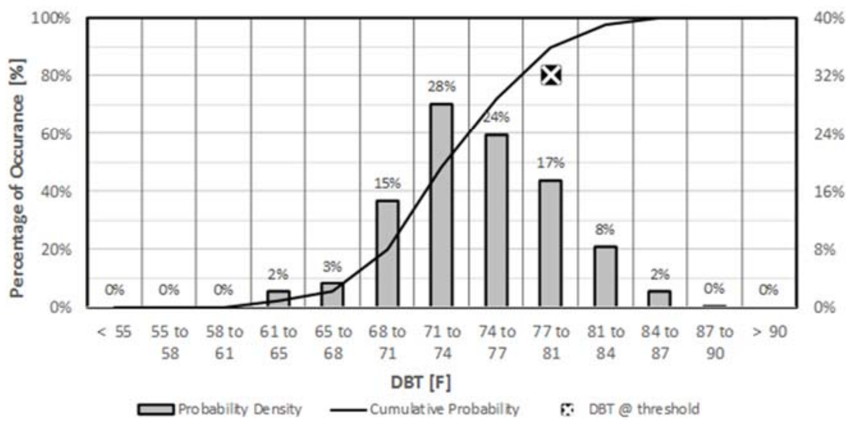


Figure A-77: Dry bulb temperature; Full record; Probability of exceedance for the month of December

December	Cum. Probability 80% = RH 79 %				
DBT [F]	Average	Max	Min	Stnd. Dev.	
	F	F	F	F	CV
	74.6	87.1	62.1	4.6	6%



Dry bulb temperature (Full record, 8760 hours per year)

	average F	Max F	Min F	stnd. Dev. F
Annual	80.9	91.9	60.1	4.6
Jan	76.1	84.9	64.9	4.4
Feb	77.1	84.0	60.1	4.0
March	78.3	86.0	64.9	3.7
April	78.9	86.0	70.0	3.4
May	82.0	88.0	72.0	3.0
Jun	82.0	87.1	75.0	2.5
July	84.5	90.0	77.0	2.8
Aug	85.4	91.0	75.9	3.1
Sept	83.8	91.9	75.0	3.1
Oct	84.2	91.9	75.0	3.5
Nov	80.4	87.1	64.9	3.8
Dec	77.7	87.1	64.9	3.9

Figure A-478:
Dry bulb temperature; 11-h record; Annual and monthly averages, min and maxvalues of RH for the full record

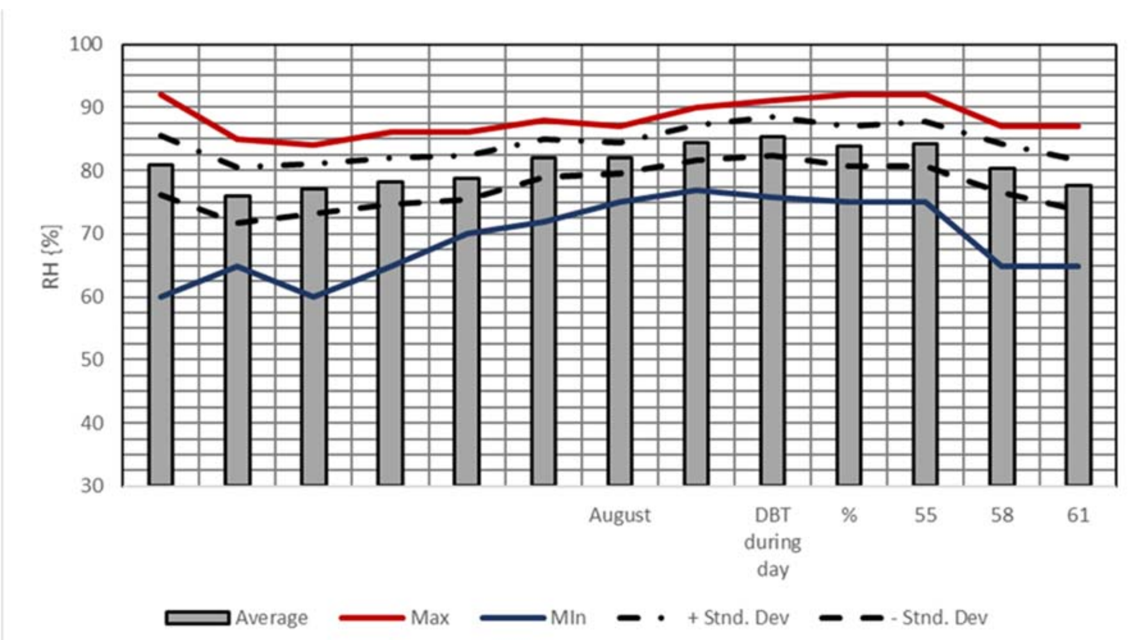


Figure A-79: Dry bulb temperature; Full record , Annual and monthly averages, min and maxvalues of RH for the full record

Cum. Probability 80% = DBT 85 F

Annual
DBT [F]

Average	Max	Min	Stnd. Dev.
F	F	F	F
80.9	91.9	60.1	4.6

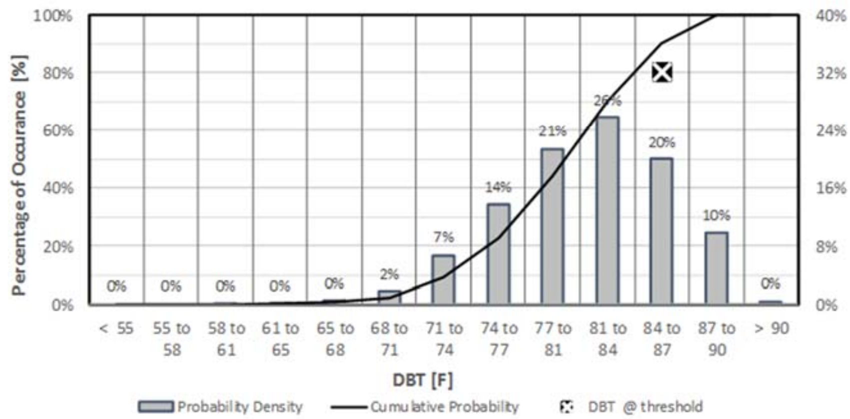


Figure A-80:
Dry bulb temperature; 11-h record; Probability of exceedance for the the entire year

Cum. Probability 80% = DBT 79 F

January
DBT [F]

Average	Max	Min	Stnd. Dev.
F	F	F	F
76.1	84.9	64.9	4.4

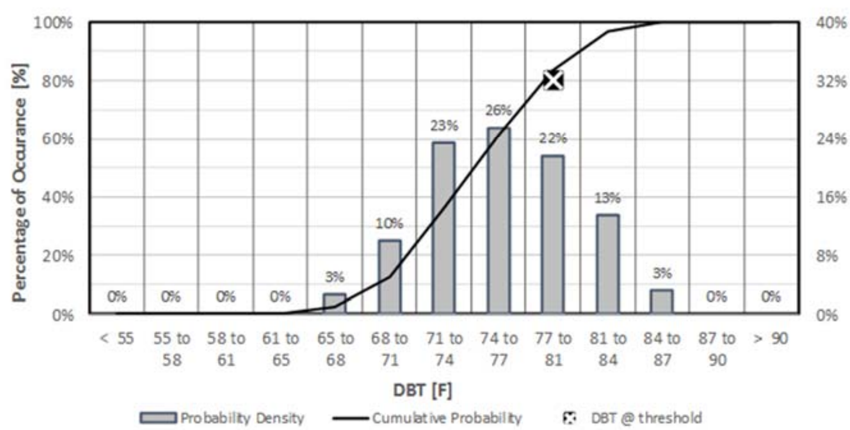


Figure A-81:
Dry bulb temperature; 11-h record; Probability of exceedance for the month of January

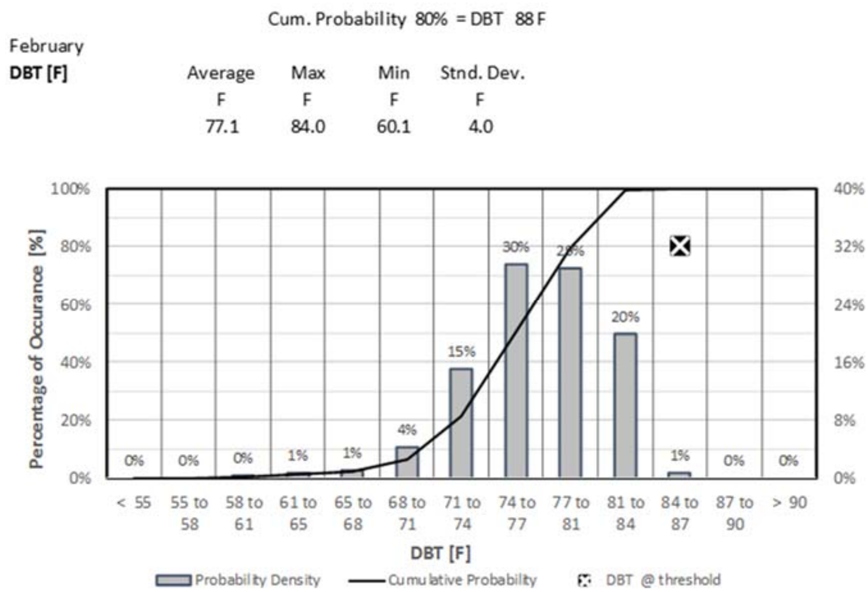


Figure A-82:
Dry bulb temperature; 11-h record; Probability of exceedance for the month of February

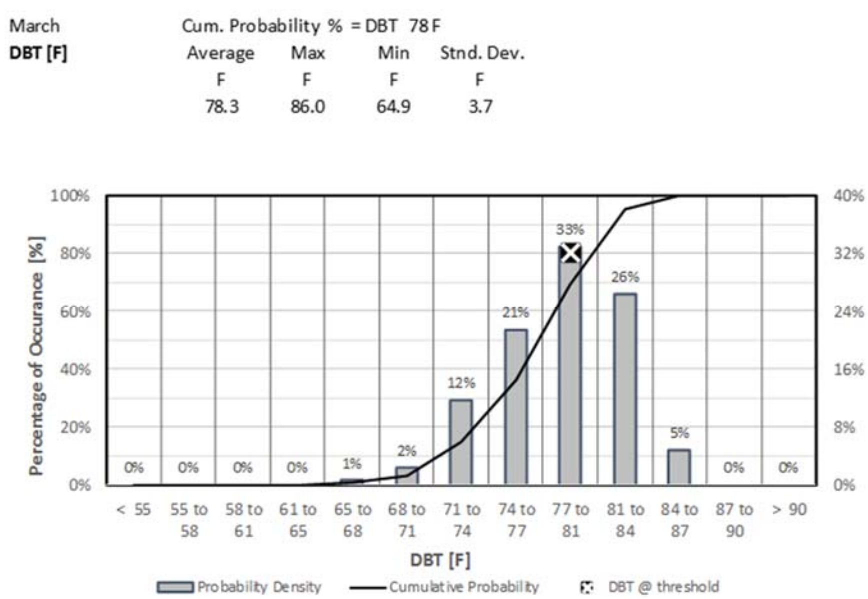


Figure A-83:
Dry bulb temperature; 11-h record; Probability of exceedance for the month of March

April	Cum. Probability % = DBT 78 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	78.9	86.0	70.0	3.4

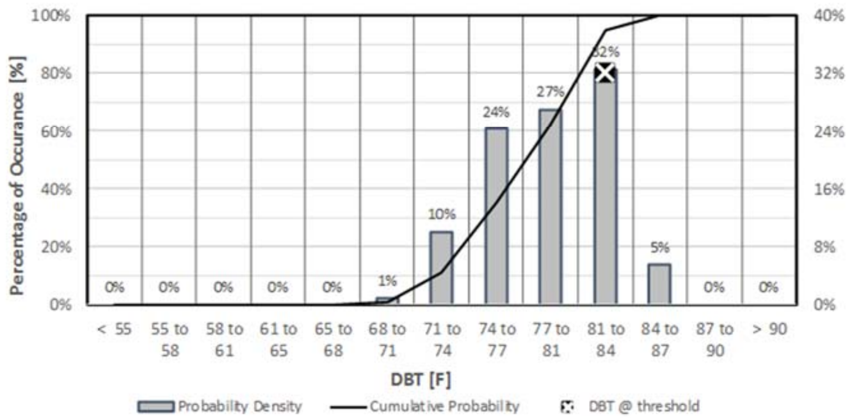


Figure A-84:
Dry bulb temperature; 11-h record; Probability of exceedance for the month of April

May	Cum. Probability % = DBT 85 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	82.0	88.0	72.0	3.0

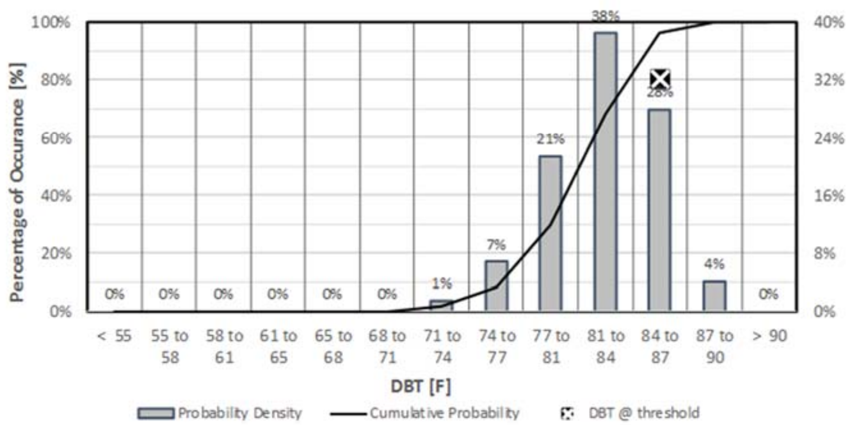


Figure A-85:
Dry bulb temperature; 11-h record; Probability of exceedance for the month of May

June	Cum. Probability % = DBT 85 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	82.0	87.1	75.0	2.5

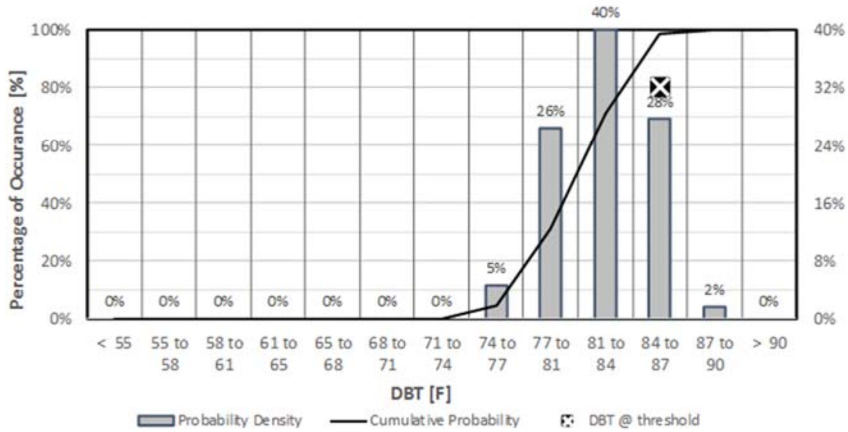


Figure A-86: Dry bulb temperature; 11-h record; Probability of exceedance for the month of June

July	Cum. Probability % = DBT 89 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	%	F	F	F
	84.5	90.0	77.0	2.8

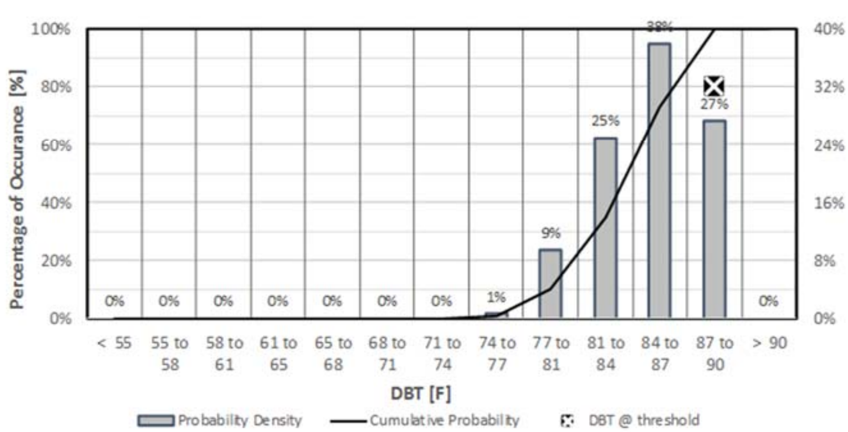


Figure A-87: Dry bulb temperature; 11-h record; ; Probability of exceedance for the month of July

