MODELLING AND TRANSIENT ANALYSIS OF A HYBRID LIQUID DESICCANT COOLING SYSTEM

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ARASH KARSHENASS

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MODELLING AND TRANSIENT ANALYSIS OF A HYBRID LIQUID DESICCANT COOLING SYSTEM

Submitted by **ARASH KARSHENASS** in partial fulfillment of the requirements for the degree of **Master of Science in Mechanical Engineering Department, Middle East Technical University** by,

Prof. Dr. Canan ÖZGEN
Dean, Graduate School of **Natural and Applied Science** ____________________

Prof. Dr. Süha ORAL
Head of Department, **Mechanical Engineering** ____________________

Assoc. Prof. Dr. Cemil Yamalı
Supervisor, **Mechanical Engineering Dept., METU** ____________________

Assoc. Prof. Dr. Derek K. Baker
Co-Supervisor, **Mechanical Engineering Dept., METU** ____________________

Examining Committee Members:

Assistant Prof. Dr. Tahsin Çetinkaya
Mechanical Engineering Dept., METU ____________________

Assoc. Prof. Dr. Cemil Yamalı
Mechanical Engineering Dept., METU ____________________

Assoc. Prof. Dr. Derek K. Baker
Mechanical Engineering Dept., METU ____________________

Assoc. Prof. Dr. İlker Tarı
Mechanical Engineering Dept., METU ____________________

Prof. Dr. Mecit Siviroğlu
Mechanical Engineering Dept., Gazi University ____________________

Date: 03/09/2014

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Name, Last Name: ARASH KARSHENASS

Signature :

ABSTRACT

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KARSHENASS, ARASH
M.S., Department of Mechanical Engineering
Supervisor: Assoc. Prof. Dr. Cemil Yamalı
Co-Supervisor: Assoc. Prof. Dr. Derek K. Baker

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Desiccant Cooling Systems (DCS) are considered as an alternative method for conventional vapor compression cooling systems (VCCS) or at least a complimentary component to them. In conventional VCCS inlet air is cooled down to blow its dew point for dehumidification and then is reheated again to obtain air flow with desired temperature and humidity, and consequently inefficient consumption of energy. In DCS, dehumidification of air is done by utilizing of desiccant material to get desirable humidity and then dry air is cooled by evaporation method or cooling coils down to suiTABLE temperature. This thesis presents a study of the feasibility of a hybrid liquid desiccant cooling system with Lithium Chloride as the desiccant material. Mathematical models of desiccant contactors are adopted from the literature. The whole system is modeled in the TRNSYS platform and is simulated using Typical Metrological Year data. The building model has developed in accordance with the building construction and operation. Simulations are performed over the summer period of the year and the results are compared to the results assuming a VCCS from system characteristics and energy saving points of view. One of the most important outputs is that, DCS has to be investigated in a transient manner rather than steady state conditions. The results also indicate that consumed energy in both systems are approximately equal in magnitudes but different in type; DCS shifts required energy from electricity to thermal energy. Also for large supply-air flow rate applications, DCS would be more beneficial than VCCS. In addition, DCS provides suiTABLE conditions in lower supply-air flow rates in which VCCS cannot. This is due to low humidity ratio requirement of supply air which cannot be accessed in typical VCCS because of temperature limit of chiller.

Keywords: Liquid Desiccant Cooling; Hybrid Desiccant; Solar Cooling; Energy Saving, TRNSYS

ÖZ

HİBRİT SIVI DESICCANT SOĞUTMA SİSTEM MODELLEMESİ VE GEÇİCİ ANALİZİ

KARSHENASS, ARASH
Yüksek Lisans, Makine Mühendisliği Bölümü
Tez Yöneticisi: Doç. Dr. Cemil YAMALI
Ortak Tez Yöneticisi: Doç. Dr. Derek K. BAKER

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Desiccant Soğutma Sistemi (DSS), geleneksel Buhar Kompresyon Soğutma Sistemlerine (BKSS) alternatif bir yöntem veya tamamlayıcı bir bileşen olarak değerlendirilmektedir. Geleneksel BKSS' de, giriş havasının nemini almak için sıcaklığı çiy noktasına kadar düşürülmekte ve sonradan istenilen sıcaklık ve neme sahip hava akışı elde etmek için tekrar ısıtılmaktadır. Bu işlem enerjinin verimsiz bir şekilde tüketilmesine yol açmaktadır. DSS'de ise istenilen nem oranına ulaşmak için havanın nemi desiccant madde kullanılarak alınmakta ve daha sonra kuru hava buharlaşma yöntemi veya soğutma bobinleri kullanılarak metotla uygun sıcaklığa kadar soğutulmaktadır. Bu çalışmada Desiccant madde olarak Lityum Klorür seçilen hibrit bir sıvı desiccant soğutma sisteminin uygunluğu sunulmuştur. Desiccant kontaktörlerin matematik modelleri literatürden uyarlanmıştır. Bütün sistem TRNSYS platformunda modellenmiş ve tipik yıllık meteoroloji verileri kullanılarak simüle edilmiştir. Binanın modeli, operasyon ve inşaatına göre çıkarılmıştır. Simülasyonlar yaz dönemi için yapılmıştır ve sonuçlar BKSS ile sistem karakteristikleri ve enerji tasarrufu açısından karşılaştırılmıştır. Elde edilen en önemli sonuçlardan birisi, DSS'nin kararlı durum analizinden ziyade geçici durum analizlerinin yapılmasının gerekli olduğudur. Ayrıca sonuçlara göre her iki sistemde de harcanılan enerji miktarı yaklaşık olarak aynıdır ancak kullanılan enerji türü farklıdır. DSS ihtiyaç duyulan enerjiyi elektrikten enerjisinden ısı enerjisine dönüştürerek elde etmektedir. Ayrıca yüksek besleme hava akış oranına sahip uygulamalar için DSS, BKSS’ den daha yararlı olabilmektedir. Ek olarak, düşük besleme hava akış oranlarında DSS uygun koşullar sağlayabilirken BKSS sağlayamamaktadır. Bunun sebebi ise geleneksel BKSS deki soğutucuların sıcaklık limitlerinden dolayı düşük nem oranına sahip giriş havası gereksiniminin sağlanamamasıdır.

Anahtar kelimeler: .Sıvı Desiccant Soğutma; Hibrit Desiccant; Solar Soğutma; Enerji Tasarrufu, TRNSYS

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Nomenclature

$\dot{M}$: mass flow rate (kg hr^{-1})

$\dot{m}$: mass flow rate (kg s^{-1})

$\dot{V}$: Volumetric flow rate (m^3 hr^{-1})

$\dot{v}$: Volumetric flow rate (m^3 s^{-1})

A: air superficial flow rate (kg m^{-2} hr^{-1})

a: packing specific area (m^2 m^{-3})

ACH: air change per hour (h^{-1})

atm: atmosphere

C: Capacitance

C_p: specific heat at constant pressure (kJ kg^{-1} K^{-1})

f: factor

g: humidity ratio (kg$_{water}$ kg$_{air}^{-1}$)

H: enthalpy (kJ)

h: specific enthalpy (kJ kg^{-1})

h_{fg}: latent heat of evaporation (kJ kg^{-1})

L: solution superficial flow rate (kg m^{-2} hr^{-1})

P: Pressure (Pa or kPa)

Q: heat input (kJ hr^{-1})

q: cooling load (kW)

T: temperature (K)

t: temperature (^{o}C)

V: volume (m^3)

W: Power (kJ hr^{-1})

X: solution concentration (-)

Z: dehumidifier height (m)

Greek Letters

ε: effectiveness (-)

β: regenerator temperature difference ratio (-)

ρ: Density (kg m^{-3})

γ: solution surface tension (N m^{-1})

η: efficiency

τ: time step

Subscripts

A: air

avg: average

b: solution from regenerator to dehumidifier

build: building

cond: condensation

cs: conditioned space

deh: dehumidifier

equ: equivalent

evap: evaporation

f: fresh air

H: enthalpy

i: inlet

L: solution

lat: latent

o: outlet

r: regenerating solution

rc: recirculation air

reg: regenerator

s: supply air

sen: sensible

stn: standard value

u: reusing solution

v: ventilation air

w: water

CHAPTER 1

INTRODUCTION

1.1. Motivation

Energy in general form plays a very important role at life of human of today and would have more significance because of more demand of modern life. Currently nearly all of the harvested energy by human kind is obtained by converting stored energy in carbon-based fossil fuels, Figure 1-1, which have several drawbacks such as limitation, emission of harmful gasses and global warming.

