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Improving the energy efficiency of dehumidification technology at a large facility in Florida

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Improving the Energy Efficiency of Dehumidification Technology at a Large Facility in Florida

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Abstract

This thesis examines energy use and management of twenty heat pipes used in dehumidification systems at a large (10,000+ acre) facility in Florida. Eleven of the twenty heat pipes use electric strip heaters that, when activated, consume 693.8 kW of electrical power from the grid. Solar photovoltaics, specifically a silicon monocrystalline cell with 22.5% efficiency, were considered as a means to provide an alternative energy source and opportunity for cost savings for 11 of the heat pipes (Sunpower, 2011). The remaining nine heat pipes use hot water heaters for which alternative energy sources were not considered.

Data gathered and analyzed include weather, solar irradiance, PV size and cost, utility incentives, emissions, fuel consumption, energy cost, and heat pipe operating parameters. These data were used to calculate the (1) annual electricity cost for the heaters, (2) installed cost for enough PV to offset electric heater energy use, (3) surface area needed to install the estimated PV system, (4) one-time and ongoing financial incentives, (5) avoided energy savings, (6) avoided fuel usage and emissions, and (7) the undiscounted payback period of the various equipment investments.

Savings were calculated to be almost $600,000 annually (approximately $145,000 attributable to the heaters) if PV were to power the heaters 125 days of the year and ancillary systems at other times. The cost of an appropriately sized PV system (4.57 acres with between 8,553 and 15,205 PV panels depending on panel size) was estimated at $3,228,806 assuming $150,000 of electric utility incentives. It was also estimated that the photovoltaic (PV) system could earn $85,087 in annual tax credits through the Florida Renewable Energy Production Tax Credit program, and that the undiscounted payback

period would be about five years. Further, the use of 728,350 US gallons of oil, and the emission of 13,656.6 lbs of SO_2, 9,104.37 lbs of NO_x, and 10,843,300 lbs of CO_2, could also be avoided.

It is recommended that the installation of PV energy generation capabilities be further investigated. It is also recommended that further research be performed to obtain accurate costs and benefits of integrating solar thermal into the hot water heaters at the facility because of the complexity of integrating solar thermal into the existing hot water heaters, the lack of readily available price information regarding solar thermal heating, and the fact that the hot water heaters consume about $170,000 per year.

Chapter 1: Introduction

Introduction

Dehumidification is the most energy intensive in any Heating, Ventilation, and Air Conditioning (HVAC) sector because it needs to cool incoming air to either condense out the excess moisture or to cool it after it has gone through a desiccant dehumidification process (ASHRAE, 2012). This will become even more of an issue as developing countries continue to raise their standard of living (thus increasing new HVAC system installations) (Conti, et al., 2013). Therefore, research has been done into making HVAC and specifically dehumidification more efficient, and one of the most elegant solutions and the one examined in detail here is the heat pipe (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011), (Brooke, Optimizing Wrap Around Heat Pipes, 2007). Heat pipes are used extensively for dehumidification in several buildings within a very large (10,000+ acre) facility in Florida. Although the heat pipes have saved a great deal of money and energy over the time they have been installed (the Florida Beach-Side Resort study below will show this), there is room for improvement, and some options for improvement are examined in this dissertation.

Purpose

This research addresses HVAC systems with an emphasis on the energy efficiency of dehumidification practices and technologies utilized within a large facility in Florida. The purpose of this work is to examine these technologies and practices (specifically where heat pipes are used) and determine whether they provide the most energy-efficient means to dehumidify conditioned space(s); if they are, describe why the alternatives are inferior; and if they are not then describe and recommend alternatives that may provide a more efficient means of dehumidification. The purpose of this effort is to consider dehumidification technologies on their own individual merits, and also to determine whether, by supplementing them with sustainable energy technologies and/or practices (which will be discussed later), the dehumidification process can be made more efficient.

Justification

As the global population increases and the climate continues to change, an increasing number of people are moving into urban areas (World Health Organization, 2013). One hundred years ago less than 20% of people lived in cities, in 1990 that number had jumped to 40%, in 2010 it had again jumped to 50%, and it is expected to reach 60% by 2030 and 70% by 2050 (World Health Organization, 2013). As urban populations grow, so does the standard of living of developing countries and with this, electricity consumption rises. According to the Energy Information Administration's *International Energy Outlook 2013*, world energy consumption grew from less than400 quadrillion Btu (quads) in 1990 to 524 quads in 2010, and is projected to increase to 820 quads by 2040 with an average worldwide increase of 1.5% annually (Conti, et al., 2013). The nations involved in the study were split into two groups – OECD countries (generally considered the "developed" countries[1]) and non-OECD countries (generally considered "developing" countries[2]), as is shown in Figure 1. Energy consumption growth in non-OECD countries has outpaced that of OECD countries and is projected to do so at a growing rate through

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[1] Australia, Austria, Belgium, Canada, Chile, Czech Republic, Denmark, Estonia, Finland, France,
[2] i.e. China, India, and the African, Asian, and Middle Eastern nations not in the OECD

2040 (Conti, et al., 2013). As these countries continue to develop, resulting in greater urbanization, more buildings will be constructed, resulting in greater utilization of HVAC systems. Since 20-40% of energy consumption in developed nations occurs in buildings,

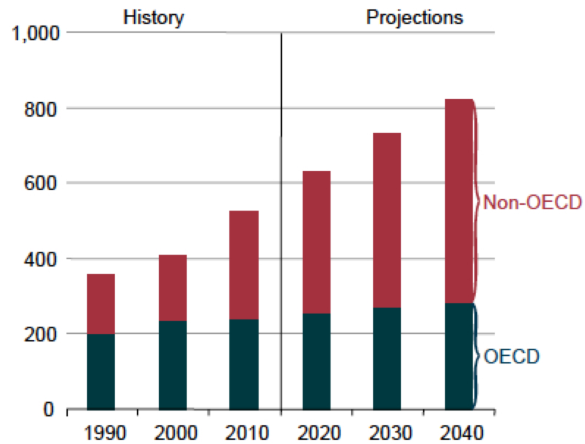

Figure 1: World Total Energy Consumption, 1990-2004 (quadrillion Btu)

and 50% of that energy is attributable to HVAC systems (Perez-Lombard, Ortiz, & Maestre, 2011), it is reasonable to assume that, in developed countries, 10-20% of all energy consumption is used to provide energy for HVAC systems. It may also be assumed that as more developing/non-OECD countries continue to develop, their energy consumption patterns may be similar to those of currently developed/OECD countries (Perez-Lombard, Ortiz, & Maestre, 2011). Although HVAC systems comprise many components and sub-systems, one of the more energy-intensive processes is dehumidification.

Dehumidification can be accomplished in various ways. In a simple HVAC application, dehumidification may represent a marginal portion of the total HVAC system energy consumption (an example would be the Florida Beach-Side Resort study below); in a more complex case such as a clean room or similar space that needs steady, ultra-low-humidity conditions, dehumidification may consume a significant portion of overall HVAC energy. In fact, typical dehumidification to sensible load ratios is 3:1 to 5:1

(ASHRAE, 2012), (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008).

The varied nature of dehumidification processes, as well as systems energy consumption which is influenced by many variables including HVAC system capacity, conditioned space requirements, ambient outside environmental conditions, and percentage of outside air being used, provides a focus for this study. The hot, humid climate of sub-tropical Florida exerts a high demand on HVAC/dehumidification systems due to extremely high year-round average relative humidity (RH) (ranging from 51 to 88% during the last 13 years, with a dry season (fall and winter) mean of ~69% and wet season (spring and summer) mean of ~79% (Diebel & Norda, 2013). Given this extremely high RH as well as the sheer size of the facility being examined, the energy consumption used for dehumidification in the study area is substantial (Study Site Energy Official, 2013).

The reason a single large facility was chosen is that data collection is simplified by gathering from one large facility comprising many smaller buildings, rather than from many unassociated smaller buildings, since there is only one source for the information and it is stored and formatted in a relatively uniform manner. The site chosen for study offers a good small- to medium-scale study opportunity for dehumidification energy efficiency in slightly varied situations (i.e. different conditioned space types, sizes, and requirements) in a climate that is similar to that of the developing countries that exhibit the fastest-growing rate of energy consumption (i.e. coastal China and India, parts of South America and Africa). This makes the data gathered, and even some conclusions, useful in solving dehumidification efficiency issues in these countries (SAGE, 1999).

Heat Pipes are considered in this study because of their inherent ability to increase efficiency in dehumidification systems (this will be explained in upcoming sections), and they represent a relatively simple, robust, and cost-effective technology that has the potential to become more efficient when enhancements are made according to specifics of the location (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011), (Brooke, Optimizing Wrap Around Heat Pipes, 2007). This study is also justified because very little research has been done on further enhancing heat pipe efficiency or combining it with renewable energy technologies after they have been installed.

Current energy consumption trends need to be curtailed; incorporation of energy-reduction strategies in HVAC systems can provide such an impact. Since dehumidification accounts for a significant portion of total HVAC energy consumption, and many of the countries that have the highest growth rate in energy consumption have hot and humid climates, dehumidification as applied in hot/humid climates is a logical focus. Heat pipes represent an appropriate technology as they offer a highly efficient and effective dehumidification-enhancement technology, are low-maintenance devices, and are scalable in application.

Goals and Objectives

This dissertation aims to achieve several things. The first objective is to examine how the facility is already reducing energy consumption, including the installation of heat pipes in new and existing buildings. The second objective is to examine alternative methods to heat pipes that may be better suited to meet the dehumidification needs of the facility. Another objective is to analyze a potential efficiency enhancement using solar photovoltaic (PV) cells to generate the energy consumed when reheating of air is necessary. The goal of all of the above is to come up with recommendations regarding the current dehumidification system and any enhancements that can be made. The recommendations will concern whether changes are needed, what should be changed, and when changes should be made, with appropriate justifications.

Methods Summary

In order to make relevant assessments and recommendations, many aspects needed to be analyzed: climate patterns, heat pipe operating parameters, conditioned space requirements, solar energy generation potential, the desires of the facility's management, solar PV efficiency and cost, solar thermal capabilities and cost, and solar energy incentives. The first step was to gather data on the following:

- The heat pipe operational parameters from the Site Energy Official (SEO) of the facility
- Weather data for the facility site
- PV efficiency
- PV price
- Solar energy incentives (both one-time and continuous)
- Emissions generated from oil-burning power plants

The next step was to preprocess the data (clean, organize, and make some calculations). The first cleaning step was the removal of irrelevant data. To aid in organizing, all data was merged into a single Excel® workbook containing PV efficiency data, weather data, and the heat pipe parameters. Next, the daily and annual electricity and fuel consumptions were calculated.

The next step was to process the data in order to obtain data that would be usable to draw conclusions from and make recommendations. The first data sets processed were the weather data and heat pipe operational parameters. These data sets were used to find:

- The number of days the heat pipe was not cycling
- The number of days heat pipes and heaters were operating at the same time
- The number of days needing heating
- The number of days that didn't need dehumidification
- The absolute humidity
- The relative humidity at room temperature given outside relative humidity

The next sets of data to be processed were PV cell efficiency, electric heater energy consumption, solar irradiation, and PV cell size. These data sets were used to calculate the number of PV cells and panels needed to completely power the electric heaters in the heat pipes and the area that would be needed for these cells.

The next major data set to be processed was the Financial data—the PV installed cost and incentives. The first thing to be calculated using this data was the total installed cost of the PV installation using the average per watt cost of PV installations over 100 kW given in (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). Later the installed cost using different prices was calculated; this will be discussed in detail in later chapters. Next, the one-time incentives were assumed to be their maximums since the installed capacity would far exceed the limits of the incentives. The annual, continuing, incentive payout was then calculated. This was more complicated than the one-time incentives and is explained in detail in the methodologies section later.

The next, and likely some of the most significant, calculations were the cost savings from prevented energy consumption of heaters and overall energy savings by using PV. These are significant because the PV would not only be able to run the electric heaters, but when the heaters were not being used, they could be used to power other sectors.

The last calculations using these data sets concern the undiscounted payback period (PBP). The PBP was also very important in deciding whether or not the installation of PV would be cost-effective. The last data set to be processed was the emissions and fuel consumption data. The data that was calculated for this data set was the annual fuel consumption for the electric heaters, the annual sulfur dioxide, nitrogen oxide, and carbon dioxide emissions. The fuel consumption of the heaters that used a natural gas fired hot water loop was also calculated. After the natural gas consumption and annual cost was calculated, it was determined the extremely low annual cost makes any changes to these systems almost unjustifiable.

Finally, the data obtained from processing was then analyzed along with information that was gathered for the coming chapter, conclusions were made, and recommendations made according to the following priorities:

- Financial savings
- Simplicity and dependability of the system

- Progress toward meeting current and future corporate citizenship goals
- Environmental impact, which would tie into the citizenship goals

Literature Review

As previously mentioned, worldwide energy consumption was 524 quads in 2010, and this amount is projected to increase to 820 quads by 2040, unless measures to reduce consumption are applied (Conti, et al., 2013). The developed world (defined as OECD member countries) have been the largest contributors to global energy consumption in the past, but now the developing world (non-OECD member countries) are consuming the most energy overall (as is shown in Figure 2) (Conti, et al., 2013), (The World Bank, 2013). This is despite the fact that developing countries have not yet exceeded our rate of consumption per capita (seen in Figure 2) (The World Bank, 2013).

With the exception of Qatar, many developing countries demonstrate a level of energy consumption per capita far less than in the developed world; as shown below, three of the largest consumers of electricity in the developing world consume less than half the energy per capita consumed in the U.S. and Canada, but if their consumption were to continue to grow in the same way the developed world did in the past, their energy consumption per capita (and therefore overall) energy consumption will increase. Because of the above fact and their extremely large populations their overall energy consumption will soon far exceed the developed world's overall energy consumption

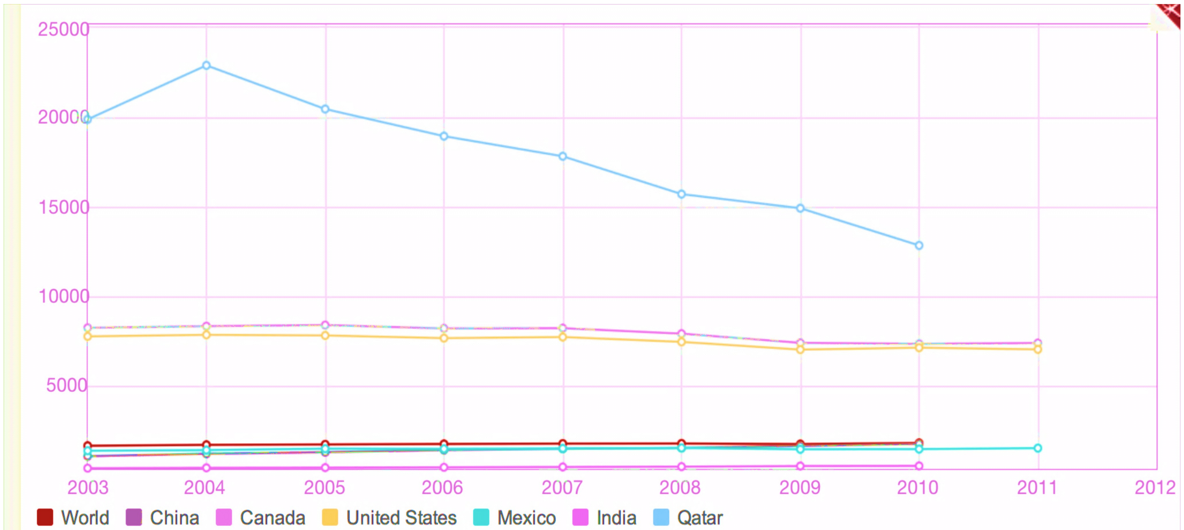

Figure 2: Energy consumption in kg of oil equivalent per capita: 2003 - 2011

unless measures to conserve energy are taken (The World Bank, 2013). Figure 2 shows that a relatively small shift in per capita energy consumption is magnified immensely by the fact that two of the three developing countries shown have populations of over 1 billion people (China: ~1.3 billion and India: ~1.2 billion) while the United States has ~300 million, so their energy consumption is magnified 4 times more than USA's (U.S. Census Bureau, 2013).

Since energy consumption is a problem much of the developed and developing world is contributing to, we must all work harder toward more efficient energy use. One of the most significant energy consumption sectors is buildings; depending on the nation, 20-40% of the national overall energy consumption is attributed to them (Conti, et al., 2013). Commercial buildings are known to have consumed 28.9 quads of energy in 2010, and this number is projected to rise to 49 quads by 2040, with the developing nations having the largest increase (Conti, et al., 2013). This can plainly be seen in Table 1 (Conti, et al., 2013). As can be seen in Figure 3, 51% of the total energy consumed in

2010 by commercial buildings was in the form of electricity; this is projected to rise to 64% by 2040 (Conti, et al., 2013). Again, a significant portion (about 2/3) of this increase is predicted to be due to the growth of developing countries (Conti, et al., 2013).