August	Cum. Probability % = DBT 89 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	85.4	91.0	75.9	3.1

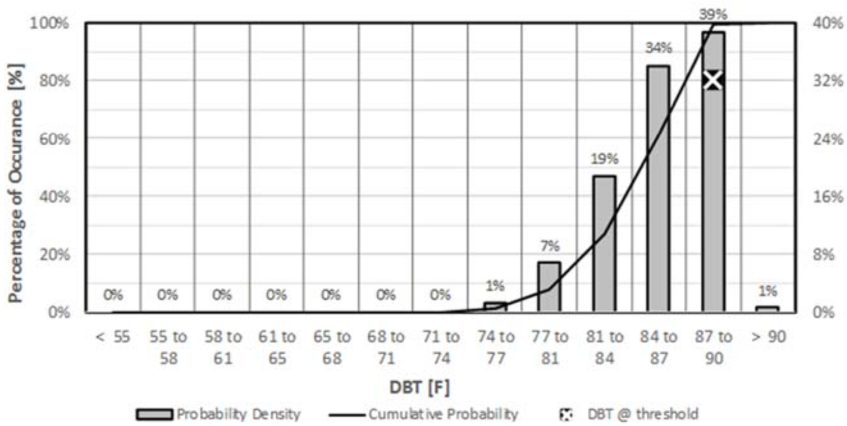


Figure A-88: Dry bulb temperature; 11-h record; Probability of exceedance for the month of August

September	Cum. Probability % = DBT 89 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	%	F	F	F
	83.8	91.9	75.0	3.1

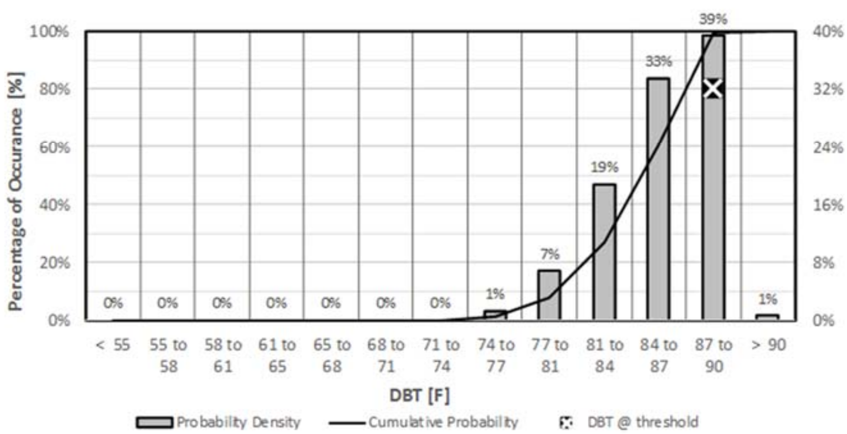


Figure A-89: Dry bulb temperature; 11-h record; ; Probability of exceedance for the month of September

October
DBT [F]

Cum. Probability % = DBT 89 F			
Average	Max	Min	Stnd. Dev.
F	F	F	F
84.2	91.9	75.0	3.5

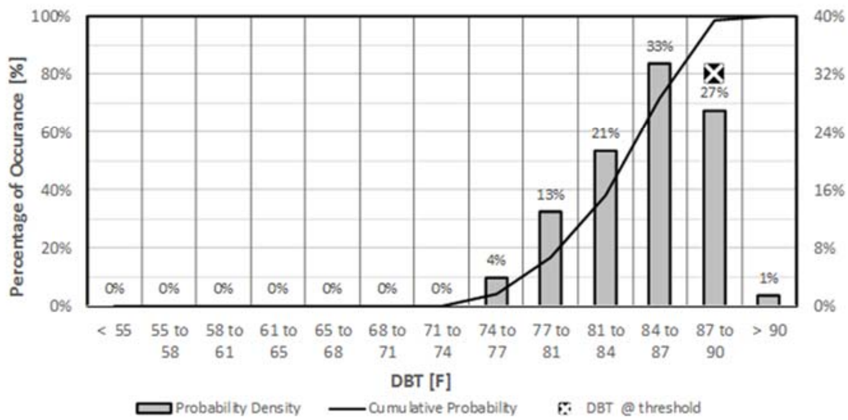


Figure A-90: Dry bulb temperature; 11-h record; Probability of exceedance for the month of October

November
DBT [F]

Cum. Probability % = DBT 85 F			
Average	Max	Min	Stnd. Dev.
F	F	F	F
80.4	87.1	64.9	3.8

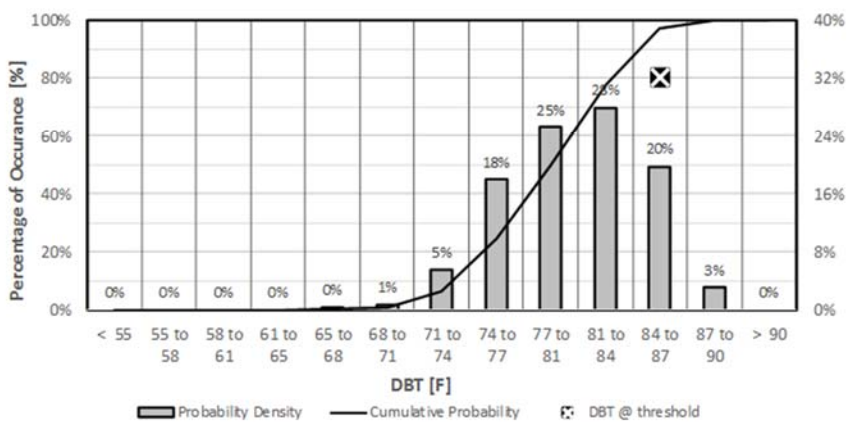
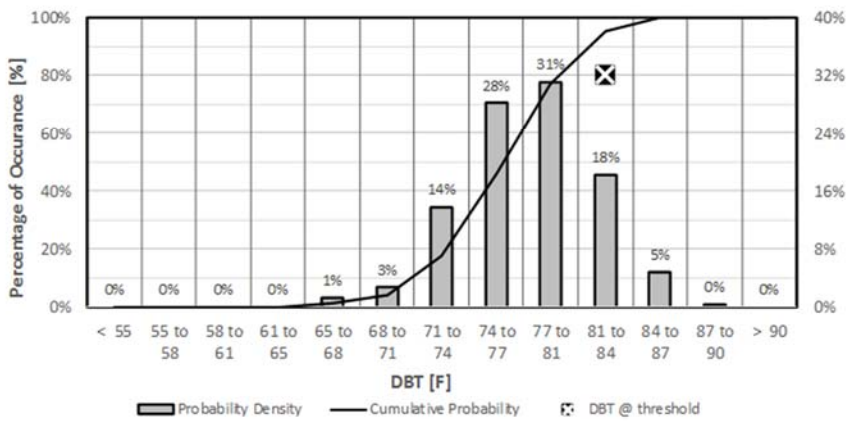


Figure A-91: Dry bulb temperature; 11-h record;; Probability of exceedance for the month of November

Figure A-92: Dry bulb temperature; 11-h record; Probability of exceedance for the month of December

December	Cum. Probability % = DBT 82 F			
DBT [F]	Average	Max	Min	Stnd. Dev.
	F	F	F	F
	77.7	87.1	64.9	3.9



Comparison of Dry bulb temperature: Full record and 11-h record

	DBT full record monthly F	DBT 11-h record monthly F
Annual	76.8	80.9
Jan	72.5	76.1
Feb	73.0	77.1
March	73.7	78.3
April	74.8	78.9
May	77.4	82.0
Jun	78.6	82.0
July	80.4	84.5
Aug	80.7	85.4
Sept	80.4	83.8
Oct	79.2	84.2
Nov	76.6	80.4
Dec	74.6	77.7

Figure A-93:
Dry bulb temperature; comparison of full record and 11-h record

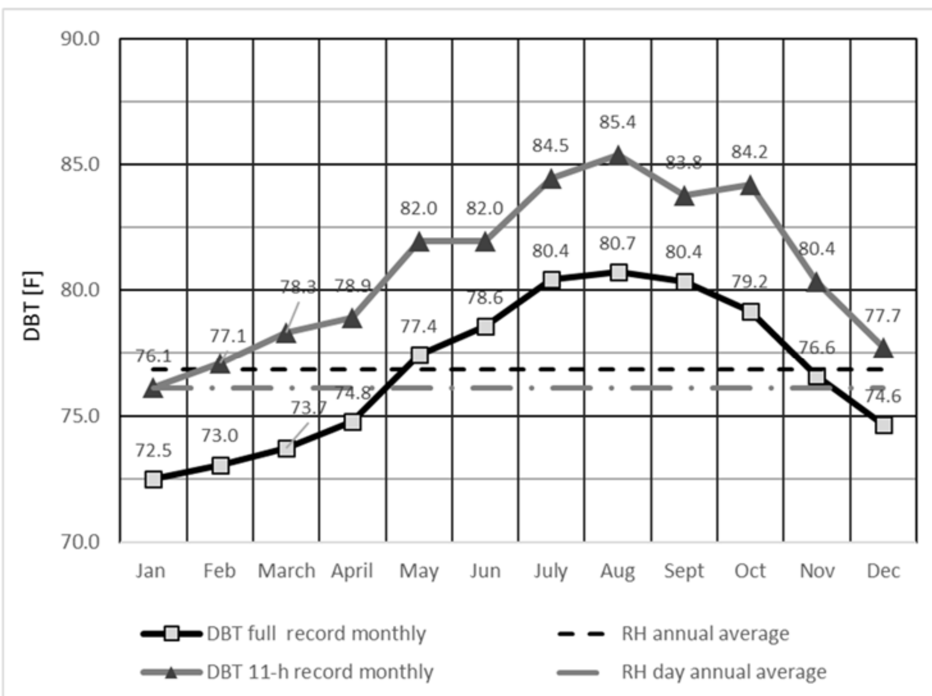
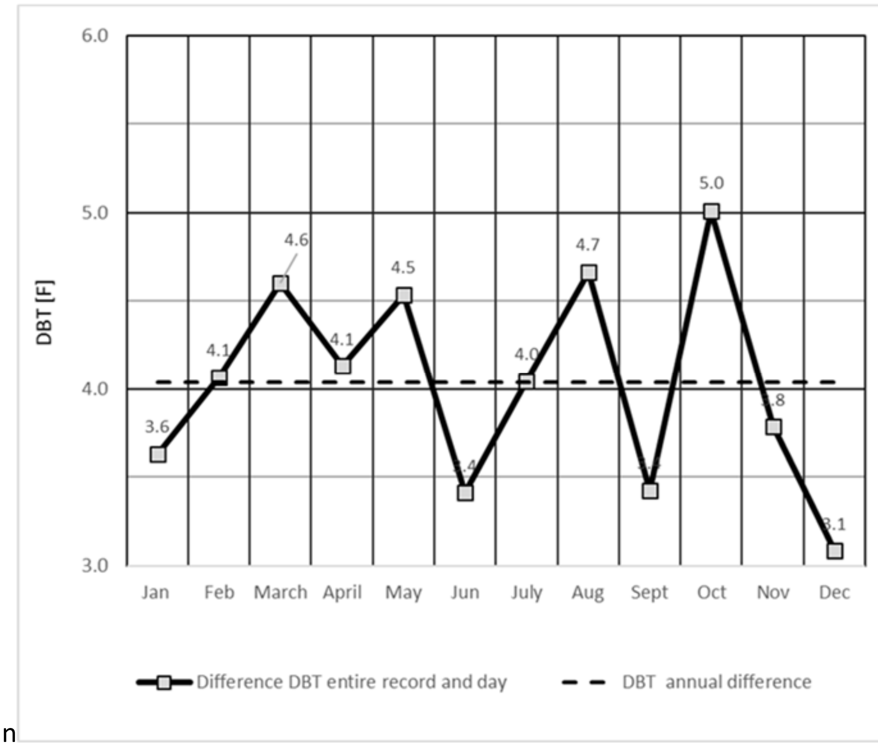


Figure A-94: Dry bulb temperature; comparison of full record and 11-h record



n

Figure A-95: Dry bulb temperature; difference between DBT of 11-h record and full record; the DBT of the 11-h recordos larger than DBT of the full record

Comparison of RH and Dry bulb temperature for Full record

	RH full record monthly averages	DTB full record monthly averages [F]
Annual	68%	77
Jan	68%	72
Feb	77%	73
March	69%	74
April	65%	75
May	67%	77
Jun	65%	79
July	65%	80
Aug	65%	81
Sept	69%	80
Oct	69%	79
Nov	70%	77
Dec	70%	75

Figure A-96:
RH and DBT; comparison of full record

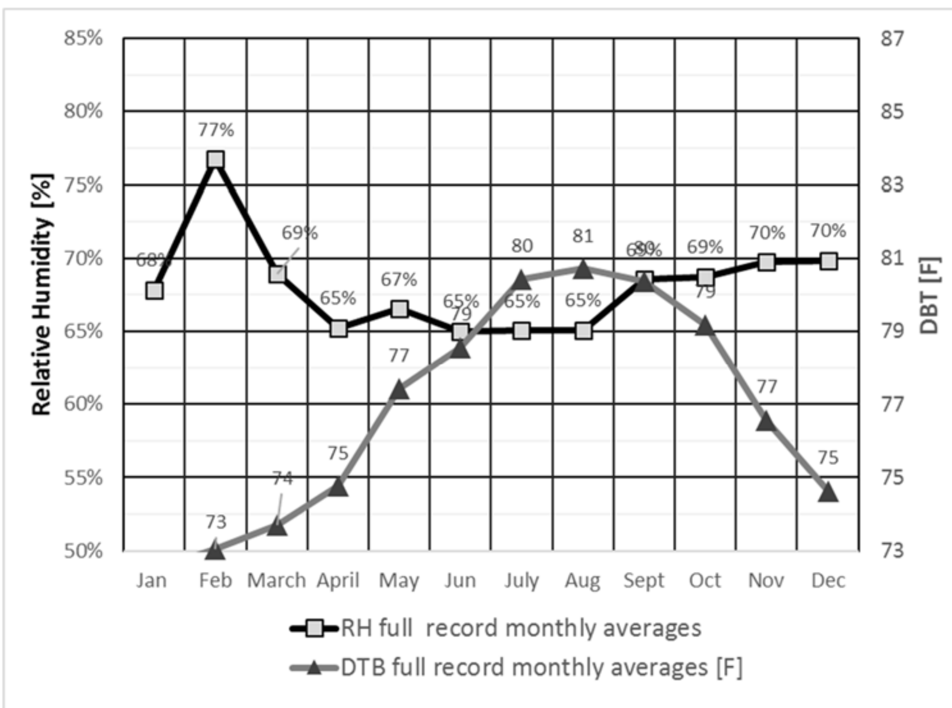


Figure A-97: RH and DBT; comparison of full record

Comparison of RH and Dry bulb temperature for 11-h record

	RH 11-h record monthly averages	DTB 11-h record monthly averages [F]
Annual	59%	81
Jan	61%	76
Feb	68%	77
March	59%	78
April	56%	79
May	57%	82
Jun	58%	82
July	56%	84
Aug	55%	85
Sept	61%	84
Oct	58%	84
Nov	62%	80
Dec	63%	78

Figure A-98:
RH and DBT; comparison of full record

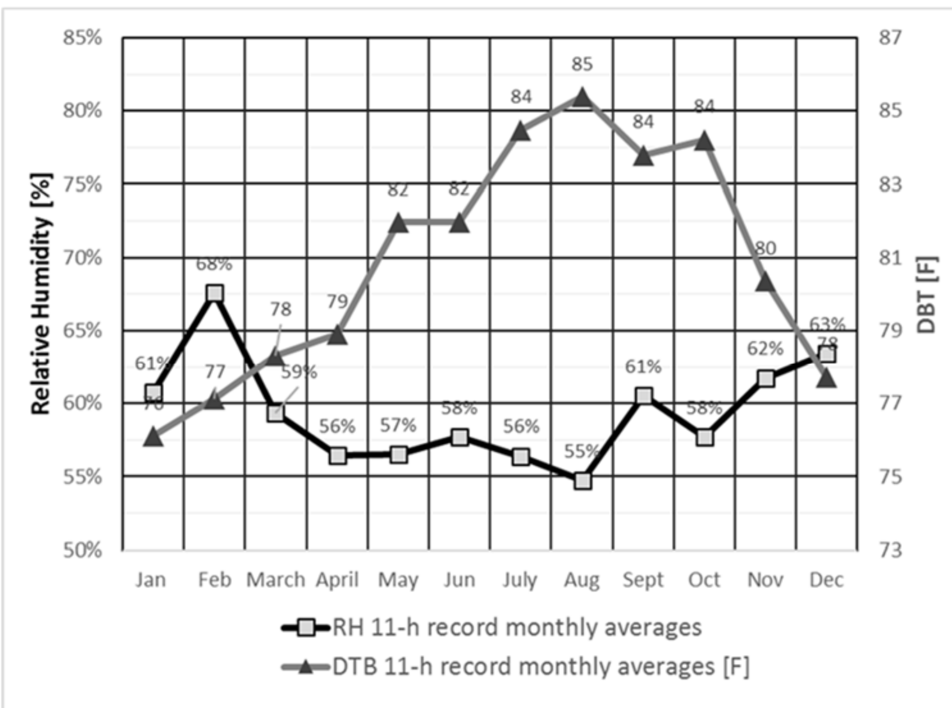


Figure A-99: RH and DBT; comparison of full record

APPENDIX B

TYPICAL WEATHER DATA FOR HONOLULU

Psychrometric charts

Appendix B

TYPICAL WEATHER DATA FOR HONOLULU

This Appendix B contains:

1. Psychrometric charts

Hourly weather data was determined for two conditions, (1) a "full record", for all 8760 hours per year, and a "11-h" record, for the time from 8:00 am (start) to 6:00 pm (end).