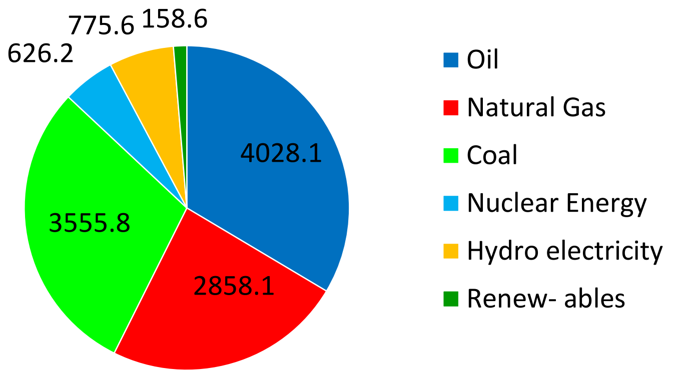


Figure 1-1Energy consumption in 2010 at entire world in million tons of oil equivalents [1]

64% and 53% of energy utilization in residential and commercial part at U.S respectively is related to building energy systems including heating, cooling, lightning and water heating in 2010 [2] and detailed information is provided in Table 1-1.

Table 1-1 1Percentage of energy usage of residential and commercial sectors in buildings at U.S in 2010 [2]

	Space Heating	Space Cooling	Water Heating	Ventilation	Lighting
Residential	25	16	13	Not Available	10
Commercial	13	10	4	9	17

Cooling is one of the Indispensable demands of buildings consuming numerous units of energy, soaring as ambient temperature and humidity go up. Refrigeration systems can be classified in three main categories according to the final energy used to operate them: electrical systems (high grade energy), thermal systems (low grade energy) and hybrid systems, particularly thermal and electrical. Among these, thermal systems have been receiving higher interest at recent years because of lower effect on global warming and ozone layer depletion [3]. Thermally operated class includes several systems but the most prominent ones are absorption, adsorption, desiccant and metal hybrid cooling systems. Absorption and adsorption systems are thermodynamic cycles using absorbents and adsorbents materials to run the system. While the cycle of two systems are completely different, in both of them water vapor is sorbet by sorbents. Although absorption systems operate at higher COP, the operating temperature of adsorption systems is lower so more opportunity for waste heat usage. Desiccant system is open cycle which reduces the humidity of air by desiccant materials, thus requires lower energy thanks to the fact that no mechanical condensation is required. Metal hybrid systems which is based on hydrogen sorption/desorption, has capability of producing very low temperatures up to -30 ^{o}C [3].

Among these technologies, recently developed DCS is receiving higher attraction by both researchers and companies and many authors consider it as the alternative method to conventional VCCS or at least a supplementary to it. In fact, desiccants have been utilized for dehumidification at agricultural and industrial processes such as textile production and drying of crops and now its applicability in HVAC industry is being proved thanks to the ability of promising control of humidity but no official published guideline exist for the design and operation of such systems.

1.2. Principle of the desiccant cooling system

Desiccants are one subset of sorbents which attract and hold water vapor between 10% and 1100% of their dry weight [4], called desiccant capacity depending on its type and conditions. Desiccants absorb/adsorb moist because of difference in water vapor pressure at surrounding and surface of desiccant and in fact this difference in vapor pressures is the driving force for the mass transfer. Desiccant material, solid or liquid, attract moisture when the vapor pressure at its surface is lower than that of surrounding air until reaching equilibrium point while some heat is released during the process and is equal to the latent heat of condensation of water plus an additional heat of dilution. The vapor pressure of air is a function of its temperature only but for desiccant it depends on temperature and concentration. Thus for a desiccant material, capacity and vapor pressure are the two most important parameters. Figure 1-2 shows the effect of temperature and concentration on vapor pressure of LiCl as a liquid desiccant and Figure 1-3 displays the capacity of some desiccants for different ambient humidity.

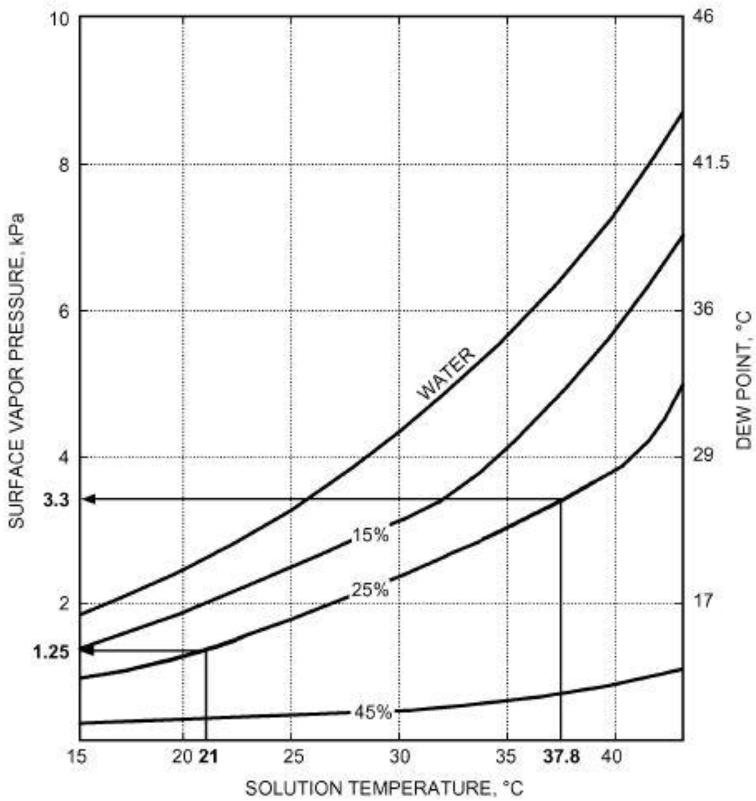


Figure 1-2 Effect of Temperature and Concentration on vapor pressure of LiCl [4]