Table 1: Commercial Building Energy Consumption by Region in quadrillion Btu - 2010 to 2040

Region	2010	2015	2020	2025	2030	2035	2040	Average annual percent change, 2010-2040
OECD	**20.2**	**20.9**	**22.0**	**23.2**	**24.4**	**25.5**	**26.5**	**0.9**
Americas	9.8	10.1	10.5	10.9	11.5	12.0	12.6	0.8
Europe	6.5	6.9	7.4	7.8	8.3	8.6	9.0	1.1
Asia	3.9	3.9	4.2	4.4	4.6	4.8	5.0	0.8
Non-OECD	**8.8**	**9.9**	**11.7**	**13.9**	**16.5**	**19.4**	**22.5**	**3.2**
Europe and Eurasia	2.2	2.3	2.5	2.8	3.1	3.5	3.8	1.8
Asia	4.2	4.9	6.0	7.4	9.1	11.0	13.1	3.9
Middle East	1.0	1.1	1.3	1.5	1.7	1.9	2.0	2.4
Africa	0.4	0.5	0.6	0.7	0.8	1.0	1.2	3.5
Central and South America	1.0	1.1	1.3	1.5	1.8	2.0	2.4	3.1
World	**28.9**	**30.8**	**33.6**	**37.1**	**40.9**	**44.8**	**49.0**	**1.8**

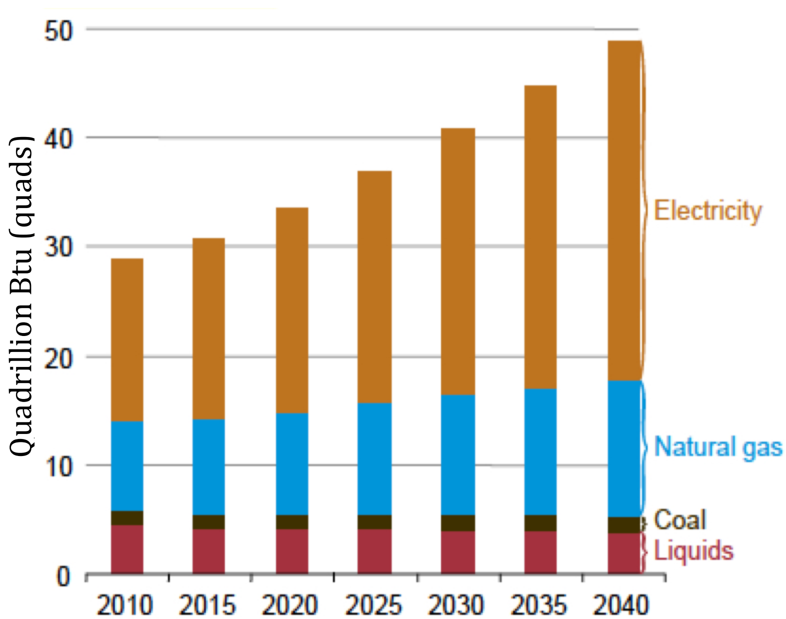

Figure 3: Total World Commercial Building Energy Consumption by Type

Developed countries' electricity use is projected to grow from 10.4 quads in 2010 to 15.7 in 2040, while developing countries' usage is projected to grow from 4.3 quads in 2010 to 15.4 in 2040 (Conti, et al., 2013). This large growth in energy consumption in developing countries (the fastest growth being in China, India, and non-OECD Asia @ 3.9% per year) is being driven by rising standards of living and greater demand for services (Conti, et al., 2013). Both of these drivers suggest greater deployment of HVAC systems, especially as more hotels are built as countries develop (Conti, et al., 2013). As mentioned earlier, HVAC systems on average account for ~50% of the total energy consumption of buildings; thus ~2.15 quads of electricity were consumed for HVAC applications in 2010 in the developing world, with between ~1.6 and 1.8 quads applied to dehumidification, and that can be extrapolated to increase to ~7.9 quads in 2040, with between ~5.9 and 6.6 attributable to dehumidification (Perez-Lombard, Ortiz, & Maestre, 2011), (Conti, et al., 2013). If we consider the above energy consumption facts and that the two countries with the fastest growing energy consumption rate (India and China) have hot humid climates (according to the Koppen-Geiger system), as they develop their HVAC load will greatly increase due to increased dehumidification loads (World of Maps, 2013), (ASHRAE, 2012).

The Significance of Air Conditioning and Dehumidification

Air conditioning (AC) was at one time considered strictly as a means for cooling and dehumidification of a space, but it has now developed to become synonymous with comprehensive environmental control and suggests "*the control of temperature, moisture content, cleanliness, air quality, and air circulation as required by occupants, a process, or a product in the space.*" – William Carrier (McQuiston, Parker, & Spitler, 2005). Carrier is credited with the first successful attempt to control humidity in 1902 and thus achieve the first true environmental control (McQuiston, Parker, & Spitler, 2005).

HVAC technology has advanced significantly with the advent of computers and modern controls and building automation systems. It has been further driven by human priorities (McQuiston, Parker, & Spitler, 2005). As people/customers have become more knowledgeable and concerned about the environment, especially ozone layer depletion and climate change, the industry has begun placing voluntary restrictions alongside imposed standards that limit what materials can be used, and has set efficiency standards, among other actions (McQuiston, Parker, & Spitler, 2005).

There has also been a trend toward requiring greater volumes of ventilation air in an indoor space, as evidenced by ASHRAE standards 62.1 and 62.2, which were updated in 2010 and 2013, respectively. These standards set requirements for ventilation air and air cleanliness among others (ASHRAE, 2013). Because of the mandate for increased ventilation air, dedicated outside/outdoor air systems (DOAS) have become increasingly popular, but with these increases in outside air comes an increase in humidity and an increased risk of mold growth and general stuffiness of the inside conditioned air (McQuiston, Parker, & Spitler, 2005), (ASHRAE, 2013). This increase in humidity and mold growth risk leads to discomfort and health risks for building occupants, as well as damage to a building itself, and therefore increased rates of dehumidification are required as compared to the scenario in which a greater volume of return air is conditioned (ASHRAE, 2013), (McQuiston, Parker, & Spitler, 2005). This increased demand for dehumidification has prompted the development of more efficient means to accomplish moisture removal, which is the central focus of this thesis.

The facility being studied has already made considerable progress in its efforts to conserve energy. Best practices in energy management have already been implemented and fine-tuned, and thus little additional room for improvement is anticipated. The HVAC systems were made more efficient in multiple ways, but one of the most successful methods involved improvements to the dehumidification processes already being used (Study Site Energy Official, 2013).

Chapter 2: Dehumidification systems – types, advantages, disadvantages, and research

Dehumidification is performed for many reasons, but in the case of the facility being examined it is performed mainly for two reasons—comfort and health (Study Site Energy Official, 2013). According to the *ASHRAE 2011 Handbook— HVAC Applications*, "Humidity control is critical to ensure satisfactory air quality and to minimize costly mold and mildew problems in hotels." This quote applies directly to health issues, as some types of mold that grow due to poor humidity control are harmful to human health and costly to clean up. Even in cases in which mold is not directly harmful, many types (as well as mildew) emit a foul odor that can create a public nuisance (ASHRAE, 2011). Humidity control is extremely important in terms of comfort; as humidity increases, a space begins to feel more "stuffy" and the heat index (perceived/felt temperature) increases as well, thus resulting in the decrease of indoor air quality and overall comfort of occupants (The Weather Channel, 2012).

Florida Energy Consumption, Climate/Weather, and Dehumidification

As discussed previously, Florida has an average relative humidity of ~69% in the dry season and ~79% in the wet season, conditions that are similar to other subtropical and tropical locations (namely India and areas of China) in the need for dehumidification (Diebel & Norda, 2013), (World of Maps, 2013). This elevated humidity throughout the year, along with relatively high temperatures, contributes to Florida's (and in general the South Atlantic region's) higher electricity consumption as compared to the rest of the nation. In fact, the South Atlantic census division has the largest commercial cooling energy load (45 billion kWh; bkWh; in 2003) (which includes dehumidification in this case). This load is higher than for the other three census regions (Northeast: 13 bkWh, Midwest: 17 bkWh, and West: 24 bkWh) (EIA, 2003).

Although more recent data are not available from the EIA (Energy Information Administration) regarding commercial energy use, they recently released residential information that shows that current trends that apply to Florida are the same as they were in 2003. According to the 2009 EIA Residential Energy Consumption Survey (RECS), Floridians consumed ~15,000 kWh of electricity per household, while the national average was less than12,000 kWh. Among Floridians, 27% of the load was dedicated to cooling (which, of course, includes dehumidification) while the national average was only 6%. This results in the average Florida household consuming ~4,000 kWh per year for cooling, whereas the average American household consumes 720 kWh or less for cooling, per year (EIA, 2013). In short, Florida households consume greater than 5.5 times more energy for cooling than does the average household in the USA (EIA, 2013).

Given that Florida does not record nearly as many heating degree days as does the majority of the nation, it consumes much less total energy (FL consumes ~55 million Btu of energy per household, per year, while the national average is ~90 MMBtu) (EIA, 2013). Since Florida consumes less total energy per household as compared to the national average, but consumes significantly more electricity, and cooling (including dehumidification load) makes up the largest share of energy consumption from a single source (the largest is appliances, electronics, and lighting which are multiple systems with 50%), focusing on enhancing dehumidification efficiency makes a great deal of sense (EIA, 2013).

As mentioned previously, the effect of humidity on heat index is very important. For example, if we consider a hot and arid location such as Las Vegas, with an average summer daytime high over 100°F but with a relative humidity of less than 20%, the heat index is 100°F, but not muggy. In Florida with an average summer humidity of 79% and average daytime highs around 90°F, the heat index is likely to be between 110 and 120°F. The air will feel uncomfortable and may even result in respiratory distress among some of the population (Diebel & Norda, 2013), (The Weather Channel, 2012), (ASHRAE, 2013). Air is considered most comfortable at between 30 and 50% relative humidity, where it is neither muggy nor dry, and temperatures in the low- to mid-70s are a comfortable temperature. These conditions present a heat index significantly higher than the dry bulb temperature of the air (The Weather Channel, 2012), (ASHRAE, 2013). It is well recognized that it is a more energy-intensive process in general to dehumidify air than it is just to cool it (which will be explained below), so increasing efficiency of the dehumidification process can significantly reduce energy consumption. The method used to dehumidify can have a significant effect on electrical loads, and the various means available will be described.

Desiccant Dehumidification

Different situations and demands require different methods of dehumidification. There are two main methods of dehumidification technology: desiccant and mechanical; each method presents different variations, but such variations are subtle. Desiccant dehumidification operates by using chemicals or materials that either adsorb (water adheres to the surface of the material, not chemically changing it) or absorb (water is taken in on a molecular level by the chemical, chemically changing it) water (ASHRAE, 2012).

There are means by which desiccants can be classified, the first of which is by state; desiccants are either solid or liquid; absorbing desiccants are generally liquid while adsorbing desiccants are generally solid (ASHRAE, 2012). Another way to classify desiccants is by reusability; if a desiccant can be reused, it is called regenerative and generally uses silica or alumina gel (ASHRAE, 2012). If the desiccant cannot be reused, it is called non-regenerative, and usually uses hygroscopic salts (ASHRAE, 2012). The desiccant choice depends on equipment, gases, and requirements of the end user (ASHRAE, 2012). The basic means by which liquid desiccant processes work is by first bringing the “wet” air in contact with the desiccant, which then absorbs the moisture from the air, after this the air is cooled and blown into the conditioned space. Once the desiccant is near saturation it is regenerated (ASHRAE, 2012). Although this process sounds relatively simple, it does involve certain limitations that make it less appealing for the facility in Florida.

The first is that it generates a fair amount of heat, which must be removed in order for the desiccant to function properly (ASHRAE, 2012). The heat produced presents an issue, because in Florida the ambient outside air temperature is already high and the study is focusing on the heat pipes in the facility that provide make-up/ventilation air. Make-up/ventilation air is air blown from outside (so it is close to 100% outside air) to aid in ventilating the space and in this case keep it positively pressurized (ASHRAE, 2012). Because of the need to condition high humidity and high temperature air, liquid desiccant systems are likely not ideal (McQuiston, Parker, & Spitler, 2005).

Another challenge is that liquid desiccant systems require reactivation, which requires a separate air stream and a means of heating the desiccant to increase its vapor pressure relative to the air stream so that the airstream carries the moisture out with the exhaust air in order for the desiccant to be reused (ASHRAE, 2012). Again, if we consider the scenario in which the desiccant would likely be used (to remove moisture from high heat, high humidity input air) it would likely reach its saturation point fairly quickly. The typical regeneration side-stream of desiccant (8%) may not be adequate; thus a larger percentage of desiccant may need to be regenerated at once, so more desiccant would be needed to make up the difference (ASHRAE, 2012).

Solid desiccants are operated differently than liquids, but involve most of the same issues. Solid desiccants are generally distributed into three categories: continuously reactivated, periodically reactivated, and non-reactivated/disposable (ASHRAE, 2012). The silica packets found in shoe, electronic, and furniture packaging best exemplify the non-reactivated/disposable packages; they are there to ensure that small amounts of moisture that penetrate the package do not damage the product (ASHRAE, 2012). Periodically reactivated cartridges are used where a constant but small load is expected, so that when the desiccant is saturated it is removed, heated and reactivated, then replaced, and the humidity load must be small enough that the duration without conditioning causes only a negligible effect on the product/space (ASHRAE, 2012). The continuous reactivation dehumidifier is the predominant desiccant type used in high-moisture-load applications, and in this group of processes, rotary solid-desiccant dehumidifiers are the most popular (ASHRAE, 2012). The process works by having two separate air streams, the process air stream (that being dehumidified) and the reactivation air stream (ASHRAE, 2012). The solid desiccant is fixed to a wheel that rotates continuously while process air flows through most of it (thus dehumidifying the air), and as the desiccant comes in contact with the process air it becomes saturated. As it becomes saturated, it is moved closer to the reactivation air stream (ASHRAE, 2012). As the desiccant enters the reactivation air stream (which is sealed off from the process stream), it is heated so that the desiccant reactivates and the reactivation air stream removes the moisture from the desiccant and exhausts it (ASHRAE, 2012).

One issue that persists with solid desiccants is associated with the heat generated. An example scenario is presented and is shown in Figure 4. If the air enters the desiccant dehumidification system at 70°F with a humidity of 56 gr/lb (~50% RH), it will leave the dehumidifier at 97°F with a humidity of 23 gr/lb (less than10% RH) (blue Room Temp & Humidity line in Figure 4). If the humidity is increased to 80 gr/lb (~70% RH), the outlet temperature increases to 106°F @ 34 gr/lb (~13% RH) (red Room Temp @ 70% Humidity line in Figure 4) (ASHRAE, 2012). If the temperature and RH are brought to

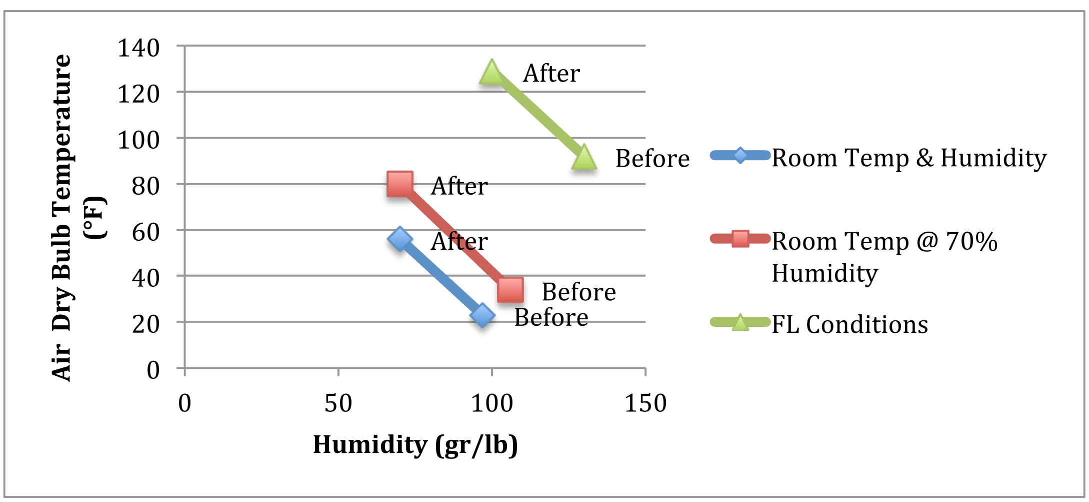

Figure 4: Desiccant Wheel Performance Chart

levels typical of Florida summer outside conditions (~90°F @ 80% RH/173 gr/lb) the estimator (created by ASHRAE technical committee 8.12) to calculate the above values can't go that high, so the equivalent heat index at 100°F (114°F) was used (100°F @ 45% RH/129 gr/lb) (ASHRAE, 2012). The result for increasing the parameters to FL conditions was a temperature increase to 130°F @ 92 gr/lb/~35% RH (green FL Conditions line on Figure 4). This will need to be cooled to ~73-75°F before entering the conditioned space, thus adding another significant load (ASHRAE, 2012). The above data is also summarized in Table 2 below, which shows the condition the air is in (Temp and humidity) before and after dehumidification in each situation.

Table 2: Summary of Desiccant Effects on Temperature and Humidity

	Room Temp (°F)	Room Humidity (gr/lb)	Increased Humidity Temp (°F)	Increased Humidity (gr/lb)	FL Temp(°F)	FL Humidity (gr/lb)
Before DH	70	56	70	80	100	129
After DH	97	23	106	34	130	92

Another issue that becomes apparent with both systems is that they are complex when compared to many mechanical methods of dehumidification (again, heat pipes are the perfect antithesis of complexity, they are uncomplicated and have no moving parts) (Allen, June 14, 2013). A desiccant system requires regular maintenance in order to function optimally; according to the *ASHRAE 2012 Systems and Equipment Handbook*, the average desiccant system requires desiccant replacement or replenishment every five to ten years, and also requires constant maintenance of the filter in the system. Otherwise, the desiccant life could be reduced to ~2 years due to contamination (ASHRAE, 2012). Although desiccant cooling has limitations, there are also advantages.

Desiccant systems can either dehumidify or humidify, depending upon the process air conditions and the desired humidity in the conditioned space (ASHRAE, 2012). A desiccant functions by vapor pressure differential; if the desiccant has a lower vapor pressure than the air, then the desiccant takes in moisture, and if it is higher, then it rejects moisture, thereby maintaining a desired humidity with a conditioned space (ASHRAE, 2012). The control of the desiccant system is managed through several processes, the first of which is to control the flow rate of process air across the desiccant —the faster the rate, the less humidity that is removed (ASHRAE, 2012). The regeneration air temperature is the next parameter. The higher the temperature, the more moisture is removed from the desiccant; thus, the more moisture it can remove from process air when it re-enters that air stream (ASHRAE, 2012). Finally, as was mentioned previously, there are numerous types of solid and liquid desiccants, each of which has specific uses, but silica gel (solid) and lithium-chloride + water (liquid) are two of the most commonly used. Silica gel has proven to be a good default desiccant as it performs predictably well in many different condition combinations (ASHRAE, 2012), (Goldsworthy & White, 2012). Another major benefit is that desiccants are usable in a wide range of situations. An extreme example of the minimum temperature that a desiccant system can be used at is –40° (same temperature in Fahrenheit and Celsius), while mechanical dehumidification is limited to a minimum of 39.2 °F (4°C) (La, Dai, Li, Wang, & Ge, 2009), (ASHRAE, 2012). Some specialized applications of desiccant dehumidification are (ASHRAE, 2012):

- drying natural gas;
- drying gases that are to be liquefied;
- drying instrument and plant air;
- drying process and industrial gases;

- dehydration of liquids;
- frost-free cooling and dehumidification;
- lowering dew point to facilitate low-temperature manufacturing;
- preservation of equipment;
- maintaining dry atmosphere;
- drying in situations where space is limited (fixed bed or disposable desiccant drying processes generally used here).