For the psychrometric charts the individual observations are plotted in the ASHRAE psychrometric chart. The centroid of all observation of the data set is determined with the 1st moment for the data sets.

Psychrometric charts (Full record, 8760 hours per year)

Figure B-1:
Psychrometric data for the Full record
The data depited represtn the centroits of the data sets

period / month	DBT F	hum. Ratio g/kg	humidity grains / lb	RH %	WBT F	Dew F
Year	76.8	13.329	93.3	67.3	68.8	65.1
1	72.5	11.470	80.3	67.0	64.9	60.9
2	73.1	13.363	93.5	76.3	67.7	65.2
3	73.8	11.470	80.3	64.1	65.3	61.8
4	74.8	11.852	83.0	64.1	66.2	61.8
5	77.5	13.219	92.5	65.2	68.9	64.9
6	78.6	13.488	94.4	64.1	69.6	65.5
7	80.5	14.318	100.2	63.9	71.2	67.1
8	80.8	14.406	100.8	63.6	71.4	67.3
9	80.4	15.110	105.8	67.5	72.1	68.7
10	79.3	14.622	102.4	67.8	71.2	67.7
11	76.6	13.550	94.9	68.8	69.1	65.6
12	74.7	12.790	89.5	69.3	67.5	64.0

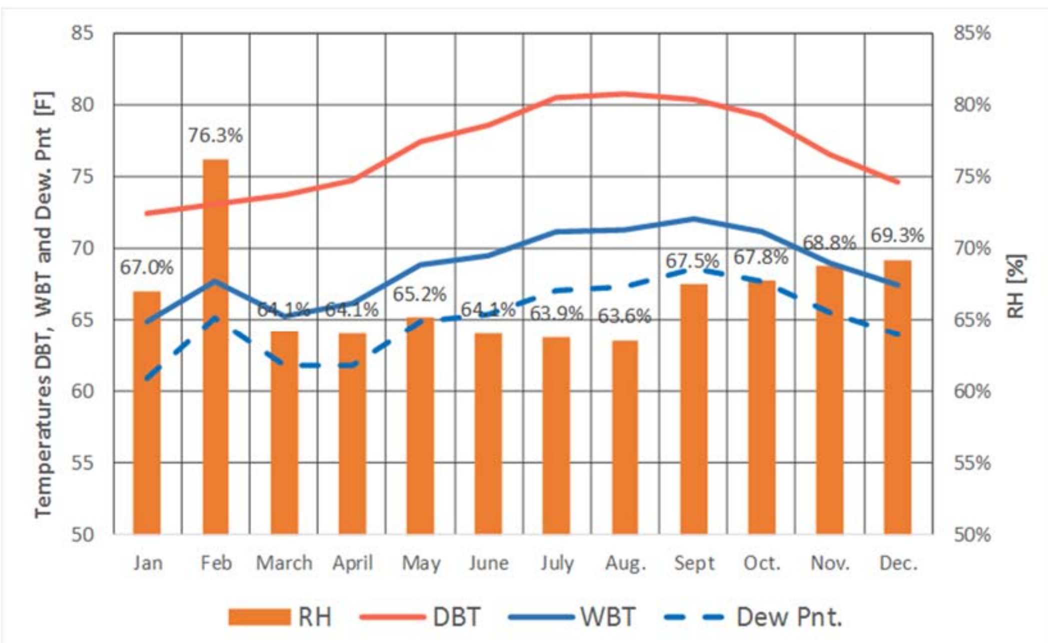


Figure B-2: Psychrometric data for the Full record; The data depited represtn the centroits of the data sets

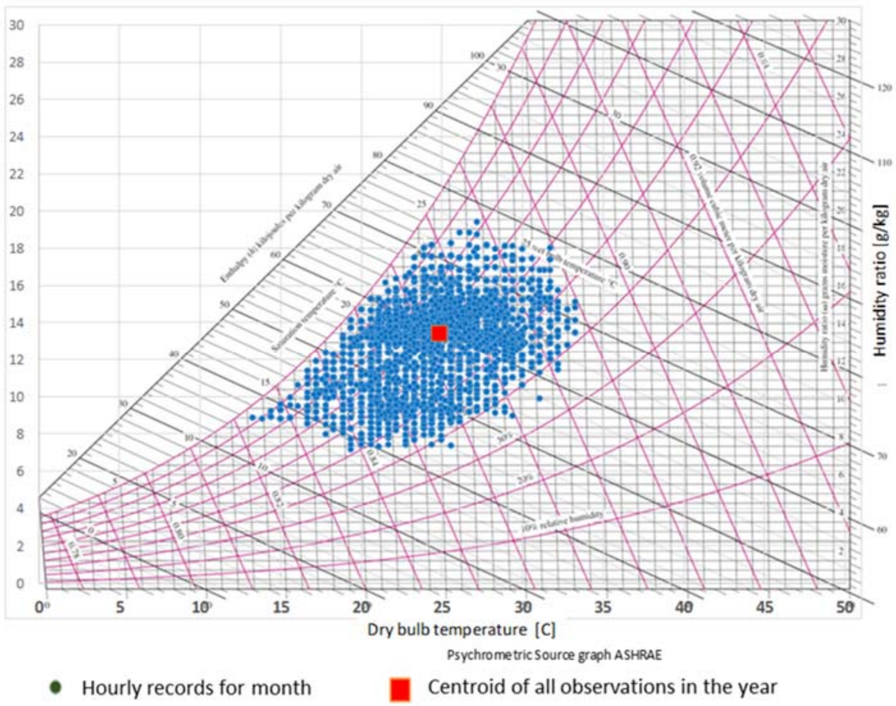


Figure B-3: Full record Psychrometric conditions – for all data points in the year;