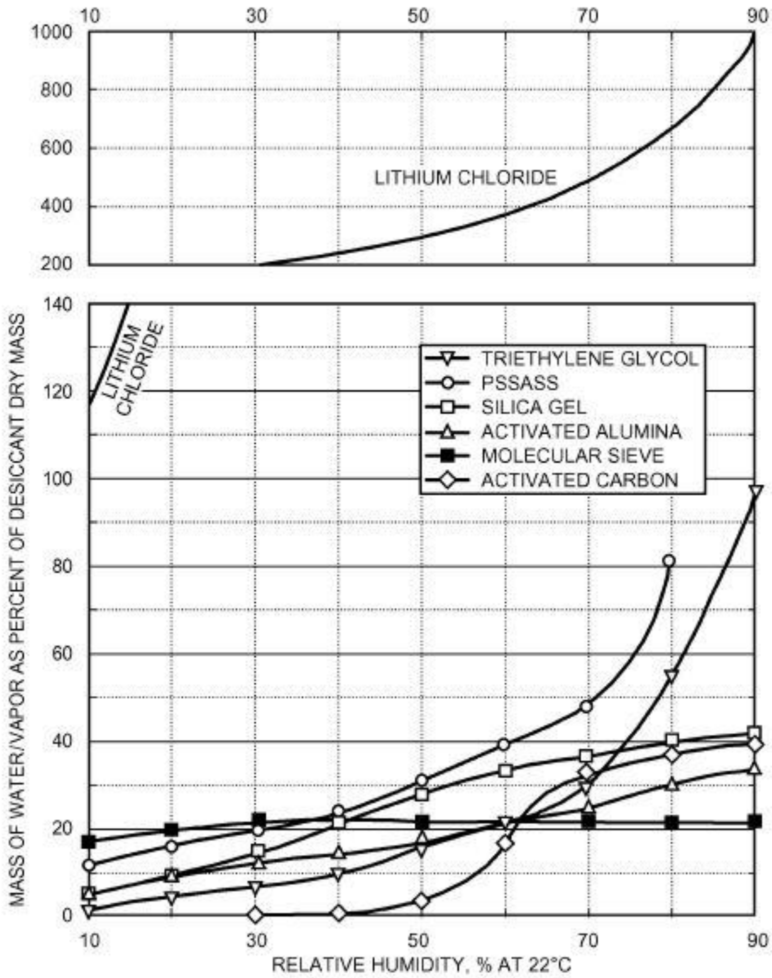


Figure 1-3 Effect of surrounding humidity on capacity of several industrial desiccants [4]

To put a comment, lower vapor pressure results in more and faster sorption process or in other words producing air with lower humidity while higher capacity means ability to hold more moisture at the same condition. Higher concentration and lower temperature results in lower vapor pressure of desiccant surface, hence by increasing its temperature, attracted moisture is released, named reactivation or regeneration of weak desiccant which needs thermal energy. The sorption process is done by either adsorption or absorption. Absorption is a process in which natures of materials are changed, either physically or chemically and are usually liquid. On the contrary, in adsorption process, desiccant material is usually solid and the nature of that does not changed such as sponge soaking up water. Usually liquid desiccant (with absorption behavior) have higher capacity and lower vapor pressure compared to solid ones (with adsorption behavior) thus removes moisture significantly but the regeneration temperature of these type is very high cease utilizing them in this way. To overcome this problem, liquid desiccants are normally used in desiccant-water solution to reduce its concentration between 40 to 60%. Even though vapor pressure increases and capacity decreases in this way, but regeneration temperature also drops considerably due to the fact that as the concentration of desiccant goes up, regeneration temperature to bring the weak desiccant to the original state soars. This occurrence is further described in liquid contactor part of literature review section. Desiccant systems could be combined with heating systems such as heat pumps, different cooling systems and also with combined cooling, heating and power (CCHP) systems for energy saving purposes.

Desiccant cooling systems, firstly developed in Sweden, are novel heat driven open cycle affording an opportunity to utilize heat which might otherwise be wasted and consists in dehumidifying inlet air (process air) by brought it to contact with desiccant materials and then cooling it down, by evaporation or sensible cooling, to desire temperature. In order to make the system working continuously, attracted vapor must be driven out of desiccant material, called regeneration (reactivation), and it is done by increasing material temperature as mentioned before and in order to be prepared for the new cycle, hot-strong mean must be cooled down in order to be able to attract moisture again. The regeneration energy for heating is equal to the sum of: 1) Required energy to rise the temperature of desiccant to the desired magnitude 2) Required energy to vaporize the moisture 3) Required energy for water desorption from desiccant. Thus whole the system can be divided into three main blocks, namely dehumidifier, regenerator and cooling unit. The schematic working principle of basic desiccant cooling system is shown in Figure 1-4. At state 1, outside air (process air) goes through desiccant wheel called core of the system where air humidity drops significantly and temperature goes up. The air after passing the desiccant material is dried and also hot due to the conversion of air latent energy to sensible energy. Following that, at state 2, warm and dry air passes the heat exchanger where air enthalpy drops due to reduction of temperature while humidity ratio remains constant (sensible cooling). After, air moves toward an evaporative cooler in which water will be sprayed on the air to decrease and increase its temperature and humidity respectively to an appropriate point. Note that the last process is isenthalpic, where total enthalpy remains constant, only air load changes from sensible to the latent type. The supply air goes into conditioned space and counterbalance sensible load which is due to heat gains and latent loads of space, appeared through mass transfer by infiltration and occupants, and finally leaves the space as return air. The return air is used to achieve two goals, as the cooler mean of the heat exchanger and also for regeneration of diluted desiccant material.

The return air is used to achieve two goals, as the cooler mean of the heat exchanger and also for regeneration of diluted desiccant material. To fulfill that, it passes through the second evaporator cooler at first step to be cooled down close to dew point temperature. After going through heat exchanger and cooling the process air, it is heated at heater and finally, hot air moves toward the other side of desiccant wheel where diluted desiccant is located in order to be regenerated and finally hot and humid air leaves the system to atmosphere. Note that while the humidity ratio of air for regeneration air is high because of evaporative cooling at the previous step, but its impact on regeneration process is very small. Several other systems could be made with different configurations and additional components based on this system.

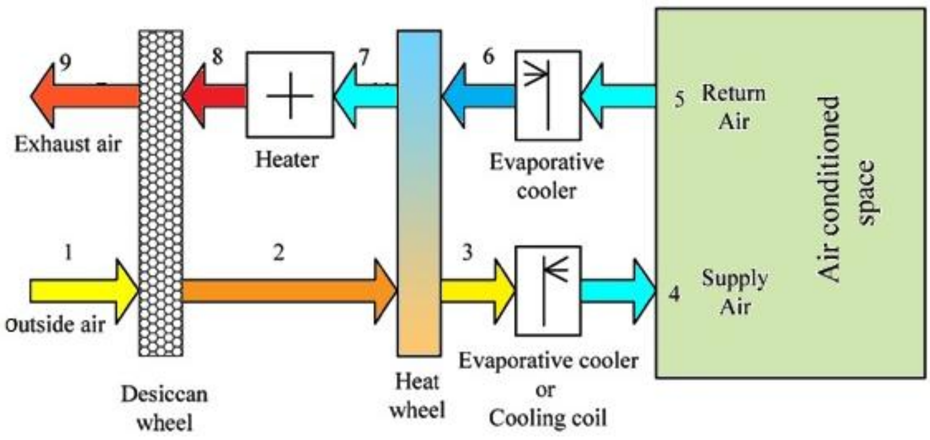


Figure 1-4 Schematic Diagram of DCS Principle

There are two types of loads that an air-conditioning system has to meet and sum of these two is the total load. Sensible load which is the result of heat transfer to space from walls, windows, roofs and etc. and latent load which is the result of generated moisture at space and mass transfer through infiltrations but it is important to say that the latent and sensible load of supply air is not accounted as space load but it can be considered as “Supply air load”. In fact a cooling system at the first step is responsible for removing supply air load and preparing an air flow at standard condition of room, e.g. 25^{o}C and 10 gr/kg of humidity ratio and after that make these magnitudes much lower to overcome the loads of space. Latent load ration (LLR) is defined as the latent load of supply air divided by total load of that and likewise sensible load ratio (SLR) is defined, Eq. (1-1) and Eq. (1-2).