These applications are indicative of the common assumption that desiccant dehumidification is best suited for specialized processes, but as will be shown below it has applications in larger-scale dehumidification processes as well (La, Dai, Li, Wang, & Ge, 2009). Another major benefit of desiccant dehumidification is that it is purported to be more environmentally-friendly than conventional mechanical methods. This is because conventional mechanical dehumidification uses refrigerants that may be chlorofluorocarbons (CFCs) or hydrochlorofluorocarbons (HCFCs), but desiccant dehumidification doesn't require the amount of refrigerants that conventional methods do unless significant cooling is required (ASHRAE, 2012), (Goldsworthy & White, 2012), (La, Dai, Li, Wang, & Ge, 2009). Desiccant dehumidification also has the benefit of being able to disinfect the process air to a degree, thus further aiding in the prevention of mold and bacterial growth in a conditioned space (La, Dai, Li, Wang, & Ge, 2009). Desiccant dehumidification also tends to be very energy efficient, because the only input energy required for the dehumidification process is to heat the regeneration air. Because this only requires the air to be heated to between 100 and 250°F, this heating can often be accomplished with low-grade heat via solar-thermal, 'waste heat' recovery, district heating, and bioenergy. Minimal electricity is needed unless electric strip heating is used (La, Dai, Li, Wang, & Ge, 2009). One of the specific advantages of the solar-thermal approach is that the regeneration capacity adjusts automatically to dehumidification load since periods of higher solar radiation generally coincide with the need of greater dehumidification (La, Dai, Li, Wang, & Ge, 2009).

Another benefit of desiccant systems is that they are easily hybridized with cooling systems to enable cost-effective solutions for achieving extremely low dew points, important in situations that require a conditioned space (and thus a dew point) below 40°F (~4°C) (ASHRAE, 2012), (La, Dai, Li, Wang, & Ge, 2009). The opposite is also true for hybridizing desiccant systems; they can be combined with conventional AC systems or one of several other systems discussed below to adapt desiccant dehumidification to use in hot and humid climates (La, Dai, Li, Wang, & Ge, 2009). Desiccant dehumidification requires hybridization/adaptation to hot and humid climates because the conventional process counteracts the increased temperature of the output process/supply air by over-drying it. Conventional evaporative cooling can cool it further along in the process, but if the air enters at an already high temperature, the over-drying is insufficient to allow for appropriate cooling (La, Dai, Li, Wang, & Ge, 2009). It is also notable that hybridizing desiccant systems with conventional AC means of cooling is best suited for hot and dry climates; if it is used in hot and humid climates an efficiency improvement may be seen, but it may also consume more energy than by using simply a conventional system configuration (La, Dai, Li, Wang, & Ge, 2009). Another form of hybridization is the desiccant and absorption chiller hybrid, which combines the two thermally driven systems to dehumidify and cool the air in hot and humid climates (La, Dai, Li, Wang, & Ge, 2009). The benefit here is that both thermally-driven systems can be powered by the same energy source, with the absorption chiller receiving the initial heat since it requires higher temperatures, and the exhaust air being used to regenerate the desiccant (La, Dai, Li, Wang, & Ge, 2009).

Another point to consider which may present a benefit is that the technology and design methods for desiccant dehumidification are improving rapidly; an example of this is the recent research into advanced desiccant materials. The composite materials (a combination of silica and haloid desiccants being the most commonly researched) are approaching the same effectiveness as for conventional desiccants (i.e. silica gels and haloids) without the disadvantages (i.e. loss of adsorption capability at high temperature in silica-based desiccants and formation of crystals that cause loss of desiccant in haloids) (La, Dai, Li, Wang, & Ge, 2009). A detailed study was conducted that used a compound desiccant in order to design a high-performance desiccant wheel system. The compound desiccant researched was a two-layer material of a porous silica gel medium impregnated with a lithium chloride (hygroscopic) substrate in the pores (Jia, Dai, Wu, & Wang, 2006). The results of the study are promising; the new desiccant first achieved a COP (Coefficient of Performance) of 1.3, the typical COP is between 0.5 and 1.0 (Jia, Dai, Wu, & Wang, 2006). A comparison of the performance of the compound desiccant with conventional silica gel is shown in Figure 5. What can be observed is that:

- the compound desiccant is about twice as effective at removing moisture than silica gel;
- the compound desiccant is more hygroscopic than silica gel at low RH;

- the rate of increase of the compound desiccants adsorption capacity increases as the RH increases, while the silica gels rate of increase is almost linear (Jia, Dai, Wu, & Wang, 2006).

The compound desiccant is also more effectively regenerated at lower temperatures than is silica gel; therefore less heat/energy can be used to achieve the desired level of

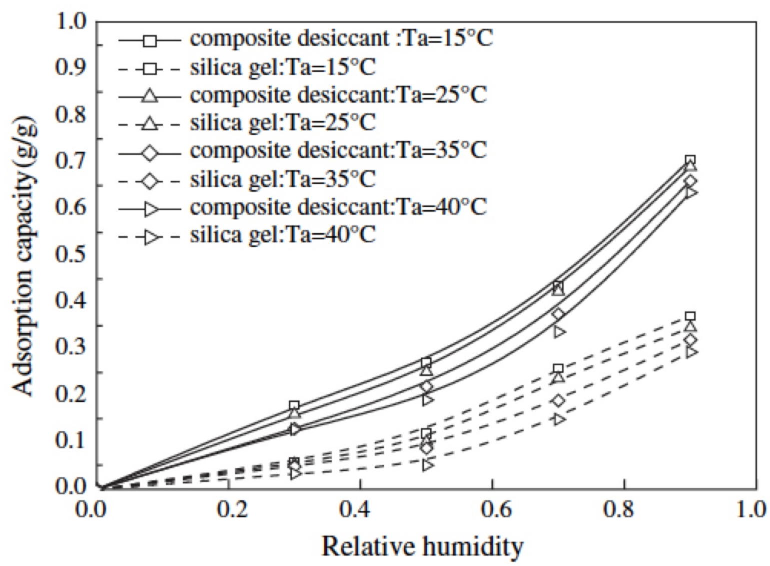

Figure 5: Compound Desiccant Performance vs. Silica Gel Performance

regeneration than would be necessary with comparable levels with silica gel (Jia, Dai, Wu, & Wang, 2006). The regeneration efficiency difference between the compound desiccant and silica gel also grows as the regeneration temperature increases (Jia, Dai, Wu, & Wang, 2006). The final improvement seen with the compound desiccant in dehumidification capability was between 20 and 40% better than with silica gel (Jia, Dai, Wu, & Wang, 2006).

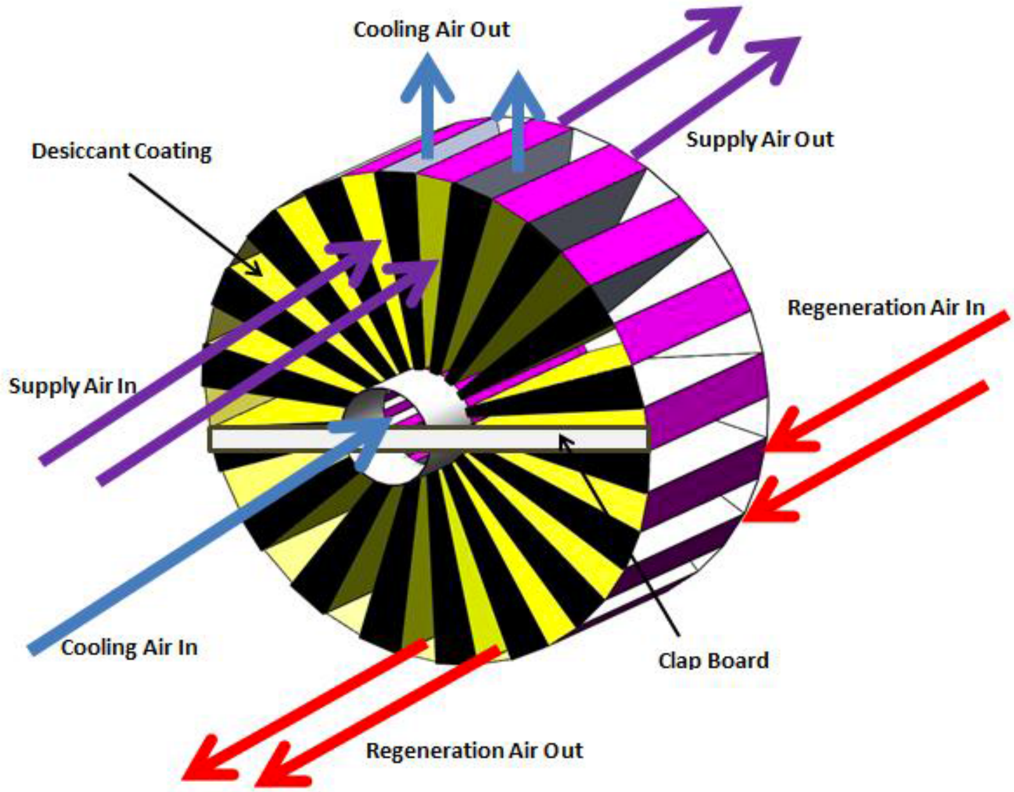

Figure 6: Non-adiabatic Desiccant Wheel Design

Yet another example of advancements in design and technology pertains to the design of the desiccant wheel. One promising proposed design is the non-adiabatic desiccant wheel, which integrates cooling air channels into the design of the wheel; the design is seen in Figure 6 (Narayanan, Saman, & White, 2013). The design operates by having the same two sections as a conventional desiccant wheel (supply air and regeneration air) but structured as shown in Figure 6 and described below:

- Cooling air enters the central hole with the closed back and distributes radially through channels.
- Supply/Process air flows axially through desiccant coated channels adjacent to the cooling air channels and is cooled/kept cool as it flows through the channels rather than having to be cooled after it exits the wheel.

- As the desiccant needs to be regenerated, the regeneration air is blown through the same channels as the process air but in the opposite direction.

The principle of this design is that the cooler the temperature at which the process operates, the greater the moisture that is collected by the desiccant: as was mentioned before, as the temperature of air increases the relative humidity decreases, and thus the higher its absolute humidity can be. The desiccant, silica gel in this case, therefore adsorbs less moisture. Desiccants also tend to have a higher vapor pressure at higher temperatures, so they release moisture into the air (ASHRAE, 2012), (ASHRAE, 2013), (Narayanan, Saman, & White, 2013). During the simulation and experiment, it was found that the non-adiabatic wheel indeed provided a more efficient method of dehumidification than with conventional adiabatic wheels (Narayanan, Saman, & White, 2013). Results are shown below in Table 3 corresponding to different supply air RH conditions (Narayanan, Saman, & White, 2013):

Table 3: Non-adiabatic Desiccant Wheel Performance Improvement over Traditional Desiccant Wheel

Relative Humidity	Performance Improvement %
50%	45%
60%	46%
70%	53%

These results pertain to the use of cooling air at 59 °F (15 °C); if this temperature were to be lowered, it is safe to assume that the performance would increase and the outlet temperature of the supply air would also be cooler (Narayanan, Saman, & White, 2013). With further development, this desiccant wheel design may become a viable option for dehumidification and cooling in hot and humid climates such as the area where the facility being researched is located (Narayanan, Saman, & White, 2013). If the two improvements described above were to be used in conjunction with each other, there may be a compounding effect on efficiency and performance that is cost-effective. Desiccant dehumidification has distinct benefits and drawbacks, depending on the specifics of a given scenario, such a system may present a favorable or unfavorable result. When desiccant dehumidification proves to be a poor choice, there are other options that are likely to be more appropriate.

Mechanical Dehumidification

The next type of dehumidification is called mechanical dehumidification and works on a different principle than desiccant dehumidification. Mechanical dehumidification works one of two ways, either by chilling or by compressing (ASHRAE, 2012). Chilling works because the moisture holding-ability of air drops as the temperature drops, thus causing the absolute humidity of the air to drop (ASHRAE, 2012). Compression occurs when air is compressed, and moisture is literally squeezed out of the air, this is possible because air can be compressed while water is virtually incompressible, so the two substances separate when under pressure (ASHRAE, 2012). Mechanical dehumidification presents many different variations, all of which function in essentially the same way (ASHRAE, 2012):

- Air is cooled below the dew point;
- Moisture condenses onto the cooling coils and is removed (usually via a simple drip pan);
- Dehumidified air is then reheated to the appropriate room temperature and blown into the conditioned space.

The methods by which these steps are accomplished are often what govern the effectiveness and efficiency of a mechanical dehumidification system. One of the major drawbacks of mechanical dehumidification is that it highly power-intensive (Allen, June 14, 2013), (ASHRAE, 2012). One reason for this is that conventional mechanical dehumidification involves two significant temperature changes—the air must first be cooled to a point lower than the dew point (which is generally in the high 60s to mid 70s °F at the site during the summer). With average temperatures in the high 80s to 90s °F, the air needs to be cooled by at least 10-20 °F, and is typically cooled to at least 59 °F in order to ensure adequate moisture removal (ASHRAE, 2012), (Diebel & Norda, 2013), (Allen, June 14, 2013). Once dehumidified, the air must be re-heated to room temperature, which is generally set to 72 –74 °F, thus requiring additional energy input (Allen, June 14, 2013), (ASHRAE, 2012).

Because of relatively recent developments in indoor air quality regulations, greater volumes of ventilation air are now required, suggesting that outdoor air is required to be blown into most new buildings (Mazzei, Minichiello, & Palma, 2005). The means by which outside air is added to and treated before entering a conditioned space reflects one of the ways in which dehumidification systems differ. The first method by which outside air can be added is by blowing return air into the ventilation air stream before it reaches the cooling/dehumidification coil, thus initiating the cooling and dehumidification process and lowering the load on the coil (Mazzei, Minichiello, & Palma, 2005). The second method directs outside air through the cooling/dehumidification coil and reheat coil, and then adds return air in order to pre-mix the air and bringing it closer to neutral (Mazzei, Minichiello, & Palma, 2005). The third method is to employ a ventilation system completely dedicated to outside air (ASHRAE, 2012). A 100% outside air system (this is what the systems being examined are in general) does accord special considerations, dampers and economizer settings will be considered first (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008), (Allen, June 14, 2013), (McQuiston, Parker, & Spitler, 2005).

The dampers and economizer settings allow some or all outside air to bypass the dehumidification coil and enter the conditioned space with minimal "treatment" depending on the outside conditions and desired indoor conditions (Allen, June 14, 2013). This approach is applied in one of the buildings that was recently renovated at the facility being examined to include economizers; these economizers allow outside air to flow into the space when the outside temperature and RH are low enough that cooling and dehumidification of the air is unjustified, thus saving energy (Allen, June 14, 2013). An important consideration for 100% outside air systems is how the air is re-heated or pre-conditioned. The *ASHRAE 2012 Handbook* recommends that exhaust air be used to pre-condition the ventilation/outside air; the exhaust air will act to make the outside air more neutral. In other words, if it is hot and humid outside the exhaust air will act to cool and dehumidify; the opposite is true as well. If the ventilation air is cool and dry, the exhaust will warm and humidify. (ASHRAE, 2012). The use of exhaust air to condition ventilation air also introduces the possibility to downsize the mechanical systems and thus save additional energy and costs (ASHRAE, 2012).

Another consideration for a 100% outside air system is the condition of the air as it enters the conditioned space (ASHRAE, 2012). That is, should the entering air be cooler, warmer, or neutral (same temperature) as compared to the conditioned air in the space

(ASHRAE, 2012). This is an important consideration because the means by which outside air enters a space governs how the air is treated in the dehumidification process (ASHRAE, 2012):

- If cooler air is required, reheat may not be required.
- If neutral air is required, reheat will be required in order to raise the dry bulb temperature of the air to that of the inside air since it leaves the dehumidification coil cooler than the conditioned space air.
- If warmer air is required, more reheat will likely be required

As with desiccant dehumidification, mechanical dehumidification presents specific benefits and issues. One of the greatest advantages of mechanical dehumidification is the customizability and variety of systems; there is an enormous range of dehumidifiers to fit almost every situation, for example (ASHRAE, 2012), (Taras, 2006):

- portable dehumidifiers;
- pool dehumidifiers (generally able to handle a constant load of saturated (100% RH) air, corrosion resistant, able to handle the chlorine load);
- factory-built dehumidifiers (essentially prefabricated then “dropped in” at the site);
- site-built dehumidifiers (generally larger, custom designed and built at the site rather than in a factory).

The customizability of the mechanical systems adds other benefits, one of which is the ability to apply novel solutions to make the systems more energy efficient. One example of this is the enthalpy wheel, which is a rotary mass energy transfer/exchange device (Mazzei, Minichiello, & Palma, 2005). The way the enthalpy wheel works is similar to a desiccant wheel; there is a quickly rotating (700 to 2400 rev/hr) usually desiccant-filled cylinder that is divided in half, where one half passes the outside air into the conditioned space and the other half passes the exhaust air through to the outside, depending on the condition of the outside air it will be either cooled and dehumidified (acting like supply air in the desiccant process) or heated and humidified (acting as regeneration air) (Mazzei, Minichiello, & Palma, 2005). The speed at which the wheel rotates is what determines the rate heat/energy transfer. It is heated or cooled by the outside air and then it moves into the exhaust stream where it is either cooled or heated, varying the speed at which the wheel rotates allows different transfer rates (Mazzei, Minichiello, & Palma, 2005).

