ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

MSc THESIS

Osman KARA

DESIGN OF AIR-CONDITIONING SYSTEM WITH DEHUMIDIFICATION

DEPARTMENT OF MECHANICAL ENGINEERING

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ÇUKUROVA ÜNİVERSİTESİ
FEN BİLİMLERİ ENSTİTÜSÜ

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YÜKSEK LİSANS TEZİ

MAKİNA MÜHENDİSLİĞİ ANABİLİM DALI

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ABSTRACT

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DEPARTMENT OF MECHANICAL ENGINEERING
INSTITUTE OF NATURAL AND APPLIED SCIENCES
UNIVERSITY OF ÇUKUROVA

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In this study, a desiccant based air-conditioning system suitable for hygienic applications is considered. The moisture of supply air is reduced with a solid desiccant wheel and its temperature is decreased by “dry coil” of a vapor-compression refrigeration cycle. To enhance the performance of the system, some technologies such as “pre-cooling with outdoor air”, “waste cool recovery”, “pre-cooling of waste air with evaporative cooling” and “use of a cheap thermal energy source such as solar energy to remove the moisture from the desiccant” are utilized. Energy analysis of the system considered is carried out using a program written in FORTRAN language and suitability of the system is investigated for the health care facilities in which hygiene is crucially important.

Keywords: Air-conditioning, desiccant cooling, dehumidification, hygiene

ÖZ

YÜKSEK LİSANS TEZİ

NEM ALMALI BİR İKLİMLENDİRME SİSTEMİNİN TASARIMI

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Bu çalışmada, nem almalı (desisif) bir iklimlendirme sistemi ele alınmıştır. İklimlendirilen mahale gönderilen taze havanın neminin önce bir nem alıcı üzerinden geçirilerek düşürüldüğü ve daha sonra "kuru soğutucu serpantini"ne sahip buhar sıkıştırmalı bir soğutma çevrimi tarafından soğutulduğu bir sistem tasarlanmıştır. Sistemin performansını artırmak için, dış havayla ön soğutma, atık soğu geri kazanımı, atık havanın buharlaştırmalı soğutma ile soğutulması ve nem alıcıdan nemin uzaklaştırılmasında güneş enerjisi türü ucuz bir ısıl enerji kaynağının kullanılması öngörülmüştür. FORTRAN programlama dilinde yazılan bir program kullanılarak tasarlanan sistemin enerji alanizleri yapılmış ve ele alınan sistemin hijyenin önemli olduğu uygulamalarda kullanılabilirliği araştırılmıştır.

Anahtar Kelimeler: İklimlendirme, desisif soğutma, nem alma, hijyen

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NOMENCLATURE

V_{fresh}	: Flow rate of the fresh air (m^3/h)
V_{waste}	: Flow rate of the waste air (m^3/h)
V_{regen}	: Flow rate of the regeneration air (m^3/h)
t_{db}	: Dry bulb temperature (°C)
t_{wb}	: Wet bulb temperature (°C)
W	: Humidity ratio (kg/kg dry air)
φ	: Relative humidity (%)
SHR	: Sensible heat ratio of the air-conditioned room (-)
Q_{total}	: Total cooling load of the air-conditioned room (kW)
η_{heatex}	: Effectiveness of the heat exchangers (-)
η_{hum}	: Effectiveness of evaporative cooler (-)
η_{fan}	: Efficiency of the fans (-)
W_{fan}	: Powers of the fans (kW)
t_{dp}	: Dew point temperature (ºC)
P_a	: Partial pressure of dry air (kPa)
P_w	: Partial pressure of water vapor (kPa)
P_{atm}	: Atmospheric pressure (kPa)
W_s	: Saturation humidity ratio (kg/kg dry air)
P_{ws}	: Saturation vapor pressure (kPa)
$Q_{sensible}$	: Sensible cooling load (kW)
Q_{latent}	: Latent cooling load (kW)
Q_{gain}	: Heat gain from fan (kW)
d	: Density (kg/m^3)
h	: Enthalpy (kJ/kg)
m	: Mass (kg)
c_p	: Specific heat (kJ/kg K)
V	: Volume (m^3)
m	: Mass flow rate (kg / s)

1. INTRODUCTION

Desiccant dehumidification technology is emerging as a technically viable alternative for comfort conditioning in many commercial and institutional buildings. An attempt to improve the indoor air quality of buildings has resulted in increasingly stringent guidelines for occupant out-doors air ventilation rates. Additionally, revised building heating, ventilating, and air-conditioning (HVAC) design criteria based on regional peak dew point data highlight the importance of the latent (moisture removal) building load relative to the sensible (temperature) building load. Desiccant-based air-conditioning equipment is ideally suited to efficient dehumidification of building ventilation air, and when used in combination with conventional, vapor compression, air- conditioning systems, indoor temperature and humidity can be controlled independently (Sand et al., 1999).

Conventional HVAC systems remove moisture from the supply air, respectively the latent load from a space, by cooling the supply air below the dew-point temperature. This results in dehumidification of the supply air due to condensation at the cooling coil pipes. This kind of dehumidification usually requires coolant temperatures of about 6 - 12 °C, making mechanical cooling inevitable. Chillers can provide these temperatures without problems, although the energy use increases with decreasing chilled water temperatures. Alternative cooling strategies usually cannot provide such temperatures, thus, dehumidification by means of cooling below the dew-point is not possible.

However, instead of using mechanical cooling which normally implies a high power demand by the compression chiller, the supply air can also be dehumidified with sorptive materials, called desiccants. Taking advantage of a continuous and regenerative process makes it possible to control the indoor air humidity without using CFC, HCFC or HFC refrigerants which are harmful to the ozone layer (Singh, 2005). The materials used for desiccant cooling are neither hazardous to the environment nor toxic to people. The electric power demand of such a system drops remarkably when compared with a conventional HVAC system utilizing compressor-driven cooling. The first costs of a desiccant cooling system might be as high as for a conventional HVAC system with compression chiller and cooling tower. All the components utilized are highly developed and have been used for many years for air-conditioning or other air-handling processes. In fact, desiccant cooling systems are a combination of an adsorptive or absorptive dehumidifier and indirect and direct evaporative coolers. For air-conditioning purposes, rotary desiccant dehumidification units (desiccant wheel) which inner surfaces are covered with a solid desiccant, e.g., silica gel, are commonly used. Instead of using mechanical energy to cool and dehumidify the supply air with a compression chiller or cooling coil, respectively, desiccants (liquid or solid) are used to attract and retain the moisture (latent load). The most common desiccant cooling system for air-conditioning buildings is a combination of a rotary air-to-air heat exchanger covered with a solid desiccant, e.g., silica gel, a heater (gas-fired or water-heated) for regenerating the desiccant and both an indirect evaporative cooler and a direct evaporative cooler.

In this study, a desiccant based air-conditioning system suitable for hygienic applications is considered. The moisture of the fresh air is reduced using a solid desiccant wheel and then its temperature is decreased by the “dry coil” of a vapor-compression cycle. To enhance the performance of the system, some technologies such as “pre-cooling with outdoor air”, “waste cool recovery” and “pre-cooling of waste air with evaporative cooling” is utilized.

Energy analysis of the system considered was carried out and suitability of the system was investigated for the health care facilities in which hygiene is crucially important.

2. LITERATURE REVIEW

JAIN et al. (1995) investigated four cycles (the ventilation cycle, the recirculation cycle, the Dunkle cycle and the wet surface heat exchangers cycle) for various outdoor conditions (dry-bulb temperature and wet-bulb temperature) of many cities in India. The study was aimed at evaluating the influence of the effectiveness of heat exchangers and evaporative coolers on the cooling coefficient of performance (COP) as well as on the air volumetric circulation rate in different climatic conditions. The authors found the Dunkle cycle to have better performance compared to recirculation and ventilation cycles in all climatic conditions. But the cycle using wet surface featured the best performance with respect to all the three other cycles investigated.

KAVAK et al. (1997) proposed to examine desiccant-based cooling systems (DBAC) to determine their sanitizing effects on airborne microorganisms (bioaerosols). Their study focused on the use of desiccant-based cooling as a mechanism to control bioaerosols. As noted by Hines et al. (1990), desiccant systems can enhance the quality of indoor air and reduce the level of microorganisms. So their study was designed to examine this theory in both field and laboratory conditions. Their study consisted of two parts; to determine if bioaerosols are being reduced through field-operating DBAC systems and whether specific infectious microorganisms were being reduced through the system. They investigated three separate field and laboratory to perform on the DBAC systems. They showed that the DBAC system reduced airborne levels of bacteria and fungi in nearly every test (% 93) from their studies. Also they demonstrated that reducing the level of humidity, indirectly reduces the concentrations of bioaerosols. Since, microbial contaminants thrive in high humidity environments, the use of the DBAC system may reduce the numbers of bioaerosols in door air both; directly, through desiccation and indirectly, through dehumidification.

OLIVEIRA et al. (2000) modeled a new air-conditioning system using a liquid desiccant and needle impeller rotors. Experimental data obtained for different components, i.e., evaporators, absorber, were used in the model. System performance was quantified through the definition of thermal coefficient of performance. Simulation results have showed the effect of different system parameters: ambient temperature, ambient humidity and heat exchanger efficiency. By performing several simulations for an open cycle (without air recirculation), it was found that system performance is very sensitive to changes in both indoor and outdoor conditions.

DAI et al. (2001) conducted a comparative study of a standalone vapor compression system (VCS), the desiccant-associated VCS, and the desiccant and evaporative cooling associated VCS. The authors found an increase of cold production by 38.8–76% and that of COP by 20–30%.

ZHANG et al. (2001) proposed a new desiccant cooling system, a pre-cooling Munters environmental control (PMEC) cycle, which combines with chilled-ceiling panels. They provided the mathematical model of the system and used that system to predict the system performance under South-east China weather conditions using hour-by-hour calculations. Their result indicate that chilled-ceiling combined with desiccant cooling could save up to 40% of primary energy consumption, in comparison with a conventional constant volume all-air system.

CEJUDO et al. (2002) presented two methodologies for modeling a solid desiccant wheel: physical and neural network models. Experimental values are used to validate the physical model and to calculate the parameters of the neural network model. An important conclusion of this work is that special attention must be paid when calculating moisture content from measured values of dry bulb temperature and relative humidity.

HALLIDAY et al. (2002) conducted independently two feasibility studies of solar driven desiccant cooling in diverse European cities representing different climatic zones on the continent. The conclusion reached by the authors revealed that primary energy savings were achieved in all climatic conditions. A decline in energy savings were noticed in highly humid zones. This decline was attributed to the high temperature required to regenerate the desiccant in the climates of high humidity.

KANOĞLU et al. (2002) developed a procedure for the energy and exergy analyses of open-cycle desiccant cooling systems and it is applied to an experimental unit operating in ventilation mode with natural zeolite as the desiccant. The same procedure and formulations may easily be applied to the units operating in recirculation mode. Energy-based performance parameters such as the coefficient of performance (COP) of the unit and the effectiveness of system components are presented. Exergy destruction and exergy efficiency relations for the system and its components as well as the reversible COP of the system are derived. The energy and exergy formulations are applied to the experimental unit using the data collected during a typical operation of the unit. The unit has a COP of 0.35, a reverseble COP of 3.11, and an exergy efficiency of 11.1%. Desiccant wheel has the greatest percentage of total exergy destruction with 33.8% followed by the heating system with 31.2%. Rotary regenerator and evaporative coolers account for the remaining exergy destructions. They showed that an exergy analysis can provide useful information with respect to the theoretical upper limit of the system performance, which cannot be obtained from an energy analysis alone. The analysis allows us to identify and quantify the sites with the losses of exergy, and therefore showing the direction for the minimization of exergy losses to approach the reversible COP.

CAMARGO et al. (2003) presented a thermoeconomic analysis method based on the first and second law of thermodynamics and applied to an evaporative cooling system coupled to an adsorption dehumidifier. The main objective is the use of a method called exergetic manufacturing cost (EMC) applied to a system that operates in three different conditions to minimize the operation costs. Basic parameters are the R/P ratio (reactivation air/process air) and the reactivation air temperature. They showed that the minimum reactivation temperature and the minimum R/P ratio corresponds to the smaller EMC. This result can be corroborated through an energetic analysis. It is noted that this case is also the one corresponding to smaller energy loss.

ZHANG et al. (2003) presented a theoretical model to predict the heat and mass transfer process of the desiccant wheel in their study. The mathematical model was validated using a real desiccant wheel, and the calculation results were in reasonable agreement with the experimental data. The model is expected for assisting of designing desiccant wheel more easily and expediently. They analyzed the temperature and humidity profiles in the wheel during both the dehumidification and the regeneration processes. It is showed that there exists a hump curve, which is indicates the variation in the maximum humidity ratio, regarding air humidity ratio along the air channel in the regeneration process. The hump curve can play an important role for improving the studied desiccant wheel. The effects of velocity of regeneration air, inlet temperature of regeneration air and velocity of process air on the hump moving speed are investigated. They found that the desiccant wheel can have a high effectiveness of dehumidification if the regeneration temperature and the regeneration air velocity are also high.

MOGHTADA (2003) explained that comfort applications must be seperated from hospital climate and conditioning system's design, operating, assembling and maintenance. Poor designing or poor operating of the comfort climate may cause the decrease of efficiency and also may cause diseases. For this reason, several acquirements about DIN 1946/4 standarts which are accepted as the main referance in Europe and also in Turkey are specified. Five parameters (temperature, humidity percentage, fresh air percentage, particle and micro-organism, air temperature) must be taken under control in hospital systems at the same time. As a result it was denoted that the aim of taking these five parameters under control at the same time can be achieved by using special equipments and creating special designs.

SUBRAMANYAM et al. (2004) studied a desiccant wheel integrated air-conditioner for low humidity air-conditioning. The desiccant wheel dehumidifies and heats the supply air. The regeneration of desiccant is accomplished by the return air, which gets cooled and humidified. They showed that the proposed system can deliver supply air at much lower dew point temperature compared to the conventional system with a marginal penalty on COP. Its performance is better than the typical reheat system to provide the same low humidify levels.

DAOU et al. (2004) recalled the principles underlying the operation of desiccant cooling systems and discussed their actual technological applications. Through a literature review, the feasibility of the desiccant cooling in different climates is proven and the advantages it can offer in terms energy and cost savings are underscored. Some commented examples are presented to illustrate how the desiccant cooling can be a perfective supplement to other cooling systems such as traditional vapour compression air conditioning system, the evaporative cooling, and the chilled-ceiling radiant cooling. They showed that the desiccant materials, when associated with evaporative cooling or chilled-ceiling radiant cooling, can render them applicable under a diversity of climatic conditions.

SUBRAMANYAM et al. (2004) studied the effects of design variables of the air-conditioner, namely supply airflow rate, compressor pumping capacity (compressor speed) and desiccant wheel speed on its performance. They found that an optimum wheel speed of about 17.5 rph exists at which both moisture removal capacity and COP are maximum.

ZHANG et al. (2005) compared silica gel (SG), calcium chloride (CaCl2) and composite desiccant (SG–CaCl2) applied to a corrugated paper (CP) based desiccant rotary wheel for their abilities to remove moisture from wet air. Their experimental data shows that the CP–SG–CaCl2 material could attain equilibrium much faster than the other samples and its hygroscopic capacity is much higher than that of CP–SG.

CARPINLIOGLU et al. (2005) presented the methodology used in the analysis of experimental data (Yıldırım et al., 2002) to a different approach for the performance evaluation of similar systems. The analysis of the experimental data was directed to determine the functional relationships between system performance parameters (coefficient of performance (COP), cooling capacity (CC)) and the operation parameters (the rotational speeds of rotary regenerator and desiccant wheel, air mass flow rate in process and regeneration lines, and the regeneration temperature). They found a major function of mass flow rate of air almost independent of other operation parameters in their covered ranges by the correlations COP of CC and introduced performance parameter. However the suggested approach needs verification with further experimental data out of the covered ranges of the operation parameters, together with the varied physical system constraints.

ELSAYED et al. (2005) present theoretical investigation on the performance of air cycle refrigerator driving air conditioning system integrated desiccant system. They showed that the system performance is better than the conventional vapor compression air conditioning system with reheating coil. Also they have reported that the system has a potential to become a good alternative for the conventional vapor compression air conditioning system with low environmental load, especially for applications need high outdoor air ratio and a precise control of indoor humidity.

NIA et al. (2006) made a simulation of the combined heat and mass transfer processes that occur in a solid desiccant wheel by using MATLAB and predicted the temperature and humidity states of the outlet air from a desiccant wheel and the optimum speed of the desiccant wheel at different conditions (using numerical method). The model is validated through comparison the simulated results with the published actual values of an experimental work. Their method is useful to study and modeling of solid desiccant dehumidification and cooling system. The authors found the maximum difference between the results from the proposed correlations and simulation results is $\pm$2%, provided that the limited range of the variables are considered.

JIA et al. (2006) investigated a high performance rotary solid desiccant cooling system using a novel compound desiccant wheel. The unique feature of the desiccant wheel is that it can work well under a lower regeneration temperature and have a higher dehumidification capacity due to the contribution of the new compound desiccant materials. They analyzed a new compound desiccant wheel. Their experimental results indicate that the new desiccant wheel under practical operation can remove more moisture from the process air by about 20-40% over the desiccant wheel employing regular silica gel and the new composite desiccant is more hygroscopic and more easily regenerated. Also they showed that the dehumidification coefficient of performance (COP) for the new wheel is greatly better than the conventional one by using simulation results. The system COP may reach 1.28, about 35% higher over the system using silica gel.

KABEEL (2006) investigated the performance of a solar powered desiccant air conditioning system using a rotary honeycomb wheel, which is utilized for the regeneration and absorption processes, at different conditions of inlet air and radiation intensity. The system is highly effective in the regeneration process for all flow rates compared with the absorption process. The moisture change reached 11.5 g/kg air in the regeneration process and 74 g/kg air for the absorption process at a flow rate of 90 kg/h. Their results obtained the moisture removed for regeneration process at solar noon as a function of inlet air flow rate.

GOLUBOVIC et al. (2007) analyzed the potential benefits from separating process air stream at the exit of rotary dehumidifier into two streams. One air stream, hot and humid, is called purge air stream and other is remaining process air stream. The remaining process air stream has a lower temperature and humidity ratio as result of separation of initial hot process air stream. It is found that as the purge angle increases the exit humidity ratio of remaining process air stream decreases up to a point where it reaches a minimum. The purge angle for which this occurs is named “effective purge angle”. They developed an existing finite-difference model for simulation of desiccant wheel performance is extended to account for the separation of the process air stream at the exit of rotary dehumidifier and later mixing of purge air stream and outside air to form the regeneration air stream. The performance of desiccant wheel with heated “effective purge angle” is evaluated and compared with performance of the same wheel without purge angle at all. They found that having heated “effective purge angle” has overall positive effect on the performance of the rotary dehumidifier.

ELSAYED et al. (2007) analyzed theoretically the performance of air cycle refrigerator integrated desiccant system used to cool and dehumidify warehouse. Simulation analysis is carried out to calculate the system coefficient of performance (COP), cooling effects and the humidity change under different conditions. They found that from the simulation analysis result, the desiccant system has the ability to supply air to the dock area at very low humidity (less than 0.4 g/kg). The COP of system increases due to the exhaust heat recovery on the desiccant system, and this enhancement can be more than 100%. Also COP of the system depends on the rotating speed of the rotor, air velocity and rotor length. They showed that COP of the proposed system is greater than that of a conventional system under the same operating conditions.

WEIXING et al. (2008) proposed a new type of modified cross-cooled compact dehumidifier (CCCD) and developed its mathematical model. The dehumidifier is constructed on the basic structure of plate fin heat exchanger with silica gel elaborately glued on all the metal surfaces (optimum 0.3-0.5 mm) of the process-flow channels. They simulated the cyclic dynamic performance of the CCCD by implicit finite difference method. The simulated dehumidification performance is compared with the experimental results, and showing good agreement within 7% deviation. Under high humidity ratio conditions, the DCCCCD (desiccant coated cross-cooled compact dehumidifier) shows very good performance and its moist removal amount within effective dehumidification period could reach as high as 12.4%.

3. MATERIAL AND METHOD

3.1. Desiccant Air Conditioning Systems

In 1990 about 4.12 EJ (exajoules) (13%) of primary energy were used for air conditioning (cooling and ventilation) of buildings. The energy used for air conditioning buildings is expected to rise in the 1990's and beyond as the population shifts to the warmer southern states and personal computer use increases in office buildings (Analysis and Technology Transfer Report, 1991). The air conditioning industry is faced with several challenges: increased energy efficiencies, improved indoor air quality, growing concern for improved comfort and environmental control, increased ventilation requirements, reduction of chlorofluorocarbons (CFCs), and rising peak demand charges. New approaches to space conditioning will be required to resolve these economic, environmental, and regulatory issues. Desiccant cooling and dehumidification, a technology known for some time, may provide important advantages in solving air conditioning problems (Pesaran et al., 1992).

Desiccant cooling technology provides a tool for controlling humidity (moisture) levels for conditioned air spaces (Holcomb et al., 2000). Desiccant cooling is a potentially environmentally friendly technology, which can be used to condition the internal environment of buildings without the use of traditional refrigerants (CFC free). It is becoming one of the most promising alternatives to conventional cooling systems. Building heating, ventilating, and air conditioning (HVAC) systems indicated that the initial cost for the conventional cooling equipment was greatly reduced by using desiccant technology because of downsized compressors, fans, and ductworks. This cost reduction was more than enough to offset the cost of desiccant equipment. Besides, the system operation cost was also reduced. All these indicate that desiccant systems are also cost effective (Hardy, 2003).

In a typical desiccant system, the moisture (latent load) in the process air is removed by a desiccant material in a dehumidifier, and then the temperature (sensible load) of the dried process air is reduced to the desired comfort conditions by sensible coolers (e.g., heat exchangers, evaporative coolers, cooling coils). The latent and sensible loads are handled separately and more efficiently in components designed to remove that load. The desiccant in the dehumidifier is reactivated by application of heat to release the moisture, which is exhausted to the outdoors. The heat for reactivated can be provided from a number of energy sources such as solar, waste heat, natural gas, and off-peak electricity. The desiccant can be either solid or liquid. In solid desiccant systems, air is passed through a bed of adsorptive material. Air is dried and moisture is adsorbed by the desiccant. When the desiccant is saturated, hot air is passed through the bed, releasing the moisture. Typically, the desiccant is loaded into a wheel that rotates between the process and reactivated airstreams.

In a liquid desiccant system, a concentrated liquid desiccant is sprayed in a contactor (containing cooling-coils or packing materials) to absorb moisture from humid air stream passing through the contactor. The liquid desiccant leaving the contactor is diluted with removed water. The diluted liquid desiccant is heated or sprayed into the regeneration air stream to remove and release the moisture and re-concentrate the liquid desiccant (Khamis, 2000).

The best circumstances for use of desiccant cooling and dehumidification technology are: need for humidity control, economic benefits from using low humidity, high latent load versus sensible load, low thermal energy cost versus high electric energy cost, and need for dry cooling coils and duct work to avoid microbial growth. Use of desiccant technology to provide dry, cool air for supermarkets, hotels and motels, office buildings, hospitals and nursing homes, restaurants, health clubs and swimming pools, and residences in humid climates (Pesaran et al., 1992).

3.2. Types of Desiccant Cooling Systems

There are two types of desiccant systems: liquid (sorbent) and solid. Liquid desiccant systems remove more moisture from ventilation air than do solid desiccant systems; the air produced by solid desiccant systems, however, is warmer than the air produced by liquid desiccant systems.