$$LLR = \frac{Latent\ load\ of\ supply\ air}{Total\ Load\ of\ supply\ air} \qquad (1-1)$$

$$SLR = \frac{Sensible\ load\ of\ supply\ air}{Total\ load\ of\ supply\ air} \qquad (1-2)$$

In conventional VCCS or none conventional sorption cooling systems (VSS) the process air has to be cooled down below its dew point to remove latent load by condensation of some water of air and then be reheated again to meet suitable temperature because the air temperature after being dehumidified is too low, resulting in high energy consumption in full-load hours. Consequently as the humidity of ambient air increases (LLR increment), required energy for condensation goes up slightly while energy demand for reheating soars and system acts inefficiently. Also more severe problem appears with such systems during part-load time which occupies more than 90% of cooling period time where sensible load of space is dropped but latent load remains constant. There are two possibilities with system behavior in these condition. At first type, when controlling of cooling coils are done by a single thermostat in conditioned space and system operates in part-load mode. At this condition temperature of cooling coils goes up and resulting in suitable temperature of space but high humidity of that and in tropical climates it is followed by uncomfortable feeling or even condensation of water at room on walls. In the second type, system operation is similar to full-load time and suitable condensation occurs at cooling coils but the thermostat is only used for controlling of reheat process resulting in too much energy consumption. In fact as sensible load drops, more energy is used by system at this mode due to more demand of reheat. To sum up, at this mode cooling coils operation depends on latent load and since latent load variation is small, cooling coils behavior remains the same all the time and it cease utilization of humidistat but it is manful only when latent load variation is significant. The thermostat in space controls the temperature of air leaving reheater to avoid deep cooling of space.

But one of the interesting benefits of DCS is that chemical condensation occurs without any dependence on temperature and following that no need for reheating, thus lower energy demand especially at high LLR compared to any other systems.

Secondly, there is no need to a high-temperature thermal sources (high exergy) to operate the system, thus system could be run by low thermal sources such as waste heat or solar energy. Fortunately, solar radiation is in the same phase with sensible cooling demand, meaning as the ambient temperature goes up, more energy from sun is available, consequently cleaner energy for required refrigeration. And the third prominent benefit is humidity and temperature independent control of the system while in others dehumidification of air is a function of air temperature. In fact in a desiccant cooling system, the role of cooling unit is handling of sensible load while the dehumidifier removes the latent load. Although DCS could be used anywhere, it is highly beneficial when [5]:

- LLR>=30%
- Energy cost to regenerate the desiccant is low
- Tight Control Over Moisture is required
- Occupant Comfort cannot be Compromised

Generally, pros and cons of the system could be summarized as:

Advantages:

- Lower Electrical Energy Consumption Capability
- High indoor quality, moisture damage control and microbial growth
- CFC Free
- Practicable at all climate conditions
- Utilizing thermal energy instead of electricity
- Capability for using low-temperate thermal sources such as solar or waste heat

Disadvantages**:**

- As ambient humidity goes up, system COP drops
- More difficult installation
- Higher Initial Cost
- Possibility of entering of harmful bacteria through ejecting of water at evaporator

For more information on liquid dehumidifier systems and types, readers can refer to TYPE OF DESICCANT COOLING SYSTEMS.

CHAPTER 2

LITERATURE REVIEW

Literature related to DCS can be divided into three major groups also. The first group focuses on the dehumidifier and regenerator (contactor) design and analysis as the core of system. The second group is those investigating on desiccant materials for development and the third group is literatures related with whole desiccant cooling system instead of only specific section like first group.

2.1. Contactors

Starting with the first group, Three models for analyses of contactor exists, namely, computational fluid dynamic (CFD), NTU and fitted algebraic equations model with CFD as the most accurate and complex one [6]. While the two first methods are mathematical modeling of contactors, utilized heat and mass transfer coefficients (h and h_m) in equations are adopted from experimental investigations which depends on geometry and material of packing next to desiccant mean. Main steps toward mathematical modeling are as follow [7]:

- Selecting appropriate control volume and assumptions due to complexity of existed heat and mass transfer.
- Deriving differential equation based on mass and energy balance.
- Providing appropriate auxiliary condition as boundary, initial and equilibrium conditions.
- Solve the equations

Following the method, several models were developed mathematically both for solid and liquid means, validated by other experimental or mathematical models to be used for optimization and or parametric investigations. Analysis of a finite difference model is very complex and iterative numerical solutions are required preventing them from adoption in transient conditions. To overcome this problem, algebraic methods can be derived which are simple equations used for quick prediction of effectiveness (ε) with the aim of determining air outlet properties using Eq. (2-1) and Eq. (2-2). Note Eq. (3) and Eq. (4) explicitly show that the contactor effectiveness is a function of the humidity ratio (g) and enthalpy (h) of the air (subscript A) and solution (subscript L) streams. Using this set of equations, air outlet humidity ratio and enthalpy is obtained if dehumidification and enthalpy effectiveness of dehumidifier are known and following them, the air outlet temperature. While accuracy of the method is slightly lower compared to finite difference, it is still acceptable for some applications such as air-conditioning and in fact their simplicity is beneficial as it enables hourly system analysis in a reasonable amount of time. Algebraic equations are obtained through curve fitting of available input and output data of dehumidifiers and regenerators, either using experimental tests or finite difference results.

$$\varepsilon_g = \frac{g_{A,i} - g_{A,o}}{g_{A,i} - g_{L,i}} \qquad (2-1)$$

$$\varepsilon_H = \frac{h_{A,i} - h_{A,o}}{h_{A,i} - h_{L,i}} \qquad (2-2)$$

2.1.1. Solid Contactors

Lots of works are done related with solid-based contactors, especially around rotary wheel system. Charoensupaya and Worek [8] described a mathematical model of adiabatic desiccant wheel and done a parametric study on dehumidifier length and mass function for optimization. Zhang and Dai [9] analyzed temperature and humidity profiles in wheel during dehumidification and regeneration using validated one-dimensional honeycombed desiccant wheel model. Nia et al. [10] presents the modeling of an adiabatic rotary desiccant wheel in MATLAB Simulink to predict temperature and humidity of outlet air and evaluation of system performance, also optimal rotational speed is determined and examined at different conditions. Also some other investigations were done numerically e.g. Zheng and Worek. [11] and Mihajlo et al. [12] used finite-difference method to model desiccant wheel considering condensation in regeneration portion of desiccant wheel due to the fact that at high pressures, condensations occurs at regeneration side [7] and occupy as much as 40% of regeneration section of the wheel in which regeneration of the desiccant is not possible [12]. Angrisani et al. [13] investigated rotary desiccant wheel experimentally and performance parameters have been evaluated as a function of the regeneration temperature, the inlet process air humidity ratio and temperature and flow rate in both the cases of fixed regeneration temperature and fixed regeneration thermal power. One of the most prominent problems in DCS is the operation of dehumidifier in hot and humid climates or deep dehumidification with low regeneration temperature. Following this demand, Haun et al. [14] investigate a modified model of rotary wheel mathematically and shows that by using a secondary contactor instead of one, regeneration temperature drops significantly, at the proposed model it reduces by 24º C from 83º C, because If the process air flow alternatively over infinite desiccant wheels and inter coolers, the thermodynamics of process air would be isothermal. In other word, other conditions being equal, the regeneration temperature of an ideal, infinite multistage desiccant cooling system is the minimum. Also it reveals that COP cannot be reasonable index of performance for comparison. Ando et al. [15] and Ge et al [16] investigate two-stage wheel and demonstrate that system adsorbs more vapor content with lower regeneration temperature in higher COP. Another type of solid desiccant dehumidifier is fixed-bed design in which no moving part exists, also unlike the previous design inner cooling is available during dehumidification resulting in higher sorption performance but incapable of continues dehumidification. Desiccant Coated Cross Cooled Compact Dehumidifier (DCCCCD) is modeled mathematically by Yuan et al. [17] and is compared to Desiccant Packed Cross Cooled Compact Dehumidifier (DPCCCD) which is a specific type of fixed-bed dehumidifier. The only difffrence between DCCCCD and DPCCCD is that the latter, disiccant particels packe drandomly instead of coated. The results show DPCCCD is better when process air flow rate is higher than 180 kg/h and its pressure drop is seven times larger than DCCCCD.