Another customization option that enhances efficiency is the addition of variable air volume (VAV) motors/fans (Mazzei, Minichiello, & Palma, 2005). VAV systems allow the air volume moving through the system to be varied; this provides a more efficient method of controlling the temperature and humidity of a space that has varying occupancy (Allen, June 14, 2013). The VAV approach allows for minimal air to be blown into a space in order to maintain an "economical" level of humidity and temperature (i.e. low enough to prevent mold growth but generally higher than would be considered comfortable) while a space is unoccupied, and then increase comfortable levels when the space is occupied (Allen, June 14, 2013), (Mazzei, Minichiello, & Palma, 2005). This is accomplished by varying the volume of cooled and dehumidified air blown into a space rather than increasing cooling/heating of the space, because varying a motor speed is more energy efficient than cooling or heating a space (Allen, June 14, 2013). This is also happens to be one of the major techniques used in the facility in order to save energy (Study Site Energy Official, 2013).

If another method is used besides VAV to condition the space, more energy is often required (Allen, June 14, 2013). In the case of always-on cooling/dehumidifying, greater energy is needed to accommodate heating in order to maintain the desired set point temperature because the always-on cooling would cause the space to drop below that set point (Allen, June 14, 2013). The on/off method requires more energy because it demands more energy to run the motors in bursts of 100% than is does to have a VAV constantly running at around 20% capacity and shifting lower when demand isn't high and increasing as demand does. This method also has the drawback of not maintaining the space in a constant condition, the space would be constantly cooling and heating and drying and humidifying, making it uncomfortable to occupy (Allen, June 14, 2013), (ASHRAE, 2013), (ASHRAE, 2012). The VAV controls conditions by monitoring and responding to temperature, humidity, and CO_2 (for occupancy) sensors (Mazzei, Minichiello, & Palma, 2005).

Yet another energy-saving method that can be used because of the flexibility of the mechanical systems is changing the arrangement of coils and/or adding new coils to the system (Mazzei, Minichiello, & Palma, 2005). The heat pipe provides an excellent example and is structured as follows: a thermally-conductive coil filled with refrigerant upstream of the main cooling/dehumidifying coil is connected via piping to another coil downstream of the cooling/dehumidifying coil (ASHRAE, 2012). The heat pipe operates in the following manner (Wu, Johnson, & Akbarzadeh, 1997), (ASHRAE, 2012):

- Air is pre-cooled in the first evaporator coil (as the air is cooled, the heat from the air is transferred to the refrigerant which vaporizes and flows to the downstream/condenser coil);

- the air is then further cooled and dehumidified by the dehumidification coil (at this point the air is now below the required temperature for the conditioned space);
- Because it is too cold, the downstream coil then re-heats the air since it contains “hot” refrigerant (as the refrigerant cools it condenses and flows back to the upstream coil where it can be heated again).

This method saves a great deal of energy (as will be discussed in more detail later) in two ways (ASHRAE, 2012), (Wu, Johnson, & Akbarzadeh, 1997), (Mazzei, Minichiello, & Palma, 2005):

1. The amount of cooling needed to dehumidify the air by providing “free” cooling with the upstream/evaporator coil is reduced.
2. Reducing or removing the reheat load by way of the downstream/condenser coil provides approximately the same amount of heat the refrigerant absorbed in the upstream coil. The amount of reheat provided depends on the thermal conductivity of the coils, the efficiency of the refrigerant, and the temperature of the incoming air before and after dehumidification.

Another major benefit of mechanical dehumidification is that it can be accomplished at relatively low cost and maintenance as compared to desiccant systems (Allen, June 14, 2013), (Wu, Johnson, & Akbarzadeh, 1997). This benefit will obviously vary greatly depending on how a system is designed and built, but in the case of heat pipes there is virtually no maintenance cost except for the occasional filter cleaning and leak repair since there are no moving parts except for fans (Allen, June 14, 2013), (Wu, Johnson, & Akbarzadeh, 1997). There are minor improvements to mechanical systems that can be realized by way of adjustment of settings and arrangements, but most tie into the benefits above and for the sake of time and space will not be mentioned here.

Mechanical dehumidification systems do have their limitations. One of the most important caveats is associated with the temperature limitation of the mechanical dehumidification process (ASHRAE, 2012). As mentioned earlier, mechanical dehumidification technology cannot usually handle air with a dew point below 40 °F (~4 °C), thus it is impractical to use in cold and humid situations (ASHRAE, 2012), (La, Dai, Li, Wang, & Ge, 2009). Another concern with mechanical dehumidification is that it generally consumes more energy than desiccant dehumidification (La, Dai, Li, Wang, & Ge, 2009), (Goldsworthy & White, 2012), (ASHRAE, 2012). The reason the mechanical approach consumes more than desiccant systems is that the main process of mechanical dehumidification requires compression of a refrigerant, and over-cooling air followed by reheating of air, while desiccant dehumidification requires only cooling and heating of the regeneration air (which is usually accomplished with low grade/waste heat) (ASHRAE, 2012). Also, the mechanical dehumidification process is generally not as environmentally friendly as desiccant dehumidification, in fact many of the most prevalent refrigerants are fairly damaging to the environment if released into the atmosphere (La, Dai, Li, Wang, & Ge, 2009), (Calm J. M., 2006). These refrigerants are, in some cases, thousands of times more potent greenhouse gasses (GHG) than CO_2. In the cases of R-22 and R-410A, they have a Global Warming Potential (how many times more potent a given substance is as a GHG than CO_2 in comparable amounts over a certain time period) of 1,810 and 2,100 respectively over 100 years. In other words, R-22 is 1,810 times more effective GHG than CO_2 and R-410A is 2,100 times more effective than CO_2 (Calm J. M., 2008). Some of the refrigerants (R-22 is an example) are actually ozone-depleting substances and are being phased out by the EPA and other world governments that ratified the Montreal Protocol which requires ozone depleting-substances to be phased out of new systems, and their use to be limited to existing systems (U.S. EPA, 2010). It is also worth noting that R-410A is considered a viable replacement for R-22, although it has a GWP ~16% greater than that of R-22 (Calm J. M., 2008). R-22 is also 6% more efficient as a refrigerant than R-410A (Calm J. M., 2008). R-22 is also the most popular refrigerant by a wide margin and is actually the refrigerant used at the facility under study (Calm J. M., 2006), (Study Site Energy Official, 2013).

Alternatively, desiccants are generally considered not as environmentally damaging as are refrigerants, and are not nearly as prone to leaks or removal from a system, and as mentioned previously are only changed every 5 to 10 years. Refrigerant systems as a whole lose an average of 0.5% of their refrigerant due to leaks per year (Calm J. M., 2008), (ASHRAE, 2012). It is apparent that both desiccant and mechanical dehumidification systems present both merits and disadvantages, but to fit the niche of dehumidification of 100% outside air in a hot & humid climate using heat pipes in with chilling dehumidifiers presents an excellent solution (Mazzei, Minichiello, & Palma, 2005), (ASHRAE, 2012), (Allen, June 14, 2013).

Chapter 3: Heat Pipes

As was mentioned earlier, heat pipes function with three heat transfer coils but where only one is actively consuming power. The configuration is shown in Figure 7 and the process is described with the Psychrometric Chart in Figure 8 (OA: outside air, RA: return air, MA: mixed air = outside + return, LA: over cooled air, SA: reheated air) (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008), (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011). The Psychrometric Chart can be read by starting at the MA and following the process (the green line) left, down, and right to the SA point. This allows you to track the state (temperature and RH) of the air as it undergoes dehumidification using heat pipes, the labels to the left also aid in designating which part of the dehumidification process the air in going through. The process is as follows (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008):

- Warm/hot and humid air enters the HVAC system, then passes through the pre-cool heat pipe coil which is thermally conductive and filled with a refrigerant (in the case of the facility it is R-22) which is in liquid form; so it absorbs heat and changes state from a liquid to a gas, thus pre-cooling the air.
- The refrigerant gas then migrates through connecting pipes to the reheat heat pipe coil,
- The air flows through the HVAC unit's cooling coil which is maintained between 50-55 °F leaving air temperature. The air is over-cooled and dehumidified when it leaves the cooling coil and thus needs to be re-heated.
- The air then flows into the reheat heat pipe coil, which reheats the air as a result of the refrigerant state change condensing from a gas to a liquid state due to the

entering cold air temperature. The reheat temperature increase is the same difference as the temperature drop in the precool heatpipe coil. This temperature difference (precool and reheat) is dependent on the entering air temperature (the warmer the air the larger the temperature difference) and he number of heatpipe rows (the more rows results in larger temperature difference). It should also be noted that if the HVAC system were to stop cooling the heat pipe sub-cool and reheat effect would also stop.

- Once the refrigerant has been condensed back into a liquid, it flows back into the pre-cool heat pipe by gravity (the refrigerant connection tubes are slopped downward from the reheat heat pipe coil to the precool heat pipe coil). At this point the heat pipe sub-cool/reheat cycle can start over again. . .
- If no additional heating is needed, the air is blown into the conditioned space, and if heating is needed it is usually provided by electric strip heating and then blown into the space.

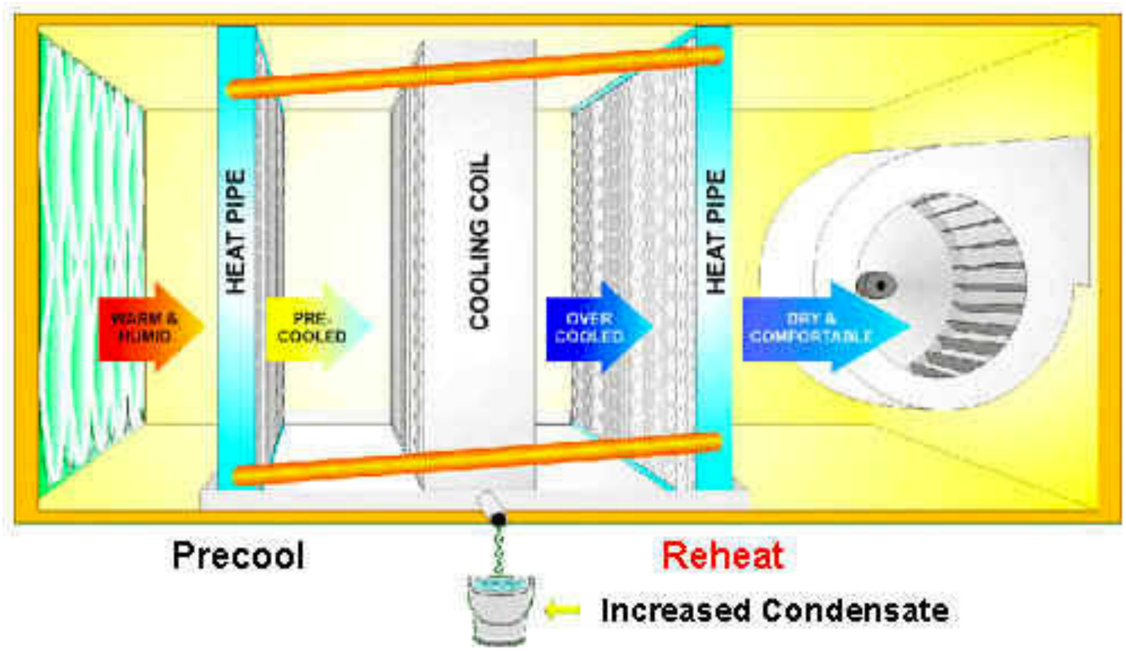

Figure 7: Heat Pipe Diagram - From (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011)

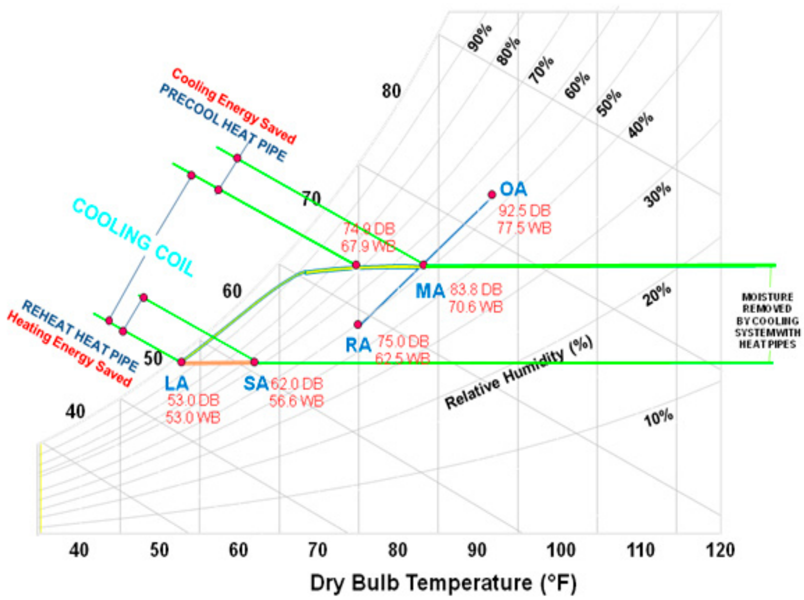

Figure 8: Heat Pipe Process Psychrometric Chart - (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011)

Florida Beach-Side Resort Heat Pipe Retrofit Case Study

Heat pipes are currently in use in a Florida beach-side resort and have been shown to significantly reduce energy consumption in most situations in which they are installed. A resort was retrofitted with heat pipes to supplement the conventional dehumidification process (sub-cool the incoming air to condense out moisture, then re-heat it using electric strip heating to the set room temperature) (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008). It was determined that, before the heat pipes were installed, an average of 733 kWh of electricity/day was used to re-heat the supply air before it entered the conditioned space, but after the heat pipes were installed the electric strip heater was not used at all (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008). Shown in Table 4 is a summary of improvements corresponding to installation of the heat pipes (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008):

Table 4: Impacts of Installing Heat Pipes at Florida Beach-Side Resort

	Electricity (kWh)	**Financial ($)**
Daily Savings	788	$100.09
Annual Savings	189,120	$24,022.00
Installed Cost	n/a	$66,920.00
Payback period in years (PBP)		2.786
Service Area Humidity Reduction	**Air Handler 8**	**Air Handler 9**
	4%	10%

As shown, the heat pipes significantly reduced energy consumption with a low PBP, and following payback the heat pipes are generating pure savings since there are minimal moving parts and little maintenance needed (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008).

Heat Pipe Efficiency Simulation Study Over Time Considering Increased Temperatures Due to Climate Change

Also worth noting is that heat pipe efficiency and effectiveness generally improve as temperature increases (Ahmadzadehtalatapeh & Yau, 2012), (Jouhara, 2009). These data were supported by simulating the running of heat pipes to dehumidify a 622 cubic ft/min load using climate data from 2000 and simulated climate data from 2020 and 2050, the resulting energy savings considering using and not using rejected heat from the process for other HVAC applications are below in Table 5 (Ahmadzadehtalatapeh & Yau, 2012), (Jouhara, 2009):

Table 5: Heat Pipe Energy Savings Due to Temperature Increase

Year	2000	2020	2050
kWh Saved	3,397	5,639	7,066
kWh Saved w/ reject heat reuse	6,794	11,278	14,132

The reason for this increase in efficiency is the fact that the ΔT (difference in temperature) between the refrigerant and the outside air is greater (Ahmadzadehtalatapeh & Yau, 2012). Because of the larger ΔT, the temperature drop across the evaporator (upstream) section of the heat pipe is expected to be greater, so there is more heat to replenish the over-cooled air in the condenser (downstream) section, thus requiring less electricity to re-heat the air (Ahmadzadehtalatapeh & Yau, 2012). Because heat pipes actually increase effectiveness and efficiency in higher temperature climates, they are particularly well suited for sub-tropical/tropical climates such as in Florida.

EPA Pensacola, FL Lab Heat Pipe Retrofit Effectiveness Case Study

A study performed by the Environmental Protection Agency (EPA) in one of their own buildings further demonstrates the effectiveness of heat pipes in Florida. In 1997, the EPA retrofitted one of their Pensacola lab buildings with heat pipes in order to aid in dehumidification and cooling as well as to research heat pipe effectiveness (U.S. EPA, 1997). The installation of heat pipes was found to reduce the relative humidity inside the building from 75% to 65% without affecting temperatures (U.S. EPA, 1997). If heat pipes had not been used, this level of dehumidification would have required an additional 20 tons of cooling capacity (U.S. EPA, 1997). The following effects in Tables 6 and 7 were seen after the heat pipes were installed at the building (U.S. EPA, 1997):

Table 6: Savings Seen During EPA Heat Pipe Study

	Cooling savings (as % of total previous installed cooling)	Reheat energy (Btu)	Projected Electricity (kWh)	Actual Electricity (kWh)	Financial	Reheat Energy (kWh)	Reheat Cost
Savings	19	4.6 million/day	153,780	230,750	$9,980.00	98,020	$4,900.00

Table 7: EPA Study Costs and PBP

	Heat Pipes	Traditional 20 ton Cooling	Premium paid
Cost	$42,000.00	$30,000.00	$12,000.00
PBP (years)	1.2		

Table 8: EPA Study Environmental Impacts

Substance	Reduction (in lbs)
CO_2	230,660
SO_2	2,330
NOx	850

Aside from the Financial savings, the positive environmental impacts were determined and are shown in Table 8 (U.S. EPA, 1997): As can be seen, if the fact that the site where this was tested was going to install additional cooling if heat pipes were not used the difference in cost is $12,000 between the two systems (U.S. EPA, 1997). Because the price difference is only

$12,000 and the annual savings is $9,980 (found in Tables 7 & 6 respectively) the PBP is only 1.2 years (found in Table 7), the period after payback would then be pure savings (U.S. EPA, 1997). These benefits described above are attributed to heat pipes not optimized for the building (U.S. EPA, 1997). It was determined that the building HVAC system had a flow rate of 19,100 cubic feet per minute (cfm), but after heat pipe installation the flow rate decreased to 16,800 cfm, this resulted in a negative pressure situation (the air pressure inside the building is lower than outside pressure, which leads to greater outside air infiltration) (U.S. EPA, 1997). If this were remedied by increasing the flow rate of the heat pipes, it would result in a 1.5 kW increase in fan load and a slight decrease in cooling and dehumidification due to the air not contacting the coil for as long. However, these are offset by an increased heat transfer rate between the evaporator and condenser which would result in a further savings of 2.7 kW (U.S. EPA, 1997). Heat Pipes also tend to have a direct correlation between their efficiency and the temperature of the air entering the system; this was demonstrated in the EPA study on September 21, 1997 between 8 am and noon when the temperature increased from 79°F to 92°F (U.S. EPA, 1997). Table 9 below shows the data from this period and demonstrates how the increased heat transfer rate caused by the increased temperature acts in favor of heat pipes and causes them to operate more efficiently (U.S. EPA, 1997):

Table 9: EPA Pensacola Heat Pipe Study - Sept 21, 1997 - 8am to noon cooling and load information

Outside dry bulb Temp °F	Total Cooling provided by systems (tons)	Load on Cooling Coils (tons)
79	78.8	67.6
92	80.7	60.8

In short, even though the demand for cooling/dehumidification increased throughout the day, due to the way heat pipes work the load demanded from the non-heat pipe systems decreased.