Solid desiccant systems operate in a manner similar to liquid desiccant systems, but use a desiccant coating on a rotary enthalpy heat exchanger. So, The desiccant in the dehumidifier is regenerated (reactivated) when heat is applied to release the moisture, which is exhausted outdoors. The heat for regeneration can be provided from a number of energy sources such as natural gas, waste heat, solar, and off-peak electricity.

3.2.1. Solid Desiccant Systems

In solid desiccant systems, air is circulated through a bed of absorptive material like silica gel or zeolite. As the moist air passes through the bed, it gives up water vapor to the desiccant. Then the saturated desiccant is heated. This releases moisture to a different air stream, drying the desiccant so it can be used once again. Typically, the desiccant is loaded into a rotating tray or impregnated into a honeycomb-form wheel, which rotates slowly between the dry air stream (process) and the heated air stream (regeneration). This constant regeneration allows the equipment to provide a continuous stream of dry air to the air-conditioned space (Mei et al., 1992).

There are four general types of solid desiccant systems available today. They are called:

a) Solid packed tower

b) Rotating horizontal bed

c) Multiple vertical bed

d) Rotating desiccant wheel

These techniques are explained briefly below.

3.2.1.1. Solid Packed Tower

The dehumidification system, shown in Figure 3.1., consists of two side-by-side cylindrical containers filled with solid desiccant and a heat exchanger acting as a desiccant cooler. The air stream to be processed is passed through dry desiccant in one of the containers, while a heated air stream is passed over the moist desiccant in the other. Adsorption (1-2) takes place in the first container, desorption (2-3) in the other container, and cooling (3-1) occurs in the desiccant cooler. The function of the two containers is periodically switched by redirecting the two air streams (Lavan et al., 1999).

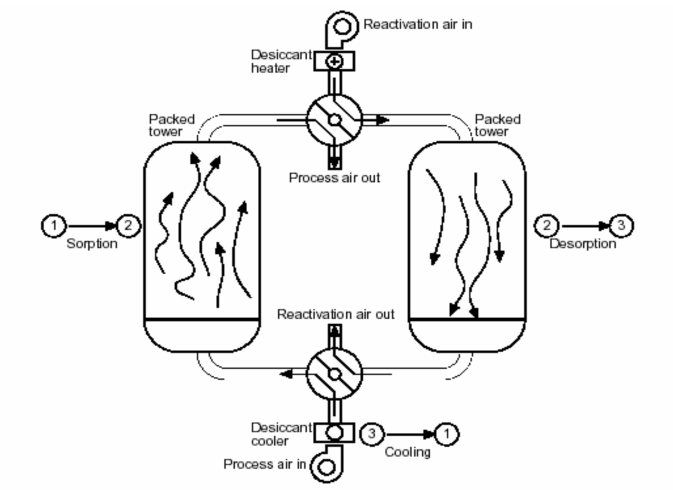

Figure 3.1. Solid packed tower

3.2.1.2. Rotating Horizontal Bed

A typical rotary solid desiccant dehumidifier is shown in Figure 3.2. In this device, dry, granular desiccant is held in a flat, segmented rotary bed that rotates continuously between the process and regeneration airstreams. As the bed rotates through the process air, the desiccant adsorbs moisture. Then the bed rotates into the regeneration air stream, which heats the desiccant, raising its vapor pressure and releasing the moisture to the air.

The process and regeneration air heats and cools the desiccant to drive the adsorption-desorption cycle. The moisture is removed through a process of continuous physical adsorption on a continuous basis (both, counter flow and parallel flow options are available).

The adsorption of moisture and regeneration of desiccant take place continuously and simultaneously without any cross mixing of the process and regeneration air streams.

To increase capacity, the diameter of the rotating bed can be increased to hold more desiccants, or increase the number of beds stacked on top of one another can be increased. Both options are not practical if very large volumes of air need to be dehumidified. If the desiccant is evenly loaded through the trays, the rotating horizontal bed provides a fairly constant outlet moisture level, and a high airflow capacity can be achieved in less floor space than with dual-tower unit. The rotating horizontal bed design offers a low first cost. The design is simple, compact and easy to produce as well as install and maintain.

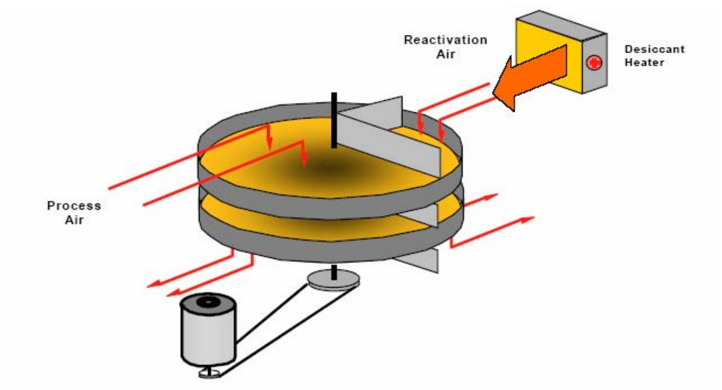

Figure 3.2. Rotating horizontal bed

3.2.1.3. Multiple Vertical Bed

The Multiple Vertical Bed (MVB) design is a 'fairly new' but 'proven' concept with the combined better features of packed tower and rotating horizontal bed designs in an arrangement that is well suited to atmospheric pressure dehumidification applications, and yet can achieve very low dew points (Figure 3.3). The single or double tower is replaced by a circular carrousel with eight or more vertical beds (towers) that rotate, by means of a drive system, between the process and regeneration air streams.

This design can achieve low dew points because leakage between process and regeneration air circuits is almost negligible. Also because the beds are separate and sealed from one another, the pressure difference between process and regeneration is not so critical; so airstreams can be arranged in the more efficient counter-flow pattern for better heat and mass transfer. Like the rotating bed, the ratcheting, semi-continuous regeneration of the desiccant provides a relatively constant outlet air moisture condition on the process side.

The “MVB” design allows for low replacement cost of desiccants as well as large savings in energy and performance improvements at low dew points, especially if the equipment incorporates a heat pipe heat exchanger in the regeneration air circuit.

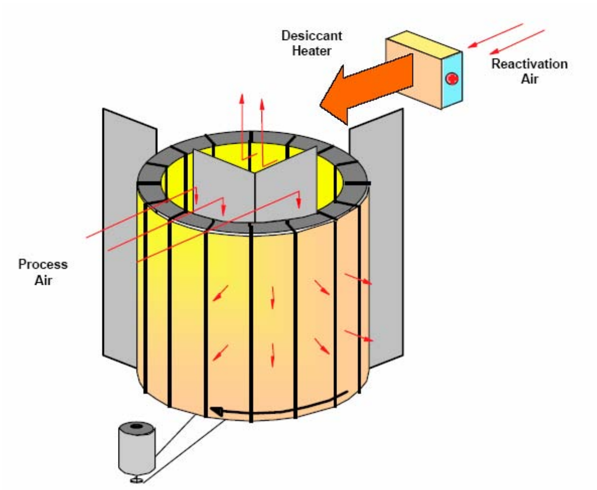

Figure 3.3. Multiple vertical bed

3.2.1.4. Rotating Desiccant Wheel

The adsorption dehumidification is a physical process that fixes molecules of an adsorbate (water, in this case) on the adsorbent surface, usually porous and granulated. Desiccants attract moisture from the air by creating an area of low vapor pressure at the surface of the desiccant. The partial pressure of the water in the air is high, so the water molecules move from the air to the desiccant and the air is dehumidified (Harriman, 1990). Thus, the essential characteristic of desiccant is their low surface vapor pressure. If the desiccant is cool and dry, its surface vapor pressure is low and it can attract moisture from the air, which has a high vapor pressure when it is moist. This process is regenerative because the adsorbent material, after saturated by the humidity, sets the water free, when submitted to a heat source (de-sorption). The thermal energy to the regeneration can be obtained by electric power, water vapor or hot air.

A rotary solid desiccant dehumidifier is shown in Figure 3.4. Unlike the intermittent operation of packed towers, rotary desiccant dehumidifiers use a wheel (or drum) that rotates continuously and delivers air at constant humidity levels.

Desiccant wheels typically consist of very fine desiccant particles dispersed and impregnated with a fibrous or ceramic medium shaped like a honeycomb or fluted corrugated paper. The wheel is divided into two segments. The process stream flows through the channels in one segment, while the regenerating (or reactivating) stream flows through the other segment (Lavan et al., 1999).

A typical configuration uses a rotary desiccant wheel that moves slowly and continuously between two crosswise air fluxes, the process and regeneration airstreams. The process air flow through the flutes formed by the corrugations, and the desiccant in the structure adsorbs the moisture from the air. As the desiccant picks up moisture it becomes saturated and its surface vapor pressure rises. Then as the wheel rotates into the regeneration air stream, the desiccant is heated by the hot regeneration air, and the surface vapor pressure rises, allowing the desiccant to release its moisture into the regeneration air. Following regeneration, the hot desiccant rotates back into the process air, where a small portion of the process air cools the desiccant so it can adsorb more moisture from the balance of the process air stream.

In typical applications, 75% of the air passage is used by the process air and 25% by the regeneration air. Desiccant rotary wheels do not reduce the air energetic load; they only change latent heat (humidity) by sensitive heat (temperature).

The addiction of a desiccant dehumidifier to an evaporative cooling air conditioning system provides a humidity control separate of the temperature control. It is especially good in applications where the latent thermal load is high comparing to the sensitive load, or when they get the maximum in different time. Usual applications are supermarkets, shopping centers, theaters, hospitals, hotels, motels and officers buildings. The energy conservation wheel (ECW) is a rotary counter flow air-to-air exchanger used to transfer both sensible and latent heat between supply and exhaust air streams (Camargo, 2005).

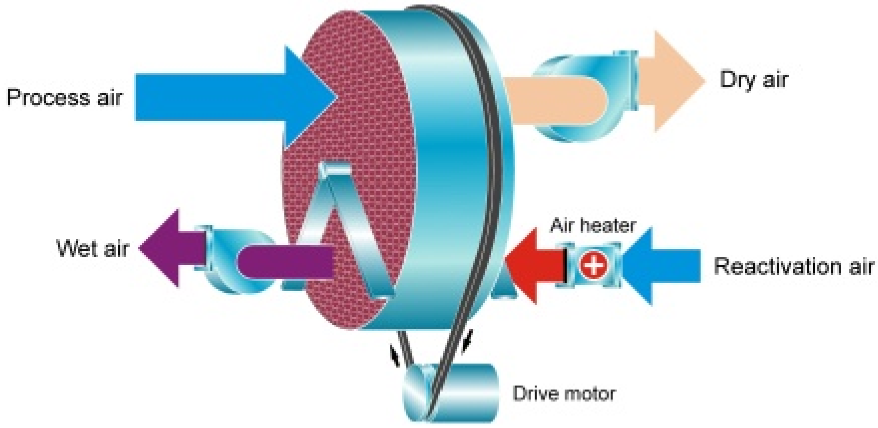

Figure 3.4. Rotating desiccant wheel

Some considerations for selection of desiccant wheels are:

- ✓ Appropriate desiccant materials
- ✓ Large desiccant content
- ✓ Wheel depth and flute size
- ✓ Size and cost

The actual performance depends on several additional factors that must be addressed. These include:

- ✓ Inlet process air temperature and humidity
- ✓ Desired exit process air humidity
- ✓ Inlet reactivating air temperature and humidity
- ✓ Face velocity of the two air streams
- ✓ Size of regeneration segment

3.2.2. Liquid Desiccant Systems

The unique characteristics of liquid desiccant systems are effective in commercial applications, especially in larger buildings, where the advantages of liquid desiccants provide cost-effective competition to both solid desiccants and to conventional cooling systems.

Liquid systems are very simple in concept, as described above. In hardware, they are somewhat more complex, because liquid desiccant solution can be corrosive, and because the components of the system can be located in different parts of a building with interconnecting piping. In the past, this flexibility of component arrangement has meant that in smaller sizes, liquid desiccant systems were more expensive to install than dry desiccant systems.

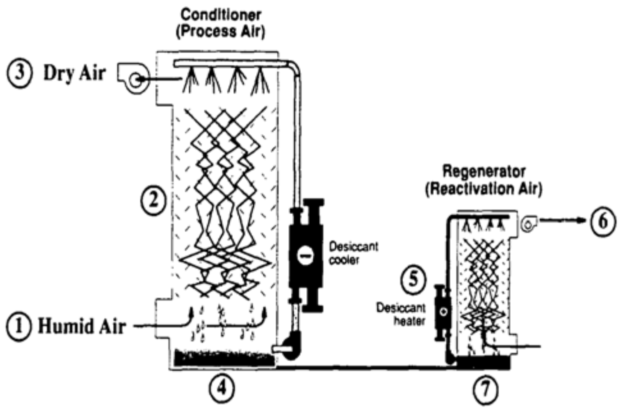

Figure 3.5. A liquid desiccant system

In liquid desiccant system is shown in Figure 3.5., humid air is passed through a cooling coil, or through a contact surface like cooling tower packing, which has been wetted with liquid desiccant. The desiccant absorbs moisture from the air, which makes the liquid solution more dilute. The dilute desiccant is sent through a heater and sprayed into a small regeneration air stream. The regeneration air carries away water vapor given off by the warm desiccant, so the re-concentrated solution can be used for drying air once again (Mei et al., 1992).

3.2.3. Hybrid Desiccant Systems

Combining components of a vapor-compression system with a desiccant system, results in a hybrid which can efficiently meet both the sensible and latent cooling loads. The vapor compression machine operates with higher evaporator temperatures (resulting in a higher thermal COP), and no reheat is required. The dehumidifier must only remove the moisture to meet the latent load; no excessive drying is required. This process reduces the amount of energy required to regenerate the desiccant. In addition, reducing the amount of moisture, which must be, removed results in a smaller dehumidification subsystem. Another advantage of the hybrid system is the reduction of required energy input (increased overall COP). Heat rejected by some components may be utilized to regenerate the desiccant, which eliminates or reduces the need for external regenerative heat from solar collectors or other heat sources. Hybrid systems have the option of utilizing the heat rejected by the vapor compression condenser for regeneration. In addition, depending on the type of driver, which provides the work for the vapor compressor, significant amounts of engine waste heat may be available.

3.3. Advantages of Desiccant Dehumidification Systems

Desiccant dehumidification systems are growing in popularity because of their ability to remove moisture from outdoor ventilation air while allowing conventional air-conditioning systems to deal primarily with control temperature (sensible cooling loads). Desiccant dehumidification is a new approach to space conditioning that offers solutions for many of the current economic, environmental, and regulatory issues being faced by facility managers. Indoor air quality is improved through higher ventilation rates, and achieving those fresh air make-up rates becomes more feasible with desiccant systems. At “low load conditions” outdoor air used for ventilation and re-circulated air from the building have to be dehumidified more than they have to be cooled. Properly integrated desiccant dehumidification systems have become cost-effective additions to many building HVAC systems because of:

• Their ability to recover energy from conditioned air that is normally exhausted from buildings.

• The lower cost of dehumidification when low-sensible load, high-latent load conditions are met.

• The greater comfort achieved with dehumidified air.

• The promotion of gas cooling for summer air-conditioning by utilities in the form of preferential gas cooling rates.

• High electric utility demand charges, which encourage a shift away from conventional, electrically driven air-conditioning (which requires a heavy daytime loading). Better control of humidity prevents moisture, mildew and rot damage to building materials. Desiccant dehumidification is particularly attractive in applications where building exhaust air is readily available for an energy-recovery ventilator (ERV, or “passive” desiccant system) or where a source of waste heat from other building operations is available to regenerate an “active” desiccant system.

• Desiccant systems offer significant potential for energy savings and reduced consumption of fossil fuels. The electrical energy consumption is small and the source of thermal energy can be diverse (i.e., solar, waste heat, natural gas).

• With desiccant systems the use of CFCs is eliminated (if used in conjunction with evaporative coolers) or reduced (if integrated with vapor compression units). CFCs contribute to depletion of the earth's ozone layer and some of them have already been banned completely and some will be banned in coming years.

• Indoor air quality is improved because of higher ventilation and fresh air rates associated with desiccant systems. Such systems also offer lower humidity levels and the capability to remove airborne pollutants.

• With desiccant systems, air humidity and temperature are controlled separately, enabling better control of humidity (Pesaran et al., 1992).

3.4. Desiccant Cooling Applications

Desiccants can dry either liquids or gases, including ambient air, and are used in many air-conditioning applications, particularly when the

- Latent load is large in comparison to the sensible load. For example; air conditioning systems for supermarkets have a very low heat load since the display cases also cool the store. The remaining load is mostly moisture.
- Energy cost to regenerate the desiccant is low compared to the cost of energy to dehumidify the air by chilling it below its dew point and reheating it.
- Moisture control level for the space would require chilling the air to subfreezing dew points if compression refrigeration alone were used to dehumidify the air.
- Temperature control level for the space or process requires continuous delivery of air at subfreezing temperatures.
- Thermal energy is available and inexpensive, or when electrical energy is limited or costly. For example, use desiccant systems where electrical demand is high and available capacity is low or where waste heat is available.
- Low humidity control levels are advantageous. For example, steel warehouses can be dehumidified rather than heated during the winter, saving energy and avoiding rust, but the dehumidification system must be operate at a low temperature and a low humidity control level.
- An air conditioning system must operate without high relative humidity in duct work and without condensed water in drain pans. For example, air distribution systems in buildings can harbor fungi, which create indoor air quality problems. Desiccant systems keep the air dry in the ductwork, preventing microbial growth.

- Desiccants can attract and hold more than simply water vapor; they can remove contaminants from airstreams to improve indoor air quality. Desiccants have been used to remove organic vapors and, in special circumstances, to control microbiological contaminants (Battelle, 1971). Hines et al. (1991) also confirmed their usefulness in removing vapors that can degrade indoor air quality. Desiccant materials can adsorb hydrocarbon vapors while they are collecting moisture from air. These sorption phenomena show promise of improving indoor air quality in typical building HVAC systems.
- Desiccants are also used in drying compressed air to low dew points. In this application, moisture can be removed from the desiccant without heat. De-sorption is accomplished using differences in vapor pressures compared to the total pressures of the compressed and ambient pressure airstreams.
- Finally, desiccants are used to dry the refrigerant circulating in air-conditioning and refrigeration systems. This reduces corrosion in refrigerant piping and prevents valves and capillaries from becoming clogged with ice crystals. In this application, the desiccant is not regenerated; it is discarded when it has adsorbed its limit of water vapor.

3.5. Types of Desiccant Materials

There are many ways to classify desiccant materials. One obvious way is liquid vs. solid (Collier et al., 1986). Another way that is not so obvious is by the sorption mechanism. Absorption refers to the process by which water is bonded within the molecular structure of the material. Adsorption refers to the process by which water is bonded to the surface of the material. Although it is mostly correct to assume that all liquid desiccant reactions are absorption and that all solid desiccant reactions are adsorption, there is one major exception. Hydrates of many metal salts are solid, yet they desiccate by absorption.

Another classification used by physical chemists is the concept of physisorption vs chemisorption. This is an arbitrary designation that reflects the strength of the bond between the adsorbed species (the adsorbate) and the surface of adsorption (the adsorbent). For all practical purposes, the class of adsorption reactions associated with moisture removal from air will always be considered physisorption, and these reactions will have low bond strength. The strength of the bond in moisture sorption must be low in order to make the energy efficiency of a cyclic operation high within the general class of solid desiccants there are several subclasses of materials: silica’s, aluminas, zeolites, hydratable salts, mixtures (Mei et al., 1992).

3.5.1. Solid Desiccants

Many solid materials such as silica, alumina, zeolite, hydratable salts and some mixtures are used as desiccant.

The silica materials are commonly referred to as "gels". They have been manufactured to obtain very high surface-to-volume ratios and have been surface-treated to produce an affinity for water. Such gels are formed by condensing soluble silicates from solutions of water or other solvents. They have the advantage of being relatively low in cost and easily customizable in terms of pore size and pore distribution.

Alumina is also referred to as "gels" for the same reasons as the silicates. They are chemically aluminum oxides and hydrides and are manufactured in much the same ways as silica gel. Generally, the alumina does not possess the ultimate sorption capacity of the silica, but they are refractory in nature and are therefore able to withstand higher-temperature environments without damage.

Zeolites fall into two categories, natural and synthetic. The natural zeolites are minerals that are mined in much the same manner as salt. The sediment beds of ancient bodies of water are the most common locations for these deposits. Synthetic zeolites are, as the name implies, manufactured materials. The characteristics that relate the natural and the synthetic materials are their chemical and structural similarities. Zeolites are alumino-silicate materials. Their crystalline structure is cage-like; the cage structure forms the sites for preferential water sorption. It is this cage-like structure that also forms the asis for the other designation of zeolites, which is the term "sieve." So-called molecular sieves are synthetic zeolites that have been engineered to possess a specific dimension of the cage. When this dimension is controlled, certain molecules will fit inside and others will be too large. This results in the effective separation of gaseous species.

Hydratable salts are a special class of solid desiccants. Salts that have soluble hydrates are called congruent salts. If the desired desiccant material is a solid, then an incongruent salt would be desired. Hydratable salts, existing as solids, are commonly used in applications where a water vapor pressure that is lower than is possible using liquid is desired. The salt will transit between the anhydrous state and the multiple hydrates state. Lithium chloride is the most common hydratable salt material used. Other hydratable salts are aluminum and copper sulfate, calcium chloride, and lithium bromide.

Using mixtures of desiccants is another common method of developing desiccant materials that have the desired sorption properties. For example, some desiccants have large total uptakes for moisture, but are not able to achieve very low humidity levels. Other materials may have marginal moisture uptake, but are capable of achieving extremely low humidity levels. For example, lithium chloride has an unparalleled capacity for moisture absorption at high relative humidities, but below 10% relative humidity, its moisture absorption is negligible. Combining that desiccant with silica gel, which has a larger capacity at low humidities, can provide adequate moisture capacity throughout a wide range of operating conditions. For best performance, desiccant mixtures are selected such that both desiccants can be reactivated at similar temperatures (ASHRAE, 1989).

3.5.2. Liquid Desiccants

Liquid desiccants used for many industrial and domestic applications have to be reconcentrated for reuse. Some liquid desiccants are lithium chloride, calcium chloride, glycols (diethylene, triethylene and tetraethylene), which are substances that can be regenerated (ASHRAE, 2004). Regeneration means that the water absorbed by these substances can be separated from them. Some liquid desiccants, such as methanol or ethylene, cannot be regenerated.

Liquid desiccants, such as lithium chloride, can absorb up to 1200 times their dry weight in water. The concentration of salt in the liquid solution determines the absorption characteristics of the liquid, which is sprayed into the process air. If the solution is concentrated, it can absorb moisture from drier air streams, and if the solution is dilute, it absorbs moisture from more humid air streams. So by controlling the concentration of the solution, one can control the humidity of the air that passes through the liquid spray. In order to control the temperature of the process air, one simply adjusts the temperature of the liquid desiccant being sprayed into that air.

Liquid desiccants have at least two major advantages over solid desiccants:

- Liquid desiccants have significantly smaller absorption-evaporation cycle hysteresis characteristics than the adsorption-desorption cycle of solid desiccants. This means that the energy losses per hydration-dehydration process cycle are smaller for liquid desiccants than they are for typical solid desiccants. A water-from-air system based on the application of liquid desiccants is inherently more energy efficient than a system based on solid desiccant surfaces.
- Liquid desiccants have a much higher relative water mass uptake capacity than solid desiccants. Upon dissolution, for example, a LiCl ion pair generates two hydration shells comprised of a total of 26 water molecules (e.g., 26 moles of water per molar equivalent of dissolved solute). Only a few water molecules may be condensed and held within a solid desiccant crystallite. As a result, liquid desiccants exhibit a 15-100 fold mass uptake advantage over solid desiccants as shown in Figure 3.6 (Sciperio, 2005).