2.1.2. Liquid Contactors

Perused by many authors, several investigations are available for liquid contactors too. Falling film contactors are studied widely and spray towers are of the early studies of liquid contactors, but recently numerous designs are done on packed bed tower with different liquid desiccant materials. Being compact is the most prominent advantage of this design over other ones.

Packed bed dehumidifiers have the most application due to its effectiveness and reliability. So many authors work on modeling of adiabatic and non-adiabatic packed bed dehumidifiers and regenerators mathematically and experimentally.

2.1.2.1. Adiabatic

2.1.2.1.1 Experimental

[19]Jain et al. [18] experimentally investigated the performance of dehumidification and regeneration of LiCl and LiBr in cross-flow structured packed bed contactor with a pre-cooler for cooling strong solution. Studies were carried over a range of desiccant flow rates, concentrations, temperatures and ambient conditions. Change in specific humidity was in the range of 0.6–1.77 gr/kg and 6.37-5.86 gr/kg for LiBr and CaCl respectively. Also dehumidification effectiveness of LiBr was found to range between 0.25 and 0.44, while that of the LiCl was between 0.36 and 0.45. Chung et al. [19] did experimental investigation on random packed bed dehumidifier and regenerator with LiCl in the concentration range of 30% to 40%. Bed was designed with 5/8 inch polypropylene rings with specific area of 342 m^2/m^3. In addition to obtaining heat and mass transfer coefficient of the bed, they found that required flow rate of solution for completely wet packing is about 800 times higher than required flow rate for dehumidification of air according to equilibrium conditions of air and solution. In other work, Chung and Ghosh [20] did experimental investigation on cross corrugated cellulose and PVC structured packed bed dehumidifier and compared the results to the data from previous work [19] next to obtaining heat and mass transfer coefficients of the beds. Data showed that air-side pressure drop and heat transfer coefficient for random packed bed is lower and higher respectively compared to structured bed. However, mass transfer coefficients of random packed beds was higher in some air flow rate ranges and lower in some other ranges. Chung and Ghosh [20] investigated poly propylene Pall rings as random packed bed with 37.5 - 42.5% LiCl solution as absorbent to obtain mass transfer coefficient experimentally, resulted in the fact that mass transfer coefficient strongly relies on solution temperature and concentration. Elsarrag [21] did

experimental investigations on structured dehumidifier with cellulose rigid media pads as the structured packing and TEG as the solution for parametric study. It was found that high liquid flow rates, (mL/ma= 1.1–1.6) do not have a significant effect on the performance variables.

2.1.2.1.2 CFD

Longo and Gasparella [22] experimentally and theoretically, using CFD method, tested dehumidification and regeneration of liquid desiccant in a counter-flow random packed bed contactor with LiBr, LiCl and the new environmentally compatible salt KCOOH, followed by developing and verifying theoretical model is by experimental data which is used to analysis sensitivity of system to parameters. The results show that conventional LiBr and LiCl present better dehumidification performance but KCOOH acts better during regeneration. Schematic diagrams of experimental setup and differential element are shown in Figure 2-1 and Figure 2-2.

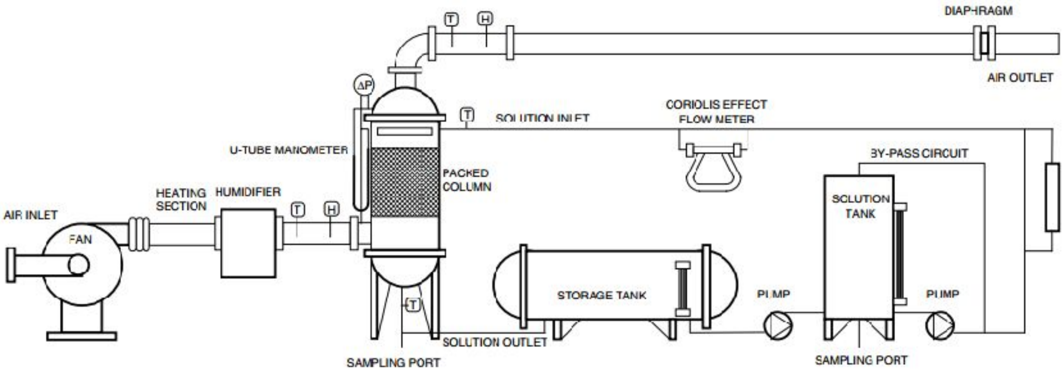


Figure 2-1 Experimental test setup

Koronaki et al. [23] developed a differential model for adiabatic counter-flow dehumidifier, Eq. (2-3) to Eq. (2-7), and did parametric investigations by utilizing three liquid desiccants, namely LiCl, LiBr and CaCl. According to the results, LiCl operates better than the others due to its lower surface vapor pressure. Also from parametric point of view several important results are obtained. Firstly, inlet air temperature increment has a negligible effect on condensation rate and efficiency in three cases but increasing inlet air humidity results in increasing condensation rate and thermal efficiency, meaning that the system is beneficial at hot and humid climates. On the other hand a rise in air flow rate results in higher and lower sorption rate and thermal efficiency respectively but it should be considered that this does not mean of lower humidity of outlet air.

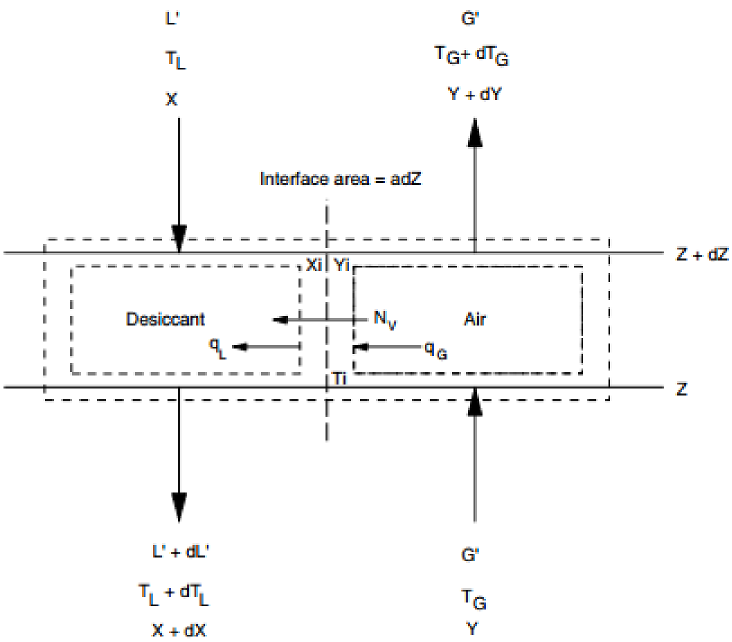


Figure 2-2 Differential element of a packed column

Also 10 times increment of air flow is followed by approximately 100 times higher pressure drop of air. Finally, desiccant flow rate does not affect condensation significantly however it must be high enough to ensure wetting of packing material meaning a minimum requirement. Following that it shows that low desiccant temperature is more effective than high concentration of that.