Early Case Study Conducted in Tampa, FL by W.H. Beckwith Showing Early Heat Pipe Potential in Hot and Humid Climates

Although heat pipes have been typically shown to be very effective in hot and humid climates, actual studies about their effectiveness in these types of climates have been limited until recently (Yau & Ahmadzadehtalatapeh, 2009). One of the earlier works that influenced the further development of heat pipes in Florida was by (Beckwith, 1997), (Yau & Ahmadzadehtalatapeh, 2009). In this study, heat pipes were tested in Tampa, Florida, and when they were retrofitted onto an existing system Table 10 shows the results (Beckwith, 1997), (Yau & Ahmadzadehtalatapeh, 2009):

Table 10: Comparison of HVAC System Effectiveness With and Without Heat Pipes

	Cooling Provided (kW)	Moisture Removal Rate (lb/min)
w/o Heat Pipe	35.2	0.487
w/ Heat Pipe	42.6	0.694

We can gather the following from the above data that the heat pipes provided 7.4 kW of "free" additional cooling and the heat pipes increased moisture removal capacity by 42.5% (Beckwith, 1997), (Yau & Ahmadzadehtalatapeh, 2009).

As can be seen with the above two studies Florida is very well suited for the installation of heat pipes because of the fact that it is hot and humid. With the likelihood that heat pipe efficiency will increase with global temperature increase (Ahmadzadehtalatapeh & Yau, 2012), it is fair to assume that heat pipes are going to become increasingly more efficient and have shorter undiscounted payback periods in Florida.

St. Petersburg, FL Museum Heat Pipe and VAV Retrofit Case Study

Another novel application of heat pipes was in St. Petersburg where heat pipes were retrofitted onto the existing HVAC system in an art museum (Shirey, 1993). The museum had the requirement that the spaces be kept at or below 50% relative humidity (Shirey, 1993). A VAV system was also installed alongside the heat pipes (Shirey, 1993). The result was a significant increase in moisture removal capability as well as a 12% drop in energy consumption (Shirey, 1993).

Dallas, Texas Heat Pipe Retrofit Simulation Case Study

Although many of the Florida studies garnered impressive results, one of the most impressive was a study that simulated in Dallas, Texas (Mathur, 1990). This study simulated the retrofitting of a 17.6 kW AC system that had an efficiency ratio of 8 with a 6-row heat pipe system to assess the improvement in cooling and dehumidification (Mathur, 1990), (Yau & Ahmadzadehtalatapeh, 2009). After the heat pipes were installed the moisture removal capability of the system rose by 0.295 lb/min, the efficiency ratio increased to 15.7 (96%), and it was figured that the retrofit would have a payback period of less than 1 year (Yau & Ahmadzadehtalatapeh, 2009), (Mathur, 1990).

Heat Pipe Technology White Papers Case Studies

Another important source of information was a study performed by one of the leading heat pipe manufacturing companies Heat Pipe Technology, which is based in Gainesville, Florida and is a subsidiary of MiTek (a Berkshire Hathaway company) (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011). This study took place in San Juan, Puerto Rico and concluded that the annual cost to run a conventional 10,000 cfm mixed air system was $53,900 with the set points in Table 11 below (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011).

Table 11: Set-points and Cost for San Juan Heat Pipe Study – Using Brute Force Dehumidification

Size (cfm)	Supply Air Dry Bulb Temp °F	Dew Point °F	Relative Humidity %	Annual Cost ($)
10,000	62	53	50	$53,900.00

This cost was obtained with the following information:

- The system was run 24/7
- The System used 50% outside air
- The return air being mixed with the outside air was 75°F dry bulb, 50% RH
- .7 kW/ton (kW/ton is the amount of energy consumed per cooling ton) cooling operated @ 70% heating system efficiency
- Electricity cost was $0.15/kWh
- Heating was $1.50/therm

Aside from being expensive, this brute-force method is generally forbidden by ASHRAE standard 90.1 with the exception of process applications (these are applications that must have a humidity within a certain range for manufacturing, medical, or safety reasons) (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011).

After the initial baseline was established with the “brute force”/conventional method of over cooling and reheating and obtaining the above results, heat pipes were installed and the annual cost dropped by 36% to $34,400 (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011). Interestingly enough the ASHRAE standard 90.1 that forbids using brute force methods for dehumidification recommends using “waste” heat and allows heat generated in the dehumidification process to count toward this “waste” heat (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011). The life cycle cost of the heat pipe was also calculated in the study and is shown in Table 12 for both a new installation and retrofitting of existing systems (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011). With the life cycle costs and the savings from installing the heat pipes calculated, it was possible to figure that the payback periods for installing heat pipes were 9 months for a new system and 13 months for retrofitting (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011).

Table 12: Heat Pipe Technology San Juan Study - Heat Pipe Life Cycle Costs

One Time Costs	
Installed Cost (New)	$13,000.00
Installed Cost (Retrofit)	$21,000.00
Recurring Costs	
Maintenance (hrs/year)	2
Maintenance Cost ($/man hour)	$80.00
Annual Parts Cost	$50.00
Total Annual Maintenance Cost	$210.00
Pay Back Period (Months)	
New	9
Retrofit	13

Another white paper from Heat Pipe Technology specifically examined the effect different variables had on the payback period of heat pipes and revealed a few interesting patterns (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The Study examined one variable/change to the base case at a time in order to determine how each variable by itself effects the payback period (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The base case that was established for the study had the following parameters found in Table 13 below (Brooke, Optimizing Wrap Around Heat Pipes, 2007):

Table 13: Heat Pipes Technologies - Heat Pipe Payback Period Base Case Parameters

Location	St. Louis, MO
Outside Air %	20
Flow rate/size (cfm)	20,000
Supply Air Temp °F	59
Heat increase needed °F	5
Return air temp °F	78
Return Air RH	50%
Central Plant Efficiencies	
kW/ton	0.75
heating	0.75
System Efficiencies	
motor	0.92
fan	0.7
Energy Costs	
Electricity ($/kWh)	$0.07
Heating ($/therm)	$0.80

It was found that with the above parameters the base case payback period is 26.1 months (Brooke, Optimizing Wrap Around Heat Pipes, 2007).

The first variable discussed was the geographic location, it was found to have a profound effect on the payback period (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The payback periods for the different locations are shown in Table 14 below (Brooke, Optimizing Wrap Around Heat Pipes, 2007).

Table 14: Heat Pipe Technologies Study – Payback Period Variation by Location

Location	Payback Period (Months)
Boston	28
Newark	23
St. Louis	26.1
Atlanta	19
Tampa	13
Puerto Rico	12

The variability in length of payback period above is due to the differences in cooling hours and the outside dry bulb temperature (which are generally related) (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The pattern found was that the closer to the tropics, the shorter the payback period (Brooke, Optimizing Wrap Around Heat Pipes, 2007). Obviously, the location of a project cannot be changed, but if a company owns properties in multiple regions this may help them prioritize projects; as is the case with the facility, the company owns and operates properties around the world (Brooke, Optimizing Wrap Around Heat Pipes, 2007).

The next variable examined was flow rate (cfm) (Brooke, Optimizing Wrap Around Heat Pipes, 2007). Flow rate was relatively straightforward: a larger flow rate means a larger project volume and thus larger project, and larger projects end up costing less on a per cfm basis due to pricing breaks (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The next factors that were examined were plant efficiencies and utilities cost, which responded predictably (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The way PBP reacted to existing efficiency (system kW/ton and heating plant efficiency) was that the lower the existing efficiency the lower the PBP (Brooke, Optimizing Wrap

Around Heat Pipes, 2007). The reason for this is that even though the heat pipes are operating the same regardless of plant efficiency, if the plant is less efficient, then the heat pipe is actually saving more energy since there is more being used than can be saved. The same principle also applies for cost; the higher the cost for electricity and heating, the lower the PBP because of the fact that even though the heat pipe operates the same regardless of cost, the higher the costs the more money can be saved (Brooke, Optimizing Wrap Around Heat Pipes, 2007).

A variable that had surprising results was face velocity; as the face velocity increased, the PBP decreased (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The reason this is surprising is that the face velocity is generally kept lower in order to prevent water blowing off the cooling coil (used for the dehumidification) and to keep airside pressure drop low (Brooke, Optimizing Wrap Around Heat Pipes, 2007). Designers and owners must decide if the decreased PBP is worth sacrificing the increased benefits of lower face velocities and find a balance that fits the needs of the project. Another interesting result was seen with the outside air % variable (Brooke, Optimizing Wrap Around Heat Pipes, 2007). What was found was that as the outside air % increased, so did the PBP, this was found to be heavily dependent on location though (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The reason for this was that the typical dry bulb temperature of return air in St. Louis was 78°F but the outside air had more hours between 55-75°F dry bulb than above, so the average entering air temperature was lower than the return air, thus the heat pipe wasn't as effective (Brooke, Optimizing Wrap Around Heat Pipes, 2007). If the location were changed as well during this test, it is likely that this result would also be different; if the outside air dry bulb temperature is higher than the return air temperature, you will likely see the opposite effect than was seen here: as the OA % increased, the PBP would decrease (Brooke, Optimizing Wrap Around Heat Pipes, 2007). The last factor that was taken into account was the reheat amount; it was found that the higher the reheat amount the shorter the PBP (Brooke, Optimizing Wrap Around Heat Pipes, 2007). What was interesting about the pattern observed is that there was a very large initial drop in PBP going from 3°F to 5°F reheat (PBP dropped from 40 months @ 3°F to 25 months @ 5°F). However, the PBP only dropped an additional 5 months when the amount of reheat was increased from 5°F to 11°F (Brooke, Optimizing Wrap Around Heat Pipes, 2007). As was shown above, many different variables factor into the PBP of heat pipes and must be considered when deciding if heat pipes are a worthwhile investment and also when designing the heat pipe.

As can be seen by all the studies above, heat pipes are an extremely effective method of dehumidification, and their effectiveness is actually enhanced by larger ΔT between the evaporator (upstream) and condenser (downstream) portion of the heat pipe. Such improvement in effectiveness is one reason why Florida and other hot and humid climates are ideal for heat pipe implementation; normally the extreme heat makes dehumidification and cooling difficult, but the extreme heat acts as a benefit for heat pipes as was shown in Table 9 (U.S. EPA, 1997). This effect will also only become more apparent in the future as the world starts to feel the effects of global climate change and the temperatures climb (Jouhara, 2009), (Ahmadzadehtalatapeh & Yau, 2012). This is well demonstrated by Table 5, which clearly shows how much energy can be saved even on a very small scale as temperatures increase over the next 37 years (Ahmadzadehtalatapeh & Yau, 2012), (Jouhara, 2009).

In addition to heat pipes becoming more effective with increasing air temperatures, they also enhance the dehumidification capabilities of the traditional dehumidification systems they are retrofitted to, as seen in Tables 4,10, and 12 (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008), (Beckwith, 1997), (Yau & Ahmadzadehtalatapeh, 2009), (Brooke, Critical Dehumidification Systems in Tropical Locations, 2011).

There is a caveat that needs to be mentioned, which is that it has been found in practice that additional reheat is generally needed if the inlet air starts at a temperature below the set point for the conditioned space (Allen, June 14, 2013), (Study Site Energy Official, 2013). This means that for a portion of the time at the facility, reheating is needed to bring the air temperature up to the desired temperature after it leaves the heat pipe. Additional electricity or natural gas is needed to run the reheat section depending on whether it is electric strip heat or a hot water loop (both are used in the facility). Because the additional reheat section is only being used to make up a few degrees difference (i.e. in one of the heat pipes at the facility the air leaves the heat pipe at 69.5°F but the required temperature is 72-74°F in every season except winter), it may be possible to make up the reheat energy deficit with renewable sources such as solar thermal or photovoltaic (Allen, June 14, 2013).

Chapter 4: Solar Energy and Dehumidification

Combining solar energy with dehumidification technology is not a new idea, as discussed previously in the desiccant system section. Low grade heat is often used to provide the heat in the regeneration air stream, and solar thermal provides an appropriate thermal match (Grossman, 2002). Solar can be and has also been used in cooling systems, but that isn't examined here (Grossman, 2002). There is one study that investigated integrating both solar PV and thermal into heat pipes and heat pumps. The paper investigated creating what is called a photovoltaic-solar-assisted heat pump/heat pipe system (PV-SAHP/HP) that could operate in three different modes: normal air/air heat pump, solar-assisted heat pump, and solar-assisted heat pipes (Fu, Pei, Ji, Long, Zhang, & Chow, 2012).

It was determined that with a total collection area of 50.2 ft^2 for solar thermal/hot water, and for PV a collection area of 31.3 ft^2 the system could heat 147.9 gallons of water to more than 94.1°F if the solar irradiation was greater than 87.4 MJ (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). With the solar radiation between 47.2 and 98.3 MJ during the test period, the heat (solar thermal) energy generated was between 7,030.9 and 29,868 Btu (7.4 and 31.5 MJ), operating at between 18.5% and 38.4% efficiency (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). The electrical energy generated under the above conditions was between 0.86 and 1.89 kWh (3.1 and 6.8 MJ), operating at 10.1 to 11.6% efficiency (Fu, Pei, Ji, Long, Zhang, & Chow, 2012).

It was also found that, although the heat pipes provided better performance at higher solar irradiation ranges, they performed rather poorly in lower irradiation conditions, but the solar-assisted heat pumps could provide residential hot water throughout the full range of solar irradiation levels (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). The cost of the system was $1904.80 for the photovoltaic and thermal (PV/T) system and $952.40 for the remainder of the system. It can therefore be assumed that the $952.40 would be significantly less if the heat pipes and heat pumps were already installed, because there would be less equipment to purchase and install) (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). The PBP for this system was calculated to be 14.1 years, but it must be stressed that this is highly dependent on the location (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). The location governs which of the three systems is operating, as well as the electricity/fuel cost which are significantly lower than the facility site at $0.03/kWh vs. ~$0.13/kWh (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). The fact that the HVAC portion (the heat pipe/pumps) was included in the cost, along with the information presented above, makes it unlikely that the PBP for such a system would be as high in the facility site (Fu, Pei, Ji, Long, Zhang, & Chow, 2012).

Since this system is very small in size, and the location is so different from the location being examined, this study works as a jumping-off point more than anything else. The work shows that it is possible to use a heat pipe system in conjunction with PV and/or solar thermal systems in order to provide hot water and electricity to a system. It is worth investigating to determine whether retrofitting the existing heat pipes with solar PV and/or thermal technology would be cost-effective. As it has already been shown to be possible to integrate solar PV/T (photovoltaic/thermal) into heat pipes, what must also be considered is the cost of the solar PV and thermal technology.

An excellent source of information regarding PV pricing and associated trends is the U.S. Department of Energy (DOE) Sun Shot—Photovoltaic (PV) Pricing Trends:

Historical, Recent, and Near-Term Projections report compiled in partnership with the National Renewable Energy Laboratory and Lawrence Berkeley National Laboratory (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). The DOE priced the installed cost of 150,000 installed PV systems in 2011 and generated the statistics shown below in Table 15 (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). What the data in Table 15 shows is that as the capacity of the installation increases as seen in the left column, the price of the PV system per watt decreases, as seen in the right column (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). This is going to be due largely to the economies of scale, as you buy more of something it becomes cheaper (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012).

Table 15: Average PV Installed Price - 2011

Capacity Range	Cost /Watt
<10 kW	$6.13
10-100 kW	$5.62
>100 kW	$4.87

The report stated that PV prices are on a downward trend, with the price decreasing an average of 5 to 7% per year between 1998 and 2011 depending on the sector in which it is installed and between 11 and 14% in 2010-11 alone (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). This amounts to the decreases in price shown in Table 16 attributable to 2010-11 (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012).

Table 16: PV Price Drop Information - 2010 to 2011

Capacity Range	Price Drop/Watt ($)	Price Drop/Watt (%)
<10 kW	$0.72	11
10-100 kW	$0.89	14
>100 kW	$0.77	14

The report also found that the average increase in PV installed capacity increased an average of 53% per year between 1998 and 2011, but increased 109% between 2010 and 2011 (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). It was also found that, on an international scale, the more mature markets (such as Germany) tended to have significantly lower prices than the U.S., so as the U.S. market matures with incentives and improved manufacturing techniques, this will lead to lower manufacturing costs and lower costs to consumers (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). With the lower manufacturing costs and costs to consumers it can be expected that the price of PV will decrease as the market matures (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). The report also determined that the installed cost of PV will continue to decrease at the 2010-2011 rate (see Table 16 above) or even faster, and that the installation of PV is also going to continue to increase (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). It can be gathered from this report, as pertains to this study, that even if prices cause integration of PV into heat pipes to be prohibitive at the current time, future likely decreases in prices may make incorporation of PV a more attractive option in the future. One must also consider not only the installed cost of a system, but incentives being offered that pertain to solar PV technology as well. One of the incentives currently in place is the Florida Renewable Energy Production Tax Credit (Fla. Stat. § 220.193, H.B. 7117). This incentive was originally enacted on June 19, 2006, and went into effect on July 1, 2006, but was allowed to expire in 2010; it was then reenacted April 13, 2012 and went into effect on July 1, 2012 (North Carolina State University; NREL, 2013). The Incentive provides for $0.01 per kWh of renewable energy produced in the form of a tax credit (North Carolina State University; NREL, 2013). The incentive is based on either total capacity for new facilities placed in service after May 1, 2012, or in the case of expanded facilities, the additional production of the expansions placed in service after May 1, 2012 (North Carolina State University; NREL, 2013). The maximum allowed credit is $1,000,000 per corporation with a statewide cap of $5,000,000 for fiscal year 2012/2013, and $10,000,000 every year thereafter (North Carolina State University; NREL, 2013). If there is not enough money to meet the total credits, a priority system is used with a maximum credit of $250,000 for the highest priority corporations (North Carolina State University; NREL, 2013). This incentive also has the stipulation that it cannot be used in conjunction with Florida's Renewable Energy Technologies Investment Tax Credit (North Carolina State University; NREL, 2013).