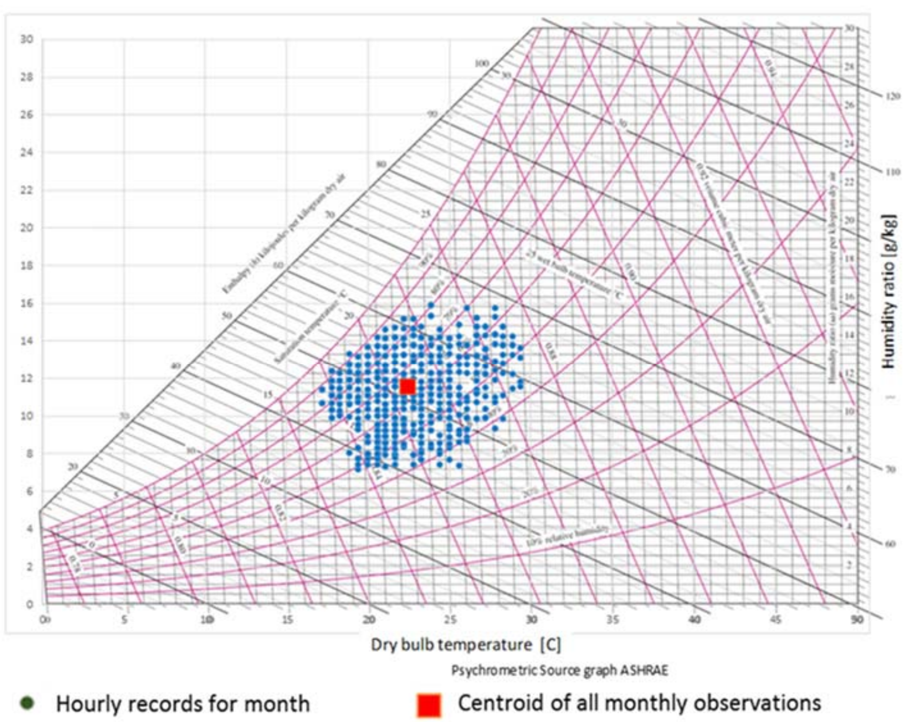


Figure B-4: Full record Psychrometric conditions – Monthly data for January

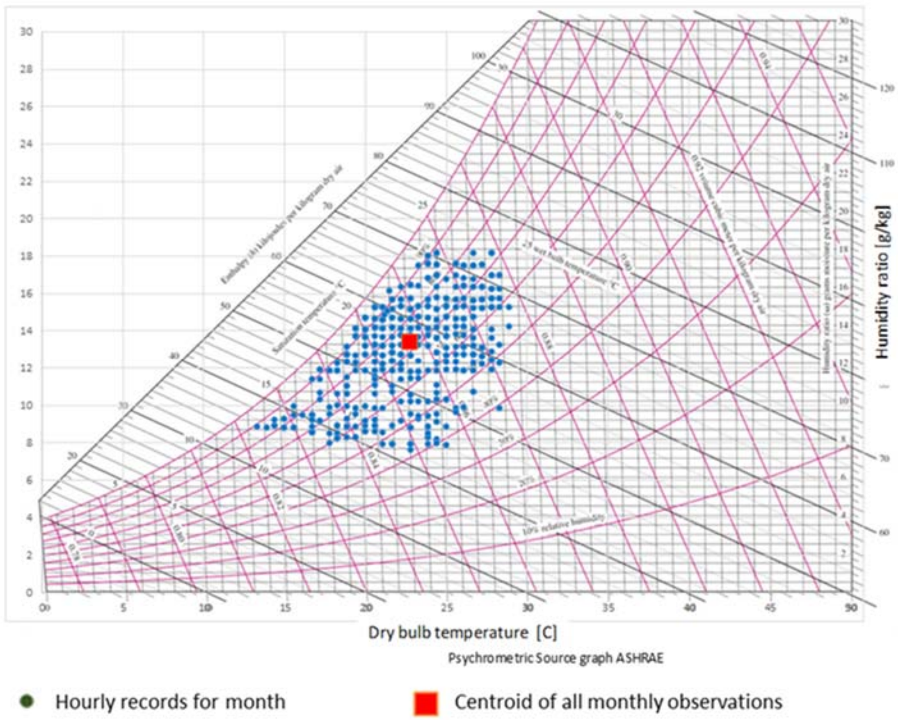


Figure B-5: Full record Psychrometric conditions – Monthly data for February

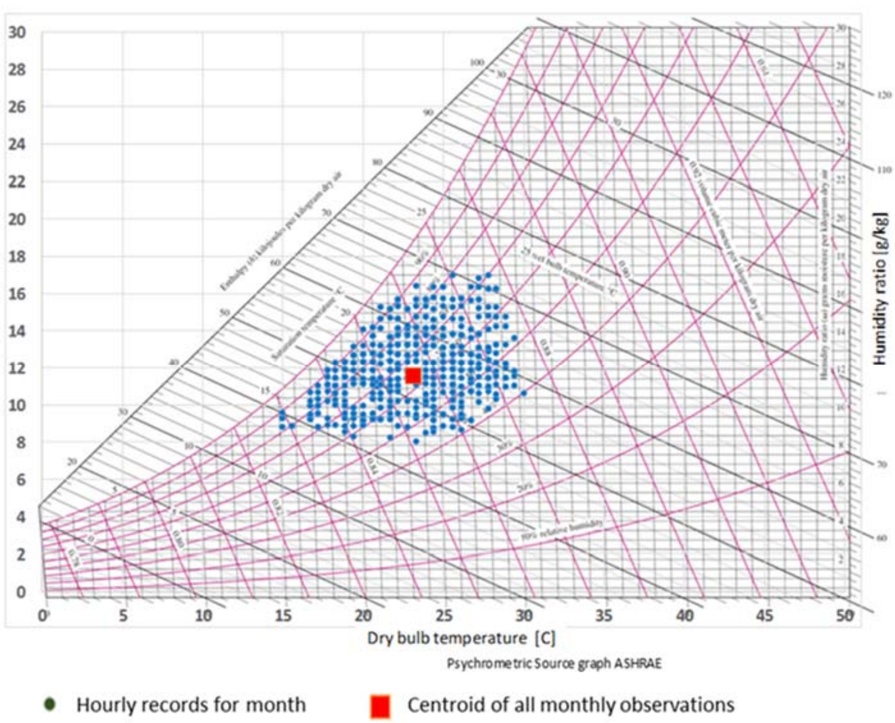


Figure B-6: Full record Psychrometric conditions – Monthly data for March

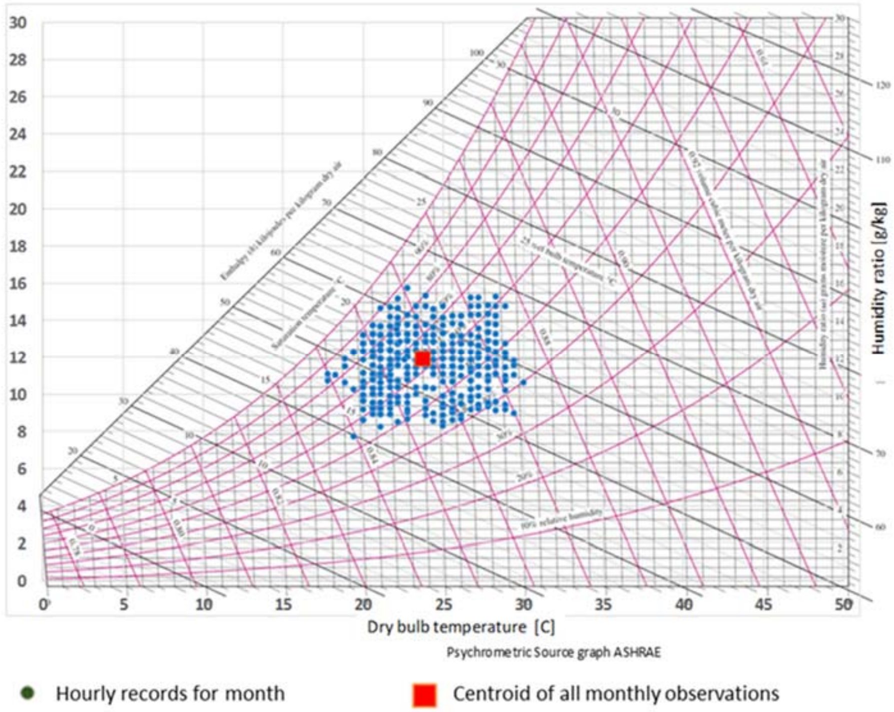


Figure B-7: Full record Psychrometric conditions – Monthly data for April

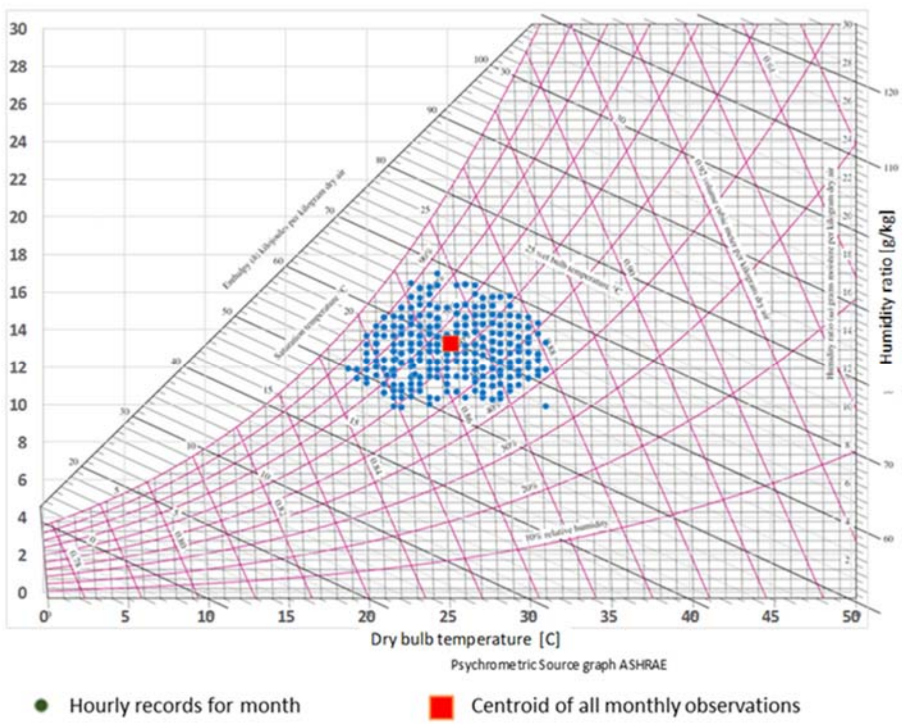


Figure B-9: Full record Psychrometric conditions – Monthly data for May

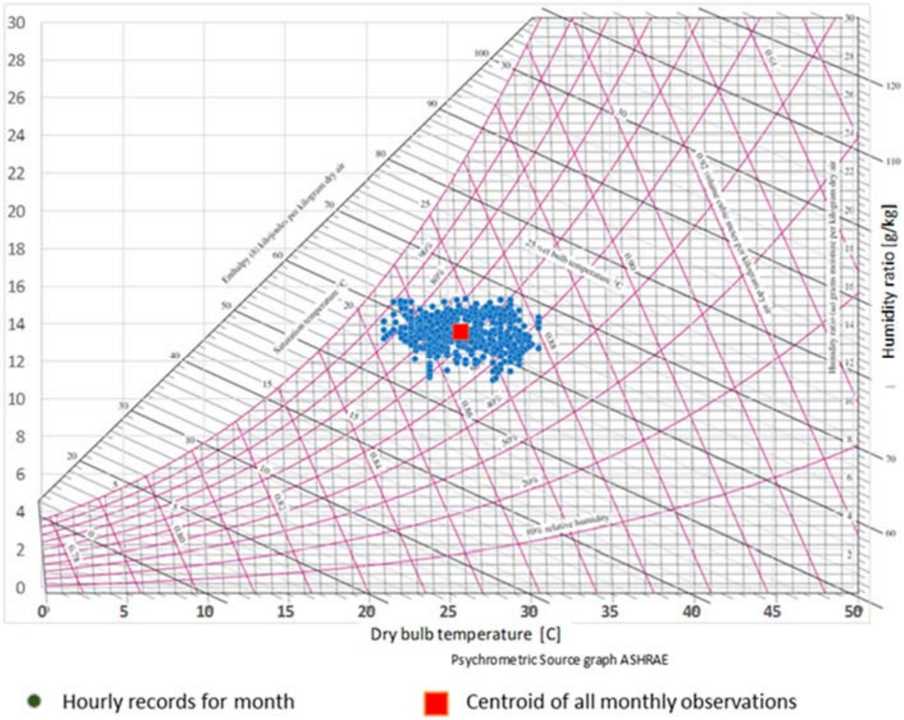


Figure B-8: Full record Psychrometric conditions – Monthly data for June

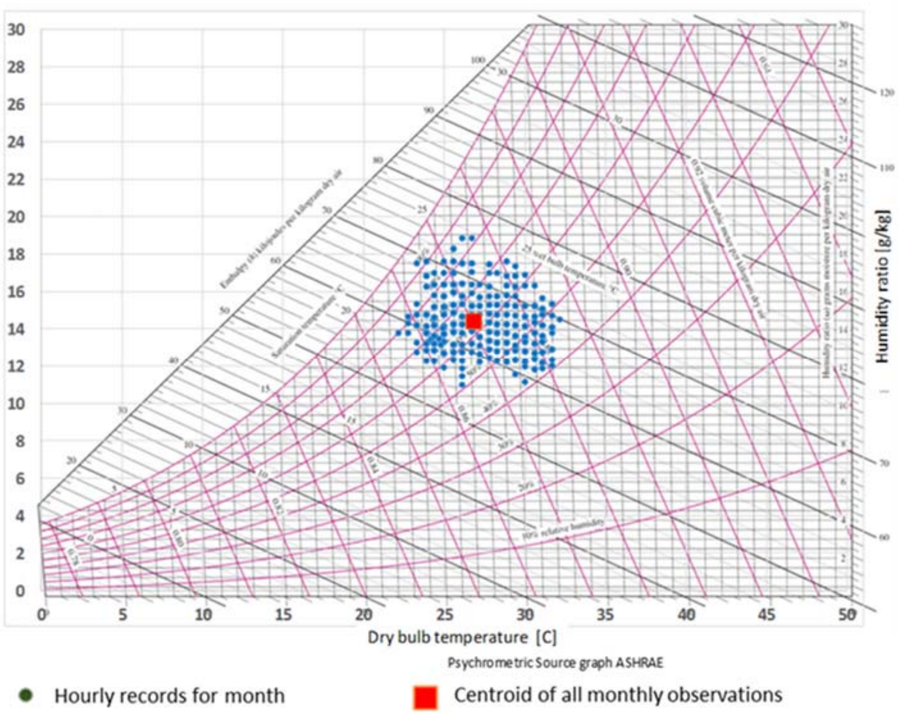


Figure B-10: Full record Psychrometric conditions – Monthly data for July

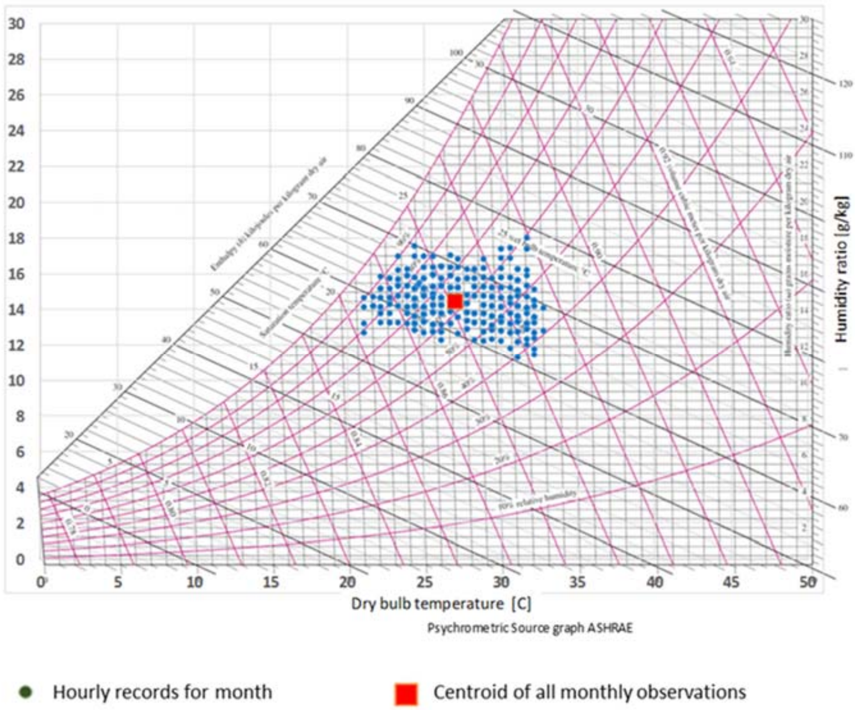


Figure B-11: Full record Psychrometric conditions – Monthly data for August

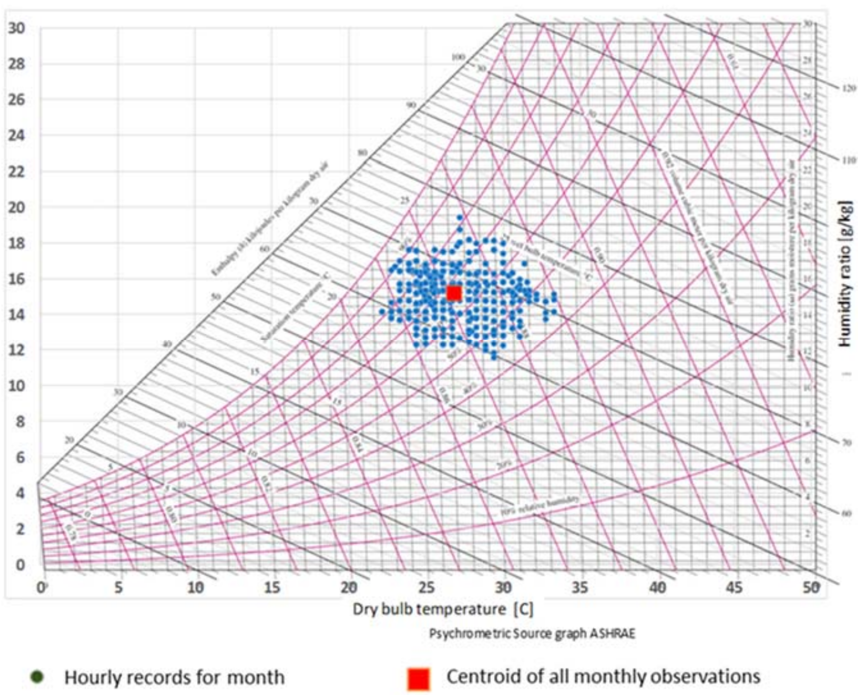


Figure B-13: Full record Psychrometric conditions – Monthly data for September

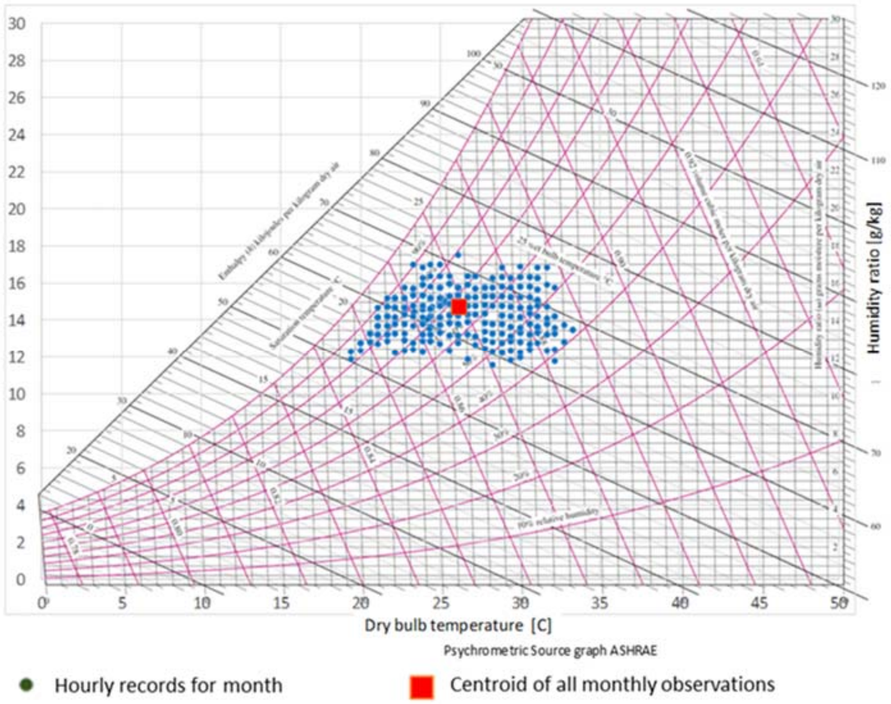


Figure B-12: Full record Psychrometric conditions – Monthly data for October

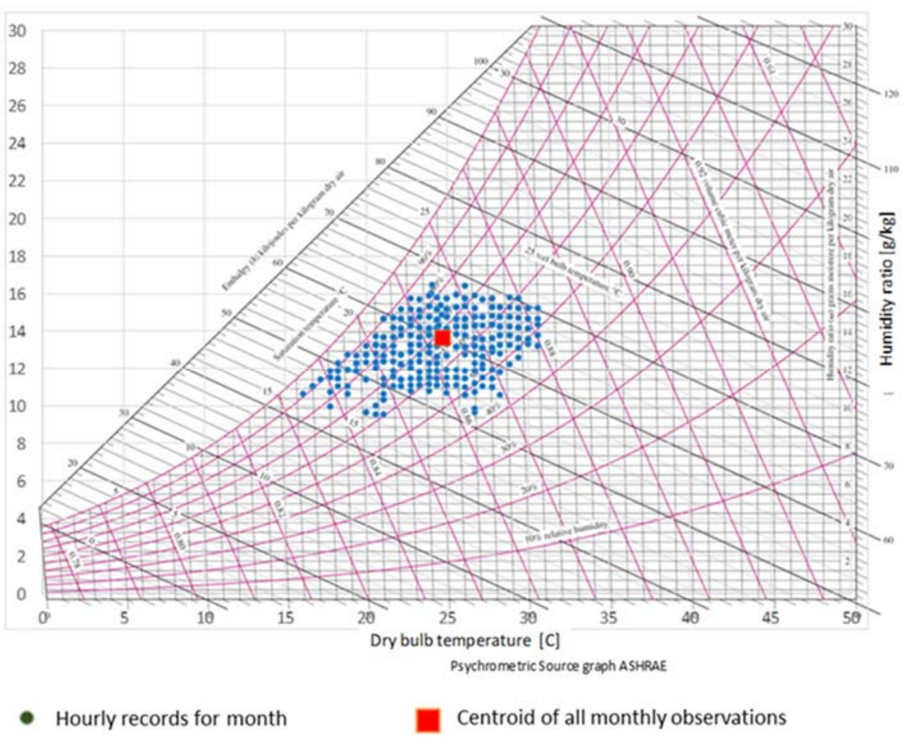


Figure B-14: Full record Psychrometric conditions – Monthly data for November

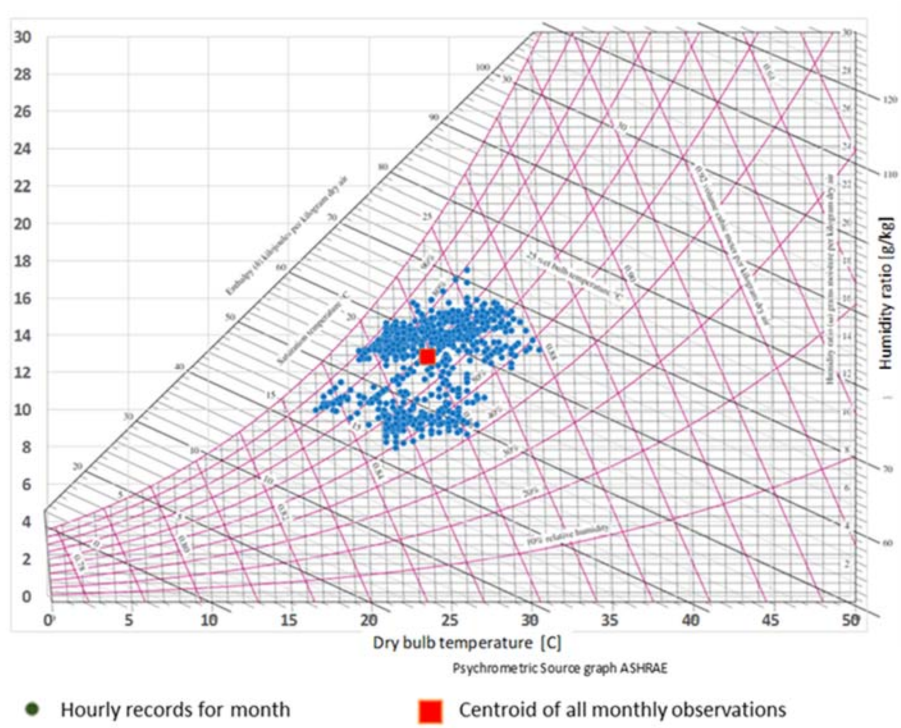


Figure B-15: Full record Psychrometric conditions – Monthly data for December

Psychrometric charts (11-h record)

period / month	DBT F	hum. Ratio g/kg	humidity grains / lb	RH %	WBT F	Dew Pnt. F
Year	76.8	13.329	93.260	58.7%	70.0	65.1
Jan	72.5	11.470	81.345	60.2%	66.3	61.3
Feb	73.1	13.363	94.525	67.4%	69.1	65.5
March	73.8	11.470	85.628	58.8%	67.8	62.7
April	74.8	11.852	82.731	55.8%	67.5	61.8
May	77.5	13.219	91.763	55.8%	70.1	64.7
June	78.6	13.488	94.822	57.6%	70.7	65.6
July	80.5	14.318	99.391	55.6%	72.2	66.9
Aug.	80.8	14.406	99.391	54.1%	72.4	66.9
Sept	80.4	15.110	104.527	59.8%	72.9	68.3
Oct.	79.3	14.622	100.733	56.9%	72.3	67.3
Nov.	76.6	13.550	95.277	61.0%	70.3	65.7
Dec.	74.7	12.790	90.158	63.1%	68.5	64.2