$$\frac{dT_a}{dz} = \frac{T_a - T_s}{C_{p,ma}}\left(\frac{Lh_G}{\dot{m}_a} - C_{p,st}{}^{sat}\frac{dW}{dz}\right) \qquad (2-3)$$

$$\frac{dT_s}{dz} = \frac{\dot{m}_a}{\dot{m}_s C_{p,s}}\left[\left(-C_{p,s}\left(T_s - T_{s,in}\right) + C_{p,st}{}^{sat}T_a + \Delta h_{abs}\right)\frac{dW}{dz} + C_{p,ma}\frac{dT_a}{dz}\right] \qquad (2-4)$$

$$\frac{dW}{dz} = \frac{LK_G}{\dot{m}_a}\left(W^{sat} - W_{in}\right) \qquad (2-5)$$

$$\frac{d\dot{m}_s}{dz} = \dot{m}_a\frac{dW}{dz} \qquad (2-6)$$

$$\frac{dX}{dz} = -\frac{\dot{m}_a}{\dot{m}_s}X\frac{dW}{dz} \qquad (2-7)$$

For more detailed information about the equations, readers should refer to the reference [40].

Also Fumo and Goswami [24] did a similar study for regenerator in addition to dehumidifier with LiCl for counter flow configuration. The results for dehumidifier are similar to the previous study as expected thus are not repeated again. From regeneration point of view, Water evaporation increases by increasing air and desiccant flow rate with slope 0.5 and 0.3 respectively but the increment of the latter one does not mean obtaining more concentrate desiccant at outlet. In opposite manner to dehumidifier, water evaporation increases with slop of 5 with increasing desiccant inlet temperature and similarly as concentration goes down, evaporation drops with slop 1.8 which means temperature is more important rather than concentration which was resulted also in dehumidifier. Finally, as air humidity goes up, evaporation goes down with slop 0.3 but temperature does not affect significantly. Likewise but with different pickings and solutions, several investigations are done on packed bed dehumidifiers and regenerators. For instance, Oberg and Goswami [25] did experimental investigations and developed a finite model for randomly packed dehumidifier with polypropylene Rauschert Hiflow rings and using TEG. Similarly, Lazzarin et al. [26] did experimental investigations and developed a computer model for counter flow, randomly packed plastic Pall rings dehumidifier with LiCL and $CaCl_2$ as sorbents. Rahimi and Babakhani [27] developed a mathematical model for a packed–bed air dehumidifier and the impact of some well–known empirical correlations available in literature is evaluated on the model’s predictions and accuracy. The results reveal that in designing an air dehumidifier, using different empirical correlations may lead to very different predictions.

2.1.2.1.3 NTU

Several studies have investigated contactors through number of transfer units method [28], [29] and [30]. Liu and Jiang [31] have focused on the combined heat and mass transfer interaction in the packed bed dehumidifier/ regenerator. It follows that the variance of the desiccant concentration within the dehumidifier/ regenerator may be neglected in most conditions but the variance of the desiccant temperature is the main factor that affects the desiccant absorption ability. Using their coupled-design model, Figure 2-3, they show that counter flow is good for dehumidification while parallel flow and counter flow streams are good for regeneration when air and desiccant is heated respectively in adiabatic packed bed dehumidifier.

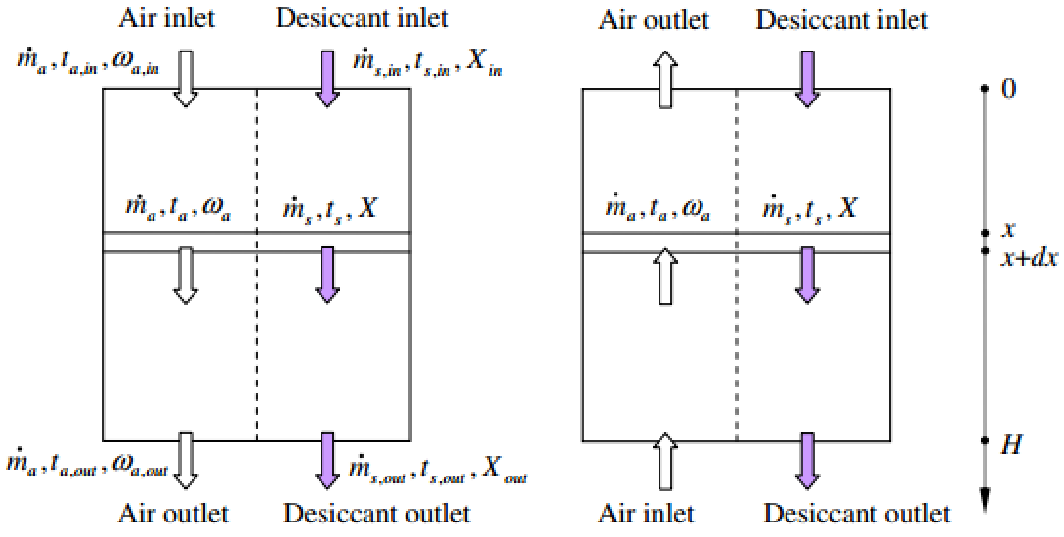


Figure 2-3 Schematic diagram of parallel flow and counter flow configuration [42]

2.1.2.1.4. Algebraic

Martin and Goswami [32] developed algebraic correlation, Eq. (2-8), for dehumidification and enthalpy effectiveness of counter-flow random dehumidifier based on several experimental works on TEG such as the work of Chung et al. [19] and one for LiCl [25]. Gandhidasan [33], [34] developed equations for estimating dehumidifier and regenerator outputs in two separate works by predicting air outlet temperature and validated it by the work of Fumo and Goswami [24]. The results show that the temperature of cooling water and efficiency of heat exchanger in order to cool down strong solution affect the water vapor sorption deeply. For instance increasing heat exchanger efficiency from 0.6 to 0.7 and decreasing cooling water temperature from 29.5 to 28.5 ^{o}C with desiccant flow rate of 5 kg/m^{2}s results in soar of sorption effectiveness by 83% and 400% respectively.

$$\varepsilon = 1 - f\left(\frac{L}{A}\right)^{a}\left(\frac{H_{A,i}}{H_{L,i}}\right)^{b}(Za)^{c} \qquad (2-8)$$