Additional incentives are also available from Florida electric power utilities; these are summarized in Table 17 below. It is worth noting that, for Tampa Electric Customers, any PV installations over 10 kW in installed capacity will still receive only $20,000, but for Duke Energy the price paid per watt simply decreases incrementally but still has a much higher maximum payout of $130,000. It is also important that FPL has no limit on the rebate but the amount paid decreases quicker than Duke Energy, and that the limit on Gulf Power is only 5 kW (North Carolina State University; NREL, 2012).

Table 17: Florida Utility Company PV Incentives

Company	$ given/Watt @ < 10 kW	$ given/Watt @ 10 - 50 kW	$ given/Watt @ 50 - 100 kW	Maximum Rebate Possible	Annual Budget
Duke Energy	$2.00	$1.50	$1.00	$130,000.00	$1,300,000.00
Tampa Electric	$2.00	n/a	n/a	$20,000.00	$1,000,000.00
	$ given/Watt @ ≤ 10 kW	**$ given/Watt @ 10 - 25 kW**	**$ given @ > 25 kW**	**Max Rebate Possible**	**Annual Budget**
Florida Power and Light (FPL)	$2.00	$1.50	$1.00	n/a	$15,500,000.00
Gulf Power	$2.00	n/a	n/a	$10,000.00	$435,000.00

Other important incentives to consider are from the Federal government. One of the better incentives currently in place with regard to solar PV is the Business Energy Investment Tax Credit (ITC) (26 USC § 48, H.R. 8 - American Taxpayer Relief Act of 2012) (North Carolina State University; NREL, 2013). The Business Energy ITC provides a tax credit of up to 30% of the installed cost for any PV for which construction is started by the end of 2013 (North Carolina State University; NREL, 2013). The Act was passed and enacted in 2008, but in January 2013 the rules governing who can claim the credit were changed. Projects that had started construction by the end of 2013 were allowed to claim the credit; previously, systems had to be operational in order to claim the credit (North Carolina State University; NREL, 2013). There are many opportunities to save money, both initially and over the lifetime of a solar energy installation, in Florida it is possible to profit from installation of PV if appropriate efficiency measures are taken and incentives utilized.

Another factor that should be considered is the actual energy production potential of a location, as can be seen in Figures 10 and 11, the global horizontal irradiance (GHI) and solar resource potential vary greatly with geography across the United States (Mapcruzin, 2012), (NREL | The Open PV Project, 2012), (Hoilett, Average Annual GHI from 2001-2012 and Installed PV Capacity in MW, 2013). It is important to note that sometimes policies bolster PV installation in places where the energy generation potential would otherwise make it cost-prohibitive. This is made more apparent in Figure 10 (which shows the GHI and installed PV capacity between 2001-2012) as well as in Figure 11, which shows the locations of the major PV installation sites and whether they would be supported by policy or the utility of the site (i.e. the ability to generate power) (Mapcruzin, 2012), (NREL | The Open PV Project, 2012), (Hoilett, Average Annual GHI from 2001-2012 and Installed PV Capacity in MW, 2013), (U.S. EPA, 2009).

As is seen in Figures 10 and 11, the two areas (the Northeast and Southwest) with the highest installed capacity are also the areas with the greatest GHI (Southwest) and the lowest GHI (Northeast) (Mapcruzin, 2012), (NREL | The Open PV Project, 2012), (Hoilett, Average Annual GHI from 2001-2012 and Installed PV Capacity in MW, 2013), (U.S. EPA, 2009). The only way the lowest GHI would likely demonstrate this level of installed capacity is if PV were incentivized (Solar Energy Industries Association, 2013), (U.S. Department of Energy, 2012). It is worth noticing in Figure 11 that policy-driven PV installations are dominant in the eastern half of the U.S. while utility scale or both are dominant in the western U.S. (U.S. EPA, 2009). It appears from examination of Figures 10 and 11 that the lack of installed PV in Florida is not a result of poor location, but rather is attributable to a lack of incentives to make an initial investment. The lack of incentives is shown by the incentive information discussed above and shown in Table 17;

Florida’s state government provides no incentives to help cover the installation costs of large PV projects, only the utility companies provide such incentives.

It may be the case that if an initial investment in PV is considered affordable with limited incentives, then commercial installations would likely see benefits from the installation of PV capacity in the form of energy savings and federal incentives. As has been shown using solar energy is steadily becoming a more viable option to generate electricity at all scales from utility to residential. This is being driven by technology improvements and cost reduction and also by policies incentivizing PV installation at the Federal, State, and Local levels. The following chapter will investigate the potential of supplementing heat pipe technology in the facility with solar energy generation technologies in order to sustainably supply the energy required to re-heat air in dehumidification systems.

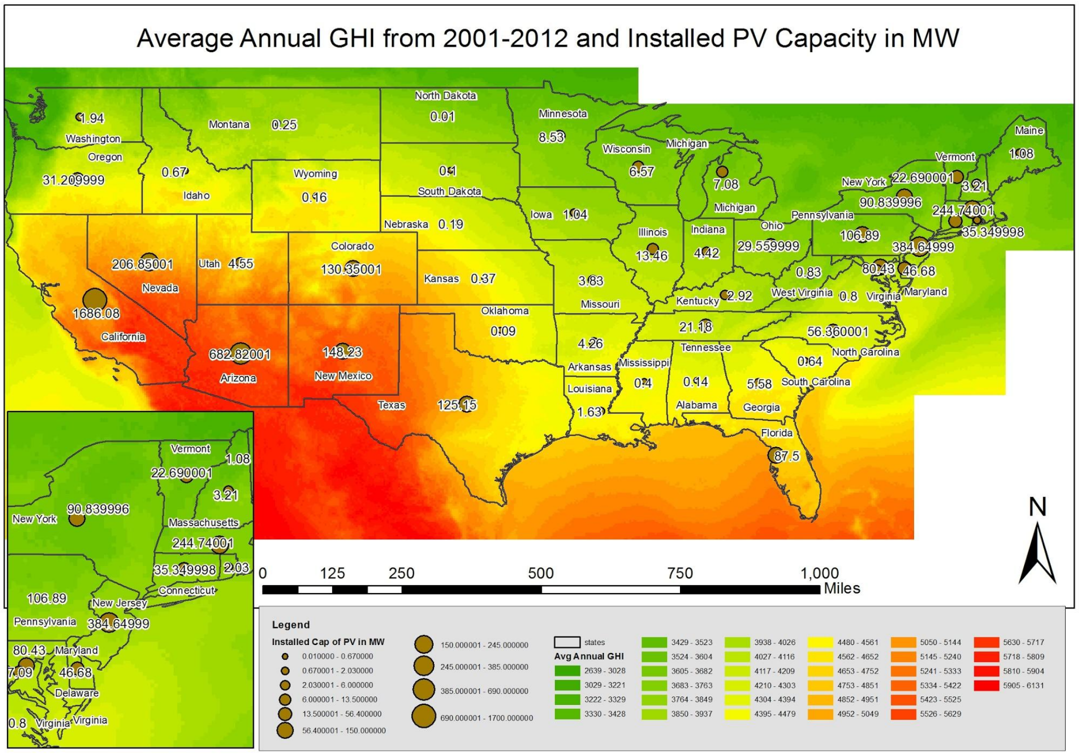

Figure 10: Average Annual Global Horizontal Irradiance (GHI) and Installed PV Capacity in MW

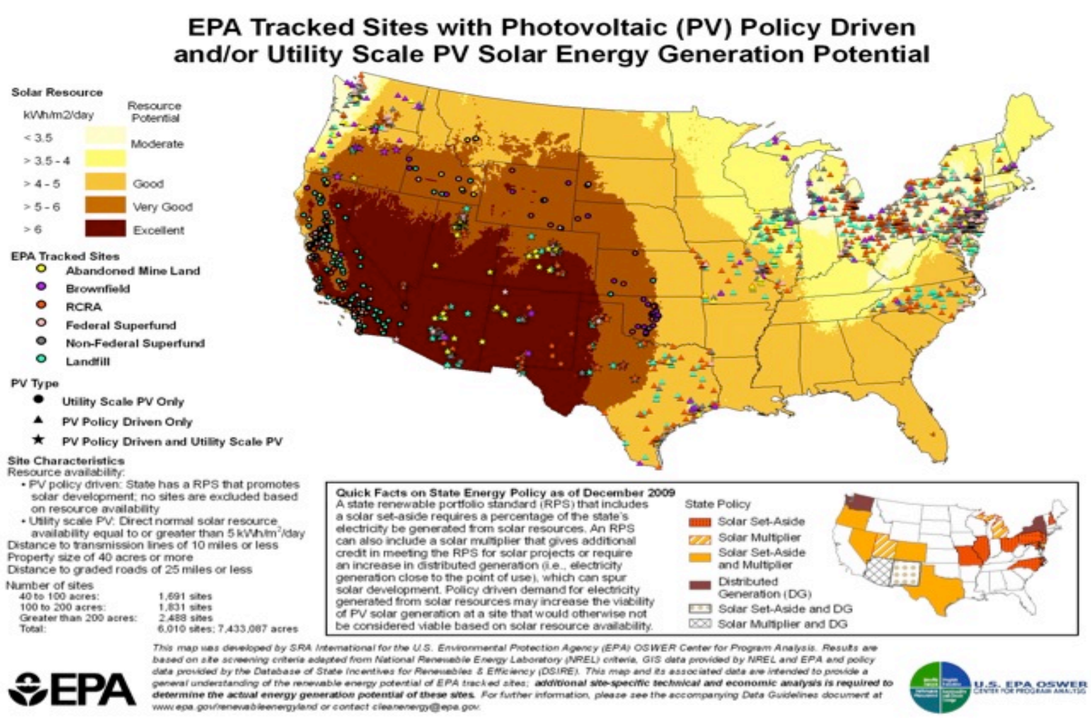

Figure 11: EPA Tracked PV Installations by Type and Location Energy Generation Potential

Chapter 5: Methodology, Results, & Analysis

Methodology

The purpose of this project is to assess the effectiveness and efficiency of heat pipes used for dehumidification at a facility in Florida. A range of parameters associated with the facility and the heat pipes were considered. These include the following:

- climate and weather patterns in the region of the facility;
- operational parameters of heat pipes including
  - minimum operating temperature;
  - type of air being provided to conditioned space by heat pipes;
  - criteria for applying re-heat;
  - method for applying re-heat;
- set points of the conditioned space including
  - desired temperature;
  - desired relative humidity;
- solar resource associated with the location.

Several other parameters were analyzed, these include

- priorities of management;
- solar PV efficiency and cost;
- solar thermal/hot water efficiency/capacities and cost;
- solar PV and thermal/hot water incentives.

Data Collection

The first element of the analysis was to gather critical data. The Site Energy Official (SEO) of the facility provided the data pertaining to the operational parameters of the heat pipes and the conditioned space set points (Study Site Energy Official, 2013). The SEO was deeply involved in choosing and designing the heat pipe systems to be installed, and therefore is highly knowledgeable about the workings of the system. He has cooperated throughout the study and is considered an extremely reliable source of data. Weather data were collected for the period between September 10, 2012 and September 13, 2013 from the website "Weather Underground" in the form of a comma separated value file (CSV) and includes the parameters found in Table 18 (Weather Underground, 2013):

Table 18: Types of Weather Data Gathered

Data Type	Units	Daily info gathered
Temperature	°F	high, low, mean
Dew Point	°F	high, low, mean
Humidity	% relative to saturation point	high, low, mean
Sea Level Pressure	inches Hg	high, low, mean
Visibility	miles	high, low, mean
Wind Speed	mph	high, low, mean
Precipitation	Inches	sum
Events	n/a	type of event (rain, fog, thunderstorm, etc.)

The weather data were collected within 12 miles of the facility and thus should be highly representative of the conditions at the facility (Weather Underground, 2013). The data for PV efficiency rankings were located on http://solarplaza.com and verified on the websites of the respective manufacturers as available (Solarplaza, 2012). The data for the solar PV energy generation potential in kWh/m^2/day were gathered from maps located on the National Renewable Energy Laboratory (NREL) website (NREL, 2013). The data were found in the form of maps that showed average measurements for kWh/m^2/day on a country-wide scale; the maps present both the annual average and monthly averages of kWh/m^2/day for 1998–2005 (NREL, 2013).

PV pricing information were gathered from the most recent (November 2012) NREL/SunShot report about the topic: "Photovoltaic (PV) Pricing Trends: Historical, Recent, and Near-Term Projections" (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). This report lists the current average prices for installed systems per Watt for 3 different sized systems – <10 kW, 10–100 kW, and >100 kW (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). The report also provides data on pricing trends for larger, utility-scale systems (2–10 MW), but it was stated that these prices were not as accurate as the smaller-scale prices due to the fewer samples and highly individualized nature of the utility systems (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012).

Information concerning the applicable incentives to install solar energy generation capabilities (mainly PV) was found on the NREL and NC State University Database of State Incentives for Renewables & Efficiency site, and then verified on the website of the body giving the incentive. All data gathered were organized and processed as described below.

Data Management and Initial Calculations

Data were pasted into and organized within a set of Excel® worksheets. The data from the SEO were provided in spreadsheet form and were processed as follows:

- Data irrelevant to this study i.e. those pertaining to locations other than the facility considered in this study were removed.
- Missing data values were acquired directly from the SEO or estimated by interpolation:
  - The Florida Beach-Side Resort location data were incomplete, but the report that discussed them (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008) included the daily total energy consumption for the heaters in kWh. After the SEO provided the hours per day that the heaters operated (recorded for a previous study), the heating capacity of the heaters was able to be calculated.
- Daily fuel and electricity usage were calculated.
- Costs (electric and fuel) to run the heaters in the systems were calculated.

The data pertaining to PV efficiencies were input into a separate sheet for later use. The weather data were imported from a CSV file into a separate sheet, thus making easily accessible all data relevant to this study.

Data Processing

After data were input and organized within Excel® worksheets, data processing was carried out as described below.

Weather Data and Heat Pipe Operational Parameters

The heat pipe operational parameters and set points were used in conjunction with weather data for the site in order to determine the following:

- the number of days that the heat pipes were not operating, because the temperature was below 65°F;
- the dates and number of days that heat pipes and heaters were operating at the same time, because outside temperature was below the set point temperature but above 65°F;
- the dates and number of days when heating was required, days that were below 68°F in winter and below 72°F during other periods;
- the dates and number of days that required dehumidification (days in which the RH was >50% at room temperature), and the number that did not.
- absolute humidity in g/m^3 (AH).
  - AH was calculated with the following formula in Excel®:

$$AH = 2.16679 \times \left[\left[\left(6.116441 \times 10^{\frac{7.591386 * D2}{D2 + 240.7263}} \right) \times \frac{H2}{100} \times 100 \right] \right) \Big/ ((273.15 + D2)] \quad (1)$$

where $D2$ represents the mean temperature in °C and $H2$ the mean relative humidity (Vaisala Oyj, 2013);!

- relative humidity at room temp given outside relative humidity by implementing the following steps (Mahidol Wittayanusorn School), (Vaisala Oyj, 2013):
  1. calculate the AH;
  2. calculate the saturated vapor density in g/m^3 (SVD) using the following empirical formula in which T_c represents temperature in °C:

     $$SVD = 5.018 + .32321T_c + 8.1847 \times 10^{-3} {T_c}^2 + 3.1243 \times 10^{-4} {T_c}^3 \quad (2)$$

  3. divide AH by SVD and multiply by 100 in order to determine RH.

Solar PV Cell Efficiency, Heater Energy Consumption, and Solar Energy Potential

The solar PV efficiency and cell size were used in conjunction with the heater energy consumption and energy generation potential data, to calculate the area and number of PV cells needed for placement of a sufficient number of solar cells to power the heaters, expressed in terms of both m^2 and acres. This was accomplished by dividing the power load of the heaters used in the heat pipes (L_h) by the efficiency of a given cell (E_{PV}), and then dividing that answer by (4 kWh/m^2/day ÷24 hours) (4 is the lowest expected kWh/m^2/day "possible" in the area). It was divided by 24 hours to account for the fact that the 4 kWh is generated over the course of 24 hours; this results in the area (in m^2) needed to generate the electricity required in a worst case scenario (where minimum average solar energy is provided at all times) (A_{PV}). The 4 kWh/m^2/day value is also used to aid in allowing for the space needed to avoid cross-shading, maintenance, inspection, and other factors that wouldn't be accounted for otherwise. It is also important to mention that the irradiance levels do not remain the same throughout the day, but the 4 kWh/m^2/day is an average throughout the day. The formula for this process is:

$$A_{PV} = \frac{L_h/E_{PV}}{\frac{4\ kWh/m^2/day}{24\ hours/day}} \quad (3)$$

Where A_{PV} is area (in m^2) needed to generate the electricity in a worst case scenario: L_h is the power load of the heaters (in kW); and E_{PV} is Efficiency of given PV cell (as a decimal).

Efficiency for the heaters was not factored in because, according to the SEO, the heaters are 100% efficient, thus no energy loss (Study Site Energy Official, 2013). The total number of cells was calculated by dividing the given cell area by the area needed to generate the needed energy.