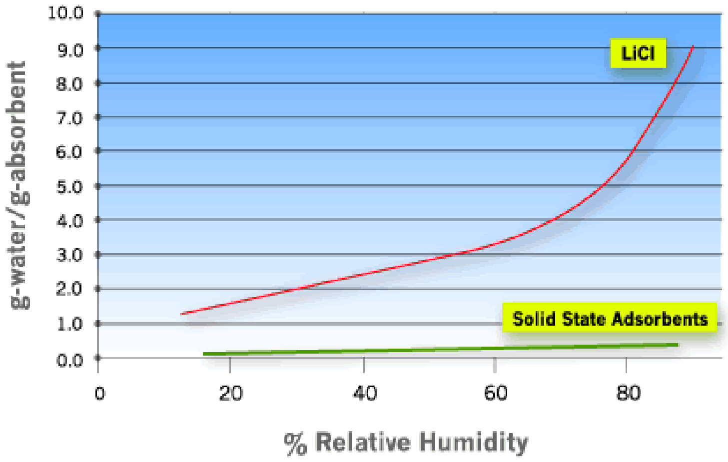

Figure 3.6. Liquid desiccants advantage over solid desiccant

3.6. Description of Desiccant Cooling System Studied

Desiccant cooling technology provides a tool for controlling humidity moisture levels for conditioned air spaces. Desiccant systems work in conjunction with conventional air conditioning systems to dehumidify the air. Desiccant materials are those that attract moisture due to differences in vapor pressure. These desiccants have been selected based on their ability to hold large quantities of water, their ability to be reactivated, and cost. The desiccant that is in the process air path has been prepared to have a lower vapor pressure than the air passing over it. Thus, the moisture in the air is transferred onto the desiccant material. As the desiccant vapor pressure increases due to the presence of the moisture that it has attracted, the desiccant material is transferred to a regeneration process. In the regeneration process, hot air is passed over the desiccant. The vapor pressure of the hot air is lower than the desiccant surface which forces the moisture to transfer from the desiccant surface into the hot air stream. The moist hot air is then exhausted from the system into the outdoor air.

Global environmental concerns, improving standards of ventilation and increasing concerns about indoor air quality have all contributed to a change in design thinking. Desiccant cooling systems are rapidly becoming established technology in most parts of the world. This growth has been brought about by the contribution of refrigerants used in conventional cooling systems to the depletion of the ozone layer. Also, the contribution towards global warming of refrigerants and fossil fuels used to generate electricity to power the refrigeration systems is very significant.

New approaches to space conditioning will be required to resolve these economic, environmental, and regulatory issues. Desiccant cooling and dehumidification, a technology known for some time, may provide important advantages in solving air conditioning problems.

One of the main characteristics of the desiccant air-conditioning systems is that they are very suitable for hygienic applications. These systems are viable alternatives for air-conditioning in health care facilities to reduce the airborne disease transmission. Nosocomial infections caused many fatal problems in some hospitals in Turkey in the past.

Since infections spread through air depend on humidity in some medical institutions like hospitals, humidity control is very important (Arundel et al., 1986). In conventional systems commonly used for air conditioning, the air sent to the place being conditioned comes through coils, cooled by a cooling unit – generally by a vapor compressing cycle. While coming through coils, both the heat (sensible heat) and the humidity (latent heat) of air are reduced. Removing humid from the air is realized by the condensation of the water steam in the air, which meets cold coil. This process creates a suitable atmosphere in the air sent to the place, for reproduction of bacteries that can cause infections. The substances used in these processes for removing humid, removes the bacteries (Escherichia coli, Enterococcus, Pseudomonas aeruginosa, Staphylococcus aureus, Coagulase negative Staphylococcus (epidermidis), Mycobacterium fortuitum, Candida albicans) that can cause illnesses and various infections in the air sent to the place being air conditioned about 35-70% (Kovak et al., 1997). However, the mechanism removing bacteries have not actually been made totally clear.

Most common illnesses spread through air are tuberculosis, splenic fever and legionnaires' disease (ORNL, 2000). At the present time, intensive studies are being made in order to find out the sources of infections seen in hospitals and take necessary precautions; the observation frequency of hospital infections are being declared as 8.4% and according to a research made in the United States of America, hospital infections comes fourth as a death cause after heart diseases, cancer and cerebral hemorrhage (Tr.Net Internet Sitesi, 2005). Hospital infections are also a serious problem in Turkey. In recent years, some baby deaths and fatal infections after operations have been reported in some health facilities. Another problem is the the amount of time patients stay at the hospitals. The stay time is much longer in Turkey than in developed countries.

Studies carried out in Turkey concluded that, hospital infections increase the amount of time patients stay at the hospitals about 2 weeks and cause approximately 1500 $ extra cost per patient (Türk İnfeksiyon İnternet Sitesi, 2005). The average amount of time a patient stays in hospital after operation is 2 days in the USA, 3 days in EU countries, and about 13-14 days in Turkey (MMO, 2003). That's why the time of using antibiotics in Turkey is 7 times more than EU countries, and 8 times more than the USA. One of the causes of these incidents are the bacteries spread through air. In order to make such infection problems spread through air minimum, desiccant systems are suggested to be used at hospital operations (ORNL, 2000).

The main reason for choosing desiccant air conditioning systems in hospital operations is not lowering the first investment cost of air conditioning system or saving up energy, but bringing the quality to hygiene and inside air, which is very important for health and economy. Besides this, in an occasion of using a cheap energy source (solar energy, waste heat, etc.), energy consumption of desiccant systems may be less than conventional systems.

Although desiccant air conditioning systems have started to become widespread in Europe and the USA in recent years, they are not known exactly in Turkey and there is not much application of it.

In this thesis, a desiccant based air-conditioning system suitable for hygienic applications is considered (Figure 3.7). The moisture of the fresh air is reduced using a solid desiccant wheel and then its temperature is decreased by the "dry coil" of a vapor-compression cycle. To enhance the performance of the system, some technologies such as “pre-cooling with outdoor air”, “waste cool recovery” and “pre-cooling of waste air with evaporative cooling” is utilized.

Energy analysis of the system considered will be carried out and suitability of the system will be investigated for the health care facilities in which hygiene is crucially important.

Figure 3.7 shows the desiccant cooling system, which is considered in this study. Since the system is considered for the health care facilities, 100 % fresh air will be used. Fresh air duct is used to supply fresh air for the air-conditioned room. The waste air sucked from the air-conditioned room is sent to outside via waste air duct. Regeneration air duct is used to remove moisture of desiccant unit. Various components (dehumidifier, heat exchangers, fans, heaters, temperature and relative humidity sensors) were located into these channels to control and adjust the conditions of the air streams. Psychrometric diagram of the system considered for the design day of Adana for summer air conditioning is shown in Figure 3.8.

Fresh outside air (state 1) is brought to the desiccant wheel and its humidity is absorbed by the desiccant material of the wheel. During dehumidification process, dry-bulb temperature of the air increases (state 2). The fresh air leaves the process side of the desiccant wheel hotter, and much drier than when it entered the system. This air is too hot to send directly to the air-conditioned space. It must be cooled down to a temperature that is suitable for the sensible cooling of the air-conditioned space. Sensible cooling of the fresh air is carried out (process 4-5) in a cooling coil (heat exchanger 3), which is fed by chilled water from a chiller unit. However, the fresh air is passed through two heat exchangers (heat exchanger 1 and 2) before coming to the cooling coil for the cool recovery. After the desiccant unit, the fresh air is passed through the first and the second recuperative type heat exchangers to decrease its temperature. Heat is removed from the fresh air to the regeneration air by the first recuperative heat exchanger (2→3). This, reduces both the amount of sensible cooling requirement of the fresh air in the cooling coil and heating energy requirement of the regeneration air. In the second heat exchanger (3→4), temperature of the fresh air is further reduced by heat transfer to the waste air that is sucked from the air-conditioned room and subsequently evaporatively cooled. In recuperators, only sensible heat is removed from the fresh air, and therefore, humidity ratio of the air is constant.

The desired blowing temperature of the fresh air (state 6) is obtained in the cooling coil (heat exchanger 3). The surface temperature of the cooling coil is always kept higher than the dew point temperature of the fresh air entering into the cooling coil (no condensation). Therefore, only sensible cooling is performed in the coil (dry-coil application). Removing moisture from the fresh air (latent cooling) is occurred only in the desiccant wheel and there is no change in the humidity ratio of the fresh air after the desiccant wheel.

Air which is sucked from the indoor (7) into the waste air duct is evaporatively cooled in an evaporative cooler (8→9) before entering into the second heat exchanger in order to increase the saving. In the second heat exchanger, due to heat transfer from the fresh air to the waste air, temperature of the fresh air decreases (3-4) and that of the waste air increases (9-10). The waste air leaving the heat exchanger is rejected to the outdoors.

At the desiccant wheel, hot regeneration air which is outdoor air is used for removing moisture. The regeneration air leaving the desiccant wheel first comes to the heat exchanger (11-12) in which heat transfer from the fresh air to the regeneration air takes place. Temperature of the regeneration air at the exit of the desiccant wheel (15) is generally higher than that of the regeneration air leaving heat exchanger 1 (12). Therefore, a rotary regenerator is used for weak heat recovery. The regeneration air leaving the desiccant wheel passes through the rotary regenerator (12-13), In which heat is transferred from the regeneration air that left the desiccant wheel (state 15) to regeneration air left the heat exchanger 1 (12). If the temperature of the regeneration air at state 16 is not higher than that of state 12, rotary regenerator is stopped.

Temperature of the regeneration air should be a critical value (usually 100 ^{o}C) to be able to remove moisture in the desiccant wheel. Although the temperature of the regeneration air is increased in heat exchanger 1 (11-12) and rotary regenerator (12-13), it is not high enough for dehumidification of the desiccant wheel. Temperature can be further increased, if there is a cheap heat source available such as solar energy, waste heat or geothermal energy. Use of such cheap heating energy is very important for the economic operation of the desiccant systems. If the system can be backed up by a cheap thermal energy source, operating cost can be decreased dramatically. The final temperature of the regeneration air (regeneration temperature) is achieved with the help of electric heaters (13-14). The air removes the humidity of the desiccant wheel (14-15) and flows to the rotary regenerator (15-16) before discharged to the outdoors.

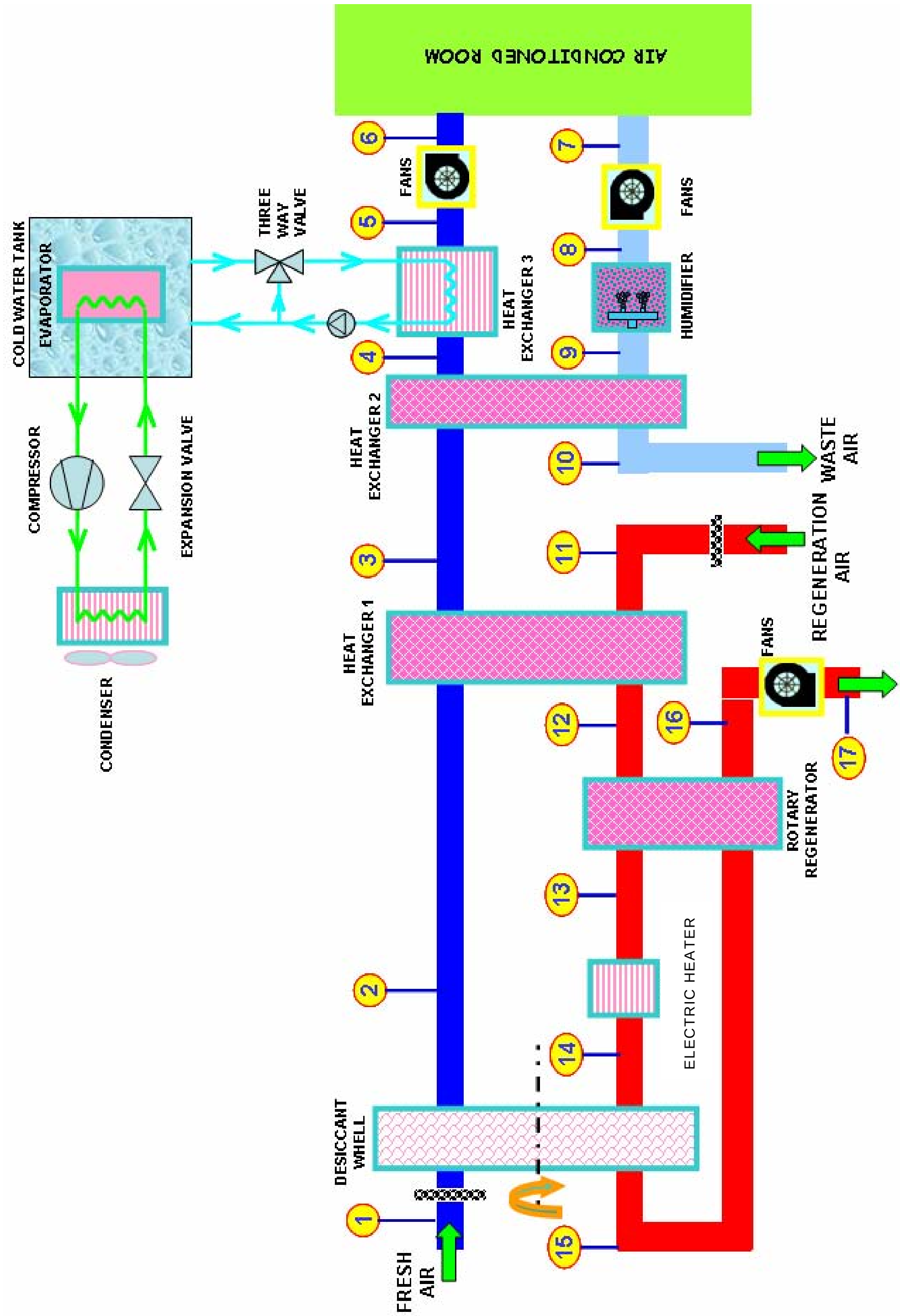

Figure 3.7. Schematic view of desiccant cooling system

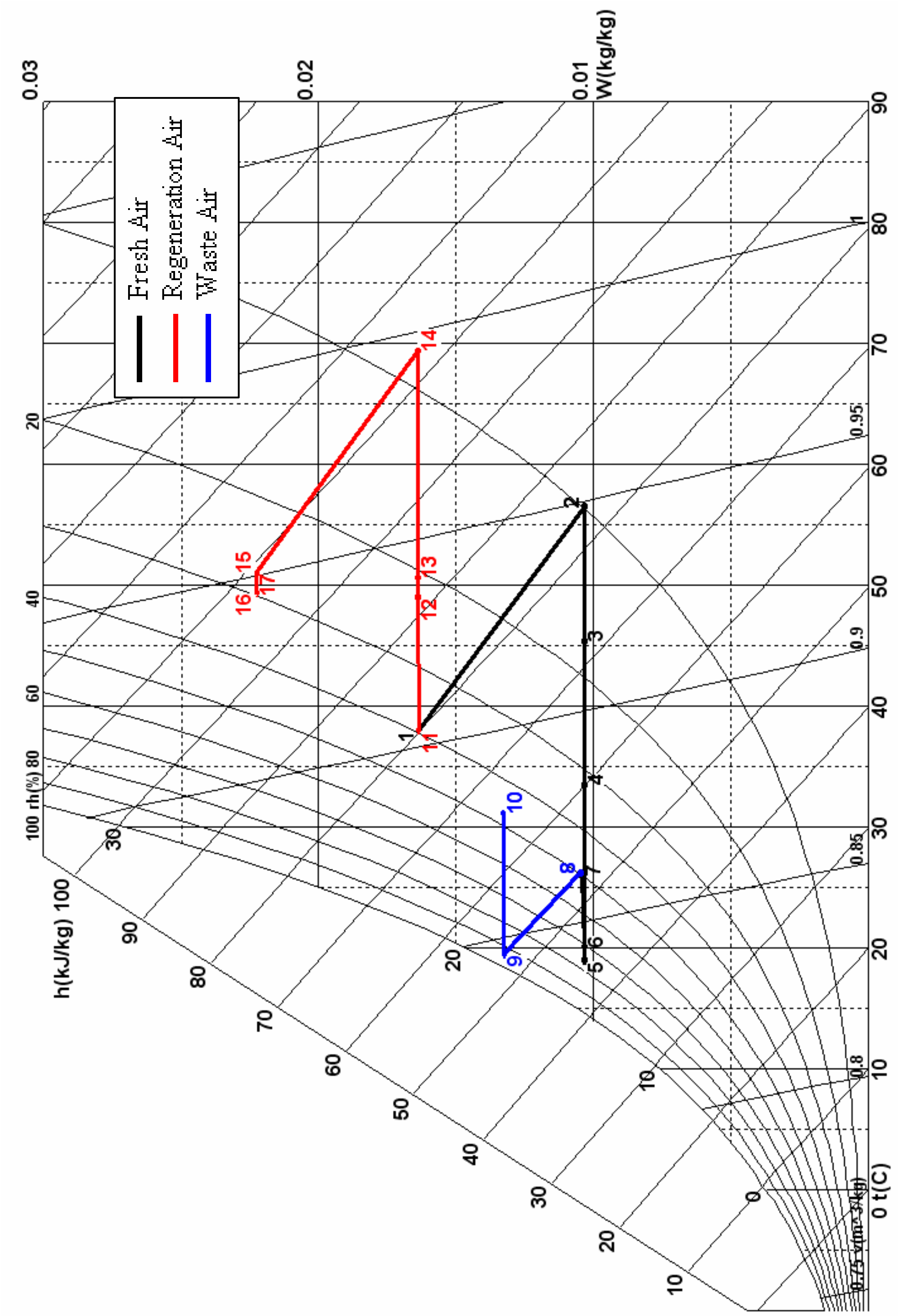

Figure 3.8. Psychrometric diagram of the system considered for the design day of Adana

3.7. Cooling Load Profile of Air Conditioned Room

In this study, a room that is located inside Mechanical Engineering Laboratories of Mechanical Engineering Department of Çukurova University was selected as the air-conditioned room. Dimensions of the room are 11.5 x 5.1 x 3.8 m. The walls of the room are not insulated. All the components such as the walls, windows, doors and roof are made of typical composite construction. Long sides of the room face to the West and the East. Only West side of the room faces to outside and other sides of the room face to the laboratory. The air conditioned room has 16.74 m^2 windows on the side.

In this section, cooling load of the room was calculated using total equivalent temperature difference / time averaging method (TETD/TA). A MS-Excel worksheet was prepared for the calculations. Calculations were performed for 23rd July, which is considered as the design day for Adana for summer air-conditioning. The outdoor design parameters for Adana are 38 ºC dry and 26 ºC wet-bulb temperature. The indoor conditions were selected as 26 ºC dry-bulb temperature and 50 % relative humidity. The view of MS-Excel worksheet used for calculating the cooling load is shown in Appendix A1.

The results showed that the maximum cooling load of the room is 10.634 kW at 04.00 p.m. As a result of the psychrometric analysis, it was determined that the amount of fresh air which must be supplied to the room is approximately 4000 m^3/h (by considering the air requirement for hygienic applications).

3.8. Description of Computer Program Written

Psychrometric analyses were carried out using the design day parameters of Adana (38 ºC dry and 26 ºC wet bulb temperature). The preliminary design of the system was based on the results of these parameters.

However, as known, an air-conditioning system operates in design values of outside environment only a few days and/or even a few hours in a season. Therefore, it is also very important to know the behavior of the system on the other days of the season and different hours in a day. In Adana, it is assumed that cooling season starts in the middle of May and ends in the middle of October. This corresponds nearly 5000 hours. Considering the variable outdoor conditions in a cooling season, analyses of the system should be made on an hourly basis.

When the number of the states (17 states) in the system considered, and number of the hours in a cooling season (5000 hours) are thought together, it is clear that analysis can not be carried out manually. Therefore, it was decided to develop a computer program for the hourly analysis of the system. FORTRAN programming language was selected for the programming. The parameters below are used as input for the program:

- Flow rate of the fresh air supplied to the air-conditioned room (V_{fresh})
- Flow rate of the exhaust air (V_{waste})
- Flow rate of the regeneration air (V_{regen})
- Dry bulb temperature of the outside air (T_{dbo})
- Humidity ratio of the outside air (W_o)
- Design dry bulb temperature of the air-conditioned room (T_{dbi})
- Design relative humidity of the air-conditioned room (φ_i)
- Sensible heat ratio of the air-conditioned room (SHR)
- Total cooling load of the air-conditioned room (Q_{total})
- Effectiveness of the heat exchangers (η_{heatex})
- Effectiveness of evaporative cooler (η_{hum})
- Efficiency of the fans (η_{fan})
- Powers of the fans (W_{fan})

Using these inputs, the psychrometric features given bellow are calculated separately for every state on the system:

- Dry bulb temperature (^oC)
- Wet bulb temperature (^oC)
- Enthalpy (kJ/kg)
- Humidity ratio (kg/kg dry air)
- Relative humidity (%)

- Dew point temperature (ºC)
- Density (kg/m^3)
- Specific heat (kJ/kg K)

These psychrometric properties can be calculated using psychrometric equations given by ASHRAE 2004.

3.9. Psychrometric Equations

Knowledge of psychrometric properties is the basic requirement for environmental measurements. An understanding of physical and thermodynamic properties of an air-water vapor mixture is called psychrometric. It is fundamental to the design of environmental control systems for plants, crops, animals and human beings. The equations presented in this study enable someone to calculate all or some of the psychrometric properties, if any two independent psychrometric properties of an air-water vapor mixture are known in addition to the atmospheric pressure. Generally, one may have the values of dry-bulb temperature and another psychrometric property such as, wet-bulb temperature, relative humidity or dew point temperature. From these parameters, one can easily determine other psychrometric properties such as relative humidity, enthalpy, humidity ratio, specific volume or vapor pressure (Singh et al., 2002).

3.9.1. Saturation Vapor Pressure

The water vapor saturation pressure (P_{ws}) is required to determine a number of moist air properties, principally the saturation humidity ratio. The saturation pressure over liquid water for the temperature range of 0 to 200 ^{o}C is given by (Wexler et al., 1980):

$$\ln(P_{ws}) = C_8/T + C_9 + C_{10} \times T + C_{11} \times T^2 + C_{12} \times T^3 + C_{13} \times \ln(T) \qquad (3.1)$$

where T is air temperature (K) and C_8, C_9, C_{10}, C_{11}, C_{12} and C_{13} are coefficients:

C_8 : -5.8002206×10^{3}, C_9 : -5.5162560, C_{10} : -4.8640239×10^{-2},
C_{11} : 4.1764768×10^{-5}, C_{12} : -1.4452093×10^{-8}, C_{13} : 6.5459673.