Figure B-16: Psychrometric data for the 11-h record
The data depited represtn the centroits of the data sets

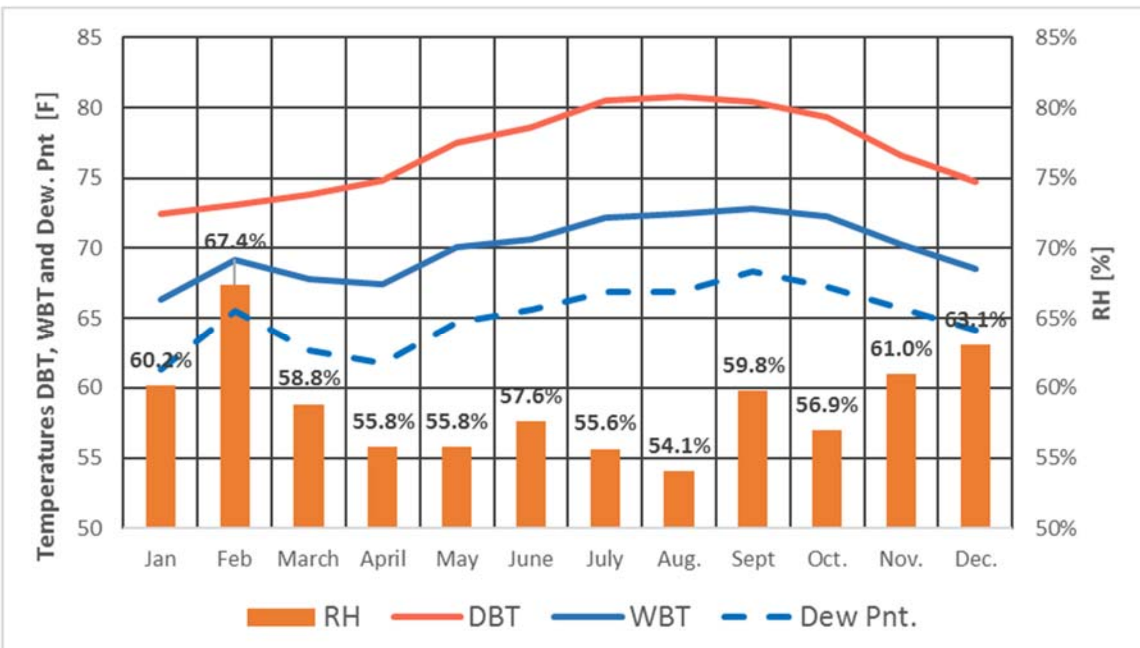


Figure B-17: Psychrometric data for the 11-h record; The data depited represtn the centroits of the data sets

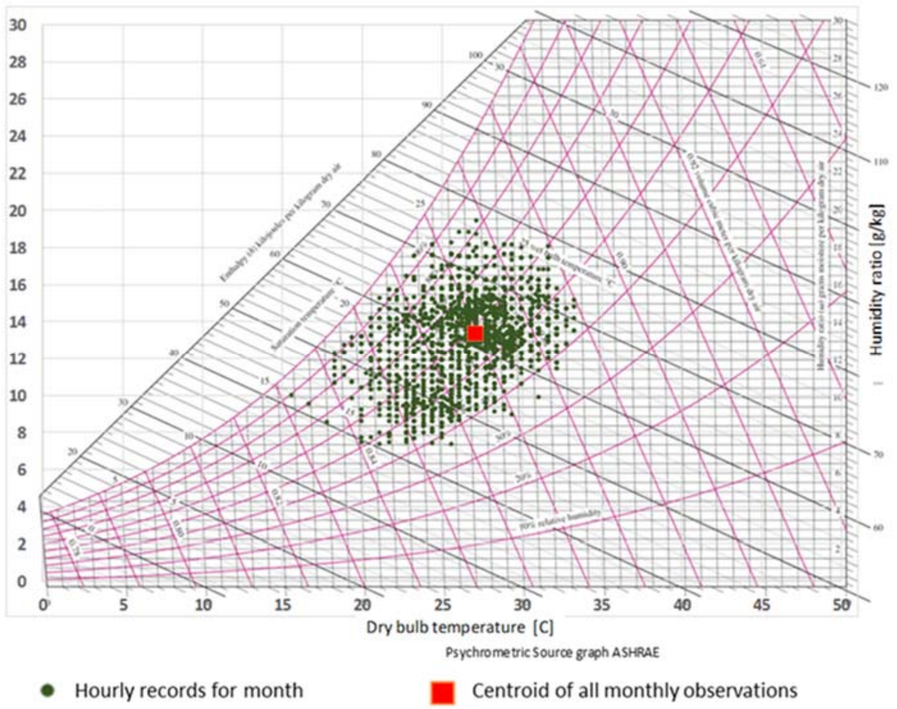


Figure B-18: 11-h record Psychrometric conditions – for all data points in the year;