Financial Data Processing

The approximate cost of PV installation was calculated using the average installed cost for solar PV installations greater than 100 kW according to Table 15, and multiplying it by the total power load of the heaters (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). This approach was taken because the costs given in the NREL report are for installed capacity, which is calculated to meet the energy needs of the location at which the PV is being installed, and already accounts for the efficiency of the cells and the PV energy generation potential of the location. The one-time utility incentives were calculated from those relevant to the facility site (North Carolina State University; NREL, 2013), (North Carolina State University; NREL, 2012).

The Florida Renewable Energy Production Tax Credit was somewhat more complicated to consider. Installed capacity and area calculated was assumed to be the minimum solar irradiance of 4 kWh/m^2/day on each day throughout the year as a conservative measure, in order to maximize assurance that the PV system accounts for all electricity required by the heaters. In reality, the average annual solar irradiance ranges typically between 5.5 and 6.0 kWh/m^2/day, dropping as low as about 4 kWh/m^2/day only in December (NREL, 2013). The most probable tax credit scenario was estimated as follows:

1. The most frequently-occurring solar irradiance value in a given month was determined within a 0.5 kWh/m^2/day range (for instance 5.5–6.0 kWh/m^2/day) the larger was used and the quotient of the most probable irradiance level and the

lowest irradiance level (4 kWh/m^2/day) was calculated, (i.e. if the given was 5.5-6 the calculation would look like 6/4 = 1.5) this quotient was used to calculate the most probable amount of electricity generated from the PV installation.

2. The monthly frequency distribution of the solar irradiance values throughout the year was then determined by calculating the number of days at each irradiance level. (i.e. the higher of the average irradiance levels was 5.0 in September and November, each has 30 days, so 30 + 30 = 60 days the irradiance level was at 5 kWh/m^2/day, so the frequency distribution for the irradiance level of 5 would be 60 days out of the year).
3. The following formula was then applied in order to obtain a dimension-less factor for the actual irradiance level versus the "worst case scenario" irradiance and also to obtain the total amount of electricity generated by the PV installation at the given irradiance level during the year:

$$\left[C_i \times \left(\frac{IR}{4}\right) \times 24\right] \times D_{IR} \quad (4)$$

Where C_i represents the installed capacity of the PV system (in kW) assuming a PV energy generation potential of 4 kWh/m^2/day; IR represents the most probable solar irradiance level for a group of days; and D_{IR} is the frequency distribution of a given solar irradiance level (IR). The result was multiplied by 24 in order to account for the fact that the system was sized to generate the energy needed to completely power the heaters all day (24 hrs), so the capacity needs to be multiplied by 24 hours to account for the fact that the system is providing the given "x" kWh/m^2/day every hour.

4. Equation 4 was then applied for each of the different irradiance levels.
5. The quantities determined in Step 4 were summed and then multiplied by $0.01 (the amount given per kWh generated by the tax credit) (North Carolina State University; NREL, 2013)
6. This final result represents the best estimate of annual tax credit accounting for the varying solar irradiance level throughout the year.

Since the heaters would likely be used even when dehumidification was not active, the annual cost savings (and thus the current annual cost) due to energy consumption is calculated according to the following steps:

1. Calculate the number of days below 72 °F for spring, summer, and fall; and below 68 °F for winter; 72 °F and 68 °F are the set point temperatures for the conditioned spaces and processed air is used to maintain these temperatures (Study Site Energy Official, 2013).
2. Calculate the electricity cost per hour to run the heaters by multiplying the sum of the total electric heating capacity (in kW) by 1.07 to account for transmission loss, then by the cost per kWh ($0.0655 per kWh) (Study Site Energy Official, 2013).
3. Multiply the result from Step 2 by 24 hours to determine the cost to run the electric heaters for one day.
4. Multiply the result from Step 3 by the answer from Step 1 which represents the cost savings attributable to reductions in energy required from the grid needed and associated energy costs to run the heaters throughout the year. In algebraic terms:

$$Cost_A = [(HC \times Cost_e \times 1.07) \times 24] \times A1 \quad (5)$$

where $Cost_A$ is the annual cost to run heaters (this would also be the amount saved due to energy savings if PV is installed); HC represents the total capacity of all electric heaters in kW; $Cost_e$ is the cost of electricity in \$/kWh; and $A1$ is the results from Step 1, the number of days during which heating was required.

The approximate initial cost with incentives was estimated by subtracting the two one-time incentives from the calculated installed cost of the PV. The undiscounted payback period (UPBP, in years) was then calculated using the following formula:

$$UPBP = \frac{(Cost_i - I_{OT})}{(FL_{TC} + S_A)} \quad (6)$$

Where $Cost_i$ is the initial cost (\$); I_{OT} is the amount paid out from the one-time incentives (\$); FL_{TC} is the amount paid out annually from FL renewable energy production tax credit (\$); and S_A is the annual savings from reduced energy usage (\$).

Emissions and Fuel Data Processing

The emissions and fuel usage values were more straightforward to calculate. Annual fuel usage was determined as follows:

1. Determine the total kWh consumed in a year by multiplying the total heating capacity in kW by 24, then by 1.07 (to account for transmission loss), then by the number of heating days

2. Multiply the result from Step 1 by the amount of oil required to generate 1 kWh as found on the EIA website (Energy Information Administration, 2012):

$$Oil\ Consumed = [(HC \times 24) \times 1.07 \times A1] \times P \quad (7)$$

Where *HC* is the total capacity of all electric heaters in kW; $A1$is the number of days that heating was needed (calculated for Equation 5 already); and P is the amount of oil (US gallons) need to generate 1 kWh of electricity.

3. Oil was assumed since it is the primary fuel used in FL for energy generation.

Annual emissions were calculated as follows:

1. Annual MWh consumption was estimated by multiplying the electrical load of the heaters (in kW) by 24 hours, then by 1.07 to account for transmission loss, then by the number of heating days, this represents the total kWh as above, then this is divided by 1000 to express as annual MWh consumption.
2. The MWh consumption is then multiplied by the emissions factor in lbs/MWh obtained from (EIA, 2010) as shown below:

$$Annual\ Emissions = \frac{[(HC \times 24) \times 1.07 \times A1]}{1000} \times EF \quad (8)$$

where *HC* is the total capacity of all electric heaters in kW; *A1*is the number of days heating was needed (calculated with Equation 5 already); and *EF* is the emissions factor in lbs/MWh.

3. Equation 8 was applied to Sulfur Dioxide, Nitrogen Oxides, and Carbon Dioxide

The natural gas used to heat the hot water used in heaters that don't use electric heaters was calculated in the following manner:

1. Capacities of the heaters were summed to determine total MBtu/hr
2. The total from Step 1 was multiplied by 1000 to get the Btu/hr
3. The total from Step 2 was multiplied by 24 to determine Btu/day
4. The result from Step 3 was then divided by 100,000 to determine the number of therms of natural gas consumed in a day
5. The number of therms was then multiplied by the billing rate for the natural gas (which is on a per therm basis and was obtained from the SEO) to get the daily cost.
6. The result from Step 5 was then multiplied by the number of days that required heating to obtain the annual heating cost
7. The formula for this would be:

$$Cost_{HW} = \left(\frac{HC_{ng} \times 1000 \times 24}{100{,}000} \times Cost_t\right) \times A1 \quad (9)$$

   where $Cost_{HW}$ is annual cost to run hot water heaters in heat pipes; HC_{ng} is total capacity of heaters (in MBtu/h) using hot water in the heat pipes; $Cost_t$ is cost per therm of natural gas; and is *A1* is the number of days heating was needed (already figured for Equation 5).

8. No additional calculations were performed with regard to this form of heating. After the annual cost to run hot water systems was calculated; the complexity of the system that would need to be installed was determined; along with the complexity involved in calculating a reasonably accurate capital cost was determined; it was decided that it would be impossible to calculate a reasonably

accurate initial cost with the available information. In order to further explain solar hot water heating and address why more detailed assessments are needed on-site, more research was performed to gain more insight into the components of a solar hot water system and how they are arranged in different types of systems. This information is presented in the first part of the results & analysis section.

Priorities for Analysis

Once all requisite values were calculated as described above, the resultant data along with the information described in previous chapters were analyzed with guidance from the facility and with consideration for specific environmental concerns that are described below:

- The SEO emphasized that financial savings were a major priority of the facility.
- The SEO also articulated that simplicity of the system is important because it minimizes the need for maintenance and reduces outages. These factors relate directly to minimization of costs and with respect to logistics given that the facility is in use twenty-four hours each day, 7 days each week (Study Site Energy Official, 2013). The facility operates in this manner throughout the entire year, with un-planned outages presenting extreme inconvenience, and where even routine maintenance requires a great deal of planning and coordination.
- The Facility also has corporate citizenship goals (described in more detail later) that must be met. The current goals have already been met but the SEO anticipates new goals being set in the near future. The current and future goals consider (or will likely consider) GHG emissions and renewable energy sources,

and the 2012 performance summary also stated that new goals would likely be announced in the near future.

- As mentioned, the environmental impacts associated with installation of PV and reduction of fuel requirements were considered.

With these priorities in mind, the data and associated information were analyzed to determine what, if any, improvements could be justified to reduce the cost of operating the heaters in the heat pipe dehumidification systems.

Results & Analysis

Hot Water Heat Pipe Heaters and Solar Hot Water

As seen in Table A1 in the appendix, there is a fairly even split between electric strip heat and central hot water being used to reheat air after it leaves the heat pipes, with 11 heat pipes using electric strip heat and 9 using hot water (Study Site Energy Official, 2013). It is also important to note that the annual cost to operate the hot water heating systems is more expensive than running the electric heaters as shown by Tables A1 and 19.

Table 19: Gas and Electricity Costs

Cost of electricity $/kWh
0.0655
Cost- $/Therm of Natural Gas
0.6038
Total Electricity cost/hr ($)
45.4439
Total Gas cost/hr ($)
56.80

This cost (282,195 Therms/year @ $0.6038/Therm with the 80% efficiency given by the SEO= $170,389.34 per year) suggests that pursuing a retrofit would likely yield economic benefit. The issue is that because of the complex nature of integrating solar thermal into an existing hot water system an accurate cost would be extremely difficult if not impossible to obtain without an onsite inspection performed by an experienced professional with access to all of the relevant system. (Study Site Energy Official, 2013).

One critical factor is that hot water flow rates through the heater in these systems can be varied (Study Site Energy Official, 2013). Such control suggests that the $170,389.34 per year in fuel cost is likely to be reduced in cases when the heaters do not need to operate at maximum capacity. Further, these systems do not often need to operate all day, as was assumed to be the case in order to estimate the annual cost of operation; these particular systems provide air to maintain the set-point temperature and need not necessarily provide neutral air or be on constantly, rather they operate as “on-demand” systems. This suggests that when the EMS/BCS determines a rise in entering air temperature is needed, it switches the system on and sets it to the appropriate heating setting, then turns the system off when it is determined that heating is no longer needed. This means that the annual cost is likely lower than the $170,839.34 estimated.

It is also important to note that these systems are already connected to a central hot water system that provides hot water for other uses. This means that the heating system only needs a heat exchanger, variable volume valve or pump, and some connections to the central system, making it much simpler than a solar system (Study Site Energy Official, 2013). This is because the type of solar thermal system needed (likely an indirect active system – a system that is actively pumped and where the water is heated by a refrigerant/coil that is heated by the sun and not directly by the sun) would need a refrigerant, multiple heat exchangers, storage tank for the water, a pump, and heating system exposed to the sun, in this case likely an evacuated tube system (U.S. Department of Energy, 2012).

The reason an active indirect solar thermal system would likely be best suited for this location is because the pumps help control the volume of water being used and with the hot water storage tank the water being heated can be stored for hours until it is needed (U.S. Department of Energy, 2012). The storage capability is important since the most heating would be needed at night (Study Site Energy Official, 2013).

Along with the above information, it is important to note that if the system were to be roof mounted structural loads would need to bear much high loads for the water storage tanks, or else have them mounted elsewhere on the premises. If they were mounted somewhere else on the premises then measures to inconspicuously install pipes and pumps would also need to be taken, adding more to the costs and complexity. The cost of a solar hot water system also varies greatly; it was found that in OECD countries the price ranges from \$460 - \$2050 per kW of installed heating capacity (REN21, 2013). This wide variance in price is due in part to the cost of labor, parts, availability of components, quality of solar resource, and specifics to the site (REN21, 2013). For example, it tended to cost more for a retrofit than a new installation due to the extra labor involved (REN21, 2013). This wide range of prices and the lack of available information regarding the hot water system and other location specific information make it impossible to assess an accurate cost.

Having the heaters in the central hot water loop may also invite a “single point of failure” situation where if the central hot water system goes down, so does the heater. However, this heating system is extremely well maintained; if it goes down, the chance of it going down on a day that would need heating is 34% of the chance of failure and 24% on a day that would require the heat pipes and heating (Study Site Energy Official, 2013). This single point of failure argument can also be rebutted with the fact that if solar thermal is installed it would add many more systems to fail (mainly pumps, tanks, and collectors for all the systems required) and add more components to maintain as opposed to one main boiler.

Another operating parameter to consider is the water temperature required for the heaters. The water in the hot water system is 160°F, which is fairly high for a solar hot water system solar as they typically heat water to the mid to high 90s°F (Fu, Pei, Ji, Long, Zhang, & Chow, 2012). This is well below the temperature required for the heaters in the heat pipes (Study Site Energy Official, 2013). There are two types of systems that can be used to heat water to the temperatures required, these are flat-plate collectors and evacuated tube collectors. For flat-plate collectors the highest temperatures are generally only reachable when the sun is directly aligned with the panels (Sunmaxx Solar Hot Water

Solutions, 2012). The evacuated tube collectors are generally capable of maintaining water at over 200°F even when outside air temperatures reaches below freezing (maximum temperatures range from 170°F to 350°F). They are also capable of generating these high temperatures at a variety of solar altitude angles; this would make them well suited for the heat pipe heaters (Sunmaxx Solar Hot Water Solutions, 2012), (U.S. Department of Energy | Energy Efficiency & Renewable Energy, 2013).

The downside with both of these systems is that they tend to be fairly expensive and large (Sunmaxx Solar Hot Water Solutions, 2012), (U.S. Department of Energy | Energy Efficiency & Renewable Energy, 2013). The size issue is especially important for the evacuated tube systems, as they are extremely efficient and run the risk of over-heating and over-pressurizing when the water gets too hot. Because of this it would need to be over-sized to avoid over-heating and pressurization or have a "dummy load" to dump excess heat into (which would add to the space requirement) (U.S. Department of Energy | Energy Efficiency & Renewable Energy, 2013). The issue of overheating is especially relevant because the heater would not likely be used when the solar irradiance is at its highest (this is because heating is usually not needed when solar irradiance is higher, this is shown in Table A2 in the appendix) unless they are connected to the existing hot water system and used for normal hot water loads (again adding more complications). Because of this the hot water would not get used as often and would build (along with pressure) for extended amounts of time; so this must either be compensated for when designing the system (as mentioned before) or the heater can be connected to the rest of the system and allow the water to be used.

One must also consider the time of year when these heaters are most often needed, which is typically during the colder months when a lower solar irradiance prevails. This means that in order to make them effective they would have to be sized to provide the hot water required during the coldest months and remain under-utilized during the warmer months when they are most efficient, or else used to supplement the boiler. This suggests that the solar thermal heating system would take up a fair amount of space, only be utilized for its original purpose about 34% of days, and be utilized to its full capacity an even lower amount of the time. This point is verified by examination of Table A2 which shows the mean temperature, date, and solar irradiance in $kWh/m^2/day$ and the pattern that is shown is that typically as the temperature drops, so does the solar power generation potential. Tables 26 & A2 also show and summarize the days during which heating would be necessary, and thus the days when the heaters would be active.

The existing technique of using hot water from a central source to meet the heat pipes re-heat requirements is cost effective. The fact that the annual cost is $170,389.34 to operate the hot water heaters makes it likely that installation of a solar thermal system to provide an alternative heating source would be a worthy investment. But without proper investigation into the buildings heating systems, it is impossible to obtain any semblance of an accurate capital cost. Further, solar thermal systems would require possible structural reinforcement, would involve a fairly large investment due to the relatively high temperatures needed, and would only be used ~1/3 of the time for their intended purpose. Thus, the current approach appears to be an appropriate one. Even though the current system requires the burning of natural gas to obtain heat, the fact that the system is already installed and simpler than any solar hot water system that could be added, suggests that the status quo is the most attractive option for the present time. But it is also possible that an in depth investigation into the costs and benefits of installing solar thermal heating capabilities would yield favorable results. Because of the complexity, lack of access to needed information (namely hot water system schematics and labor costs from local installers), the wide range of possible costs, and the lack of incentives, PV was focused on for this thesis since a usable conclusion and recommendation could not be drawn with available information.

Electric Strip Heat Pipe Heaters and Solar PV

As was observed in Table A1, eleven of the twenty heat pipes use electric strip heating (this is demonstrated by the fact that eleven heat pipes shown in Table A1 have values in the "Heating kW" column and nine have values in the "Heating MBtu/h" column), which would consume a great deal of energy and cost only slightly less than the hot water heating system as can be seen in Table 21. In Table 21, the Annual energy savings is equal to the annual cost of electricity to run the electric strip heaters. There is the potential for significant energy and cost savings.

Table 20: Solar PV Needed Capacity Summary

PV summary
PV Cell type
Sunpower - Gen3 Maxeon Cell
Cell area (mm^2)
16900
Cell efficiency (%/100)
0.225
Assumed kWh/m^2/day
4
Area of solar cells/panels needed (m^2)
18,501.33
Area of solar cells/panels needed (acres)
4.57
Cells needed
1,094,754

As shown in Table 20, the most efficient production PV cell model is the Sunpower Gen3 Maxeon cell, which can operate at a peak efficiency of 22.5% (Solarplaza, 2012), (Sunpower, 2011). Table 20 also describes the number of cells and associated area would be required to generate sufficient electricity at the most conservative value of irradiance to power all heaters at full capacity. The number of panels needed would be 15,205, 11,404, or 8553 depending on which size panel was used. Table 21 shows a summary of the financial implications of installing PV, as can be seen there are a possible $150,000 worth of one-time incentives to be used toward the installation price of the system. The cost of the system with incentives was calculated to be approximately $3.2 million.