3.9.2. Humidity Ratio

Humidity ratio (W) of a given moist air sample is defined as the ratio of the mass of water vapor (m_w) to mass of dry air (m_a) contained in the sample:

$$W = \frac{m_w}{m_a} = \frac{(P\times V/R\times T)_w}{(P\times V/R\times T)_a} = \frac{P_w\times R_a}{P_a\times R_w} = \frac{(R_a/R_w)\times P_w}{P_{atm}-P_w} = \frac{0.62198\times P_w}{P_{atm}-P_w} \tag{3.2}$$

where;

P_a : Partial pressure of dry air, [kPa]

P_w : Partial pressure of water vapor, [kPa]

P_{atm} : Atmospheric pressure (P_a+P_w), [kPa]

R_a/R_w : Ratio of mole masses of air and water (0.62198)

V : Volume, [m^3]

The values of V and T in Equation (3.2) are same for water vapor and air. Partial pressure of water vapor (P_w) can be written as:

$$P_w = \frac{P_{atm}\times W}{0.62198+W} \tag{3.3}$$

The humidity ratio (W) can be determined using Equation (3.4) if dry bulb (t_{db}) and wet bulb (t_{wb}) temperatures are known.

$$W = \frac{\left[(2501-2.381\times t_{wb})\times W_s\right]-1.006\times(t_{db}-t_{wb})}{2501+1.805\times t_{db}-4.186\times t_{wb}} \tag{3.4}$$

Saturation humidity ratio (W_s) is the humidity ratio of moist air saturated with respect to water at the same temperature T and pressure P:

$$W_s = \frac{0.62198 \times P_{ws}}{P_{atm} - P_{ws}} \quad (3.5)$$

3.9.3. Relative Humidity

Relative humidity (φ) is the ratio of the actual water vapor pressure to the saturation water vapor pressure at the same temperature:

$$\varphi = \frac{P_w}{P_{ws}} \quad (3.6)$$

φ is usually expressed as a percentage rather than as a fraction. It does not define the water content of the air unless the temperature is given. The reason φ is so much used in conversation is that most organic materials have equilibrium water content that is mainly determined by the φ and is only slightly influenced by temperature.

3.9.4. Wet Bulb and Dry Bulb Temperature

Dry air exists when all water vapor and contaminates have been removed from atmospheric air. The composition of dry air is relatively constant, but small variations in the amounts of individual components occur with time, geographic location, and altitude. Moist air is a binary mixture of dry air and water vapor. The amount of water vapor in moist air varies from zero to a maximum that depends on temperature and pressure. The wet-bulb temperature (t_{wb}) can be determined using Equation (3.7) (ASHRAE, 2001):

$$t_{wb} = \frac{1.006 \times t_{db} + W \times \left(2501 + 1.805 \times t_{db}\right) - 2501 \times W_s}{4.186 \times W - 2.381 \times W_s + 1.006} \quad (3.7)$$

Dry-bulb temperature (t_{db}) can be calculated using the Equation (3.8) if wet-bulb temperature and humidity ratio are known (ASHRAE, 2001):

$$t_{db} = \frac{W_s \times [2501 - (2.381 \times t_{wb})] + t_{wb} \times [1.006 + (4.186 \times W)] - (2501 \times W)}{[1.006 + (1.805 \times W)]} \quad (3.8)$$

3.9.5. Dew Point Temperature

Dew point temperature is defined as the temperature to which the air would have to cool (at constant pressure and constant water vapor content) in order to reach saturation. A state of saturation exists when the air is holding the maximum amount of water vapor possible at the existing temperature and pressure.

The dew-point temperature range of 0 to 93°C is given by (ASHRAE, 2001):

$$t_{dp} = C_{14} + C_{15} \times \ln(P_w) + C_{16} \times (\ln(P_w))^2 + C_{17} \times (\ln(P_w))^3 + C_{18} \times (P_w)^{0.1984} \quad (3.9)$$

where C_{14}, C_{15}, C_{16}, C_{17} and C_{18} are coefficients :

C_{14} : 6.54, C_{15} : 14.526, C_{16} : 0.7389, C_{17} : 0.09486, C_{18} : 0.4569.

3.9.6. Enthalpy

Entalpy is the heat energy content of an air-water vapor mixture. The enthalpy of moist air can be given as for (-50 °C $\leq t_{db} \leq$ 110 °C) :

$$h = 1.006 \times t_{db} + W \times (2501 + 1.805 \times t_{db}) \quad (3.10)$$

3.9.7. Specific Heat

The specific heat (c_p) is the amount of heat per unit mass required to raise the temperature by one degree celsius. c_p can be given as :

$$c_P = 1.005 + 0.006 \times (T/100)^{1.73} \tag{3.11}$$

3.9.8. Density

The density (d) of a moist air mixture is the ratio of the total mass to the total volume :

$$d = \frac{m_a + m_w}{V} \tag{3.12}$$

Inserting Equation (3.2) into Equation (3.12) the following equation is obtained:

$$d = \frac{1 + W}{\nu} \tag{3.13}$$

where ν is the specific volume of the moist air (m^3/kg, dry air). ν can be given as:

$$\nu = \frac{0.2871 \times (t_{db} + 273.15) \times (1 + 1.6078 \times W)}{P_{atm}} \tag{3.14}$$

Inserting Equation (3.14) into Equation (3.13) the following equation for density is obtained :

$$d = \frac{(1 + W) \times P_{atm}}{0.2871 \times (t + 273.15) \times (1 + 1.6078 \times W)} \tag{3.15}$$

3.10. Application of Equations

The equations from (3.1) to (3.15) are used for the determination of psychrometric properties, if any two psychrometric properties of an air-water mixture are known. Four subroutines were written to find the psychrometric properties of moist air at different states of the system considered. The explanations of these sub-programs (subroutines) are given below:

3.10.1. Subroutine DW

Subroutine DW is used when the dry bulb (t_{db}) and wet bulb temperature (t_{wb}) of the air are known. The parameters calculated and the equations used for the calculations are shown in Table 3.1.

Table 3.1. Calculated parameter and equations used for the calculations in Subroutine DW

No	***Calculated Parameter***	***Equations Used***
1	W	Equation (3.1), (3.4) and (3.5)
2	φ	Equation (3.1), (3.3) and (3.6)
3	h	Equation (3.10)
4	t_{dp}	Equation (3.3) and (3.9)
5	c_p	Equation (3.11)
6	d	Equation (3.15)

3.10.2. Subroutine DA

Subroutine DA is used when the dry bulb (t_{db}) and humidity ratio (W) of the air are known. The parameters calculated and the equations used for the calculations are shown in Table 3.2.

Table 3.2. Calculated parameter and equations used for the calculations in Subroutine DA

No	***Calculated Parameter***	***Equations Used***
1	t_{wb}	Equation (3.1), (3.5) and (3.7)
2	φ	Equation (3.1), (3.3) and (3.6)
3	h	Equation (3.10)
4	t_{dp}	Equation (3.3) and (3.9)
5	c_p	Equation (3.11)
6	d	Equation (3.15)

3.10.3. Subroutine AW

Subroutine AW is used when the humidity ratio (W) and wet bulb temperature (t_{wb}) of the air are known. The parameters calculated and the equations used for the calculations are shown in Table 3.3.

Table 3.3. Calculated parameter and equations used for the calculations in Subroutine AW

No	***Calculated Parameter***	***Equations Used***
1	T_{db}	Equation (3.3), (3.5) and (3.8)
2	φ	Equation (3.1), (3.3) and (3.6)
3	h	Equation (3.10)
4	t_{dp}	Equation (3.3) and (3.9)
5	c_p	Equation (3.11)
6	d	Equation (3.15)

3.10.4. Subroutine QD

Subroutine QD is used when the relative humidity (φ) and dry bulb temperature (t_{db}) of the air are known. The parameters calculated and the equations used for the calculations are shown in Table 3.4.

Table 3.4. Calculated parameter and equations used for the calculations in Subroutine QD

No	*Calculated Parameter*	*Equations Used*
1	t_{wb}	Equation (3.1), (3.5) and (3.7)
2	W	Equation (3.1), (3.4) and (3.5)
3	h	Equation (3.10)
4	t_{dp}	Equation (3.3) and (3.9)
5	c_p	Equation (3.11)
6	d	Equation (3.15)

3.11. Determination of the Psychrometric Properties at the States of the System (State 1 to State 17)

3.11.1. State 1 and State 11

These states represent outdoor air. At these states (1, 11), dry bulb temperature (t_{db1}, t_{db11}) and humidity ratio (W_1, W_{11}), which are inputs in the program, are known. Other psychrometric properties (enthalpy (h_1, h_{11}), wet bulb temperature (t_{wb1}, t_{wb11}), relative humidity (φ_1, φ_{11}), dew point temperature (t_{dp1}, t_{dp11}), density (d_1, d_{11})) are calculated using Subroutine DA.

3.11.2. State 7

State 7 represents indoor air. At state 7, dry bulb temperature (t_{db7}) and relative humidity (φ_7) (design conditions), which are inputs in the program, are known. Other psychrometric properties (enthalpy (h_7), wet bulb temperature (t_{wb7}), humidity ratio (W_7), dew point temperature (t_{dp7}), density (d_7)) are calculated using Subroutine QD.

3.11.3. State 6

The sensible heat ratio (SHR), which is input in the program, is used to analyze air conditioning processes (Aktacir, 2005). It is defined as the ratio of the sensible heat removed from an air stream to the total heat removed :

$$SHR = \frac{Q_{sensible}}{Q_{total}} \tag{3.16}$$

Total cooling load (Q_{total}), which is input in the program, is the sum of latent cooling load (Q_{latent}) and sensible cooling ($Q_{sensible}$). They are defined with the following equations:

$$\dot{Q}_{total} = \dot{Q}_{latent} + \dot{Q}_{sensible} \tag{3.17}$$

$$\dot{Q}_{latent} = \dot{m}_{fresh} \times h_g \times (W_7 - W_6) \tag{3.18}$$

$$\dot{Q}_{sensible} = \dot{m}_{fresh} \times c_p \times (t_7 - t_6) \tag{3.19}$$

where,

h_g : Enthalpy of the saturated water vapor, [J/kg]

t_{db7} : Indoor air dry bulb temperature, [^oC]

t_{db6} : Supply air dry bulb temperature, [^oC]

W_7 : Indoor air humidity ratio, [kg/kg dry air]

W_6 : Supply air humidity ratio, [kg/kg dry air]

$\dot{m}_{fresh}$: Mass flow rate of fresh air, [kg/s]

Using Equations (3.17) to (3.19), Equation (3.16) can be rewritten as:

$$SHR = \frac{c_p \times \Delta t}{c_p \times (t_7 - t_6) + h_g \times (W_7 - W_6)} = \frac{1}{1 + \frac{h_g}{c_p} \times \frac{W_7 - W_6}{t_7 - t_6}} \tag{3.20}$$

where $\frac{\Delta W}{\Delta t}=\left(\frac{W_7 - W_6}{t_7 - t_6}\right)$ is slope of the cooling process on psychrometric chart and it can be calculated as:

$$\frac{\Delta W}{\Delta t}=\frac{c_p}{h_g}\times(\frac{1}{SHR}-1) \tag{3.21}$$

Total cooling load (Q_{total}) is also defined as:

$$\dot{Q}_{total}=\dot{m}_{fresh}\times(h_7-h_6) \tag{3.22}$$

Enthalpy of the supply air (h_6)is calculated using Equation (3.23):

$$h_6=h_7-\frac{\dot{Q}_{total}}{\dot{m}_{fresh}} \tag{3.23}$$

Using Equations (3.10) and (3.21), supply air temperature (t_{db6}) can be calculated by solving second degree equation.

At state 6, other psychrometric properties (wet bulb temperature (t_{wb6}), relative humidity (φ_6), dew point temperature (t_{dp6}), density (d_6)) are calculated using Subroutine DA.

3.11.4. State 2

The most important component of dehumidification air conditioning system is desiccant wheel. Some of the publications and introduction brochures related with the dehumidification regenerators point out that dehumidification and removal of humidity operations (regeneration, reactivation) realize approximately at constant enthalpy (approximately at constant wet bulb temperature) (Jıa et al., 2006).

These processes are represented with the curves (A→B) and (C→D) in

Figure 3.9. However, dehumidification and regeneration processes do not occur at constant wet bulb temperature according to the data given by the rotary desiccant wheel manufactures. Actual increase in dry bulb temperature during dehumidification (A→B’) is higher than that of the constant wet bulb case (A→B). This is due to fact that a chemical thermal energy arises and the energy carried by the matrix of the wheel dehumidification regenerator from the regeneration air, which is hotter from the process air (ASHRAE Systems and Equipment Handbook, 2000). Because of this additional energy, dry bulb temperature further increases. The ratio of additional dry bulb temperature increase ($t_{B'}$-t_B) to total dry bulb temperature increase ($t_{B'}$-t_A) was defined as:

$$F_d = \frac{t_{B'} - t_B}{t_{B'} - t_A} \tag{3.24}$$

Similarly, dry bulb temperature decrease higher than constant wet bulb temperature case during regeneration (C→D’), due to sensible heat transfer to the process air. The ratio of additional dry bulb temperature decrease ($t_{D'}$-t_D) to total dry bulb temperature decrease ($t_{D'}$-t_C) was also defined as:

$$F_r = \frac{t_{D'} - t_D}{t_{D'} - t_C} \tag{3.25}$$

It is important to determine the values of F_d and F_r accurately to able to model the system. In this study the performance data given by manufactures of the desiccant rotary wheel was used to calculate F_d and F_r . Analysis of the data released that F_d and F_r are functions of dry bulb temperature and humidity ratio of the dehumidified (process) air and the regeneration air.

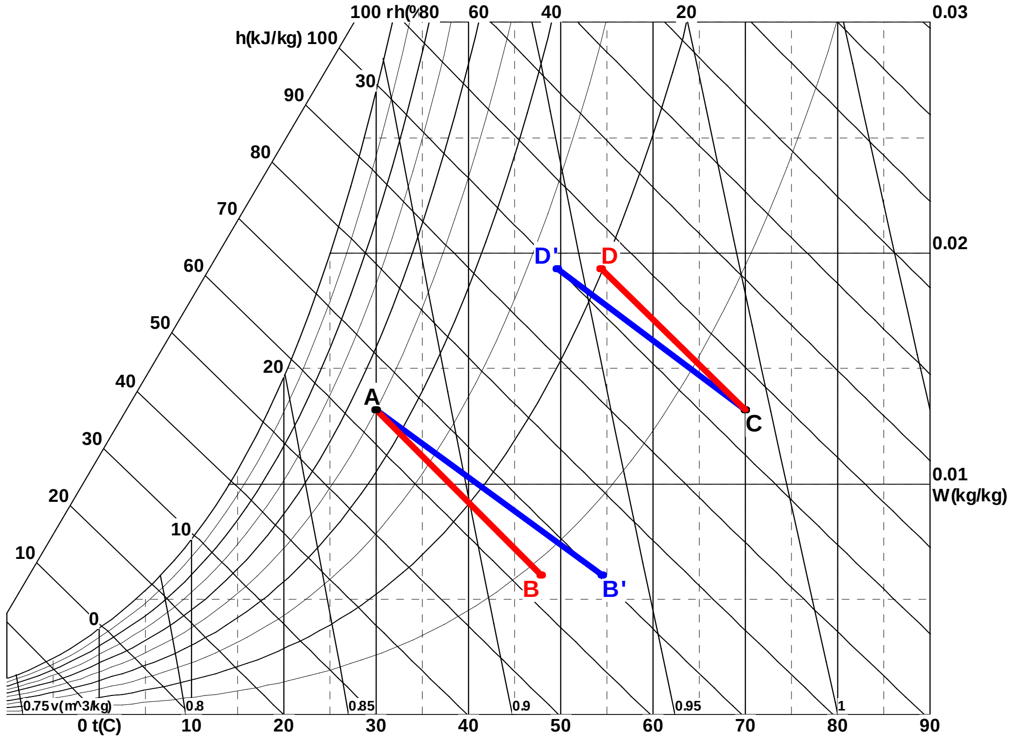

Figure 3.9. The process of dehumidification (A-B) and removal of humidity (C-D)

For the system considered in this thesis; the humidity ratio of the process air and the regeneration air are equal, since the same outdoor air was used as the process air and the regeneration air. Figure 3.10 shows the variation of F_d values with the humidity ratio of the outdoor air for a 70 ^oC regeneration air temperature. In the figure, F_d values are plotted for different outdoor air dry bulb temperatures varying between 25 ^oC and 40 ^oC. As can be seen from the figure, F_d decreases smoothly with increasing humidity ratio of outdoor air and dry bulb temperature of the outdoor air has no significant influence on F_d. Similar results were obtained for different regeneration air temperatures.

Figure 3.11 shows the variation of F_r values with the humidity ratio of the outdoor air for a 70 ^oC regeneration air temperature at different outdoor air dry bulb temperatures. As can be seen from the figure, results similar to with F_d were obtained for F_r.

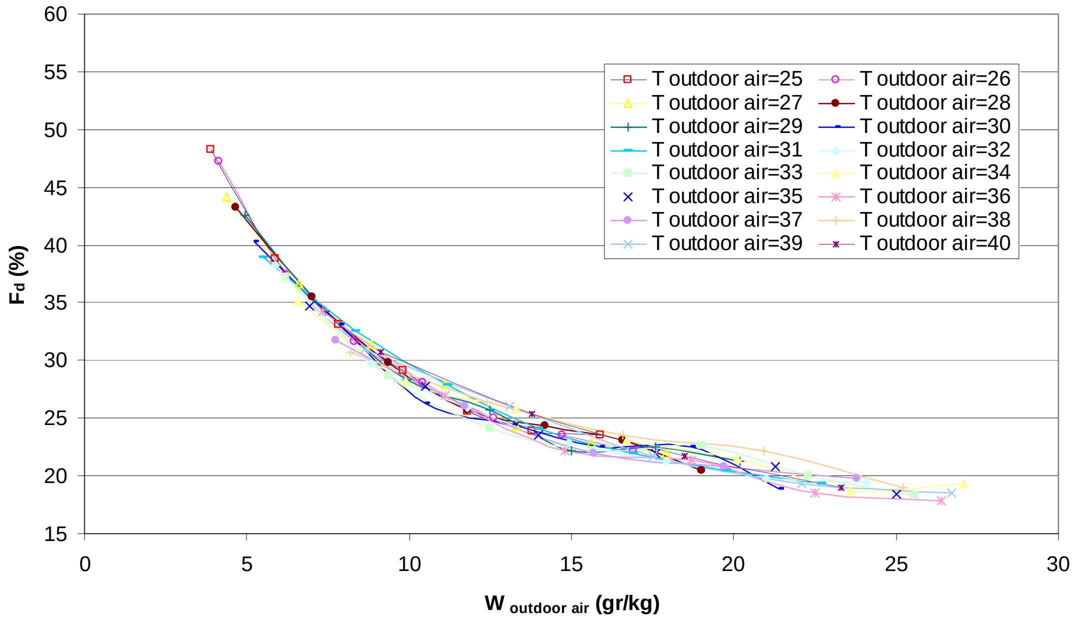

Figure 3.10. Variation of F_d at 70 ^oC regeneration air temperature with humidity ratio of the outdoor air for different outdoor air temperatures

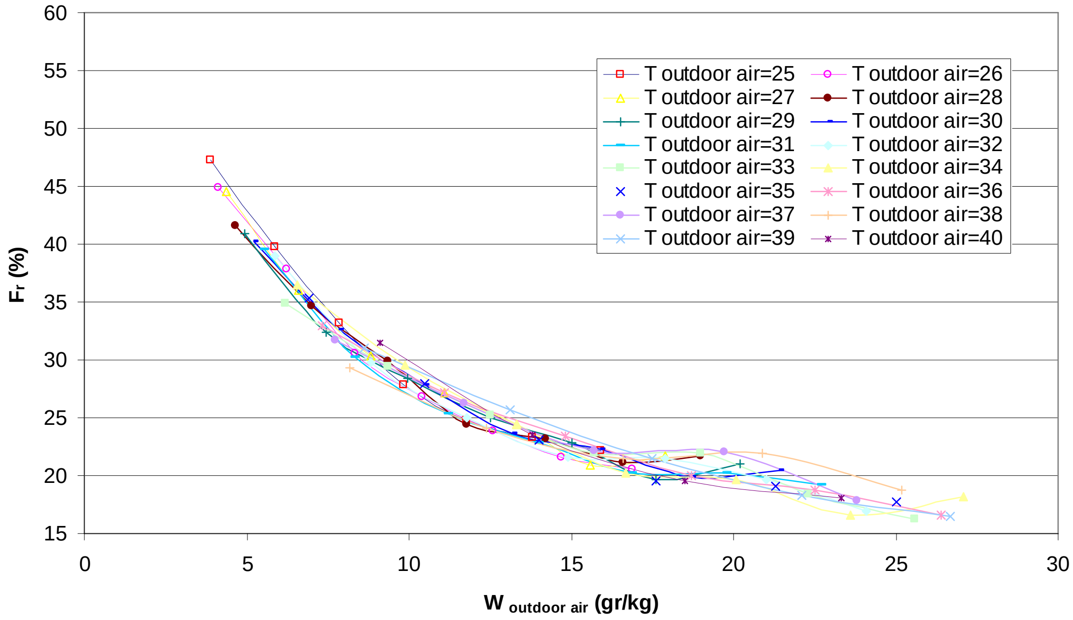

Figure 3.11. Variation of F_r at 70 ^oC regeneration air temperature with humidity ratio of the outdoor air for different outdoor air temperatures

Figure 3.12 shows values of F_d and F_r, which were calculated for different regeneration air temperatures for dehumidification and humidity removal processes. As it can be seen from the Figure, F_d and F_r are almost the same for a given regeneration temperature. Therefore, the dehumidification and humidity removal data were handled together in order to obtain equations required for the computer modeling.