2.1.2.2. Non-Adiabatic

In addition, non-adiabatic packed bed dehumidifier/ regenerator are investigated by many researches in which desiccant is cooled or heated in contactors during sorption and desorption process respectively. Anmin et al. [35] experimentally tested cross-flow internally cooled and adiabatic $CaCl_2$ structured packed bed dehumidifier. The results show that there is an optimum point of desiccant flow rate for each specific air flow rate and it is much prominent in internally cooled design. Liu et al. [36] investigated different configurations of LiCl inter-cooled dehumidifier and compare it to adiabatic dehumidifier. Results show that counter-flow and parallel-flow configuration has the highest and poorest effectiveness respectively. Also in adiabatic case, desiccant temperature increment is the limitation but in internally-cooled case, decreasing desiccant concentration is. It shows that inter cooled systems, operates at very smaller desiccant mass flow rate to absorb the same amount of water vapor. Yun and Mingheng [37] analyzed the effect of solution to air mass ratio during dehumidification and provided a standard for optimization both for adiabatic and non-adiabatic dehumidifiers. It is important to say that too low magnitudes of this ratio results in undesirable distribution of strong solution over packing material and as the ratio increases, so the irreversibility thus an optimum point is preferred. Bansal et al. [38] experimentally tested cross-flow internally cooled CaCl2 structured packed bed dehumidifier and compare it to an adiabatic one. The results show that by keeping air flow rate and desiccant concentration constant, effect of desiccant flow rate on dehumidifier effectiveness in adiabatic design is small compared to internally cooled design but there exist an optimum point for them as shown at Figure 2-4. Because as the desiccant flow increases in internally cooled case, more heat is transferred to cooling water but on the other hand it results in increasing the temperature of chilled water. Since cooling capacity is constant, after a specific magnitude of desiccant flow rate, heat transformation cannot exceed and desiccant temperature goes up resulting in lower effectiveness. The reason in the adiabatic case may be due to higher resistance to air flow through the packing material at higher desiccant flow rates, and some air may be bypass without any mass transfer. Yin and Zhang [39] study a vast parametric investigation of internally heated parallel-flow regenerator using 37% concentration LiCl as liquid desiccant and compare it to an adiabatic design. The results show that for adiabatic regenerator, desiccant and air flow rates must be high and low respectively to have an effective evaporation and high thermal efficiency of regenerator; else at low flow rates of desiccant or high flow rates of air, the other case must be adopted. Also at low regeneration temperatures, evaporation rate and thermal efficiency of internal-heated design is significantly higher than the other. In addition, decreasing concentration of diluted desiccant results in better regeneration at both cases. On the other hand increasing air temperature rises and decreases the regeneration rate of adiabatic and internally-heated regenerators respectively. In the internally-heated case, increasing heating water flow rate causes to increase of evaporation rate and thermal efficiency but it its impact is high at low water flow rates. One interesting result of this comparison is that by increasing air flow rate, regenerator thermal efficiency plummet down significantly in adiabatic design while in internally heated design, it goes down slightly.

Khan and Martinez [40] investigate internally cooled dehumidifier utilizing cold coils as contacting area instead of cooling layers and refrigerant as cooling mean for coils and have evaluated humidity and temperature of process air temperature and concentration of desiccant.

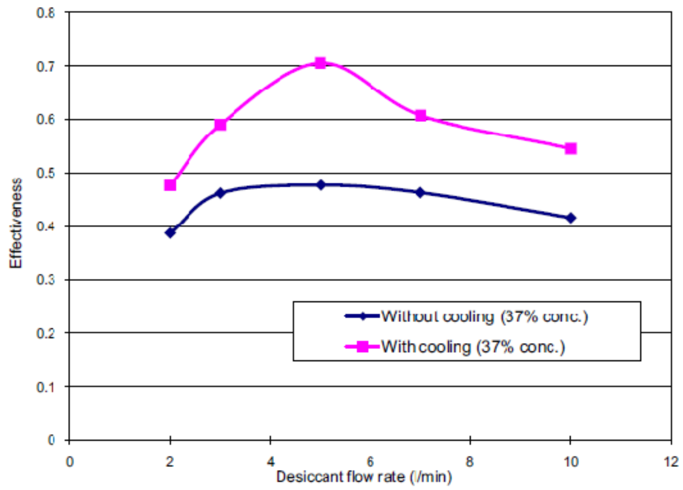


Figure 2-4 Effect of desiccant flow rate on dehumidifier effectiveness

2.2. Material

The second type is desiccant material investigation in literature. Selecting liquid desiccant is decisive in the performance of contactor and its selection depends on many factors such as regeneration temperature, surface vapor pressure (capacity to absorb vapor), stability, viscosity and ability of storage in liquid desiccants and finally the cost. Desiccant material could be devised into adsorbents and absorbents. Note that in absorbents, which are usually liquid, Water is trapped inside of absorbent like sponge but in adsorbent, which are usually solid, Water is caught in Nano pores at the surface of adsorbent and will release by a small amount of energy. The most common adsorbents are activated alumina, activated carbon, Molecular

Sieves (Zeolite) and silica gel. The most common absorbents are solutions of Calcium Chloride, Lithium Chloride, Lithium Bromide and Triethylene glycol. Cui et al. [41] compared several solid desiccants and showed that DH-5 and DH-7 adsorbents are superior to that other. Aristov et al. [42] and Tokarev et al. [43] impregnated a host porous with hygroscopic salt resulting in three-times-higher capacity of developed hybrid material compared to the capacity of host. Following the method, Liu et al. [44] impregnated silica gel with Calcium Chloride and used it for drying air. Belding et al. [45] focus on the aging phenomena of desiccants due to cyclic process of sorption/desorption and showed that as desiccant ages, the speed of cycle must be increased.

2.3. Systems

In the Third group, all the system is modeled instead of focusing on contactor and the aim is feasibility, improvement and optimization of the whole system at different combinations and ambient conditions. At this type, the equations, as the results of approved mathematical models of contactor literatures are used to modeling but the range of inputs for used equations, for which the mathematical model was approved, must be concerned. DCS can be divided into two three groups, standalone, hybrid and desiccant-aided radiant cooling. The two latter groups are mostly designed for applications in more harsh climates. Also all three system types could be run in three different air flow configurations; recirculation, ventilation and Dunkle cycle. In recirculation mode, the supply air is the return air from conditioned space and regeneration air is outdoor air. Ventilation is referred to the configuration in which supply air is outdoor air and regeneration air could be outdoor or return air and the latter one is called Pennington cycle. Dunkle cycle is the recirculation cycle but utilizing a heat exchanger for pre-cooling supply air before entering cooling unit and the flow on the other side of heat exchanger is cooled outdoor air by evaporation heading toward regenerator. It should be mentioned that in recirculation and Dunkle cycle, when all return air is adopted for regeneration, usually 20% of air bypasses the heater and regenerator to reduce heat consumption [45].

2.3.1. Standalone

Standalone desiccant cooling system benefits the evaporative cooling phenomena, either direct or indirect evaporator. In addition to desiccant dehumidification, evaporation process depend on ambient condition thus strong reliance to surrounding situation becomes the Achilles heel of that and limiting its applicability. There are so many literatures about standalone systems mostly in temperate climates. Dhar et al [46] have evaluated thermodynamic analysis for different air configuration in a standalone desiccant cooling system, including recirculation, ventilation and Dunkle cycle, with solid contactor in different Indian cities and found out that Dunkle cycle has the best performance for a wide range of outdoor conditions. White et al. [47] modeled a solar-derived standalone DCS by TRNSYS, desiccant wheel coated by silica gel with no thermal back up and an indirect and direct evaporative cooling as 1st and 2nd cooling units respectively and the indirect evaporative cooler uses a fraction of the cooled primary air stream exiting the indirect cooler as the secondary air flow for other side. The results show that two-steps cooling is high effective at when low humidity air is available in indirect evaporator. Also it shows that high flow of low-temperature air for regeneration would prepare comfort condition in temperate but not tropical climate. Katejanekarn and Kumar [48] did a parametric study on solar-assisted liquid standalone DCS in tropical climate with indirect evaporating cooling to overcome ventilation load. Heidarinejad and Pasdarshahri [49] modeled standalone solid desiccant cooling with two direct evaporators, one for process air and one for regeneration air and mathematical formulas are mentioned. The aim of study is to check the effect of ambient condition on system COP and the temperature of supply air while the cooling load and regeneration temperature considered being constant. It was investigated on ventilation and makeup mode for system. In Ventilation mode of system operation, return air, used as regeneration air stream. This system is practiced where the cost of return ducts is low or the return air is available as regeneration air. In Makeup mode of system operation, outdoor air, utilized as regeneration air stream. The results show that an increase in air dry bulb temperature, both cooling capacity and regeneration energy decrease in both cycles. Comparing ventilation and makeup types, supply temperature is lower for Ventilation cycle compared to Makeup cycle at the same outdoor conditions but The

COP of Ventilation mode is lower than Makeup in higher values of outdoor air temperature and humidity.