Table 21: PV Financial Implications

Financial/Fiscal Results	
Approx. Initial cost for solar cells ($) w/o incentives	$3,378,806.00
Total Utility Incentives	$150,000.00
Florida Renewable Energy Production Tax Credit Max/year	$85,087.63
Approx. Initial cost for PV w/ incentives	$3,228,806.00
Annual energy savings (from reduced energy consumption)	$145,875
Undiscounted payback period (years)	13.98

One caveat to consider is that this analysis assumed an installed cost of $4.87/watt installed consistent with the NREL PV Price report by (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012) for systems >100 kW. The same report also mentioned that utility-scale installations (defined as installations between 2 and 10 MW in capacity) are subject to more uncertain prices depending on the specific situation (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). The NREL report gave the following prices per watt for utility-scale systems:

- $6.25 was the highest price
- $3.42 was the capacity-weighted average price in 2011
- $2.97 was the lowest price

If the initial costs (fewer incentives) are based upon the prices shown above, the results vary according to Table 22.

Table 22: Undiscounted Payback Period and Initial Cost for PV System Adjusted for Different Prices/W.

	Cost - $/Watt	Initial Cost for PV system ($)	PBP (years)
Minimum	$2.97	$1,549,810.00	6.71
Maximum	$6.25	$4,186,250.00	18.13
Average	$3.42	$2,222,796.00	9.62

As shown, there is significant cost variation to be considered when working on a large scale. As the installed capacity increases, the installed cost per watt generally decreases (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012). It is not unreasonable, therefore, to assume that a system of the size in question (693.8 kW) would have an installed cost closer to $3.42/Watt as opposed to $4.87/Watt (Feldman, Barbose, Margolis, Wiser, Darghouth, & Goodrich, 2012).

The initial cost is not the only financial aspect to be concerned with; one must also consider the annual energy savings, the continuous incentives, and ultimately the PBP. As shown in Table 21, the annual energy savings from reduced heater energy consumption were the same as the annual energy costs ($145,875) (Study Site Energy Official, 2013). Also shown in Table 21, the annual tax credit was found to be $85,087.63 per year, which would likely vary year to year due to variations in solar irradiation. When these annual savings are added together the total repeating annual savings/credit pertaining to the installation of PV cells ($230,963) is determined. From this the PBP of the system considering only the energy saved from reduced heater energy use is determined, which for an installed cost of $4.87/W would be 13.98 years, although this is a conservative estimate because of the economics of scale pattern mentioned above.

The PBP for three alternate prices are also shown in Table 22 above. There is a variation of over 10 years between the shortest and longest PBP, and a difference of 4.36 years between the 2 median PBPs; this suggests that it is critical to receive a highly accurate estimate from the company that installs the system.

A major benefit of PV in this application is that, when not being used to power the heaters, the PV system they can still be utilized for other purposes. This suggests that the PBP is likely to be even shorter than estimated due to additional energy savings. These savings are described in Table 23. As seen in Table 23, if the PV installations are connected to the balance of the electricity grid in the facility and utilized full-time, they have the potential to generate significant savings annually ($596,337). This would reduce the PBP value to the ones shown in Table 24.

Table 23: Financial fringe benefits of PV installation

Florida Renewable Energy Production Tax Credit Max/year	**$85,087.63**
Approx. Initial cost for PV w/ incentives	**$3,228,806.00**
Annual energy savings (from heater reduced energy consumption)	**$145,875**
Undiscounted payback period (years)	**13.98**
Possible annual Savings w/ PV being utilized full-time	**$596,337**
Undiscounted payback period - assuming best case PV energy generation (years)	**4.74**

Table 24: PBP for PV System Adjusted for Savings From Utilizing PV Full-time

	Cost - $/Watt	Adjusted PBP (years)	Difference (years)
Average @ >100kW Cap	$4.87	4.74	9.24
Minimum	$2.97	2.27	4.44
Maximum	$6.25	6.14	11.99
Average	$3.42	3.46	6.16

This potential for significant savings must be considered when considering a PV installation. Once a system pays back, future energy savings results directly in new revenues to the organization.

As shown, there is a considerable Financial benefit to be realized if solar PV system were implemented. If a system were to perform according to the estimations presented, it would be cost effective to pursue PV since the area required is only 4.57 acres (according to Table 20) out of more than 10,000 acres across the entire facility.

It is also important to consider environmental benefits. As shown in Table 25, the fuel oil used to generate the electricity required to power the heaters each year has associated with it emissions (EIA, 2010). Oil-burning power plants are the most common in the area where the facility is located, and in the state of Florida in general, unlike the majority of the country throughout which coal is the more common fossil fuel used for power generation (EIA, 2010), (Duke Energy, 2013).

Table 25: Oil Consumption and Emissions usage and possible prevention

Oil required to meet annual energy demands (US Gallons)	
	178,168
Sulfur Dioxide Emissions (lbs)	
	3,340.65
Nitrogen Oxide Emissions (lbs)	
	2,227.10
Carbon Dioxide Emissions (lbs)	
	2,652,470
Possible Oil savings w/ PV utilized full-time (US Gallons)	
	728,350
Possible Sulfur Dioxide Emissions Prevented (lbs)	
	13,656.6
Possible Nitrogen Oxide Emissions Prevented (lbs)	
	9,104.37
Possible Carbon Dioxide Emissions Prevented (lbs)	
	10,843,300

If it is considered that PV is utilized even when the heaters are not, than the potential oil and emissions savings can be seen in the green text in Table 25. An offset of pollutant emissions and fuel usage would result. There are several implications from this. First, improved air quality in the area nearby and downwind of the power plant(s) generating the electricity used at the facility would be realized. Second, positive publicity for improved “green” practices and becoming one of the first large facilities in the area to install large-scale renewable energy could result. Third, such implementation could be used to meet future citizenship goals.

Table 25 quantifies the magnitude of emissions that can be avoided by powering only the heaters with PV (shown in orange), and how much pollution and fuel usage can be offset by maximizing use of PV (shown in green).

Another important factor that must also be considered is the manner in which the heaters are controlled. The heaters are not simply “on/off” systems; they operate on a 4-stage protocol, meaning that they have 1 “off” and 4 “on” settings. The settings are as follows: off, 25% capacity, 50%, 75%, and 100% (full) capacity (Study Site Energy Official, 2013). This suggests that the heaters are likely not going to operate at full capacity every time they are activated, but instead at one of the lower capacities. The issue presented with this type of system is that the EMS/ECS controls the capacity at which the heater operates and does so “on the fly” (Study Site Energy Official, 2013).

The concern is that the EMS/ECS doesn’t record how often the heaters are at a certain capacity step, in fact, there are no meters on the heaters at all, electricity usage is just per building or system, not parts of a system (Study Site Energy Official, 2013). This implies that there is no way to procure an exact reading of the amount of energy the heaters consume without installing meters (which is time-, cost-, and permission-prohibitive in this case). Given that there is at present no way to install meters, assumptions had to be made about how often they operated, and thus it was assumed that when operating the heaters would be all day at full capacity. The reason for this assumption is that, in order to estimate the amount of time spent at each capacity, the author would have needed to be well versed in the operating parameters of the EMS/ECS system specific to the different heat pipes. Time constraints prevented such familiarization and thus the access to the system needed was unavailable.

This suggests that it is very likely that the heaters did not use as much energy as was assumed, thus estimations of emissions prevented, energy saved, fuel saved, and financial savings associated with heater consumption of non-renewable energy is likely to be lower than predicted. However, it is important to observe that power generated by PV, even if demanded less by the heaters, is still available to serve other purposes. Thus, there is the possibility of similar savings, but from a different energy usage sector. In summary, the fact that the heaters operate at incremental capacities and do not necessarily operate all day when they are on does not necessarily negate the savings calculated, but shifts a portion of the savings to another energy usage sector.

Weather Data and Heat Pipes vs. Desiccant Dehumidification

As seen in Tables 26 & A2 (Table A2 in Appendix) there was a wide range of temperatures and humidity levels observed. It can be determined from the data in Table 26 that almost every day the site requires dehumidification to some degree (346 of 368 days),

Table 26: Summary of Heating and Dehumidification Demands: 9/10/2012 - 9/11/2013

# Days Heat Pipe not Cycling (OA<65°F)	83
Heat Pipe and Heating Days	89
Days That Needed Heating (T<72°F or T<68°F in winter)	125
Non-Dehumidification Days	21
Dehumidification Days	346
Days Requiring Dehumidification w/o Heat Pipes	26
Days Using Heat Pipes and Heating	88

and that for those days needing dehumidification only 88 of the 346 days require heating. Another observation is that among the days requiring dehumidification, only 26 were too cold for the heat pipes to operate, this suggests that the heat pipes only use energy for heating 88 days of the period examined (Study Site Energy Official, 2013), (Weather Underground, 2013). All this implies that the facility has chosen one of the more appropriate dehumidification techniques for their location. One of the reasons for this assessment is that, as was mentioned in the previous chapters, the other major type of dehumidification that has been generally deemed suitable for this type of climate (desiccant dehumidification) has the disadvantage of requiring excessive cooling after the air has exited the dehumidification chamber (ASHRAE, 2012). The reason this excessive cooling is needed is because the air exits the dehumidification stage significantly hotter than it entered (ASHRAE, 2012). The need for such cooling increases the energy demand to a level much greater than when powering the heaters for the heat pipes since active cooling generally requires more energy than for active heating of a space (ASHRAE, 2012).

The last statement speaks to why Florida outpaces the rest of the nation in terms of electricity consumption, and why 27% of the electricity consumed in Florida is to power air conditioning systems (the largest percentage in the nation) (EIA, 2013). Another reason why the heat pipes are best suited for the facility being examined is that they offer a much simpler design than for a typical desiccant system. The simplicity of heat pipes becomes apparent when comparing Figures 6 and 7; in Figure 5, we note that there are at least four major moving parts in this system (Narayanan, Saman, & White, 2013)**:**

- the fan that moves the regeneration air;

- the fan that moves the cooling air;
- the fan that moves the supply air;
- the desiccant wheel (rotates).

In Figure 6, we see only one major moving part, the fan that drives the air through the heat pipe and into the space (Allen & Boll, [Florida] Heat Pipes for 100% Outside Air Units, 2008). Desiccant systems are also more complex in terms of maintenance (ASHRAE, 2012). As mentioned above, typical desiccant systems require desiccant replenishment or replacement every 5 to 10 years if routine filter cleaning and maintenance are done; if proper maintenance is not applied then the desiccant can become contaminated and may need to be replaced every ~2 years (ASHRAE, 2012). Alternatively, heat pipes require only routine inspection, leak repair (if/when it occurs), and cleaning; these maintenance steps amount to much lower annual labor and part (maintenance) costs (Allen, June 14, 2013), (Wu, Johnson, & Akbarzadeh, 1997). One potentially problematic aspect of heat pipes at the facility is that they use R-22 as the refrigerant (Study Site Energy Official, 2013). The small amount used and the fact that the system is closed and sealed, however, reduces the risk of refrigerant escaping and causing damage to the ozone layer and adding to GHG emissions (Study Site Energy Official, 2013).

Prior analyses and the weather data presented in Tables 26 & A2 show that cold weather dehumidification is not required in the facility because of the few number of days that are below the minimum operational temperature of the heat pipes that require dehumidification. The complexity of the desiccant systems also disqualifies it from being suitable for use in the facility given the simplicity of the heat pipes. The desiccant systems are also not well suited for use at the facility because they don't work well with 100% outside air systems, and all but four of the heat pipes are 100% OA (Study Site Energy Official, 2013), (ASHRAE, 2012). Desiccant systems are not suitable with 100% OA because of the heat gain. If the inlet air temperature is assumed to be 90°F @ 80% RH (typical daily conditions seen at the site during the summer) the outlet temperature would be 130°F, this would then need to be cooled to 72°F at the highest to be usable in the conditioned space.

Chapter 6: Conclusions and Recommendations for Possible Improvements to Heat Pipes

Heat Pipes with Hot Water Heaters

As discussed in the previous section, the hot water heat pipe heaters are already operating at an annual cost close to that of the electric heaters. Because of the annual cost (approximately $170,000) and the fact that it is integrated into central systems that operate with or without the heat pipes connected, any attempts to improve upon the effectiveness of these systems would likely be cost effective and is recommended under one condition. The condition is that a thorough investigation be conducted into the costs and benefits of retrofitting the heating system(s) with solar hot water capabilities as well as new and more efficient boilers (no information about the actual central boilers was provided aside from the information provided in Table A1 regarding the capacity of the heaters). Some of the costs for the solar hot water system may include the capital cost of the system itself, the labor involved installing the system, and the cost to reinforce the structures if needs be (this is far more likely than with PV since the storage tanks and pumps would weigh much more).

Heat Pipes with Electric Strip Heaters and Implementation of Solar PV

The electric heat pipe heaters present a major opportunity for improvement in efficiency and implementation of renewable energy technologies. The opportunity exists to implement renewable technologies to offset the energy required to operate the heat pipe heaters by incorporating solar photovoltaic. PV would have the potential to provide power for the heaters, and while the heaters do not operate at all times, the PV can be used to provide power for other applications as well. It is shown in Table 23 that the implementation of PV has the potential to save the facility a total of almost $600,000 including the roughly \$146,000 saved just for the heaters. The financial analysis presented justifies the recommendation that solar PV be installed with a PBP between 6.14 and 2.27 years, assuming an electricity cost between \$2.45/watt and \$6.25/watt with the likeliest cost falling between \$4.87/watt and \$3.42/watt). In terms of environmental benefits, to the PV system could save about 728,000 US gallons of residual fuel oil and prevent roughly 11,000,000 lbs of CO^2, 9,100 lbs of NO_x, and 13,600 lbs of SO_2 from being emitted and aid in meeting the citizenship goals for the facility.

Another consideration is the area needed to install the PV system, which for the facility being examined is not a barrier. The large size of the buildings involved would also make it possible to install a portion of the PV cells on the building roofs, reducing the ground area needed (Study Site Energy Official, 2013). Table 20 reinforces the notion that area requirements would not affect the recommendation to pursue PV as a viable option to implement renewable energy technology to save energy.

As mentioned above, implementation of PV technology would aid in meeting two of the current cooperate citizenship goals, and may even aid in meeting more depending on the details of future goals (Study Site Energy Official, 2013). It is likely that installation of PV at this time could be applied toward the next round of corporate citizenship goals, further strengthening the recommendation to install PV. The majority of corporate citizenship goals expire in 2013. It is therefore recommended that the installation of PV should be timed to occur when new citizenship goals are released, in order to meet the new goals and gain positive publicity.

It is strongly recommended that solar PV be pursued in order to provide the power needed to operate the heat pipe reheat coils and provide a clean source of power for other ancillary systems. This analysis identifies multiple benefits: almost $600,000 in energy savings per year with an undiscounted payback period of 6.14 years, significant GHG offsets, 728,000 US gallons of fuel oil conserved annually, and contributions toward two citizenship goals. The required land areas for the PV installation is estimated at 4.57 acres of land, It is also recommended that the SunPower Maxeon Gen3 silicone monocrystalline cells be considered for installation as they are the most efficient cells commercially and readily available (more efficient cells exist but are not in mass production at this time) (Solarplaza, 2012), (Sunpower, 2011).

Heat Pipe vs. Desiccant Dehumidification

Because of the simplicity and versatility of the existing system, it is recommended that the facility continue to use heat pipes for dehumidification rather than desiccant dehumidification. First, the major advantage of desiccant dehumidification is the extremely low minimum operating temperature as compared to mechanical systems (i.e. heat pipes); this benefit does not apply given the location of the facility. For example, there were only 23 days that required dehumidification where the temperature was below the operational temperature range of heat pipes.

Second, the desiccant system is more complex, featuring at least four major moving parts as opposed to only one for heat pipes (which isn't necessarily part of the heat pipe but outside the heat pipe). Third, the significant energy requirement to cool the air after it goes through the desiccant system (assuming typical site conditions the air would need to be cooled from ~130°F to 72°F after it exits the desiccant system due to all outside air being used) is not needed with mechanical systems.

Maintenance time and costs are also much lower for heat pipes as compared to desiccant systems given the lack of moving parts and the fact heat pipes are a closed, sealed system (Allen, June 14, 2013). Only occasional cleaning and leak repairs are required on the heat pipes, while desiccant systems require routine filter cleaning multiple times per year, and desiccant replenishment/replacement every two to ten years depending on the type of desiccant and whether the filters are properly maintained (Allen, June 14, 2013), (ASHRAE, 2012).

Finally, replacement of the system would require a fairly complex operation and major investment of time and labor, thus unlikely that management would consider it. This is attributable to the high regard in which they hold their appearance and the fact that guests are in many spaces throughout the day and year. To conclude, it is not recommended that desiccant systems be pursued; rather, heat pipes should continue to serve as the major dehumidification apparatus throughout the facility.

Final Remarks

Heat pipes represent one of the more elegant solutions to dehumidification ever devised as they have no moving parts, and can re-heat the incoming air to within a few degrees of the temperature required without any extra energy input. This is the main reason why the facility uses them. They operate in conjunction with hot water heaters and electric strip heaters in order to heat air to the required temperature on the occasion that heat pipes alone cannot support a load. It is recommended that the heat pipes equipped with hot water heaters be left to function as they do at present. But it is also recommended that an investigation be conducted to accurately assess the costs and benefits of installing solar hot water to replace the natural gas fired boiler currently providing hot water to the heat pipes.

It is also recommended that solar PV be considered as a means to implement renewable energy technologies and provide a renewable source of power for the electric heaters in heat pipes as well as to reduce the power drawn from the grid for other applications when the heaters are not being used. Finally it is recommended that heat pipes be kept in place since desiccant systems are more complex, require more energy, and their low-temperature operation is not relevant at this facility. The facility has already executed plans to conserve energy by implementing heat pipes and using single vendor ECS/EMS/BCS, among other measures. PV-driven heaters in heat pipes used for dehumidification provide an opportunity for the facility to further advance as a leader in implementing green/renewable energy technology.

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