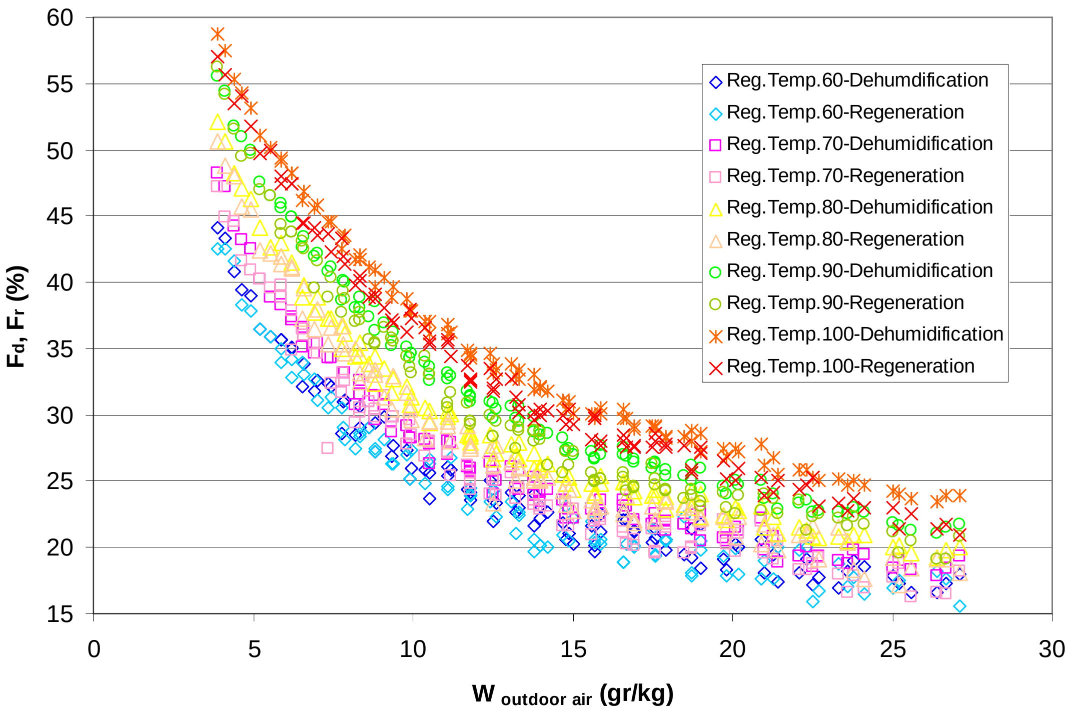

Figure 3.12. The values of F_d and F_r which were calculated at different regeneration air temperature for dehumidification and humidity removal processes

Figure 3.13 shows the values of F_d and F_r, which were obtained at different regeneration air temperatures. The curves which were fitted (by using least square method) to the calculated data for each regeneration temperature were also shown in the figure. The equations of the curves which were fitted to the calculated data for each regeneration temperature were listed below:

$$T_{reg.\ temp.} = 60\ ^{o}C : \qquad F_d = F_r = 82.136 \times W^{-0.4901} \qquad (R^2{=}0.957) \tag{3.26}$$

$$T_{reg.\ temp.} = 70\ ^{o}C : \qquad F_d = F_r = 92.289 \times W^{-0.509} \qquad (R^2{=}0.970) \tag{3.27}$$

$$T_{reg.\ temp.} = 80\ ^{o}C: \qquad F_d = F_r = 102.45 \times W^{-0.518} \qquad (R^2=0.977) \tag{3.28}$$

$$T_{reg.\ temp.} = 90\ ^{o}C: \qquad F_d = F_r = 112.41 \times W^{-0.5188} \qquad (R^2=0.983) \tag{3.29}$$

$$T_{reg.\ temp.} = 100\ ^{o}C: \qquad F_d = F_r = 116.80 \times W^{-0.4979} \qquad (R^2=0.983) \tag{3.30}$$

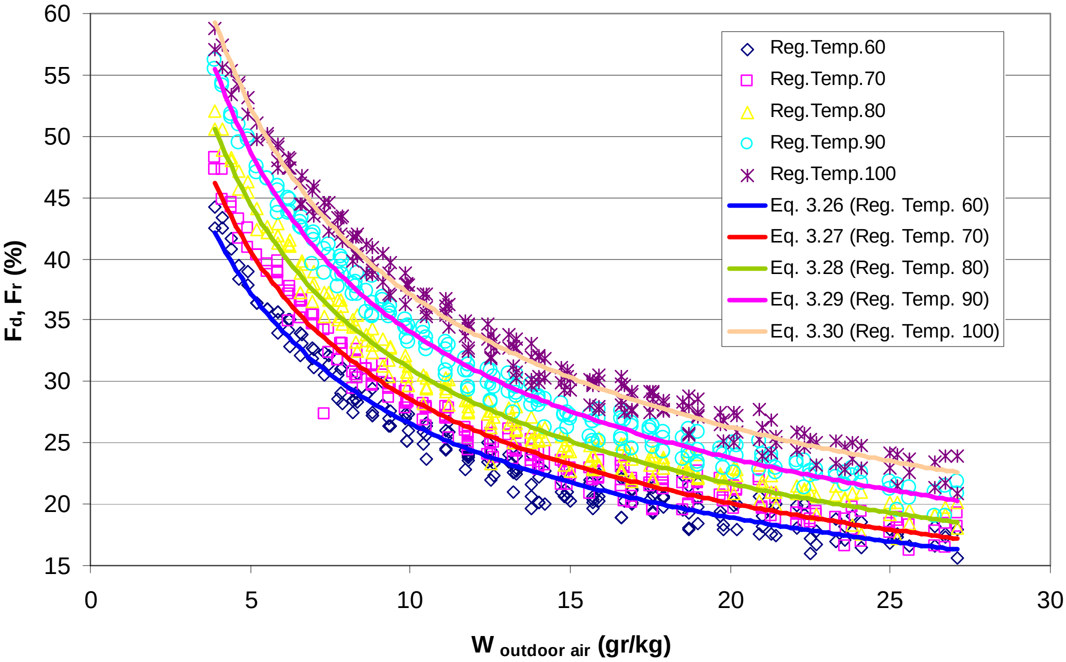

Figure 3.13. The values of F_d and F_r, which were obtained at different regeneration air temperature, related with the humidity ratio of the outdoor air (Equations 3.26-3.30).

Use of only one equation that is valid for all regeneration temperatures would be easier than use of separate equations for each regeneration temperature.

In the next step of the study, possibility of representing all the data with only one equation was investigated. As a result, Equation 3.31 that includes influence of both humidity ratio of outdoor air and regeneration temperature was obtained:

$$F_d = F_r = 5.18 \times W^{-0.507} \times T_{reg.\ temp.}^{\ 0.652} \qquad (R^2=0.96) \tag{3.31}$$

Figure 3.14 shows the comparison of the values obtained from the curve and the real data. As seen from the figure, the equation follows the data successfully.

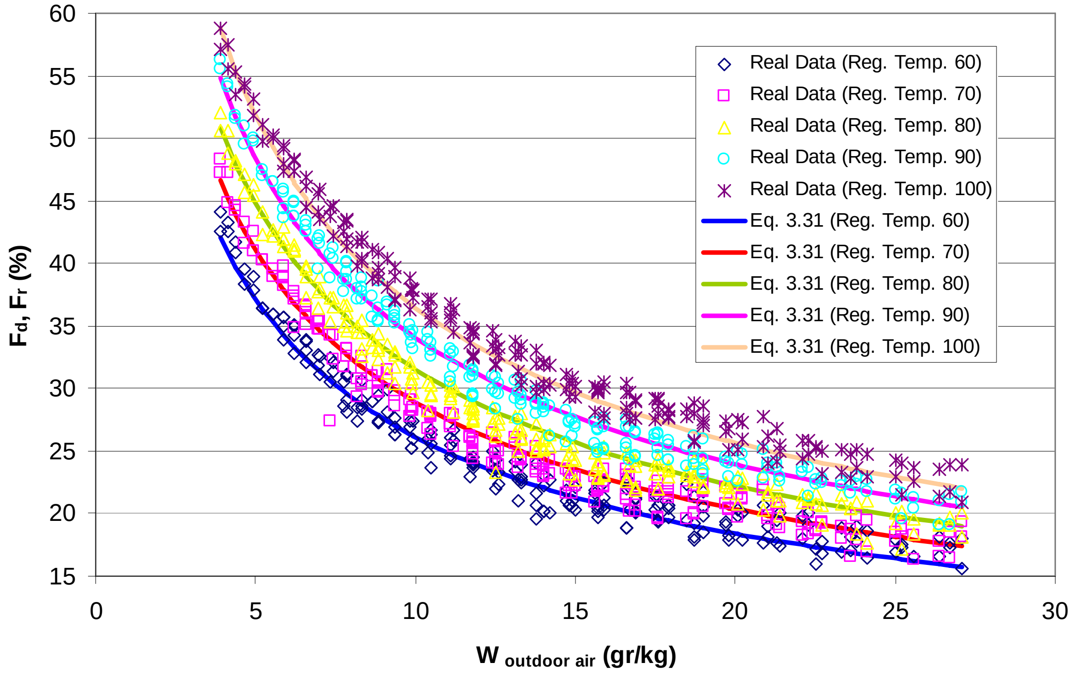

Figure 3.14. The values of F_d and F_r which were obtained at different regeneration air temperature related with the humidity ratio of the outdoor air (Equation 3.31).

By using the data received from the rotary wheel manufacturer, an equation of regeneration temperature ($t_{reg.temp}$.) which depends on the humidity ratio of the fresh air at the inlet (W_1) and outlet (W_2) of the rotary wheel dehumidifier was composed (Equation 3.32).

$$t_{reg.\,temp.} = 114.78 \times \left(W_1^{\,0.65419 \times W_2^{-0.099848}} \right) - 149.77 \times \mathrm{Alog}(W_1) \qquad (3.32)$$

Humidity ratio at state 2 is the same as the humidity ratio at state 6. Dry bulb temperature at state 2 (t_{db2}) can be calculated using Equations (3.24), (3.31) and (3.32). Other psychrometric properties (enthalpy (h_2), wet bulb temperature (t_{wb2}), relative humidity (φ_2), dew point temperature (t_{dp2}), density (d_2)) are calculated using Subroutine DA.

3.11.5. State 5

The fans that circulate air in the system cause a temperature rise in air, because fan motors are located inside the air channels. This temperature rise (T_o-T_i) can be calculated using the equations below:

$$Q_{gain} = \left(1 - \frac{\eta_{fan}}{100}\right) \times W_{fan} \tag{3.33}$$

$$T_o - T_i = \left(\frac{\dot{Q}_{gain}}{c_p \times \dot{m}} \right) \tag{3.34}$$

where,

Q_{gain} : Heat gain from fan, [kW]

η_{fan} : Efficiency of the fan

W_{fan} : Power of the fan, [kW]

T_o : Dry bulb temperature of air at the exit of the fan, [^{o}C]

T_i : Dry bulb temperature of air at the inlet of the fan, [^{o}C]

c_p : Specific heat of air [kJ/kg K]

$\dot{m}$: Mass flow rate of air, [kg/s]

Humidity ratio at state 5 is the same as the humidity ratio at state 6. Dry bulb temperature at state 5 (t_{db5}) can be calculated using Equations (3.33) and (3.34) and dry bulb temperature at state 6 (t_{db6}):

$$T_{db5} = T_{db6} - \frac{\dot{Q}_{gain}}{c_p \times \dot{m}_{fresh}} \tag{3.35}$$

Other psychrometric properties (enthalpy (h_5), wet bulb temperature (t_{wb5}), relative humidity (φ_5), dew point temperature (t_{dp5}), density (d_5)) are calculated using Subroutine DA.

3.11.6. State 3

The effectiveness of heat exchanger (η_{heatex}) is the ratio of actual ($\dot{Q}$) to maximum ($\dot{Q}_{max}$) possible heat transfer rates. They are defined with the following equations:

$$\eta_{heatex} = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{3.36}$$

$$\dot{Q}_{max} = C_{min} \times (t_{h,i} - t_{c,i}) \tag{3.37}$$

$$C_h = \dot{m}_h \times c_{p,h} \tag{3.38}$$

$$C_c = \dot{m}_c \times c_{p,c} \tag{3.39}$$

where,

$c_{p,h}$: Specific heat of hot fluid, [kJ/kg K]

$c_{p,c}$: Specific heat of cold fluid, [kJ/kg K]

C_h : Fluid capacitance rate of hot fluid, [W/K]

C_c : Fluid capacitance rate of cold fluid, [W/K]

C_{min} : Minimum fluid capacitance rate of fluids, [W/K]

$t_{h,i}$: Dry bulb temperature of the hot air at the inlet, [^oC]

$t_{c,i}$: Dry bulb temperature of the cold air at the inlet, [^oC]

$\dot{m}_h$: Mass flow rate of hot air, [kg/s]

$\dot{m}_c$: Mass flow rate of cold air, [kg/s]

Humidity ratio at state 3 is the same as the humidity ratio at state 6. Dry bulb temperature at state 3 (t_{db3}) can be calculated using Equations (3.36) to (3.39) and dry bulb temperatures at states 2 and 11 (t_{db2}, t_{db11}):

$$t_{db3} = t_{db2} - \eta_{heatex1} \times \frac{C_{min}}{C} \times (t_{db2} - t_{db11}) \tag{3.40}$$

Other psychrometric properties (enthalpy (h_3), wet bulb temperature (t_{wb3}), relative humidity (φ_3), dew point temperature (t_{dp3}), density (d_3)) are calculated using Subroutine DA.

3.11.7. State 8

Humidity ratio at state 8 is the same as the humidity ratio at state 7. Dry bulb temperature at state 8 (t_{db8}) can be calculated using Equations (3.33) and (3.34) and dry bulb temperatures at state 7 (t_{db7}):

$$T_{db8} = T_{db7} + \frac{\dot{Q}_{gain}}{c_p \times \dot{m}_{waste}} \tag{3.41}$$

Other psychrometric properties (enthalpy (h_8), wet bulb temperature (t_{wb8}), relative humidity (φ_8), dew point temperature (t_{dp8}), density (d_8)) are calculated using Subroutine DA.

3.11.8. State 9

The effectiveness of humidifier (η_{hum}) is defined as the dry-bulb temperature depression divided by the difference between the entering dry and wet bulb temperatures (Kreider et al., 1994):

$$\eta_{hum} = \frac{T_{d,i} - T_{d,o}}{T_{d,i} - T_{w,i}} \tag{3.42}$$

where,

η_{hum} : Effectiveness of the humidifier (%)

$T_{d,i}$: Dry bulb temperature of air at the inlet of humidifier, [^oC]

$T_{d,o}$: Dry bulb temperature of air at the exit of humidifier, [^oC]

$T_{w,i}$: Wet bulb temperature of air at the inlet of humidifier, [^oC]

Wet bulb temperature at state 9 is equal to the wet bulb temperature at state 8 (humidification occurs at constant wet bulb temperature). Dry bulb temperature at state 9 (t_{db9}) can be calculated using Equation (3.42), dry bulb temperature (t_{db8}) and wet bulb temperature (t_{wb8}) at state 8:

$$t_{db9} = t_{db8} - \eta_{hum} \times \left(t_{db8} - t_{wb8}\right) \tag{3.43}$$

Other psychrometric properties (enthalpy (h_9), humidity ratio (W_9), relative humidity (φ_9), dew point temperature (t_{dp9}), density (d_9)) are calculated using Subroutine DW.

3.11.9. State 4

Humidity ratio at state 4 is the same as the humidity ratio at state 6. Dry bulb temperature at state 4 (t_{db4}) can be calculated using Equations (3.36) to (3.39) and dry bulb temperatures at states 3 and 9 (t_{db3}, t_{db9}):

$$t_{db4} = t_{db3} - \eta_{heatex2} \times \frac{C_{min}}{C} \times \left(t_{db3} - t_{db9}\right) \tag{3.44}$$

Other psychrometric properties (enthalpy (h_4), wet bulb temperature (t_{wb4}), relative humidity (φ_4), dew point temperature (t_{dp4}), density (d_4)) are calculated using Subroutine DA.

3.11.10. State 10

Humidity ratio at state 10 is the same as the humidity ratio at state 7. Dry bulb temperature at state 10 (t_{db10}) can be calculated using Equations (3.36) to (3.39) and dry bulb temperatures at states 3 and 9 (t_{db3}, t_{db9}):

$$t_{db10} = t_{db9} - \eta_{heatex2} \times \frac{C_{min}}{C} \times \left(t_{db9} - t_{db3}\right) \quad (3.45)$$

Other psychrometric properties (enthalpy (h_{10}), wet bulb temperature (t_{wb10}), relative humidity (φ_{10}), dew point temperature (t_{dp10}), density (d_{10})) are calculated using Subroutine DA.

3.11.11. State 12

Humidity ratio at state 12 is the same as the humidity ratio at state 11. Dry bulb temperature at state 12 (t_{db12}) can be calculated using Equations (3.36) to (3.39), dry bulb temperatures at states 2 and 11 (t_{db2}, t_{db11}):

$$t_{db12} = t_{db11} - \eta_{heatex1} \times \frac{C_{min}}{C} \times \left(t_{db11} - t_{db2}\right) \quad (3.46)$$

Other psychrometric properties (enthalpy (h_{12}), wet bulb temperature (t_{wb12}), relative humidity (φ_{12}), dew point temperature (t_{dp12}), density (d_{12})) are calculated using Subroutine DA.

3.11.12. State 14

Humidity ratio at state 14 is the same as the humidity ratio at state 11. Dry bulb temperature at state 14 (t_{db14}) can be calculated using Equations (3.32).

Other psychrometric properties (enthalpy (h_{14}), wet bulb temperature (t_{wb14}), relative humidity (φ_{14}), dew point temperature (t_{dp14}), density (d_{14})) are calculated using Subroutine DA.

3.11.13. State 15

In the desiccant enthalpy wheel, the amount of humidity ratio difference in dehumidification process (W_1-W_2) was assumed equal to the amount of humidity ratio difference in regeneration process (W_{15}-W_{14}):

$$W_1 - W_2 = W_{15} - W_{14} \tag{3.47}$$

The humidity ratio at state 15 (W_{15}) can be calculated using Equation 3.47:

$$W_{15} = \left(W_1 - W_2\right) + W_{14} \tag{3.48}$$

Regeneration process that is explained in chapter 3.11.4 do not occur at constant wet bulb temperature according to the data given by the rotary desiccant wheel manufactures. This process is represented with the curve (C→D) in Figure 3.9. Dry bulb temperature decrease higher than constant wet bulb temperature case during regeneration (C→D'), due to sensible heat transfer to the process air. The ratio of additional dry bulb temperature decrease to total dry bulb temperature decrease (F_r) was calculated using Equation 3.31.

Dry bulb temperature at state 15 (t_{db15}) can be calculated using Equations (3.25), (3.31) and (3.32). Other psychrometric properties (enthalpy (h_{15}), wet bulb temperature (t_{wb15}), relative humidity (φ_{15}), dew point temperature (t_{dp15}), density (d_{15})) are calculated using Subroutine DA.

3.11.14. State 13

Humidity ratio at state 13 is the same as the humidity ratio at state 11. Dry bulb temperature at state 13 (t_{db13}) can be calculated using Equations (3.36) to (3.39), dry bulb temperatures at states 12 and 15 (t_{db12}, t_{db15}):

$$t_{db13} = t_{db12} - \eta_{heatex} \times \frac{C_{min}}{C} \times \left(t_{db12} - t_{db15}\right) \tag{3.49}$$

Other psychrometric properties (enthalpy (h_{13}), wet bulb temperature (t_{wb13}), relative humidity (φ_{13}), dew point temperature (t_{dp13}), density (d_{13})) are calculated using Subroutine DA.

3.11.15. State 16

Humidity ratio at state 16 is the same as the humidity ratio at state 15. Dry bulb temperature at state 16 (t_{db16}) can be calculated using Equations (3.36) to (3.39) and dry bulb temperatures at states 12 and 15 (t_{db12}, t_{db15}):

$$t_{db16} = t_{db15} - \eta_{heatex} \times \frac{C_{min}}{C} \times \left(t_{db15} - t_{db12}\right) \tag{3.50}$$

Other psychrometric properties (enthalpy (h_{16}), wet bulb temperature (t_{wb16}), relative humidity (φ_{16}), dew point temperature (t_{dp16}), density (d_{16})) are calculated using Subroutine DA.

3.11.16. State 17

Humidity ratio at state 17 is the same as the humidity ratio at state 16. Dry bulb temperature at state 17 (t_{db17}) can be calculated using Equations (3.33) and (3.34) and dry bulb temperature at state 16 (t_{db16}):

$$T_{db17} = T_{db16} - \frac{\dot{Q}_{gain3}}{c_p \times \dot{m}_{regen}} \tag{3.51}$$

Other psychrometric properties (enthalpy (h_{17}), wet bulb temperature (t_{wb17}), relative humidity (φ_{17}), dew point temperature (t_{dp17}), density (d_{17})) are calculated using Subroutine DA.

3.11.17. Heat Transferred in Heat Exchangers

The equations which were used in the calculation of the amount of heat transfer that take place in heat exchangers are presented in Table 3.5.

Table 3.5. Equations used for the calculation of heat transfer

No	***Calculated Parameter***	***Equations Used***
Heat exchanger 1	$Q_{2\rightarrow 3}$	$\dot{m}_{fresh} * (h_2 - h_3)$
Heat exchanger 2	$Q_{3\rightarrow 4}$	$\dot{m}_{fresh} * (h_3 - h_4)$
Heat exchanger 3	$Q_{4\rightarrow 5}$	$\dot{m}_{fresh} * (h_4 - h_5)$
Electric Heater	$Q_{14\rightarrow 13}$	$\dot{m}_{reac} * (h_{14} - h_{13})$
Heat exchanger 4	$Q_{15\rightarrow 16}$	$\dot{m}_{reac} * (h_{15} - h_{16})$

In these heat exchangers used here:

Heat exchanger 1 : Dry bulb temperature of fresh air at the exit desiccant wheel rotary decreased using Heat Exchanger 1. The amount of heat that is transferred from the fresh air to the regeneration air is shown as $Q_{2\rightarrow 3}$.

Heat exchanger 2 : In order to further decrease temperature of the fresh air leaving Heat exchanger 1, the waste air that is cooled in the humidifier is used. The amount of heat that is transferred from the fresh air to the waste air is shown as $Q_{3\rightarrow 4}$.

Heat exchanger 3 : Desired blowing temperature of the fresh air is obtained in the cooling coil. The amount of heat extracted from the fresh air is shown as $Q_{4\rightarrow 5}$.

Heat exchanger 4 (Rotary regenerator) : Temperature of the regeneration air at the exit of the desiccant wheel is generally higher than that of the regeneration air leaving heat exchanger 1. The regeneration air leaving the desiccant wheel (15) passes through the heat exchanger (12-13), In which heat is transferred from the regeneration air that left the desiccant wheel (state 15) to regeneration air left the heat exchanger 1 (state 12). If the temperature of the regeneration air at state 16 is not higher than that of state 12, rotary regenerator is stopped. The amount of heat that is extracted from the regeneration air is shown as $Q_{15\rightarrow16}$.

Electric Heater : Temperature of the regeneration air that is increased in heat exchanger (1) and rotary regenerator is not high enough for dehumidification of the desiccant wheel. Temperature can be further increased with the help of electric heaters. The amount of heat that is given by the electric heater is shown as $Q_{14\rightarrow13}$.

Flowchart of the computer program written is given in Figures 3.15 and 3.16.