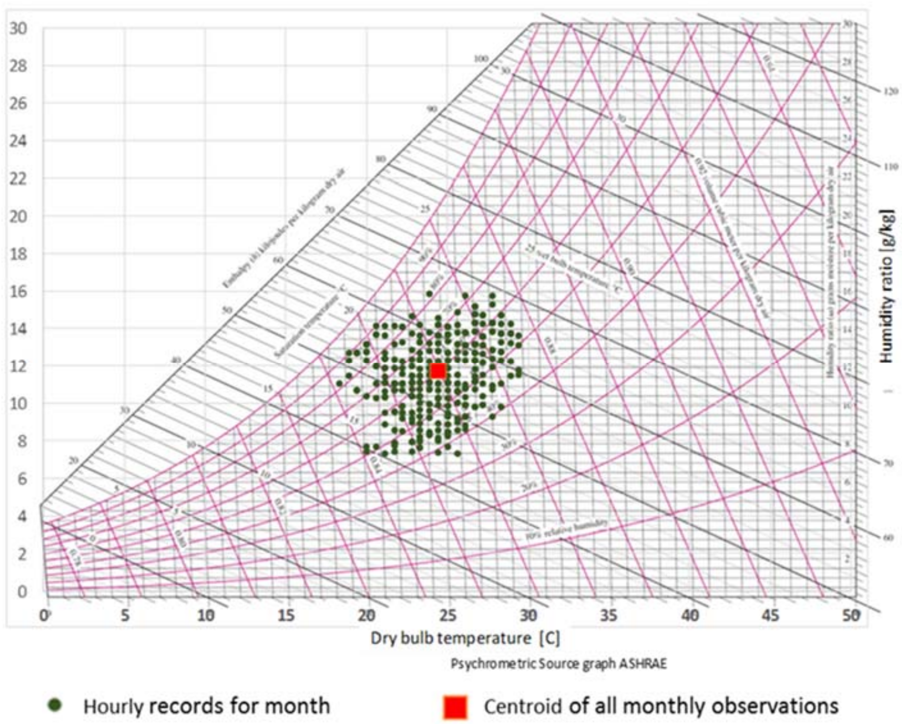


Figure B-19: 11-h record Psychrometric conditions – Monthly data for January

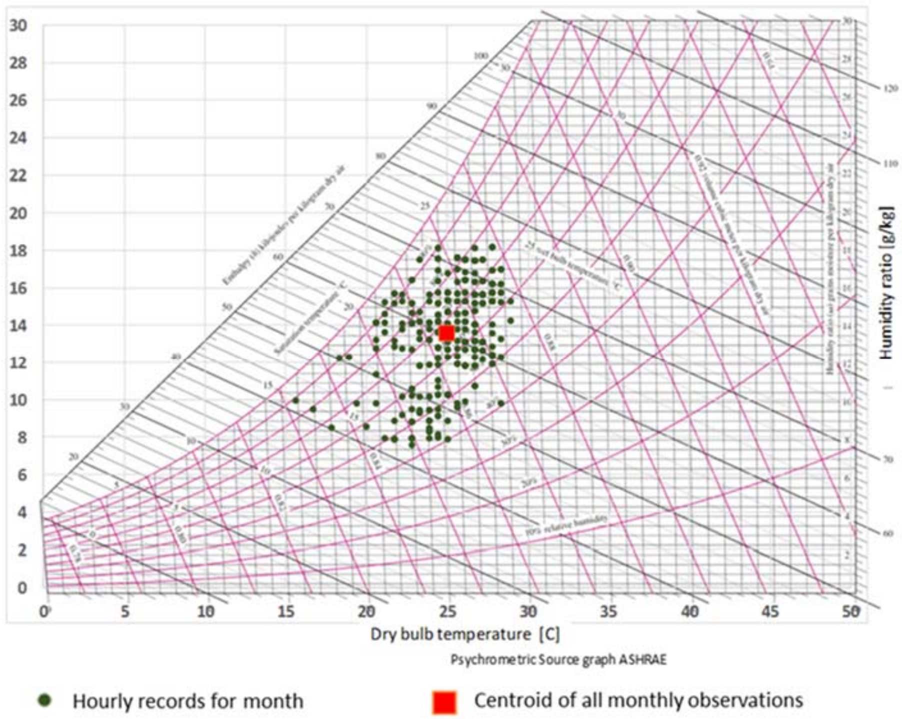


Figure B-20: 11-h record Psychrometric conditions – Monthly data for February

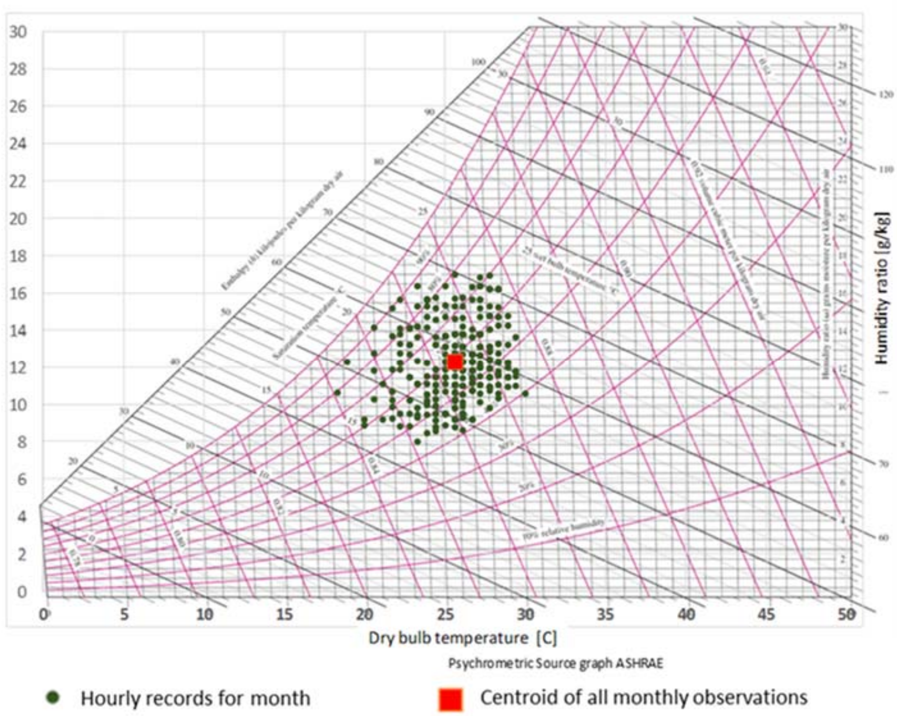


Figure B-21: 11-h record Psychrometric conditions – Monthly data for March

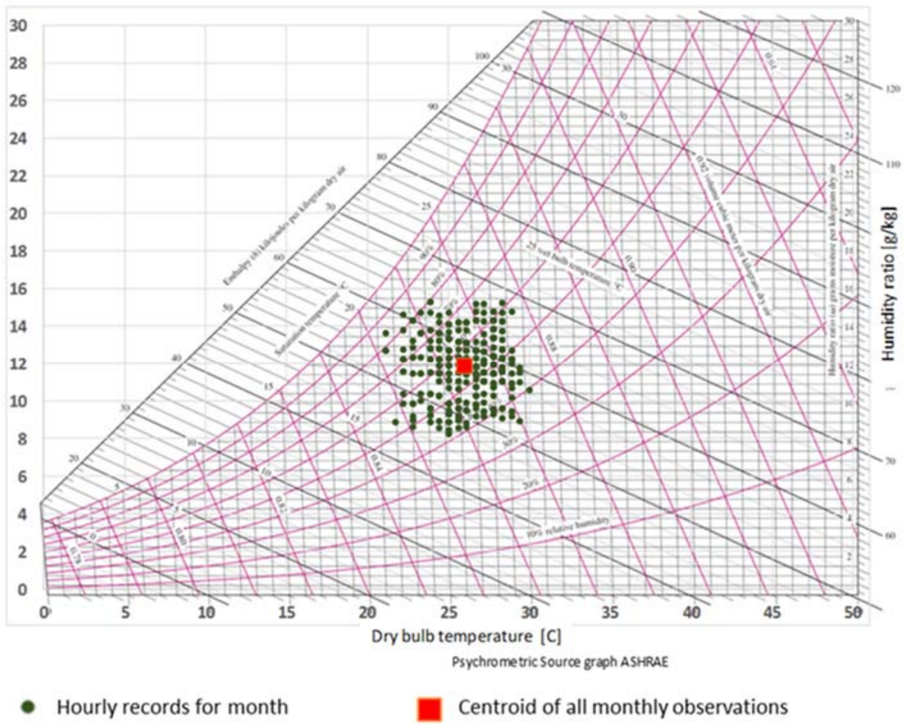


Figure B-22: 11-h record Psychrometric conditions – Monthly data for April

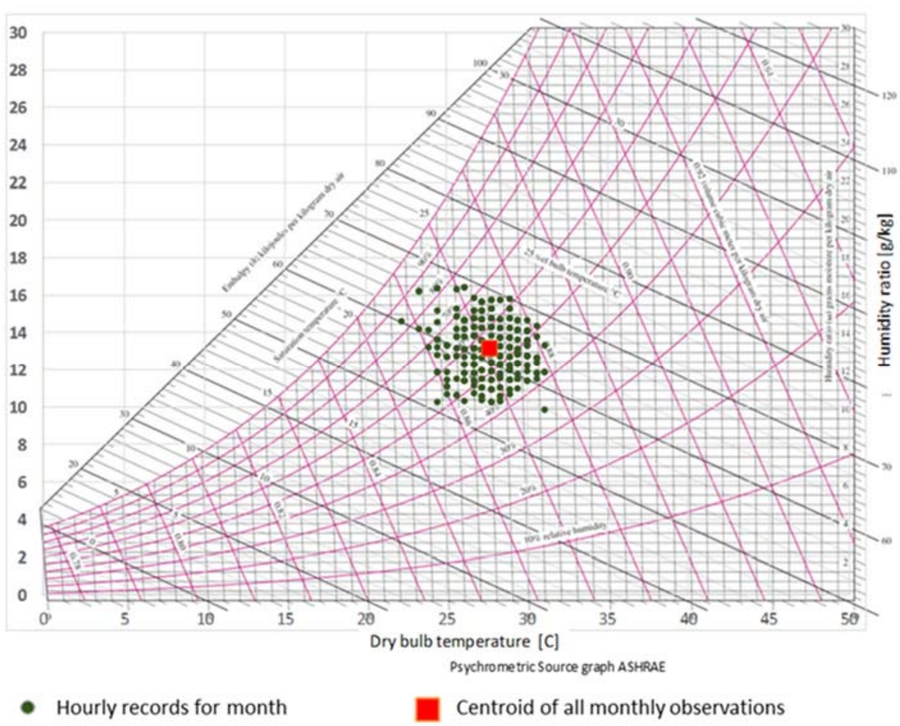


Figure B-23: 11-h record Psychrometric conditions – Monthly data for May

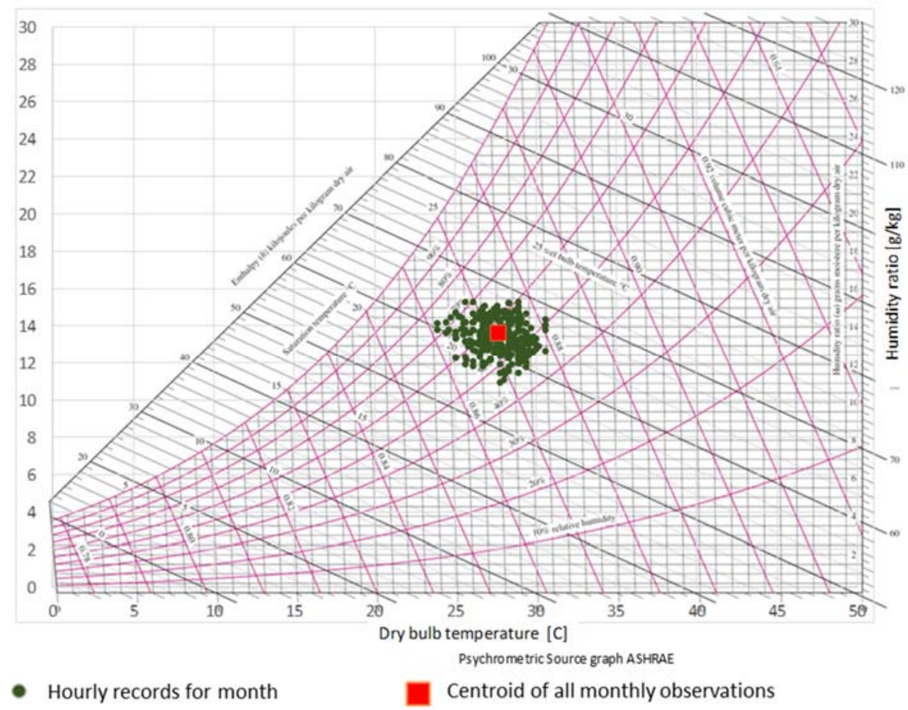


Figure B-24: 11-h record Psychrometric conditions – Monthly data for June

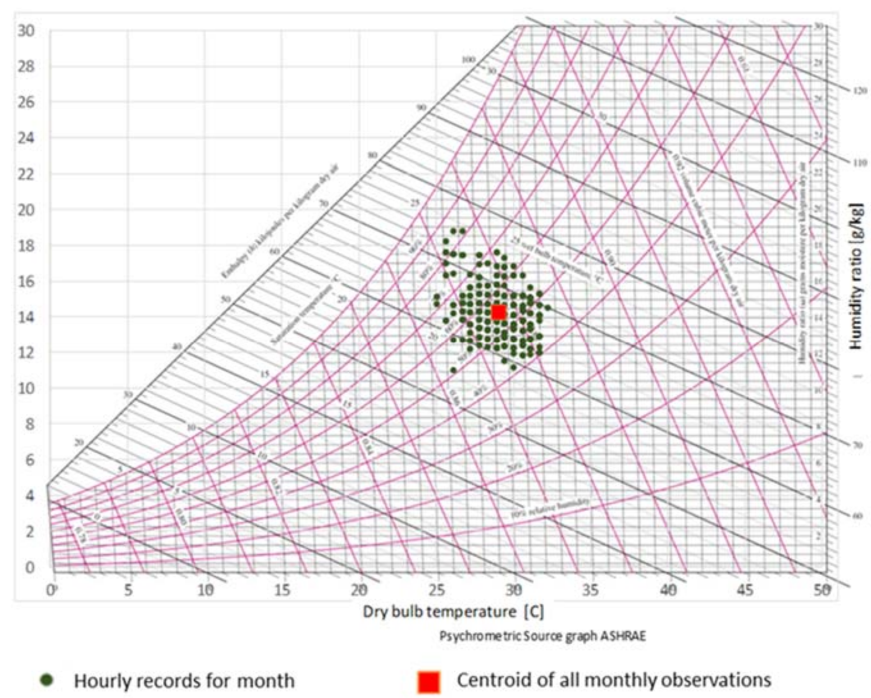


Figure B-25: 11-h record Psychrometric conditions – Monthly data for July

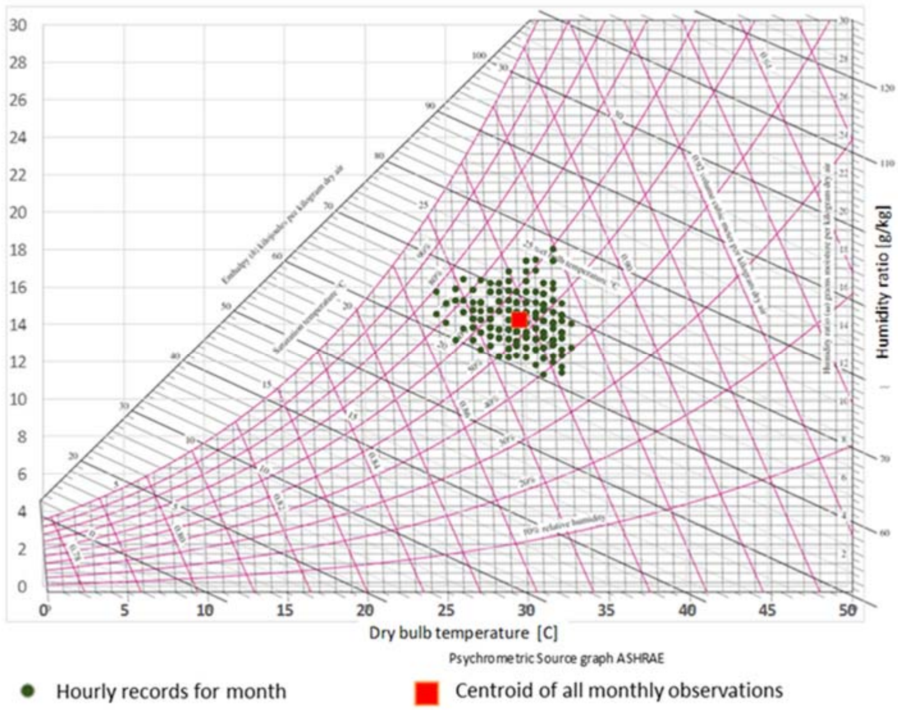


Figure B-26: 11-h record Psychrometric conditions – Monthly data for August

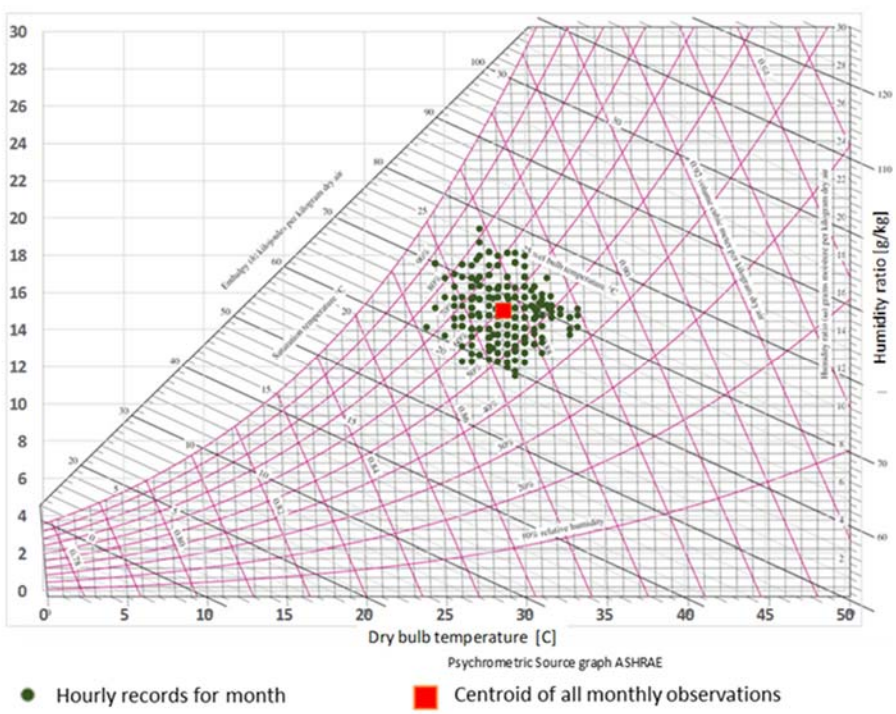


Figure B-27: 11-h record Psychrometric conditions – Monthly data for September

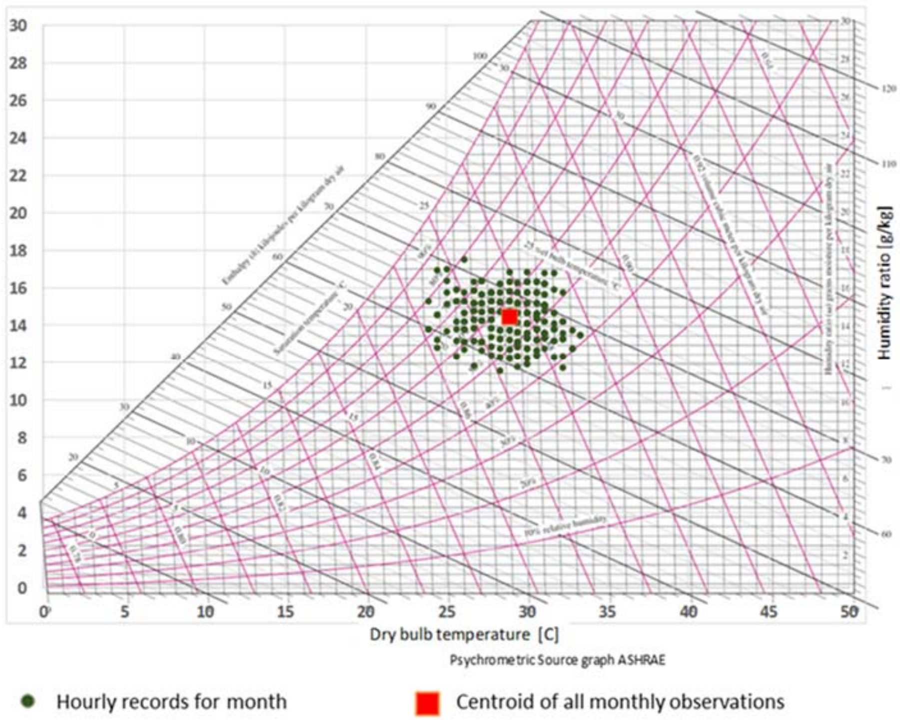


Figure B-28: 11-h record Psychrometric conditions – Monthly data for October

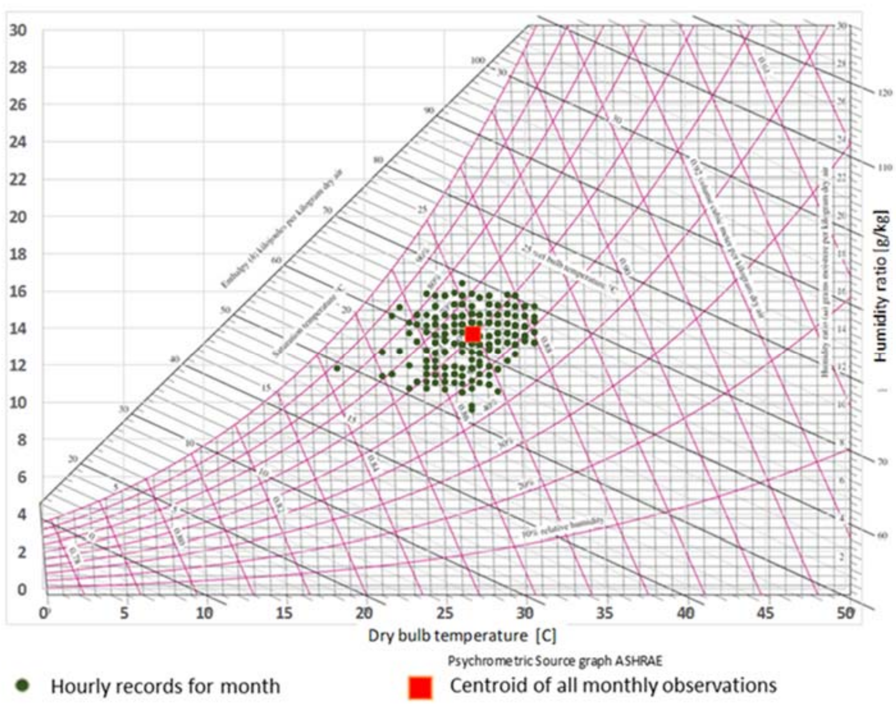


Figure B-29: 11-h record Psychrometric conditions – Monthly data for November

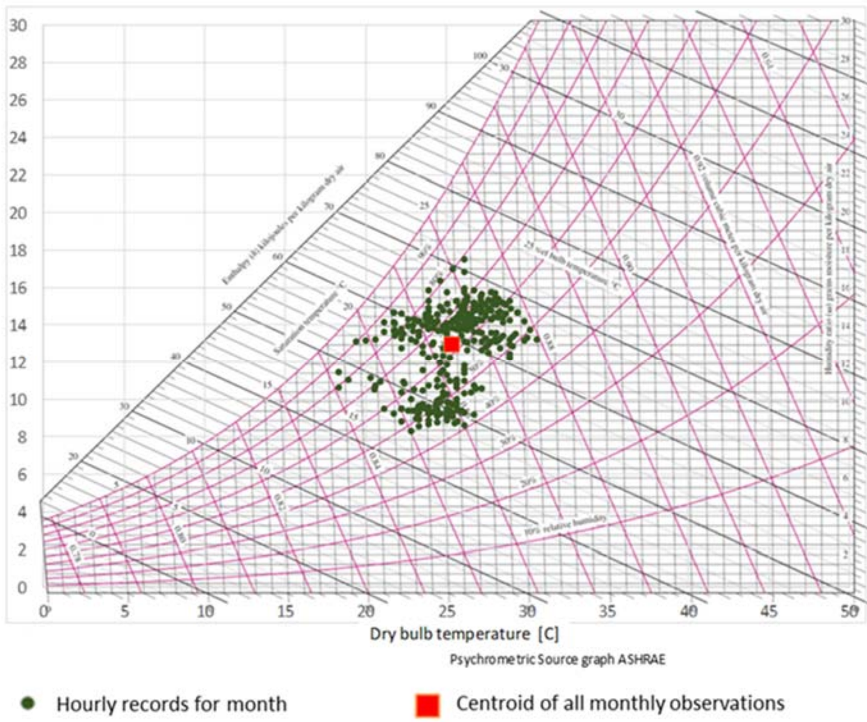


Figure B-30: 11-h record Psychrometric conditions – Monthly data for December

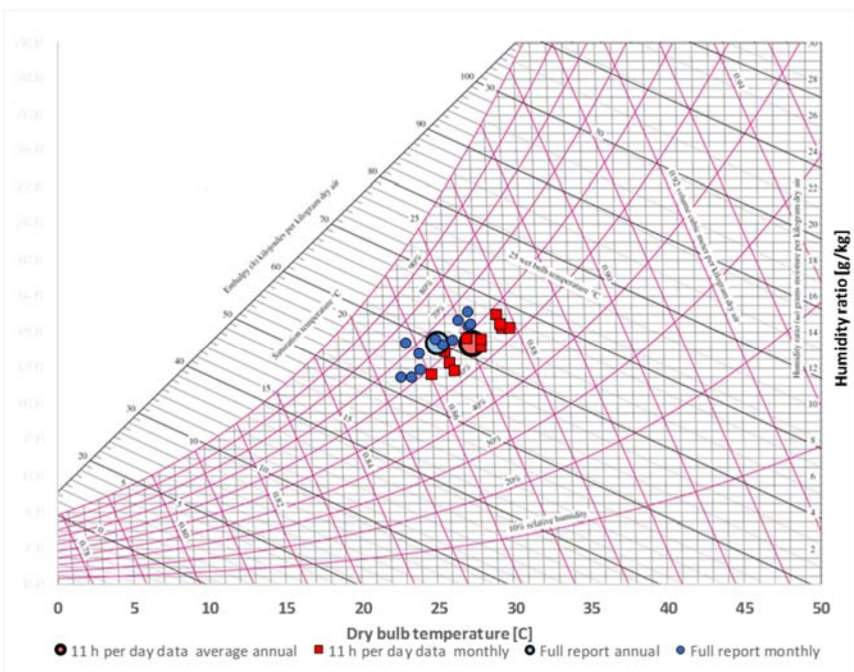


Figure B-31: Psychrometric conditions Centroids for the full record and the 11-h record

APPENDIX C

TYPICAL WEATHER DATA FOR HONOLULU:

Direct Normal Irradiance (DNI)

Appendix C

Typical weather data for Honolulu

This Appendix C contains:

Direct Normal Irradiance (DNI)

Direct Normal Irradiance (DNI)

	average [Wh/m^^{2}]	Max [Wh/m^^{2}]	Min [Wh/m^^{2}]	Stnd. Dev [W/hm^^{2}]	sum [Wh/m^^{2}]	hours [h]
Jan	394	964	1	299	142,513	362
Feb	415	936	1	286	123,950	299
March	388	908	1	261	139,772	360
April	383	986	2	261	148,896	389
May	388	926	1	288	164,311	424
June	379	880	1	267	170,275	449
July	425	934	1	292	184,112	433
Aug	472	944	1	260	195,868	415
Sept	420	913	5	255	156,144	372
Oct	405	966	1	285	157,113	388
Nov	372	954	1	269	132,866	357
Dec	409	964	1	307	139,147	340
					1,854,967	4,588

Figure C-1:
DNI averages for the month

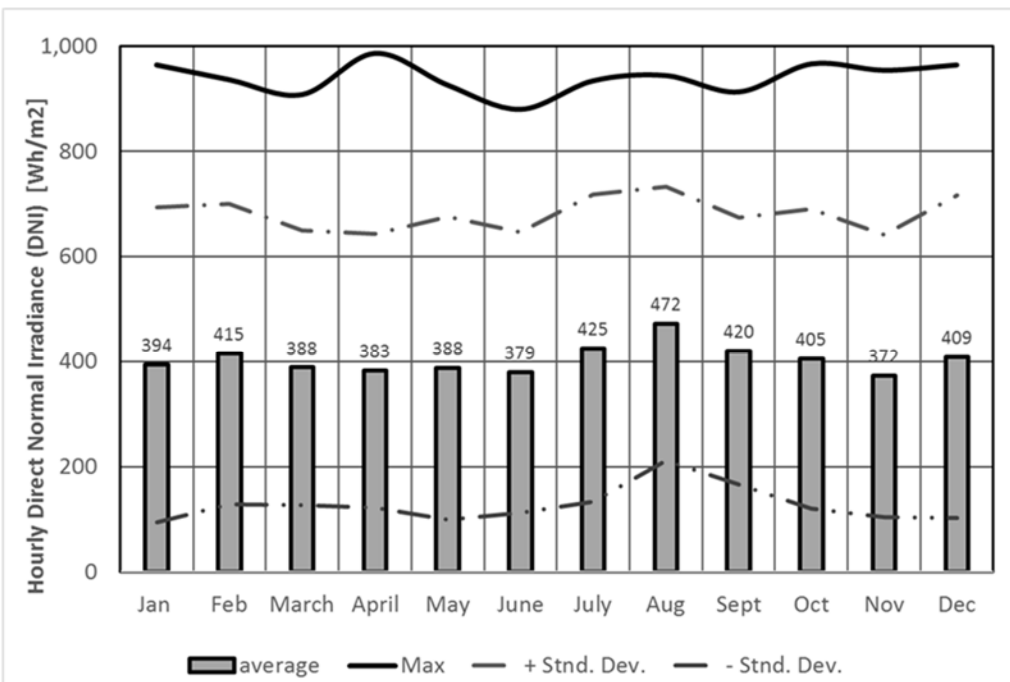


Figure C-1: DNI averages for the month

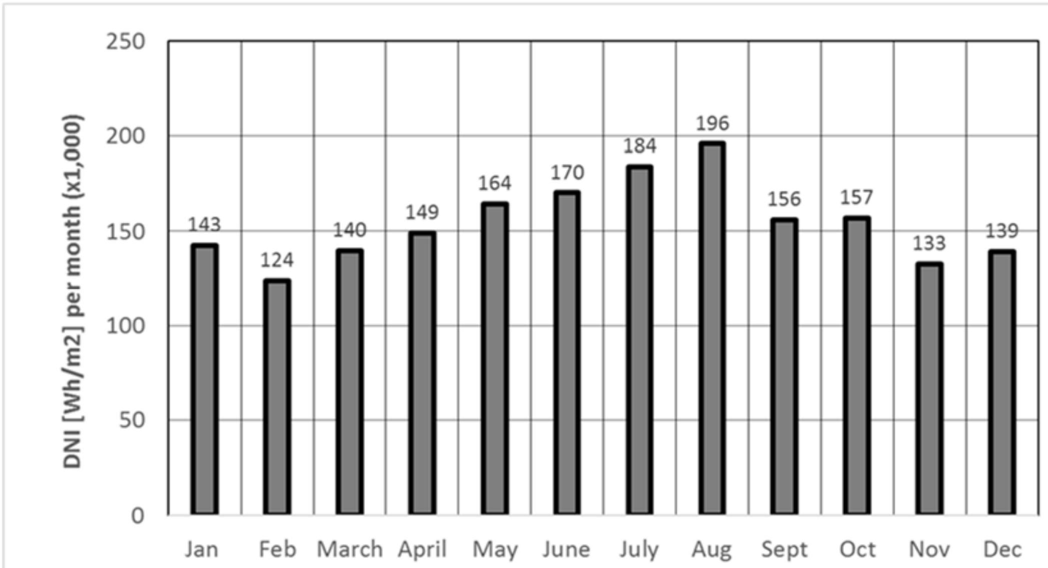


Figure C-3: DNI Full record; The data depicted are the total solar energy in the month