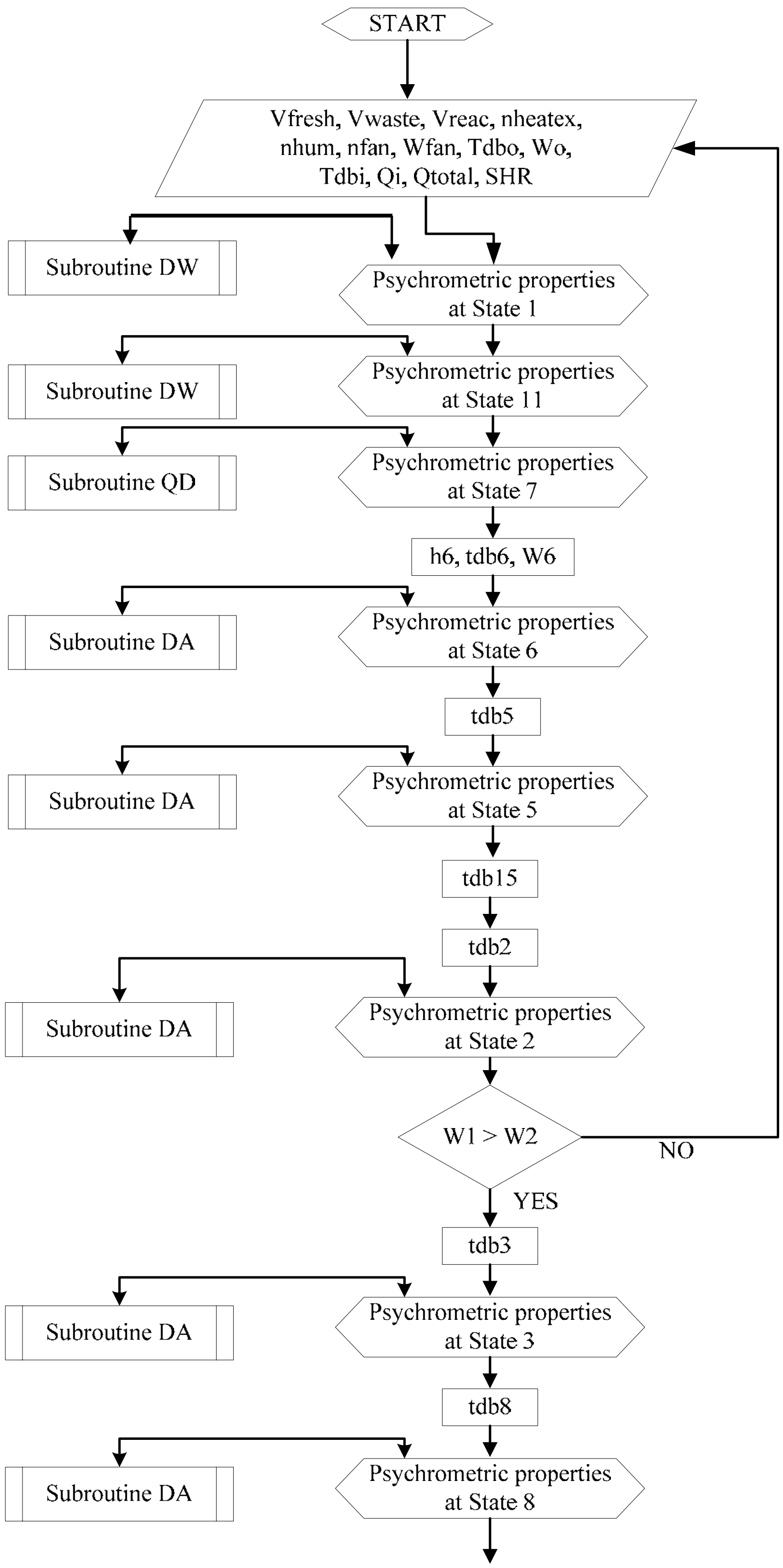

Figure 3.15. Flowchart of the computer program written

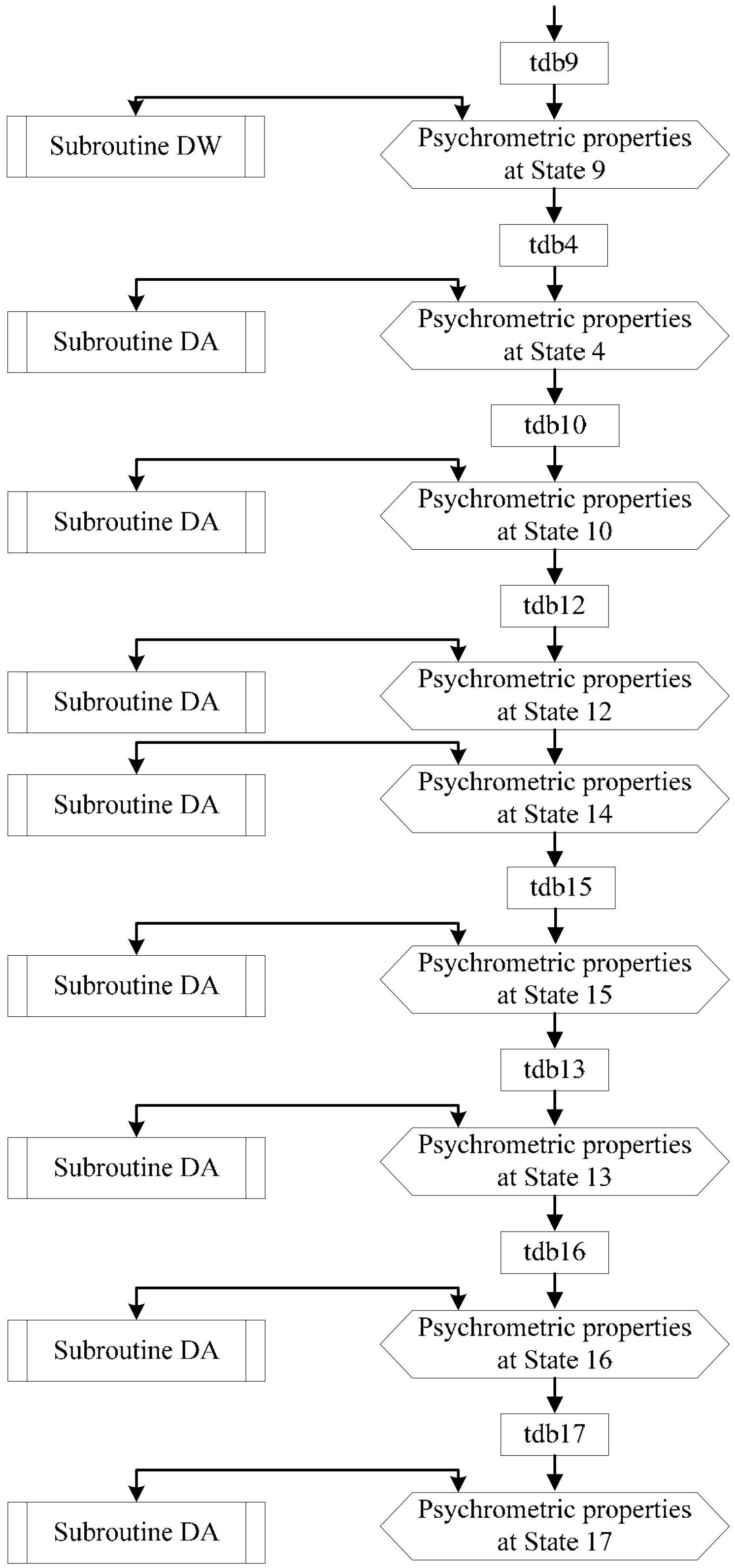

Figure 3.16 (Continued) Flowchart of the computer program written

4. RESULTS AND DISCUSSION

In this study, the performance of the system designed under the conditions of Adana is analyzed using the hourly climate data (dry bulb temperature and relative humidity), which were measured by DMI (Turkish State Meteorological Service) for the period 1983-2006. In order to analyze the performance of the system under summer conditions, the climate data, which were measured between the months June and September for the hours 07:00, 14:00 and 21:00 (relative humidity is measured terdiurnal by DMI), are used in the calculations.

The inputs of the programme used are dry bulb temperature and relative humidity of outside air (state 1). The psychrometric properties of the each state of the system are calculated using the equations which are given above (Equations 3.1-3.51) and the values given in Table 4.1. It should be noted that the humidity ratio changes only during the processes of dehumidification (1→2), humidity removal (15→16) and humidification (9→10). It is not changed during the other processes. In the analysis, the climate data are not taken into consideration, if the humidity ratio of the outside is less than that of the air-conditioned room.

Tablo 4. 1. The values used in the calculations

Parameter	Value
Total cooling load of the air-conditioned room (kW)	10
Design dry bulb temperature (^oC) and relative humidity (%) of the air-conditioned room	26 - 50
Flow rate of the fresh air (m^3/h)	4000
Flow rate of the exhaust air (m^3/h)	4000
Flow rate of the regeneration air (m^3/h)	4000
Effectiveness of the recuperators (%)	65
Effectiveness of the regenerator (%)	85
Sensible heat ratio of the air-conditioned room	0.9
Effectiveness of the evaporative cooler (%)	90
Efficiency of the fans (%)	60
Powers of the fans (kW) (Fresh-Waste-Regeneration)	3-1-4

The dry bulb temperature and the relative humidity measured at 07:00 h, 14:00 h and 21:00 h on 21st July in a 24 year period (1983-2006) are used to obtain average dry bulb temperature and relative humidity for the 21st July of each year. These average values are used to calculate the psychrometric properties of the each state of the system at 07:00, 14:00 and 21:00 hours on 21st of July. The amount of heat transferred in the heat exchangers is calculated using Table 3.5.

Table 4.2 shows results of the psychrometric analysis of the each state of the system at 07:00 h. Fresh outside air (26.3 ^{o}C dry bulb temperature, 85.9 % relative humidity) is sucked to the desiccant wheel. After the dehumidification process, the dry bulb temperature of fresh air increases to 52.48 ^{o}C and relative humidity decreases to 11.81 %. The fresh air leaves the first heat exchanger at 35.56 ^{o}C and leaves the second heat exchanger at 25.15 ^{o}C. The fresh air must be brought into the third heat exchanger in order to obtain the desired blowing temperature of the fresh air (18.10 ^{o}C). The air, which is sucked from the air-conditioned room, gets hotter (26.29 ^{o}C) while passing through the fan. The waste air is humidified (93.28 % relative humidity) in order to increase savings during heat transfer. In the evaporative cooler the waste air cools down to 19.54 ^{o}C. The waste air leaving the heat exchanger 2 is rejected to the outdoors (29.93 ^{o}C dry bulb temperature and 50.13 % relative humidity).

In regeneration process, the dry bulb temperature of regeneration air is increased to 43.32 ^{o}C in heat exchanger 1 and to 56.18 ^{o}C in the rotary regenerator. In order to remove moisture in the desiccant wheel, the dry bulb temperature of the regeneration air is increased to 84.44 ^{o}C by the means of electric heater. The regeneration air leaving the desiccant wheel comes to the rotary regenerator and cools down to 46.81 ^{o}C, before discharged to the outdoors.

Table 4.3 shows results of the psychrometric analysis of the each state of the system at 14:00 h. The outdoor fresh air conditions at 07:00 h and 14:00 h are different. At 14:00 h, dry bulb temperature increases up to 32.91 ^{o}C and relative humidity decreases down to 53.14 %. This difference affects the fresh air process and regeneration process of the system at 14:00 h. The amount of the moisture removed by dehumidifier decreases. Therefore, the dry bulb temperature of regeneration air, which is used to remove moisture, decreases.

Table 4.4 shows that the results of the psychrometric analysis of the each state of the system at 07:00 h and 21:00 h are almost the same. This is because of the fact that outdoor air conditions (dry bulb temperature and relative humidity) are very close to each other.

Psychrometric diagrams of the system for 07:00 h, 14:00 h and 21:00 hours on 21 July are shown in Figures 4.1, 4.2 and 4.3.

Table 4.2. Some psychrometric properties of the each state at 7:00 h for July 21st (outdoor air temperature is 26.3 ^oC and relative humidity is 85.9 %).

State	t_{db} (^oC)	W (gr/kg)	t_{wb} (^oC)	RH (%)	t_{dp} (^oC)	h (kj/kg)	d (kg/m^3)
1	26.30	18.59	24.42	85.90	23.75	73.83	1.17
2	52.48	10.29	26.00	11.81	14.50	79.50	1.08
3	35.56	10.29	21.48	28.39	14.50	62.17	1.14
4	25.15	10.29	18.25	51.55	14.50	51.49	1.18
5	18.10	10.29	15.81	79.37	14.50	44.27	1.20
6	18.99	10.29	16.14	75.07	14.50	45.18	1.20
7	26.00	10.50	18.70	50.00	14.81	52.90	1.17
8	26.29	10.50	18.79	49.14	14.81	53.20	1.17
9	19.54	13.29	18.79	93.28	18.45	53.38	1.20
10	29.93	13.29	21.96	50.13	18.45	64.08	1.16
11	26.30	18.59	24.42	85.90	23.75	73.83	1.17
12	43.32	18.59	28.52	33.44	23.75	91.53	1.10
13	56.18	18.59	31.20	17.64	23.75	104.90	1.06
14	84.44	18.59	36.20	5.19	23.75	134.28	0.98
15	58.45	26.89	35.19	22.63	29.79	128.90	1.05
16	45.63	26.89	32.93	42.38	29.79	115.38	1.09
17	46.81	26.89	33.15	39.91	29.79	116.62	1.09

Table 4.3. Some psychrometric properties of the each state at 14:00 h for July 21st (outdoor air temperature is 32.91 ^oC and relative humidity is 53.14 %).

State	t_{db}(^oC)	W(gr/kg)	t_{wb}(^oC)	RH(%)	t_{dp}(^oC)	h(kj/kg)	d(kg/m^3)
1	32.91	16.78	25.02	53.14	22.11	76.07	1.14
2	53.20	10.28	26.17	11.40	14.50	80.22	1.07
3	40.07	10.28	22.76	22.23	14.50	66.77	1.12
4	26.73	10.28	18.76	46.94	14.50	53.10	1.17
5	17.93	10.28	15.75	80.17	14.50	44.09	1.21
6	18.84	10.28	16.08	75.74	14.50	45.02	1.20
7	26.00	10.50	18.70	50.00	14.81	52.90	1.17
8	26.29	10.50	18.79	49.14	14.81	53.20	1.17
9	19.54	13.29	18.79	93.28	18.45	53.38	1.20
10	32.60	13.29	22.72	43.09	18.45	66.82	1.15
11	32.91	16.78	25.02	53.14	22.11	76.07	1.14
12	46.10	16.78	28.17	26.23	22.11	89.74	1.09
13	51.71	16.78	29.40	19.81	22.11	95.55	1.08
14	72.81	16.78	33.51	7.56	22.11	117.42	1.01
15	52.70	23.28	32.67	25.92	27.41	113.44	1.07
16	47.10	23.28	31.60	34.24	27.41	107.57	1.09
17	48.30	23.28	31.84	32.22	27.41	108.83	1.08

Table 4.4. Some psychrometric properties of the each state at 21:00 h for July 21st (outdoor air temperature is 27.17 ^oC and relative humidity is 83.1 %).

State	t_{db}(^oC)	W(gr/kg)	t_{wb}(^oC)	RH(%)	t_{dp}(^oC)	h(kj/kg)	d(kg/m^3)
1	27.17	18.94	24.86	83.10	24.05	75.63	1.16
2	54.52	10.29	26.49	10.70	14.50	81.59	1.07
3	36.85	10.29	21.86	26.45	14.50	63.48	1.13
4	25.60	10.29	18.40	50.19	14.50	51.95	1.17
5	18.07	10.29	15.81	79.49	14.50	44.24	1.20
6	18.96	10.29	16.13	75.17	14.50	45.15	1.20
7	26.00	10.50	18.70	50.00	14.81	52.90	1.17
8	26.29	10.50	18.79	49.14	14.81	53.20	1.17
9	19.54	13.29	18.79	93.28	18.45	53.38	1.20
10	30.74	13.29	22.19	47.87	18.45	64.91	1.15
11	27.17	18.94	24.86	83.10	24.05	75.63	1.16
12	44.95	18.94	29.05	31.30	24.05	94.12	1.10
13	57.38	18.94	31.60	16.97	24.05	107.05	1.06
14	86.73	18.94	36.70	4.84	24.05	137.58	0.97
15	59.57	27.59	35.65	22.02	30.22	131.91	1.04
16	47.18	27.59	33.51	40.14	30.22	118.83	1.08
17	48.36	27.59	33.73	37.82	30.22	120.08	1.08