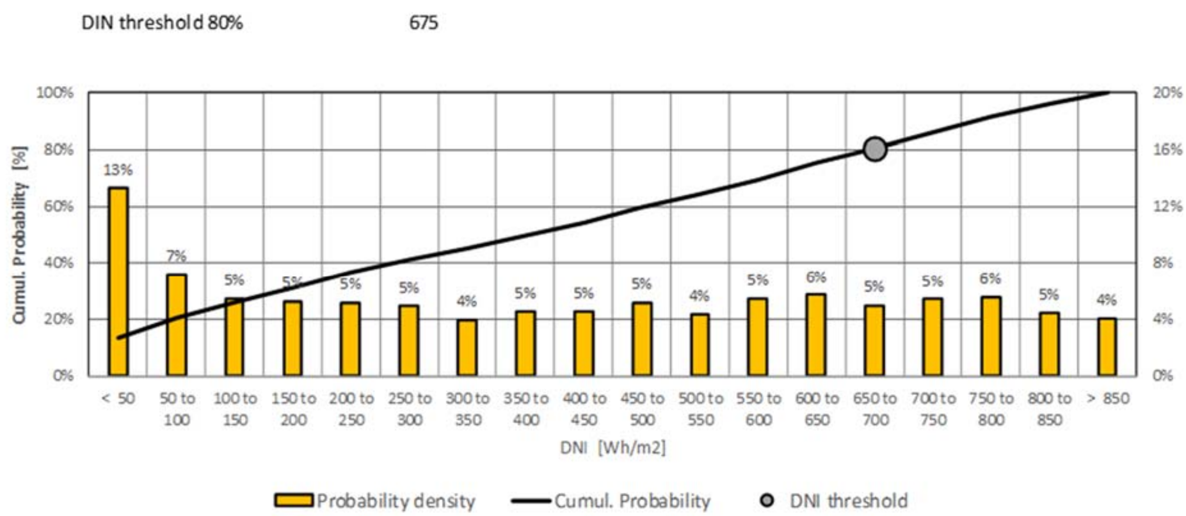


Figure C-4: DNI Full record – for all data points in the year

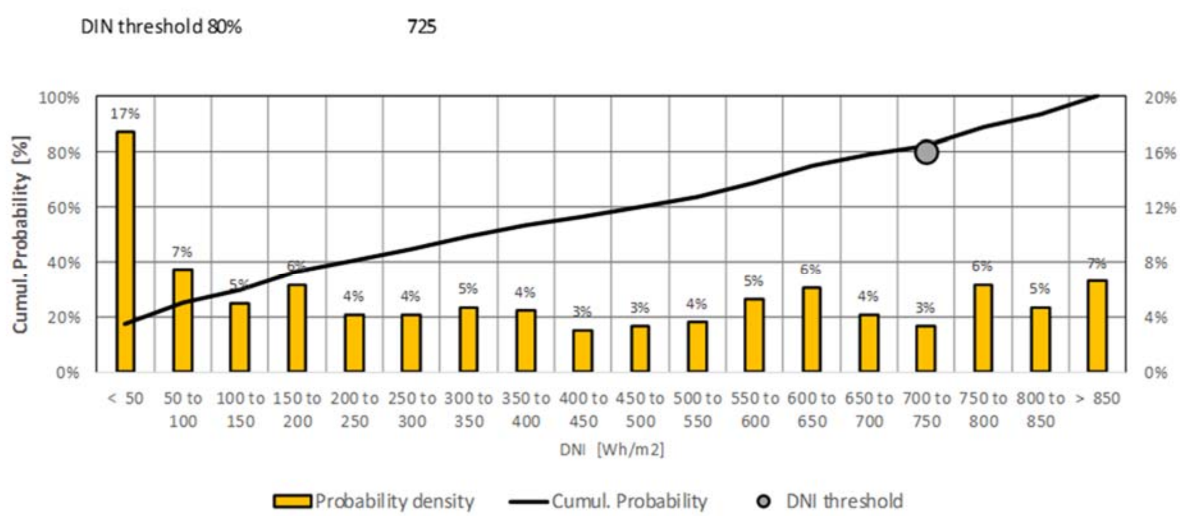


Figure C-5: DNI Full record – Probability for the month of January

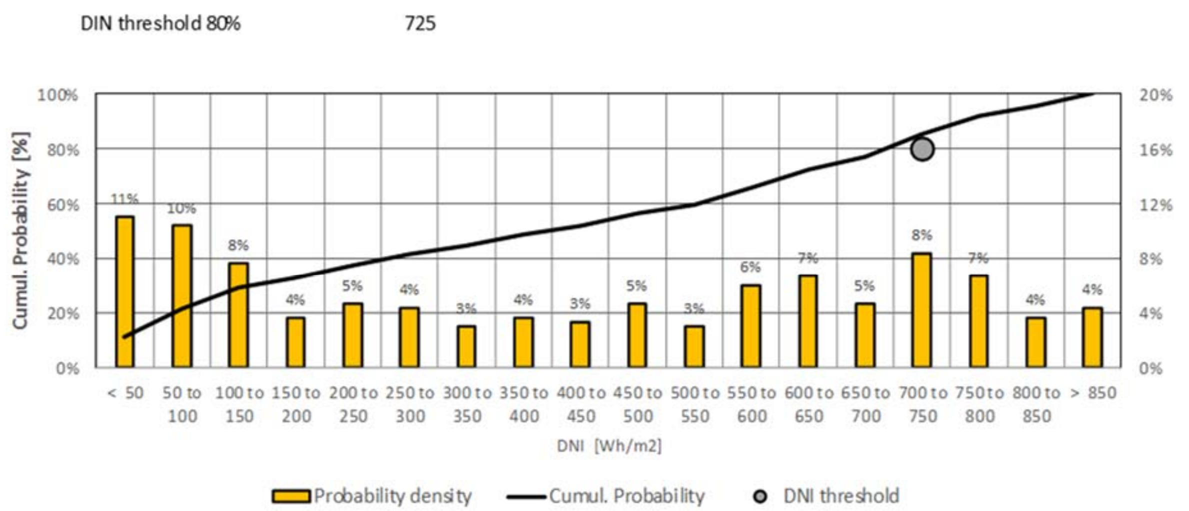


Figure C-6: DNI Full record – Probability for the month of February

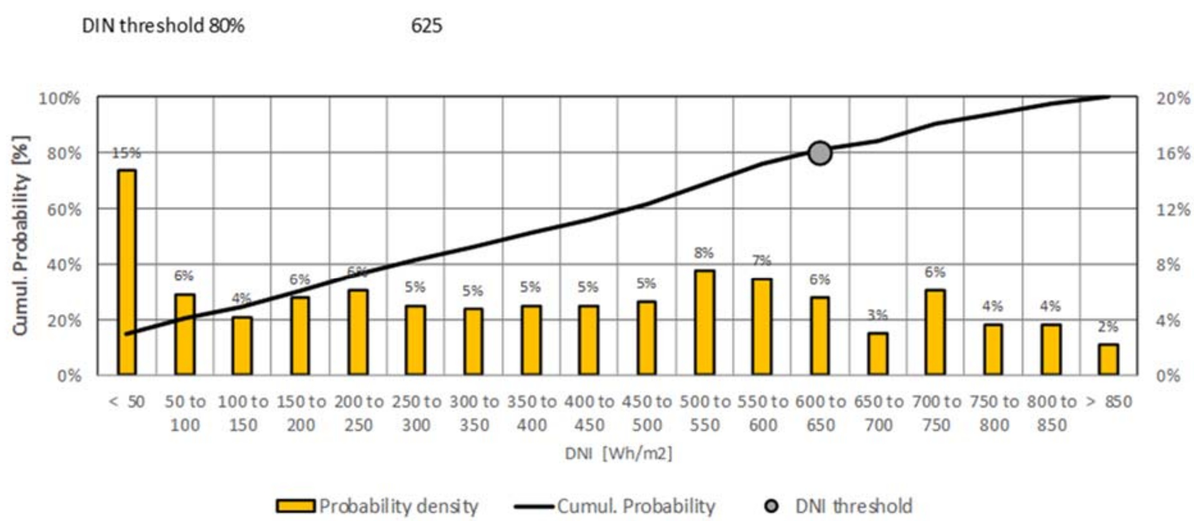


Figure C-7: DNI Full record – Probability for the month of March

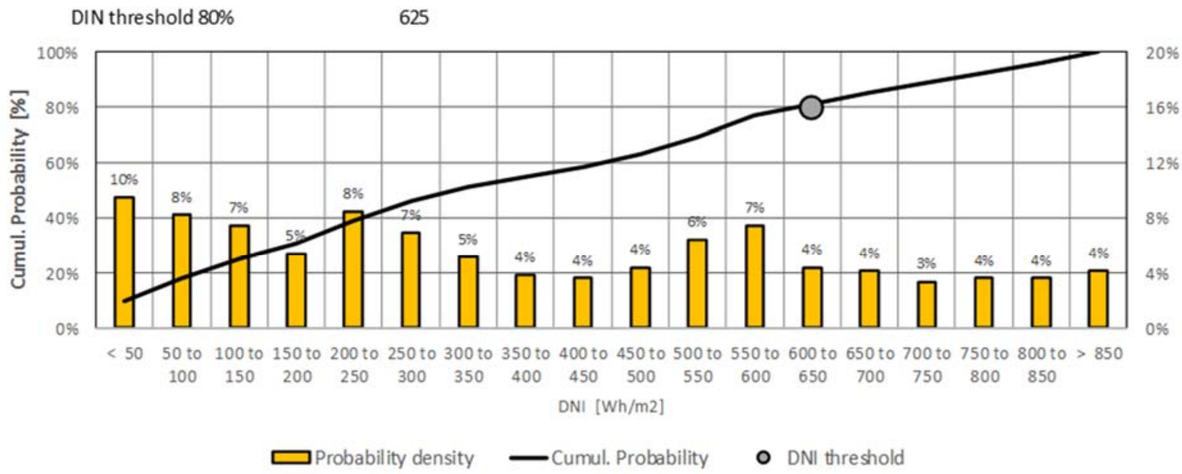


Figure C-8: DNI Full record – Probability for the month of April

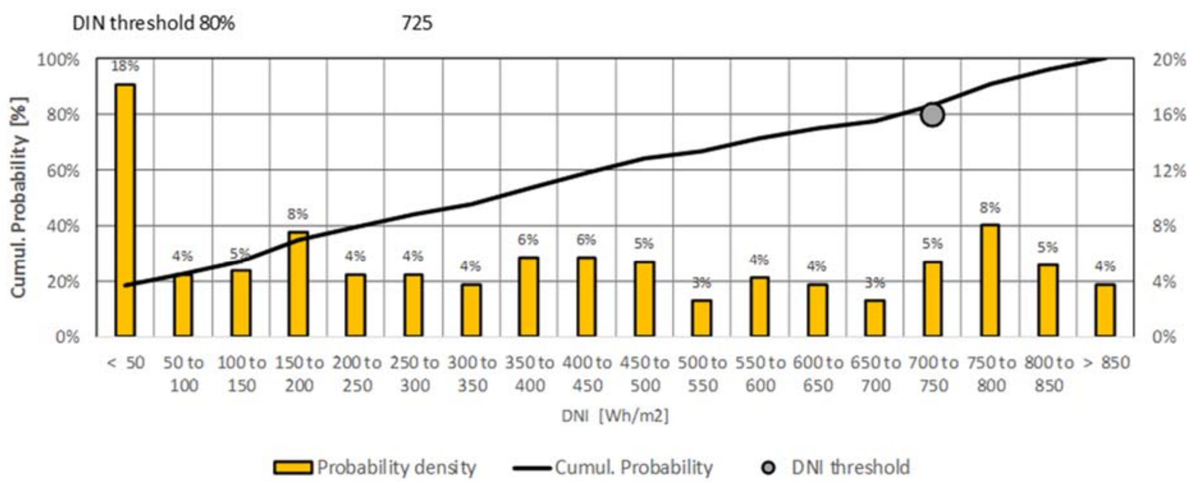


Figure C-9: DNI Full record – Probability for the month of May

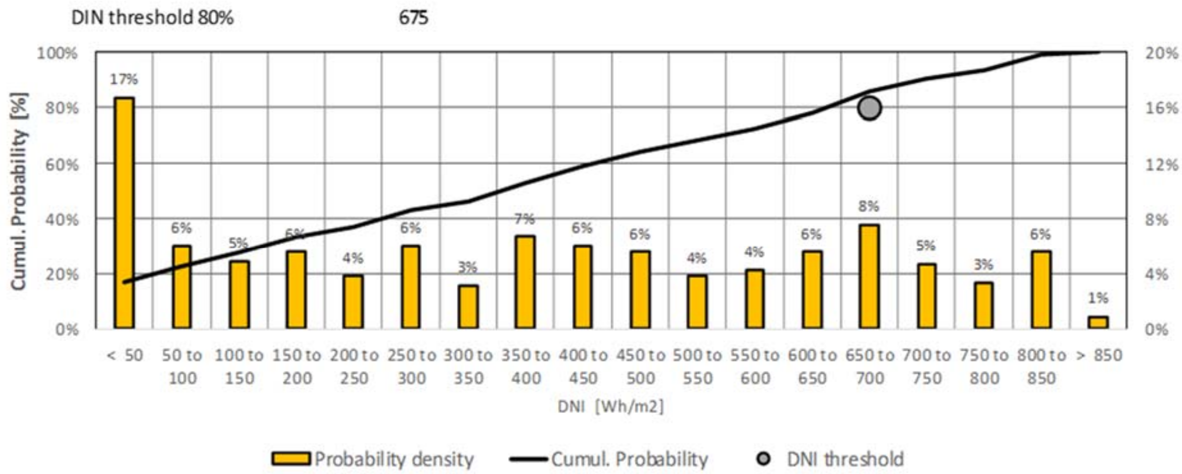


Figure C-10: DNI Full record – Probability for the month of June

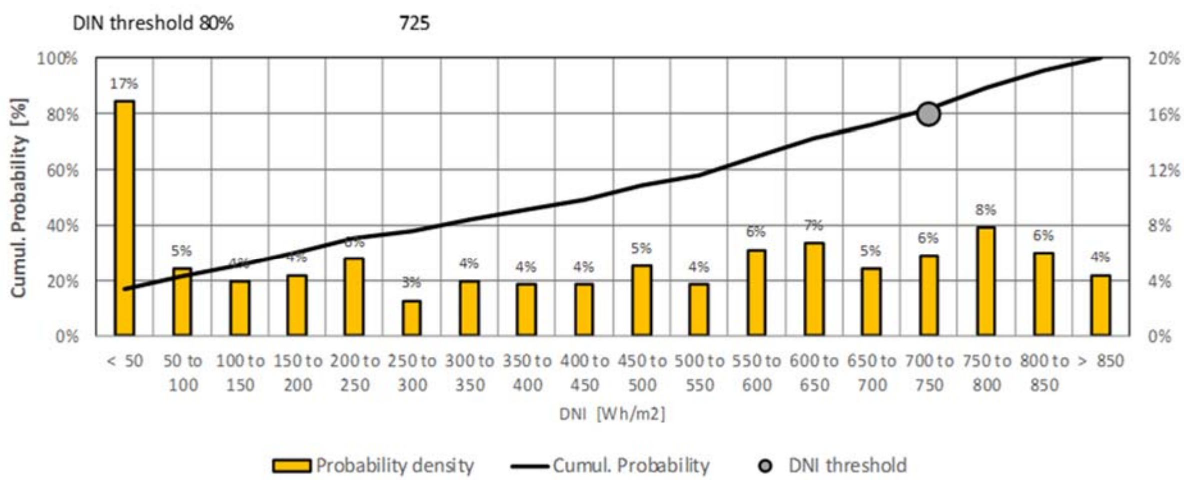


Figure C-11: DNI Full record – Probability for the month of July

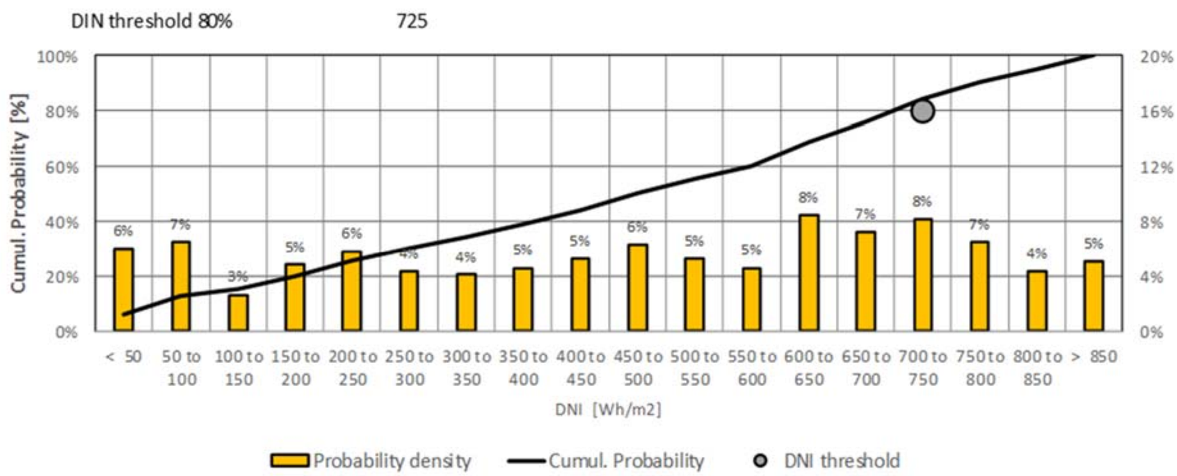


Figure C-12: DNI Full record – Probability for the month of August

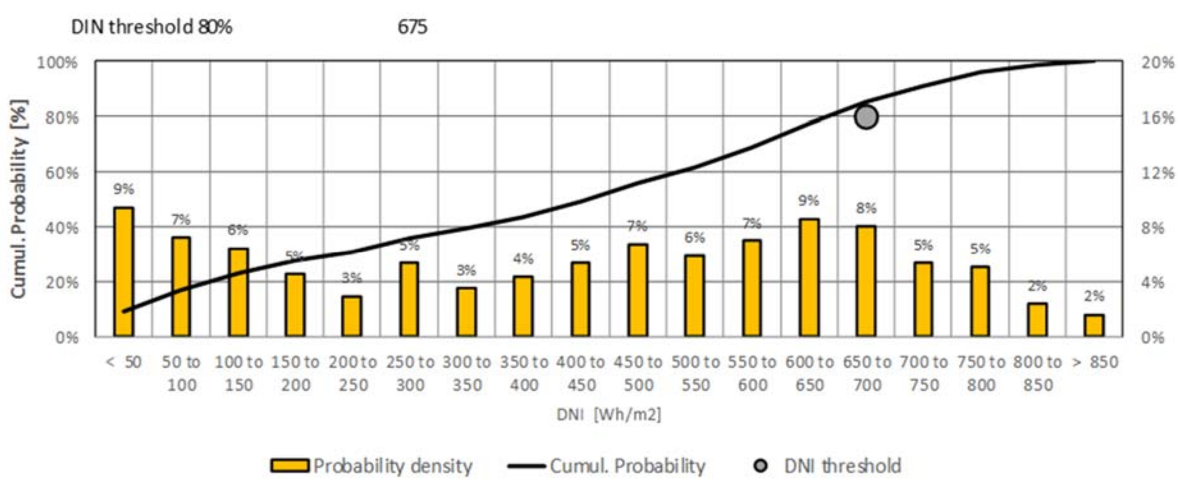


Figure C-13: DNI Full record – Probability for the month of September

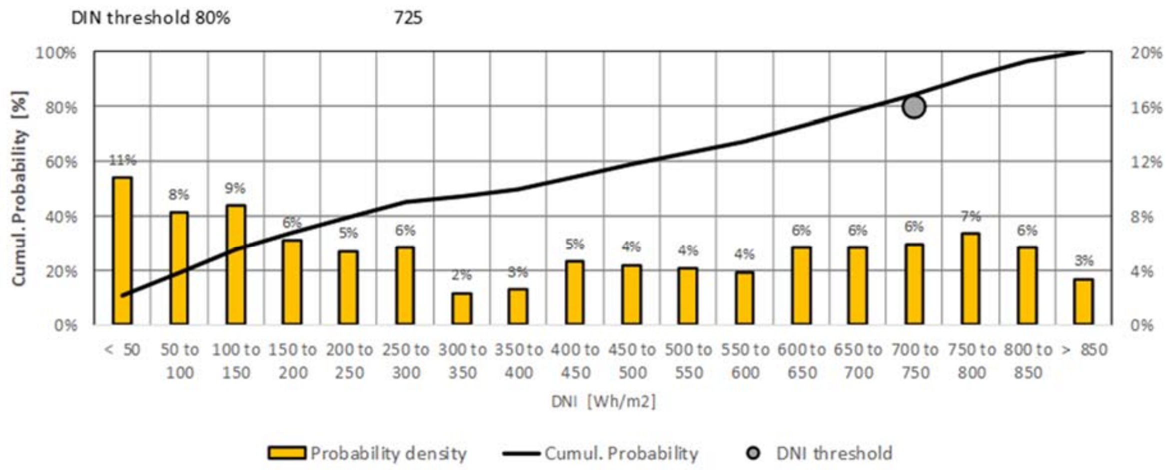


Figure C-14: DNI Full record – Probability for the month of October

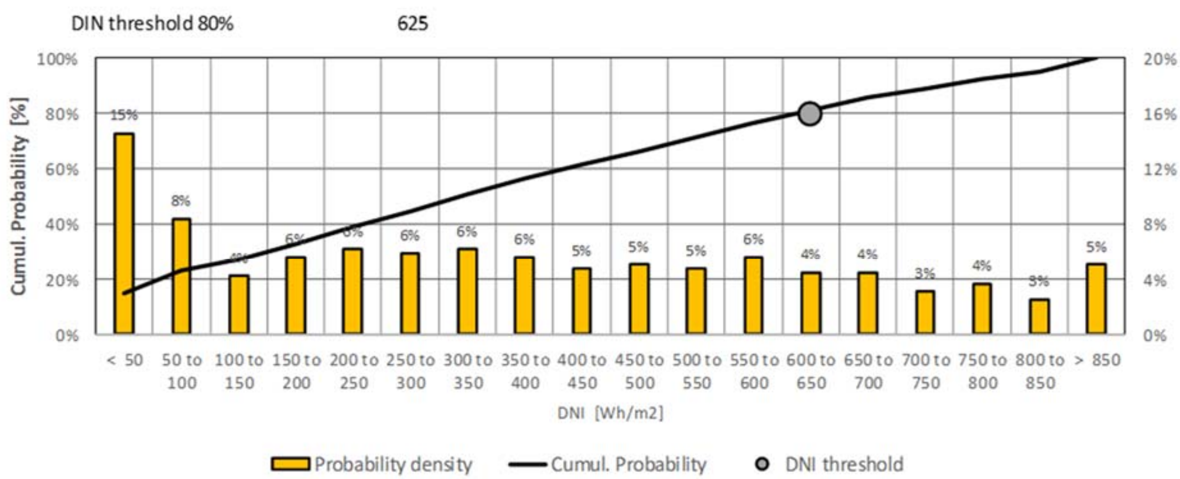


Figure C-15: DNI Full record – Probability for the month of November

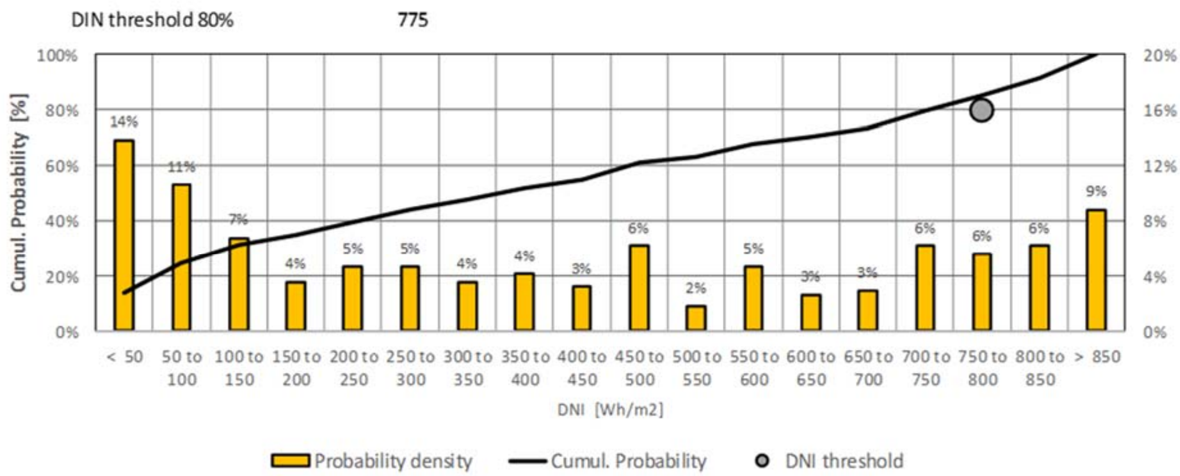


Figure C-16: DNI Full record – Probability for the month of December

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