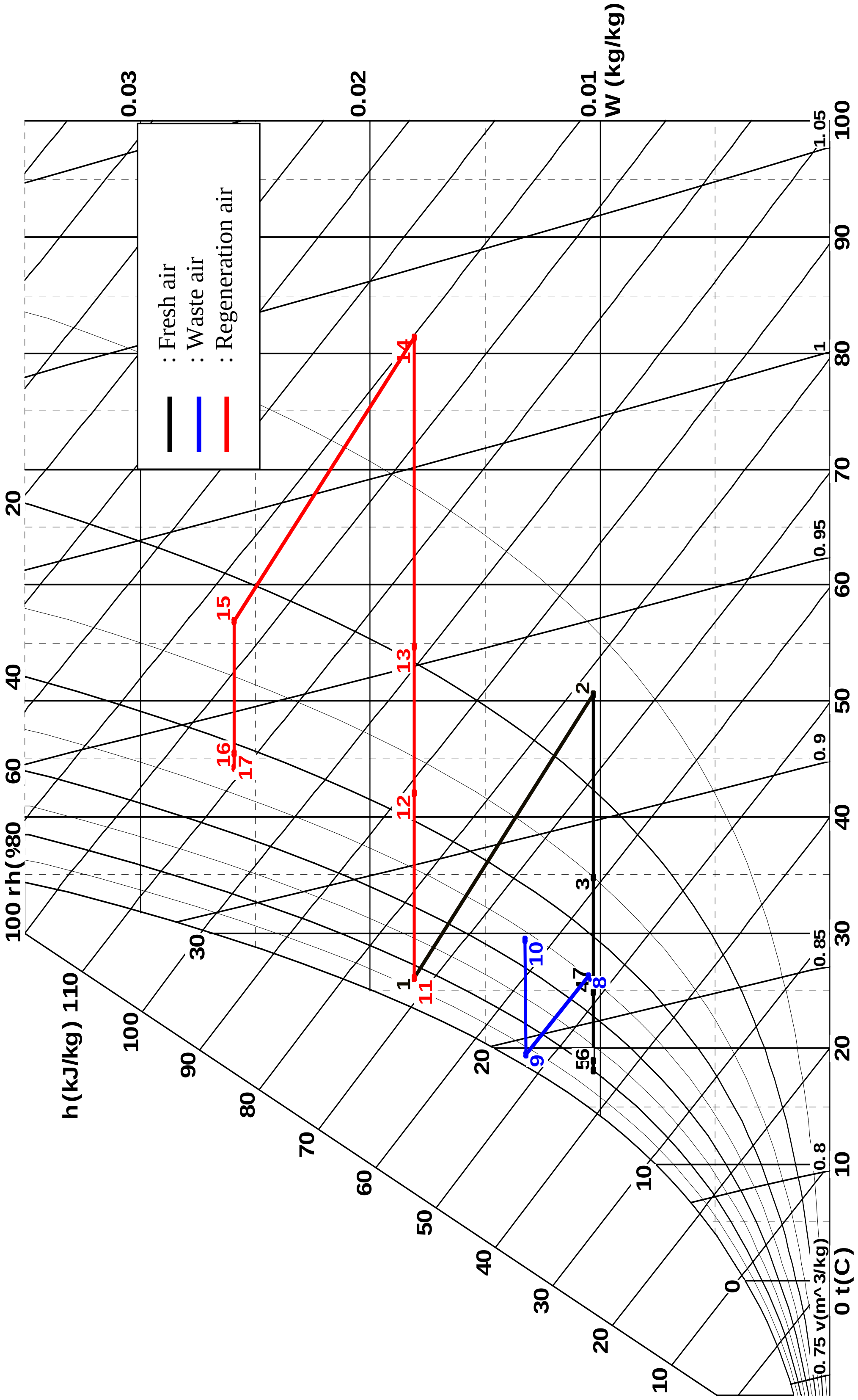

Figure 4.1. Psychrometric diagram of the system at 07:00 h on 21 July

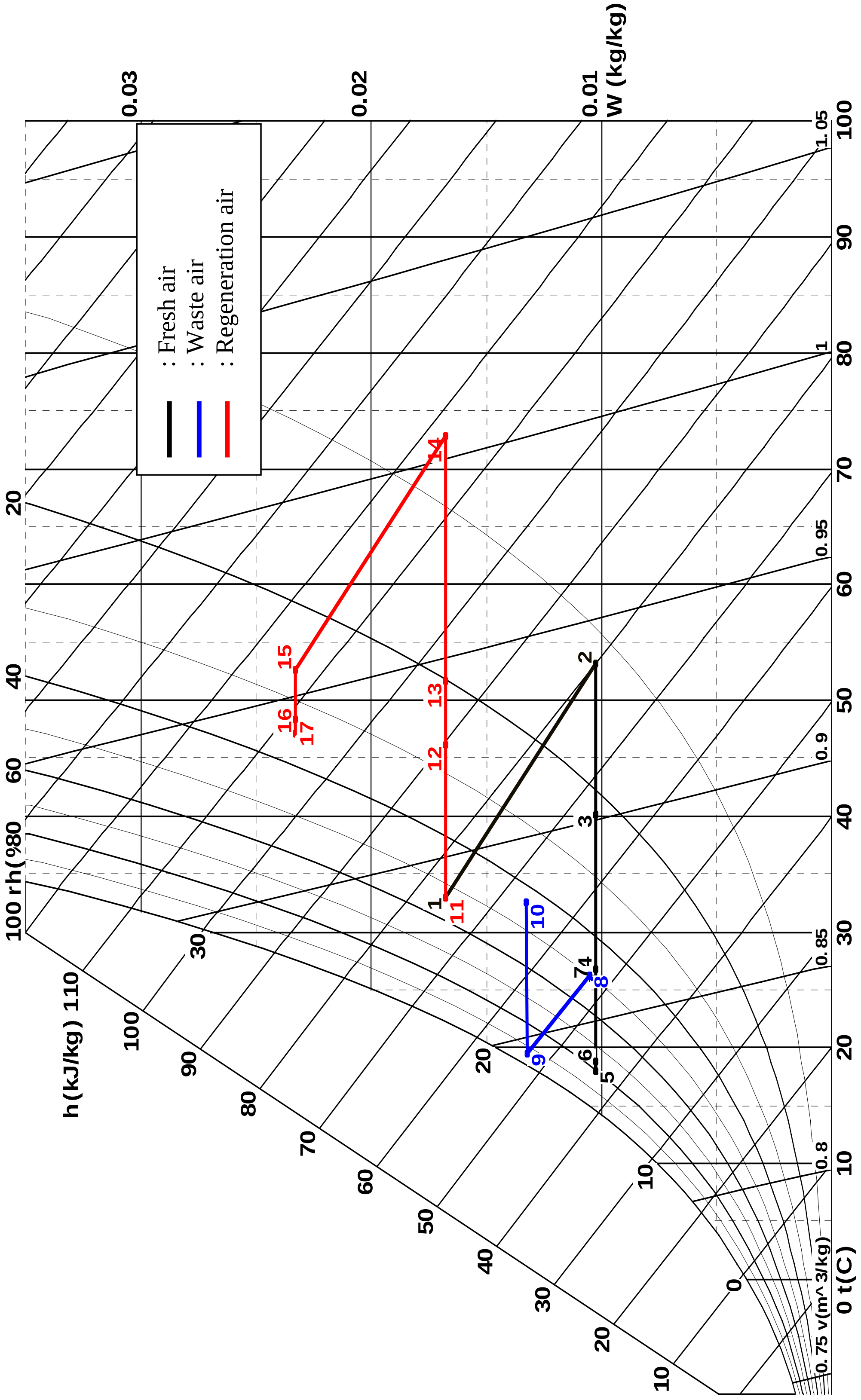

Figure 4.2. Psychrometric diagram of the system at 14:00 h on 21 July

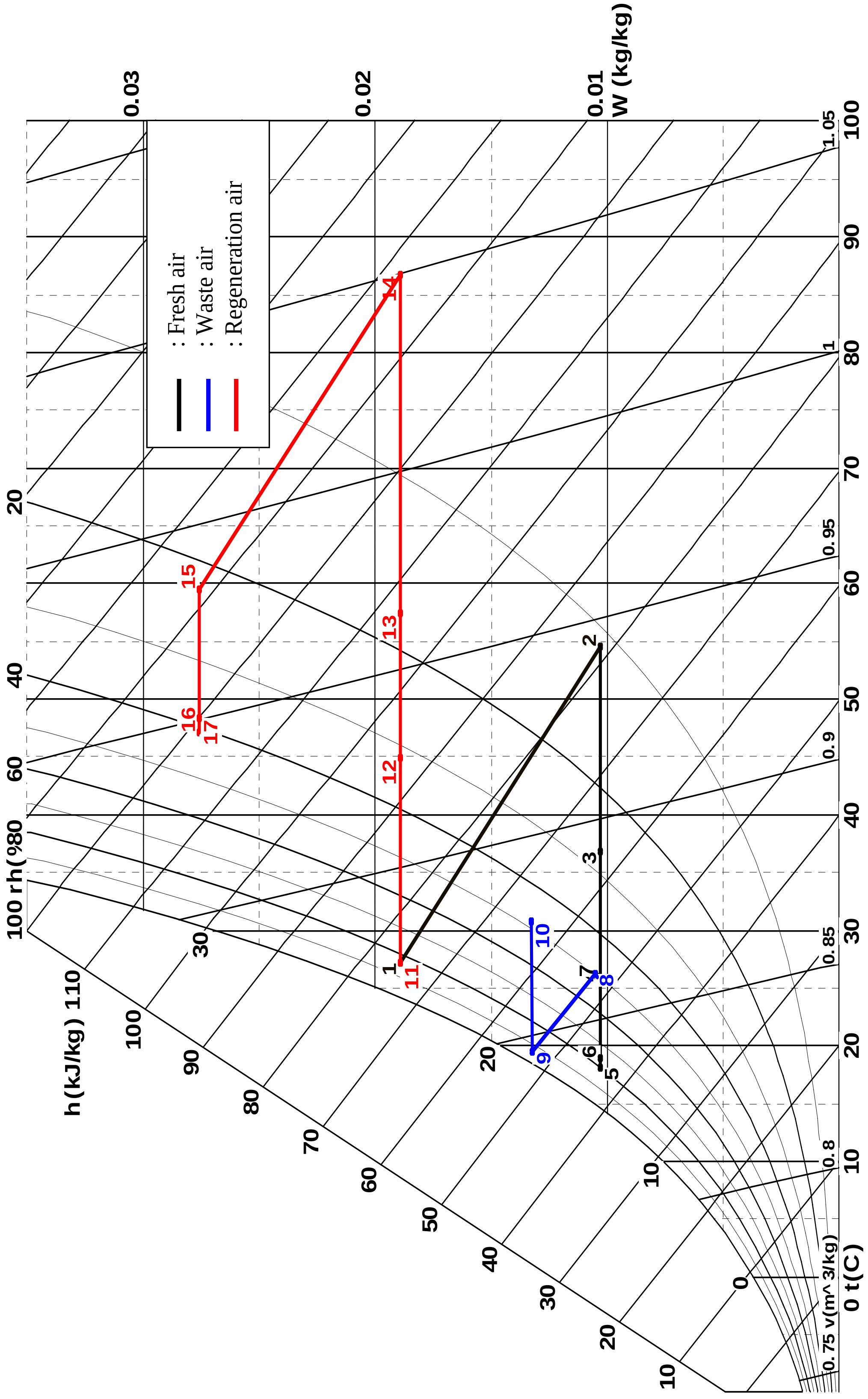

Figure 4.3. Psychrometric diagram of the system at 21:00 h on 21 July

Figure 4.4 shows the amount of heat transferred in the heat exchangers at 07:00, 14:00 and 21:00 hours for July 21$^{st.}$. As can be seen from the figure, the amount of heat transferred in the heat exchanger 1, electric heater, rotary regenerator at 07:00 and 21:00 hours is greater than at 14:00 h. This is due to fact that the relative humidity of the outdoor air is higher and dry bulb temperature of outdoor air is lower at 07:00 and 21:00 hours. The amount of heat transferred in the heat exchangers 2, 3 at 14:00 h is greater than at 07:00 and 21:00 hours. The reason of this that the dry bulb temperature difference between regeneration air and fresh air in the heat exchanger 1 at 07:00 and 21:00 hours is greater than at 14:00 h. Therefore, the maximum heat transfer in the heat exchanger 1 is 24 kW at 21:00 h. The amount of heat transferred in the rotary regenerator is higher at 07:00 and 21:00 hours, since dry bulb temperature of the air leaving the desiccant wheel at 07:00 and 21:00 hours is higher than 14:00 h. Average heat transfer in the rotary regenerator is 17 kW at 07:00 and 21:00 hours. Desiccant wheel absorbs more moisture from the fresh air at 07:00 and 21:00 hours than at 14:00 h. In order to remove moisture from the desiccant wheel, more heat must be given to regeneration air by the electric heater. The amount of heat given by the electric heater is approximately 38 kW at 07:00 and 21:00 hours. The reason why heat transfer in the heat exchanger 2, 3 is maximum at 14:00 h is that the dry bulb temperature of the fresh air at the exit of heat exchanger 1 is higher at 14:00 h than at 07:00 h and 21:00 h.

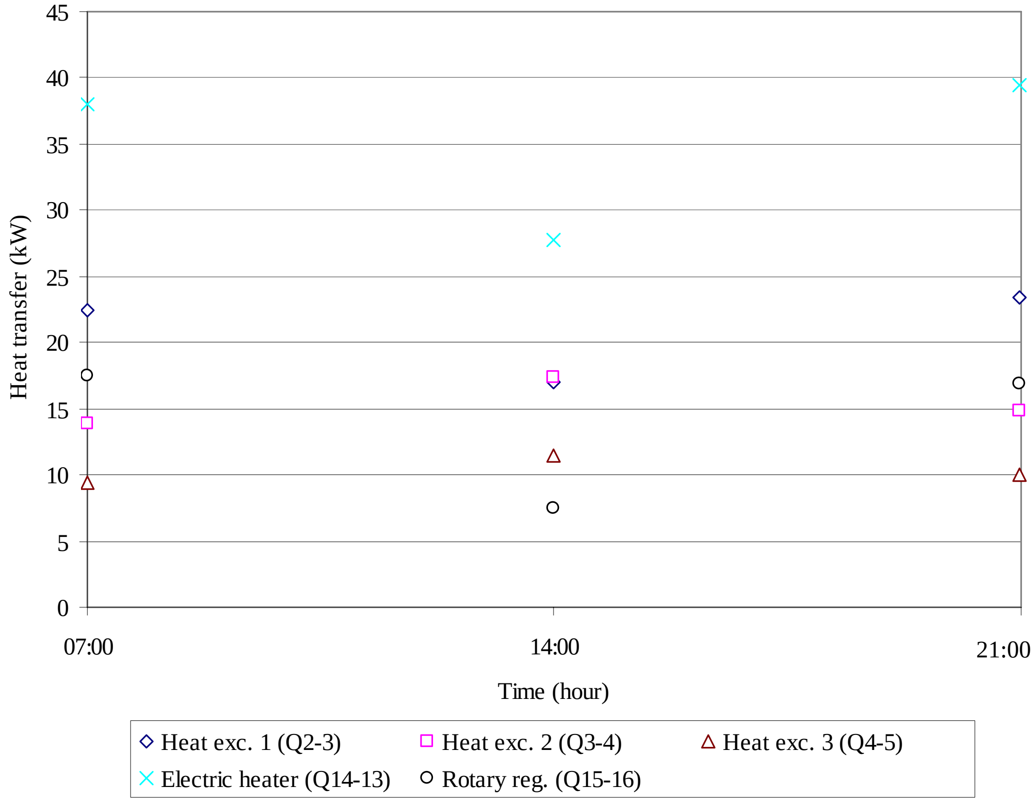

Figure 4.4. Amount of heat transferred in the heat exchangers at 07^{00}, 14^{00} and 21^{00} hours for July 21^{st}

Figure 4.5 shows the amount of heat transferred in the heat exchangers between 07:00 h and 21:00 h in July 21^{st}. The outside dry bulb temperature, which is used as input in the program, was obtained from the hourly measurements. However, the relative humidity values were obtained calculating daily average values. The humidity values measured at 07:00, 14:00 and 21:00 hours were used to calculate daily average values. As shown in Figure 4.5, daily variation of heat transfer in the rotary regenerator is greater than other heat exchangers. The amount of heat transferred in the rotary regenerator is minimum at 14:00 h. Daily average heat load of the electric heater is approximately 34 kW.

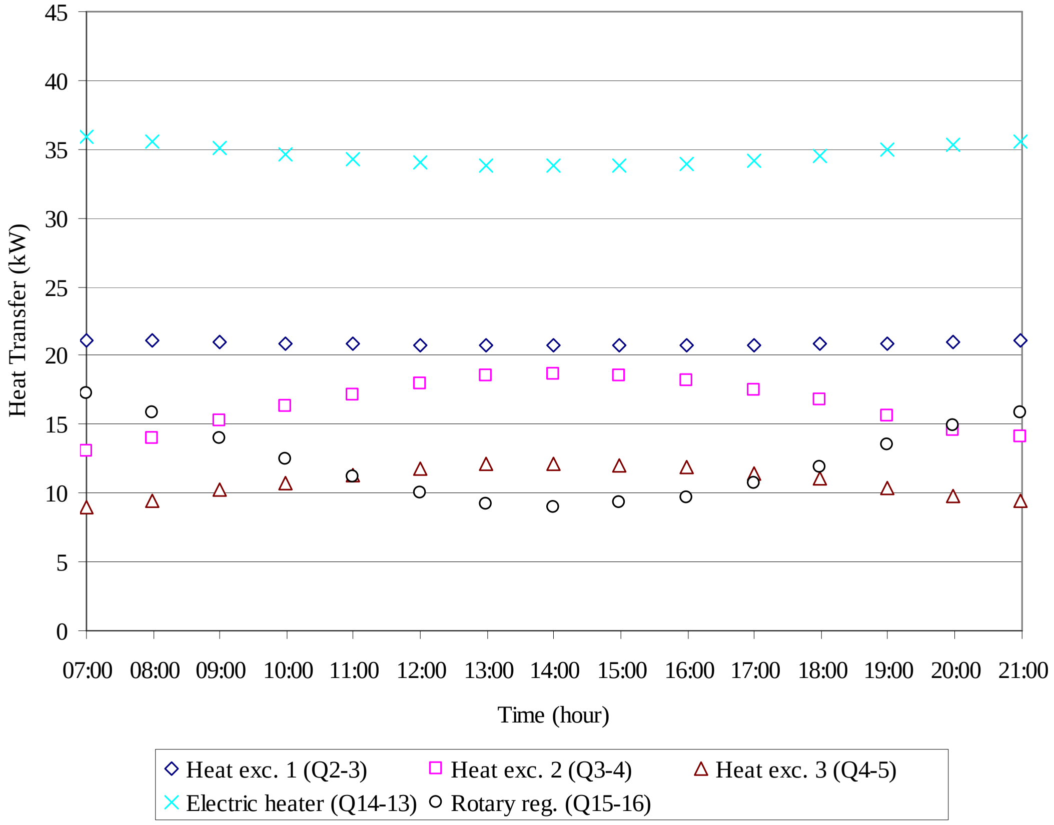

Figure 4.5. Amount of heat transfer in each heat exchangers between 07^{00} and 21^{00} hours in July 21^{st}

Figures 4.6 to 4.8 show the amount of heat transferred in the heat exchangers during summer season (from June to September). Heat load required for removing moisture (State 13-14) on desiccant wheel is between 8 to 42 kW at 07:00 h, between 3 to 30 kW at 14:00 h and between 13 to 43 kW at 21:00 h. It can also be seen from these figures that the amount of heat transferred in the heat exchangers starts to increase in June, reaches to the maximum level in July and August and decreases in September. If the daily period is considered, the amount of heat transfer in each heat exchanger at 07:00 h and 21:00 h is greater than at 14:00 h.

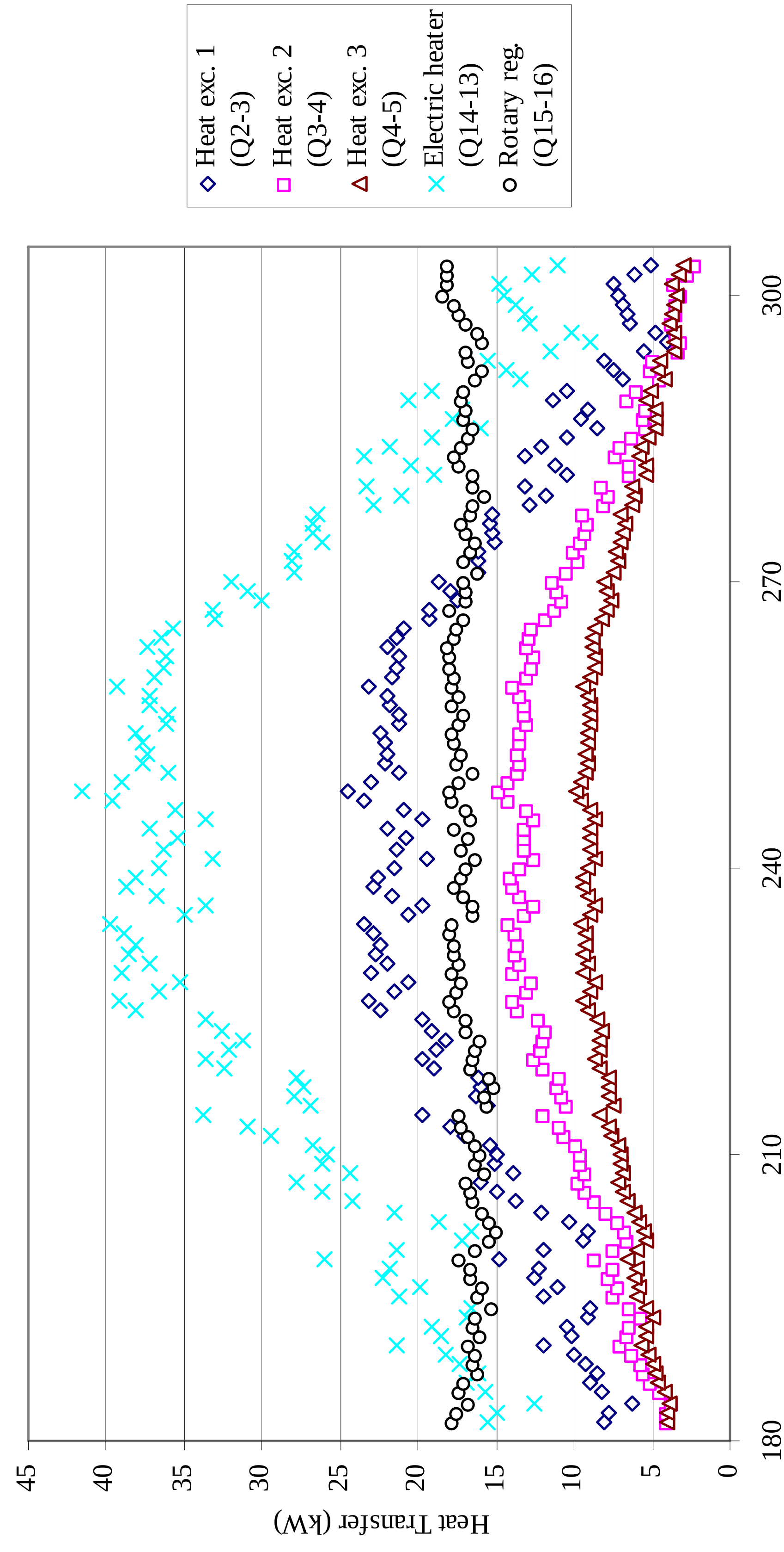

Figure 4.6. Distribution of the heat transfer at each heat exchanger for 4 months (from June to September) at 07^{00} h

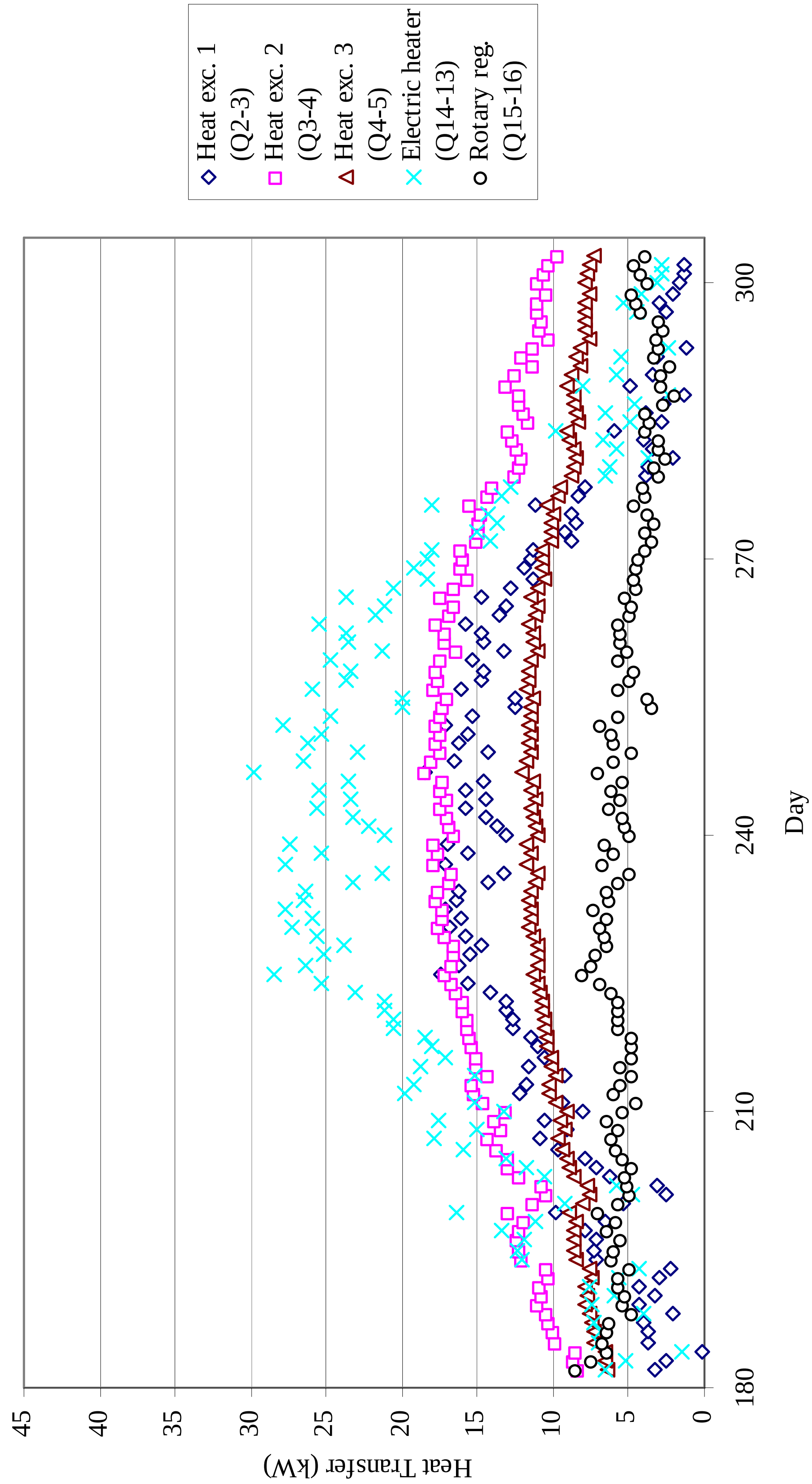

Figure 4.7. Distribution of the heat transfer at each heat exchanger for 4 months (from June to September) at 14^{00} h

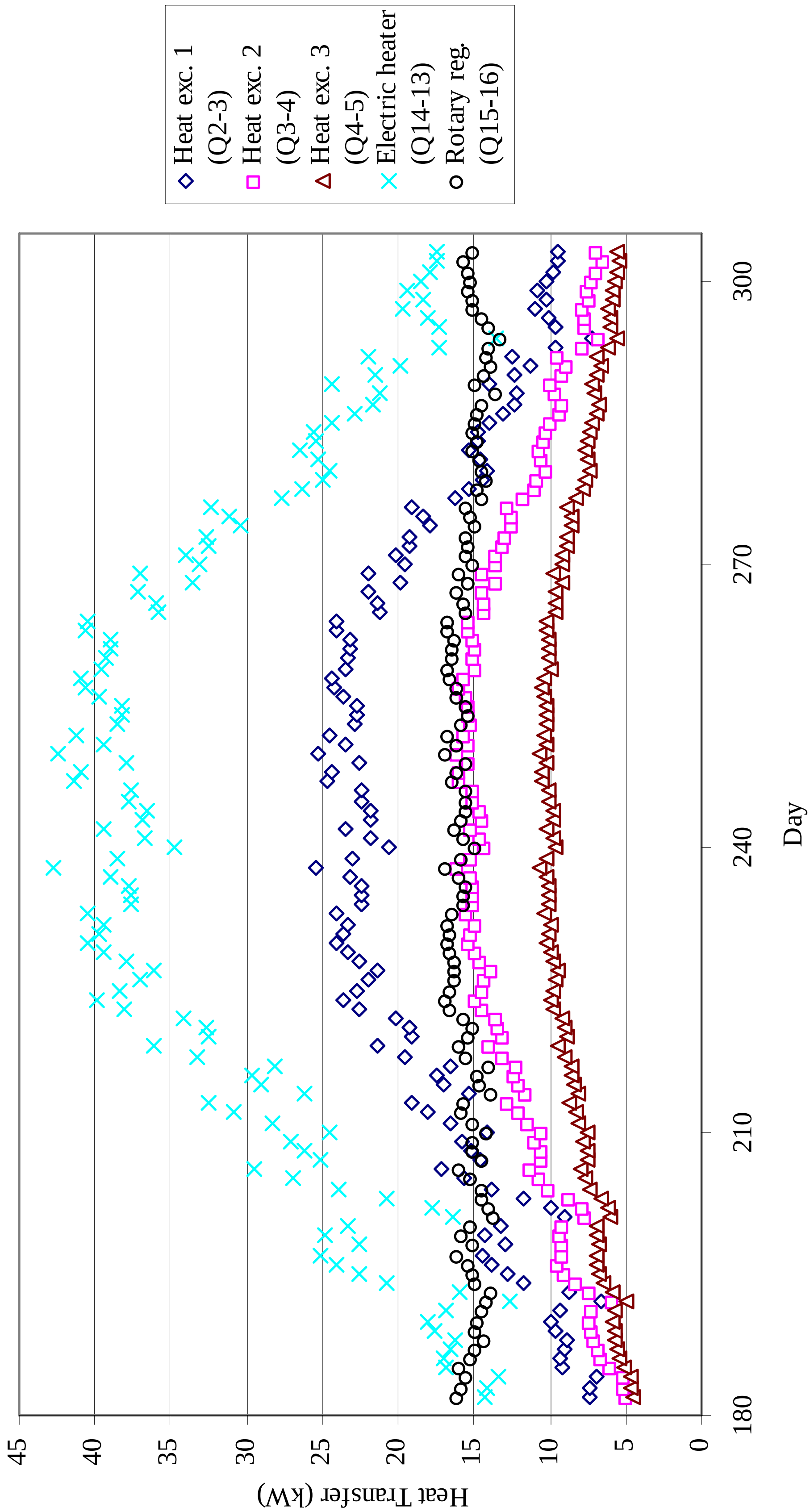

Figure 4.8. Distribution of the heat transfer at each heat exchanger for 4 months (from June to September) at 21^{00} h

5. CONCLUSIONS

In order to have a good result from medical treatments, all harmful organisms for human health should be removed; clean and ideal conditions should be provided in the hospitals that the main aim is to protect the human health and healing. Because, the patients have more risk to catch an infection comparing with healthy people. This can cause to put patients who stays in hospital, into danger and even can cause their deaths. Architecture and mechanical installations and air-condition system of the hospital are of great importance for the hygiene in the hospitals.

This study aimed to identify the role of desiccant systems to reduce the infection risks of intensive care patients in the hospital. Using desiccant cooling systems, the humidity of the hospitals can be controlled.

In this study, a computer program was written in order to simulate hourly operating of the air-conditioning system considered with desiccant wheel. The program calculates the parameters for each state and the amount of heat transferred in each heat exchanger.

The amount of heat transfer with the values given in Table 4.1 for the design day during summer season were analyzed by the program developed for variable outdoor conditions (variable dry bulb temperature, relative humidity). Hourly psychometric properties of each state in the system and hourly amount of heat transfer were calculated using computer program. Influence of the outdoor air conditions on the amount of heat transfer takes place in the desiccant cooling system was discussed. The conditions of air-conditioned room are significantly affected by changing outside conditions. The results of the analysis shown that the heat load required for the dehumidification of desiccant wheel (state 13-14) during summer period in Adana is between 8 to 42 kW at 07:00 h, between 3 to 30 kW at 14:00 h and between 13 to 43 kW at 21:00 h.

The results shown that the thermal energy required for removing moisture of desiccant wheel is very important parameter for the economic operation of the desiccant systems. If the thermal energy can be obtained by cheap thermal energy sources such as natural gas, waste heat, solar energy, operating costs can be decreased considerably.

In this study, electric energy is used for the removal of the moisture the desiccant wheel. In the future studies, the use of alternative energy sources such as solar energy and natural gas should be investigated in detail. The design of such systems, feasibility and possible advantages and disadvantages during operation of the system should be discussed.

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CURRICILUM VITAE

Osman KARA was born in Mersin, 1982. After being graduated from Tevfik Sırrı College, he enrolled in Mechanical Engineering Department of Ataturk University. He started his Master of Science education in Mechanical Engineering Department of Çukurova University in 2004. He has been working as an engineer in ADVANSA SASA.

APPENDIX A1

CUKUROVA UNIVERSITY ENG. - ARCH. DEPARTMENT
MECHANICAL ENG. DEPARTMENT LABORATORY HEAT GAIN TABLE

Location :	Mechanical Eng. Laboratory	Block :	1	Flat number:	1	Room name:	Air-Conditioning R.

A. HEAT GAINED FROM SOLAR RADIATION

1. From Window Glass

DIRECTION	F(m^2)		SOLAR TIME (HOUR) 8	10	12	14	16	18
N		q	18	20	20	20	18	33
		Qr						
E		q	231	138	20	19	17	7
		Qr						
S		q	17	50	78	50	17	7
		Qr						
W	16.74	q	17	20	20	138	231	157
		Qr	284.6	334.8	334.8	2310.1	3866.9	2628.2
		TOTAL	284.6	334.8	334.8	2310.1	3866.9	2628.2

2. From External Rooms

DIRECTION	k	F(m^2)		8	10	12	14	16	18
N			ΔTeş	1.1	2.2	5.5	8.9	11.1	10.0
			Q						
E			ΔTeş	15.8	18.7	17.7	9.7	11.1	10.8
			Q						
S			ΔTeş	3.3	3.3	3.3	6.6	11.1	12.2
			Q						
W	1.64	24.44	ΔTeş	1.1	3.1	6.4	13.2	22.3	24.8
			Q	44.1	124.3	256.5	529.1	893.8	994.0
		TOTAL		44.1	124.3	256.5	529.1	893.8	994.0

3. From Roofs

k	F(m^2)		8	10	12	14	16	18
		ΔTeş	5.2	7.1	12.4	18.9	23.6	23.7
		Q						
		A TOTAL	**329**	**459**	**591**	**2839**	**4761**	**3622**

B. HEAT GAINED FROM CONVECTION

	k	F	ΔT	Q
1. All window glass :	4.3	16.74	12	863.8
2. Internal walls :	0.987	79.65	6	471.7
3. Roofs :				1677.0
4. Doors :	0.5	3.6	6	10.8
4. Floors :				
		B TOTAL		3023

C. HEAT GAINED FROM PEOPLE

	People	Heat Load (kcal/h)	Total (kcal/h)
Sensible	4	54	216
Latent		59	236

D. HEAT GAINED FROM VARIOUS DEVICES

		Heat Load
Sensible	Electric Lighting	559
	Various Equipments	350
Latent	Various Equipments	

E. VENTILATION REQUIRED

Fresh Air Required	People	4	Required	30	120	m^3/h

TOTAL	SOLAR TIME (HOUR) 8	10	12	14	16	18
Sensible	4477	4607	4740	6987	8909	7770
Latent	236	236	236	236	236	236
TOTAL (kcal/h)	4713	4843	4976	7223	9145	8006
TOTAL(W)	5480	5632	5786	8399	10634	9310

APPENDIX A2

```
C********************************************************************************
C************************* DESICCANT COOLING SYSTEM *************************
C*******************************   (September 2008) *********************************
C********************************************************************************
C++++++++ Subroutine DW calculates properties of air stream +++++++++++++++++++++++++++
C++++++++ when dry bulb and wet bulb temperatures of air are given ++++++++++++++++++++++
C++++++++ Subroutine DA calculates properties of air stream +++++++++++++++++++++++++++
C++++++++ when dry bulb temperature and humidity ratio of air are given ++++++++++++++++++
C++++++++ Subroutine WA calculates properties of air stream +++++++++++++++++++++++++++
C++++++++ when wet bulb temperature and humidity ratio of air are given ++++++++++++++++++
C++++++++ Subroutine QD calculates properties of air stream +++++++++++++++++++++++++++
C++++++++ when relative humidity and dry bulb temperature of air are known ++++++++++++++++

      program main
      real mfresh, mwaste, mreac
      real nheatex1, nheatex2, nheatex5, nhum, nfan
      integer TT
      nheatex1=0.65
      nheatex2=0.65
      nhum=0.90
      nheatex5=0.85
      nfan=0.6
      Vfresh=4000
      Vwaste=4000
      Vreac=4000
      Wfan1=3
      Wfan2=1
      Wfan3=4
      Qtotal=10
      SHR=0.93
      tdb7=26
      Q7=0.5
```

```
      OPEN(4, FILE='Desiccant Cooling.txt', status='old')
      OPEN(5, FILE='Desiccant CoolingH.txt', status='unknown')
      I=6
      DO 12 TT=1, 15
      I=I+1
C################################# STATE 1 ########################################
      read(4,*)tdb1,W1
      call DA(tdb1,W1,twbi,twb1,tdp1,h1,Q1,d1,cp1)
C################################# STATE 11 ########################################
      tdb11=tdb1
      W11=W1
      twbi=30
      call DA(tdb11,W11,twbi,twb11,tdp11,h11,Q11,d11,cp11)
C################################# STATE 7 ########################################
      twbi =twb2
      call QD(Q7,tdb7,twbi,W7,twb7,tdp7,h7,d7,cp7)
      Q7A=Q7*100
C################################# STATE 6 ########################################
      mfresh=d1*(Vfresh/3600)
      h6a=h7-(Qtotal/mfresh)
      W7A=W7/1000
      AK=(1.006/2531.7)*1.805*(1-SHR)
      B1K=1.006*SHR+1.805*W7A*SHR
      B2K=(1.006/2531.7)*2501*(1-SHR)-(1.006/2531.7)*1.805*tdb7*(1-SHR)
      BK=B1K+B2K
      CK=-((1.006/2531.7)*2501*tdb7*(1-SHR)+h6a*SHR-W7A*SHR*2501)
      EK=BK**2-4*AK*CK
      if(EK.EQ.0) THEN
        X=-BK/(2*AK)
      elseif(EK.GT.0) THEN
        X1=(-BK+(SQRT(EK)))/(2*AK)
        X2=(-BK-(SQRT(EK)))/(2*AK)
        if(X1.GT.X2) THEN
          X3=X1
        endif
      elseif(EK.LT.0) THEN
      endif
```

```
      tdb6=X3
      W6=W7-(((1.006/2531.7)*(tdb7-tdb6)*(1/SHR-1))*1000)
      call DA(tdb6,W6,twbi,twb6,tdp6,h6,Q6,d6,cp6)
C############################## STATE 5 ########################################
      Qgain=(1-nfan)*Wfan1
      tdb5=tdb6-(Qgain/(cp6*mfresh))
      W5=W6
      twbi=twb1
      call DA(tdb5,W5,twbi,twb5,tdp5,h5,Q5,d5,cp5)
C############################### STATE 2A ######################################
      W2a=W6
      twb2a=twb1
      call AW(W2a,twb2a,tdb2a,tdp2a,h2a,Q2a,d2a,cp2a)
      WP=W1-W2a
C###################################################################################
C############### CALCULATION OF REACTIVATION TEMPERATURE #################
C###################################################################################
      C=0.65419*((W6)**(-0.099848))
      tdb14=114.78*((W1)**C)-149.77*ALOG(W1)
C############################### STATE 2 #######################################
      W2=W6
      F1=(5.82*((W1)**(-0.507))*(tdb14**0.652))
      F=F1/100
      twbi=twb1
      tdb2=((F*tdb1)-tdb2a)/(F-1)
      call DA(tdb2,W2,twbi,twb2,tdp2,h2,Q2,d2,cp2)

C############################### STATE 3 #######################################
      mreac=d11*(Vreac/3600)
      W3=W2
      Ch2=cp2*mfresh
      Ch11=cp11*mreac
      if(Ch2.LT.Ch11) then
      tdb3=tdb2-nheatex1*(Ch2/Ch2)*(tdb2-tdb11)
      else
      tdb3=tdb2-nheatex1*(Ch11/Ch2)*(tdb2-tdb11)
      endif
```

```
      twbi=twb2
      call DA(tdb3,W3,twbi,twb3,tdp3,h3,Q3,d3,cp3)

C################################ STATE 8 #######################################
      mwaste=d7*(Vwaste/3600)
      Qgain=(1-nfan)*Wfan2
      tdb8=(Qgain/(cp7*mwaste))+tdb7
      W8=W7
      twbi=twb1
      call DA(tdb8,W8,twbi,twb8,tdp8,h8,Q8,d8,cp8)
C################################ STATE 9 #######################################
      twb9=twb8
      tdb9=tdb8-nhum*(tdb8-twb8)
      call DW(tdb9,twb9,W9,tdp9,h9,Q9,d9,cp9)
C################################ STATE 4 #######################################
      W4=W3
      Ch3=cp3*mfresh
      Ch9=cp9*mwaste
      if(Ch3.LT.Ch9) then
      tdb4=tdb3-nheatex2*(Ch3/Ch3)*(tdb3-tdb9)
      else
      tdb4=tdb3-nheatex2*(Ch9/Ch3)*(tdb3-tdb9)
      endif
      twbi=twb3
      call DA(tdb4,W4,twbi,twb4,tdp4,h4,Q4,d4,cp4)
C################################ STATE 10 ######################################
      W10=W9
      if(Ch9.LT.Ch3) then
      tdb10=tdb9-nheatex2*(Ch9/Ch9)*(tdb9-tdb3)
      else
      tdb10=tdb9-nheatex2*(Ch3/Ch9)*(tdb9-tdb3)
      endif
      twbi=twb9
      call DA(tdb10,W10,twbi,twb10,tdp10,h10,Q10,d10,cp10)
C################################ STATE 12 ######################################
      W12=W11
      if(Ch11.LT.Ch2) then
```

```
      tdb12=tdb11-nheatex1*(Ch11/Ch11)*(tdb11-tdb2)
      else
      tdb12=tdb11-nheatex1*(Ch2/Ch11)*(tdb11-tdb2)
      endif
      twbi=twb11
      call DA(tdb12,W12,twbi,twb12,tdp12,h12,Q12,d12,cp12)
C################################# STATE 14 ##########################################
      W14=W11
      twbi=twb11
      call DA(tdb14,W14,twbi,twb14,tdp14,h14,Q14,d14,cp14)
C################################## STATE 15A #######################################
      W15a=W14+WP
      twb15a=twb14
      call AW(W15a,twb15a,tdb15a,tdp15a,h15a,Q15a,d15a,cp15a)
C################################## STATE 15 #########################################
      W15=W15a
      tdb15=((F*tdb14)-tdb15a)/(F-1)
      twbi=twb14
      call DA(tdb15,W15,twbi,twb15,tdp15,h15,Q15,d15,cp15)
C################################## STATE 13 #########################################
      W13=W12
      Ch12=mreac*cp12
      Ch15=mreac*cp15
      if(Ch12.LT.Ch15) then
      tdb13=tdb12-(nheatex5*(Ch12/Ch12)*(tdb12-tdb15))
      else
      tdb13=tdb12-(nheatex5*(Ch15/Ch12)*(tdb12-tdb15))
      endif
      twbi=twb12
      call DA(tdb13,W13,twbi,twb13,tdp13,h13,Q13,d13,cp13)
C################################## STATE 16 #########################################
      W16=W15
      if(Ch15.LT.Ch12) then
      tdb16=tdb15-nheatex5*(Ch15/Ch15)*(tdb15-tdb12)
      else
      tdb16=tdb15-nheatex5*(Ch12/Ch15)*(tdb15-tdb12)
      endif
```

```
      twbi=twb15
      call DA(tdb16,W16,twbi,twb16,tdp16,h16,Q16,d16,cp16)

C################################ STATE 17 ######################################
      Qgain3=(1-nfan)*Wfan3
      tdb17=(Qgain3/(cp16*mreac))+tdb16
      W17=W16
      twbi=twb1
      call DA(tdb17,W17,twbi,twb17,tdp17,h17,Q17,d17,cp17)
C## CALCULATION OF TRANSFERED SENSIBLE HEAT AT ROCESSES##
      Q23=mfresh*(h2-h3)
      Q34=mfresh*(h3-h4)
      Q45=mfresh*(h4-h5)
      Q910=mwaste*(h10-h9)
      Q1112=mreac*(h12-h11)
      Q1213=mreac*(h13-h12)
      Q1314=mreac*(h14-h13)
      Q1516=mreac*(h15-h16)
C##################################################################################
C############################## WRITING OF THE RESULTS #########################
C##################################################################################
      write(5,*)
      write(5,*)'Saat',I,':00'
      write(5,2)tdb1
2     format(7x,'Dry-Bulb Temp. of Fresh Air (C)                 :',F7.2)
      write(5,3),W1
3     format(7x,'Absolute Humidity of Fresh Air (gr/kg)     :',F7.2)
      write(5,4)Q23
4     format(7x,'Sensible Heat Transfer at Process 2-3 (kW)       :',F7.2)
      write(5,6)Q34
6     format(7x,'Sensible Heat Transfer at Process 3-4 (kW)       :',F7.2)
      write(5,7)Q45
7     format(7x,'Sensible Heat Transfer at Process 4-5 (kW)       :',F7.2)
      write(5,8)Q910
8     format(7x,'Sensible Heat Transfer at Process 9-10 (kW)      :',F7.2)
      write(5,9)Q1112
9     format(7x,'Sensible Heat Transfer at Process 11-12 (kW)     :',F7.2)
```

```
      write(5,11)Q1213
11    format(7x,'Sensible Heat Transfer at Process 12-13 (kW)        :',F7.2)
      write(5,13)Q1314
13    format(7x,'Sensible Heat Transfer at Process 13-14 (kW)
     &  :',F7.2)
      write(5,16)Q1516
16    format(7x,'Sensible Heat Transfer at Process 15-16 (kW)        :',F7.2)
      S=1
      write(5,*)
      write(5,*) 'State    Td      W      Tw      RH      Tdp
     &    H       D'
      write(5,*)
      write(5,10)S,tdb1,W1,twb1,Q1,tdp1,h1,d1
      write(5,10)S+1,tdb2,W2,twb2,Q2,tdp2,h2,d2
       write(5,10)S+2,tdb3,W3,twb3,Q3,tdp3,h3,d3
      write(5,10)S+3,tdb4,W4,twb4,Q4,tdp4,h4,d4
      write(5,10)S+4,tdb5,W5,twb5,Q5,tdp5,h5,d5
       write(5,10)S+5,tdb6,W6,twb6,Q6,tdp6,h6,d6
      write(5,10)S+6,tdb7,W7,twb7,Q7A,tdp7,h7,d7
      write(5,10)S+7,tdb8,W8,twb8,Q8,tdp8,h8,d8
       write(5,10)S+8,tdb9,W9,twb9,Q9,tdp9,h9,d9
      write(5,10)S+9,tdb10,W10,twb10,Q10,tdp10,h10,d10
      write(5,10)S+10,tdb11,W11,twb11,Q11,tdp11,h11,d11
       write(5,10)S+11,tdb12,W12,twb12,Q12,tdp12,h12,d12
       write(5,10)S+12,tdb13,W13,twb13,Q13,tdp13,h13,d13
       write(5,10)S+13,tdb14,W14,twb14,Q14,tdp14,h14,d14
       write(5,10)S+14,tdb15,W15,twb15,Q15,tdp15,h15,d15
      write(5,10)S+15,tdb16,W16,twb16,Q16,tdp16,h16,d16
       write(5,10)S+16,tdb17,W17,twb17,Q17,tdp17,h17,d17
10    format(x,F4.1,5x,F7.2,3x,F7.2,3x,F6.2,3x,F7.2,3x,F6.2,5x,F7.2,3x,F
     &7.2)
12    continue
      end
C###############################################################################
C################################ SUBROUTINES ################################
C###############################################################################
      subroutine DW(tdb,twb,AP,tdp,h,Q,d,cp)
```

```
        real,parameter::PATM=101.325
        real,parameter::C8=-5800.2206
        real,parameter::C9=-5.516256
        real,parameter::C10=-0.048640239
        real,parameter::C11=0.000041764768
        real,parameter::C12=-0.000000014452093
        real,parameter::C13=6.5459673
        real,parameter::C14=6.54
        real,parameter::C15=14.526
        real,parameter::C16=0.7389
        real,parameter::C17=0.09486
        real,parameter::C18=0.4569
        TdbK=tdb+273.15
        TwbK=twb+273.15
        Pwsu1=((C8/TwbK)+C9+(C10*TwbK)+C11*(TwbK**2))
        Pwsu2=((C12*TwbK**3)+(C13*ALOG(TwbK)))
        Pwsu=EXP(Pwsu1+Pwsu2)
        Wsu=0.62198*(Pwsu/(PATM-Pwsu))
        W=((2501-2.381*twb)*Wsu-(tdb-twb))/(2501+1.805*tdb-4.186*twb)
        AP=W*1000
        Pw=(PATM*W)/(0.62198+W)
        Z=ALOG(Pw)
        tdp=C14+C15*Z+C16*(Z**2)+C17*(Z**3)+C18*(Pw**0.1984)
        h=1.006*tdb+W*(2501+(1.805*tdb))
        Pws1=((C8/TdbK)+C9+(C10*TdbK)+C11*(TdbK**2))
        Pws2=(C12*(TdbK**3)+(C13*ALOG(TdbK)))
        Pws=EXP(Pws1+Pws2)
        Q=(Pw/Pws)*100
        d=((1+W)*PATM)/(0.2871*(tdb+273.15)*(1+1.6078*W))
        cp=1.005+0.006*(TdbK/100)**1.73
        return
        end
C##################################################################################
        subroutine DA(tdb,Y,twbi,twb,tdp,h,Q,d,cp)
        W=Y/1000
        twb=twbi
        TwbK=273.15+twb
```

```
      itr=0
20    continue
      itr=itr+1
      TwbK=twb+273.15
      Pwsu1=(C8/TwbK)+(C9+C10*TwbK)+(C11*(TwbK**2))
      Pwsu2=(C12*TwbK**3)+(C13*ALOG(TwbK))
      Pwsu=EXP(Pwsu1+Pwsu2)
      Wsu=0.62198*(Pwsu/(PATM-Pwsu))
      twby=(tdb+W*(2501+1.805*tdb)-2501*Wsu)/(4.186*W-2.381*Wsu+1)
      bhx=abs((twby-twb)/twby)
      if (bhx.LT.1E-4) GOTO 30
      twb=(0.9*twb+0.1*twby)
      if (itr>100) go to 30
      goto 20
30    continue
      twb=twby
      TdbK=tdb+273.15
      Pw=(PATM*W)/(0.62198+W)
      Z=ALOG(Pw)
      tdp=C14+C15*Z+C16*(Z**2)+C17*(Z**3)+C18*(Pw**0.1984)
      h=1.006*tdb+W*(2501+(1.805*tdb))
      Pws1=((C8/TdbK)+C9+(C10*TdbK)+C11*(TdbK**2))
      Pws2=(C12*(TdbK**3)+(C13*ALOG(TdbK)))
      Pws=EXP(Pws1+Pws2)
      Q=(Pw/Pws)*100
      d=((1+W)*PATM)/(0.2871*(tdb+273.15)*(1+1.6078*W))
      cp=1.005+0.006*(TdbK/100)**1.73
      return
      end
C#################################################################################
      subroutine AW(AG,twba,tdba,tdpa,ha,Qa,daa,cpa)
      Wa=AG/1000
      TwbKa=twba+273.15
      Pwsu1=(C8/TwbKa)+(C9+(C10*TwbKa))+(C11*(TwbKa**2))
      Pwsu2=((C12*(TwbKa**3))+(C13*ALOG(TwbKa)))
      Pwsu=EXP(Pwsu1+Pwsu2)
      Wsua=0.62198*(Pwsu/(PATM-Pwsu))
```

```
      A=(Wsua*(2501-(2.381*twba))+twba*(1+(4.186*Wa))-(2501*Wa))
      B=(1+(1.805*Wa))
      tdba=A/B
      Pw=(PATM*Wa)/(0.62198+Wa)
      Z=ALOG(Pw)
      tdpa=C14+C15*Z+C16*(Z**2)+C17*(Z**3)+C18*(Pw**0.1984)
      ha=1.006*tdba+Wa*(2501+(1.805*tdba))
      TdbKa=tdba+273.15
      Pws1=((C8/TdbKa)+C9+(C10*TdbKa)+C11*(TdbKa**2))
      Pws2=(C12*(TdbKa**3)+(C13*ALOG(TdbKa)))
      Pws=EXP(Pws1+Pws2)
      Qa=(Pw/Pws)*100
      da=((1+Wa)*PATM)/(0.2871*(tdba+273.15)*(1+1.6078*Wa))
      cpa=1.005+0.006*(TdbKa/100)**1.73
      return
      end
C#######################################################################################
      subroutine QD(Q,tdb,twbi,AT,twb,tdp,h,d,cp)
      TdbK=tdb+273.15
      Pws1=((C8/TdbK)+C9+(C10*TdbK)+C11*(TdbK**2))
      Pws2=(C12*(TdbK**3)+(C13*ALOG(TdbK)))
      Pws=EXP(Pws1+Pws2)
      W=(Q*Pws*0.62198)/(PATM-(Q*Pws))
      AT=W*1000
      h=1.006*tdb+W*(2501+(1.805*tdb))
      Pw=(PATM*W)/(0.62198+W)
      Z=ALOG(Pw)
      tdp=C14+C15*Z+C16*(Z**2)+C17*(Z**3)+C18*(Pw**0.1984)
      twb=twbi
      TwbK=273.15+twb
      itr=0
110   continue
      itr=itr+1
      TwbK=twb+273.15
      Pwsu1=((C8/TwbK)+C9+(C10*TwbK)+(C11*(TwbK**2)))
      Pwsu2=(C12*(TwbK**3)+(C13*(ALOG(TwbK))))
      Pwsu=EXP(Pwsu1+Pwsu2)
```

```
      Wsu=0.62198*(Pwsu/(PATM-Pwsu))
      A7=(tdb+W*(2501+1.805*tdb)-2501*Wsu)
      B7=(4.186*W-2.381*Wsu+1)
      twby=A7/B7
      bhxd=abs((twby-twb)/twby)
      if (bhxd.LT.1E-4) GOTO 120
      twb=(0.9*twb+0.1*twby)
      if (itr>100) go to 120
      goto 110
120   continue
      twb=twby
      d=((1+W)*PATM)/(0.2871*(tdb+273.15)*(1+1.6078*W))
      cp=1.005+0.006*(TdbK/100)**1.73
      return
      end
```