2.3.2. Hybrid

Hybrid desiccant cooling system takes advantage of air dehumidification and cooling by desiccant materials and vapor compression system (VCS) or vapor sorption system (VSS) respectively but initial cost requirement is higher compared to evaporation cooling, available in standalone type. The most important privileges are lower dehumidification level and less effected by climate conditions thanks to VCS and VSS systems resulting into more stability and application at harsh circumstances where evaporation is ineffective or even not possible. Riley [50] Mentions an experimental research performed at Mississippi state university on energy consumption of a residential desiccant unit. According to results, a standard VCCS without reheat runs 32% times with 2279W electricity consumption while no control over humidity is done and 49% of latent load is unmet. Adding a heater to this system in order to control humidity in addition to temperature, results in 63% time fraction run and 4504W of electricity. In turn of hybrid desiccant cooling, 3489W electricity is consumed while humidity is controlled and using a heat exchanger with effectiveness of 0.7 lowers that to 2961W. This proves the importance of hybrid desiccant cooling when control over humidity is required but not that at this investigation regenerating of desiccant is done by electricity, thus using solar or waste heat sources decreases electricity consumption of the system significantly. Jia et al. [51] modeled experimentally a hybrid desiccant cooling system, a combination of rotary desiccant wheel run by heater and a VCS and compare it a conventional VCCS. Results show that for ambient condition of 30ºC and relative humidity of 55%, electricity consumption reduced by 37.5% but the main aim of exploration is to investigate the effect of outdoor air condition on behavior of evaporator of VCS. Thus sensible heat factor (SHF) is defined as sensible cooling load carried by evaporator to total cooling load carried by evaporator and a parametric study is done about that. Since the main purpose of evaporator is to reduce the temperature, ideal magnitude of this parameter is one. The results show that for the same outdoor condition, SHF is higher at hybrid system especially at high ambient humidity and temperature and also when air temperature drop is manifest (large evaporator) as shown in Figure 2-5 and Figure 2-6. It means that higher decrement of air temperature is gained in evaporator of hybrid system in the same ambient condition, in other word when supply flow rate assumed to be constant, smaller evaporator ability for decreasing the temperature of supply air in hybrid cycle is the same with a larger one provides in conventional cycle while heat load they could overcome is the same. In addition, smaller evaporator means higher temperature of that and consequently higher COP. On the other hand utilizing the same evaporator in hybrid and VCCS system, results is lower air flow demand at the former to neutralize the heat load and consequently better dehumidification at dehumidifier.

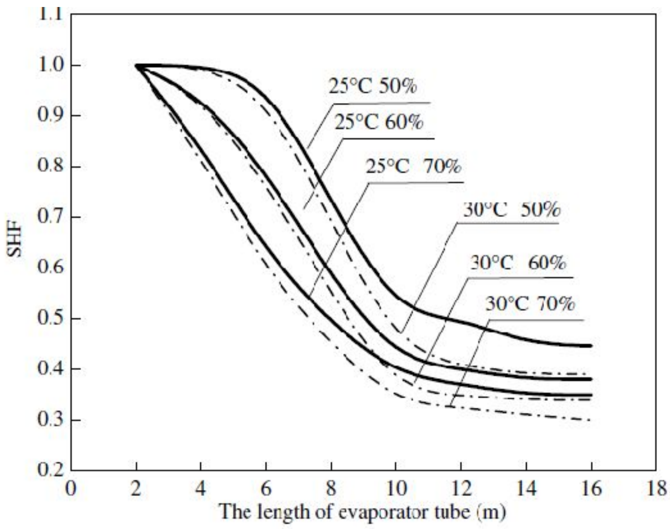


Figure 2-5 Effect of evaporator tube length (or temperature decrement of air) and ambient temperature and humidity on SHF of evaporator in conventional VCCS [51]

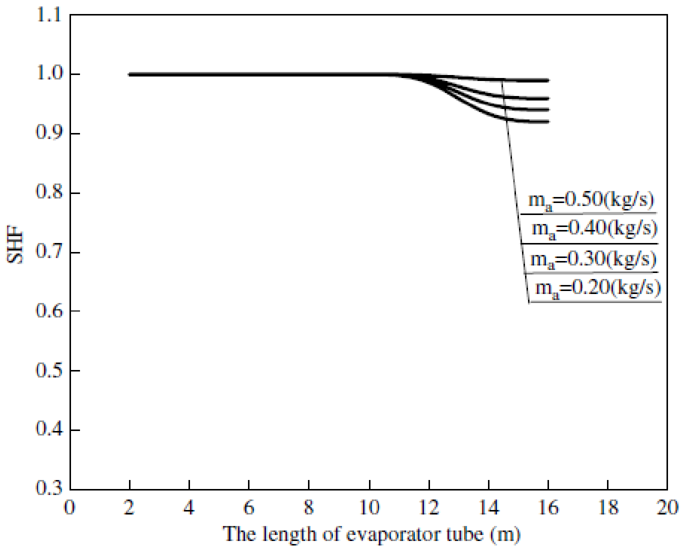


Figure 2-6 Effect of evaporator tube length (or temperature decrement of air) and ambient temperature and humidity on SHF of evaporator in conventional VCCS [51]

Investigating the effect of air flow rate by considering inlet and outlet condition the same and increasing air flow rate, results show that electricity consumption growth is quicker at conventional system than hybrid. To put a comment on this result, rise of air flow and other parameters constant, higher heat load could be concluded. Thus by remaining heat load constant, by increasing air flow rate, resulting in lower evaporator temperature requirement, the same conclusion is expected, Figure 2-7, while SHF remains constant by increasing air flow rate. Investigating inlet conditions, better saving of electrical energy would be obtained by hybrid system compared to VCCS at conditions with higher humidity according to Figure 2-8 but note that total energy consumption is not calculated. Finally studying regeneration temperature effect, increasing this parameter results in lower humidity of supply air and following that higher SHF in evaporator when air leaving dehumidifier is so cold which is done by very low evaporation temperature or huge size of that.

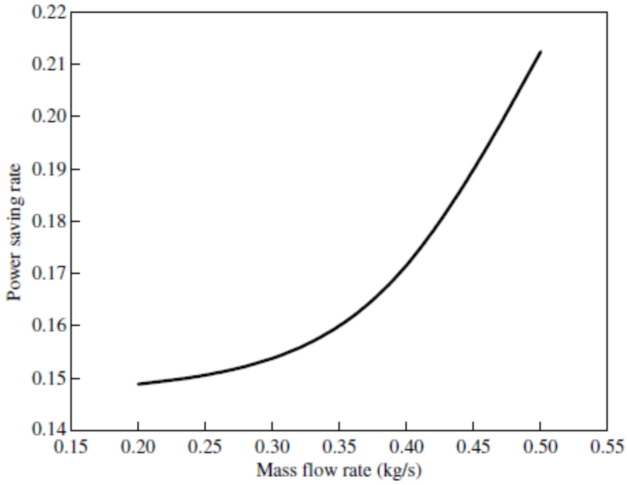


Figure 2-7 Effect of air flow rate on electricity saving rate [51]

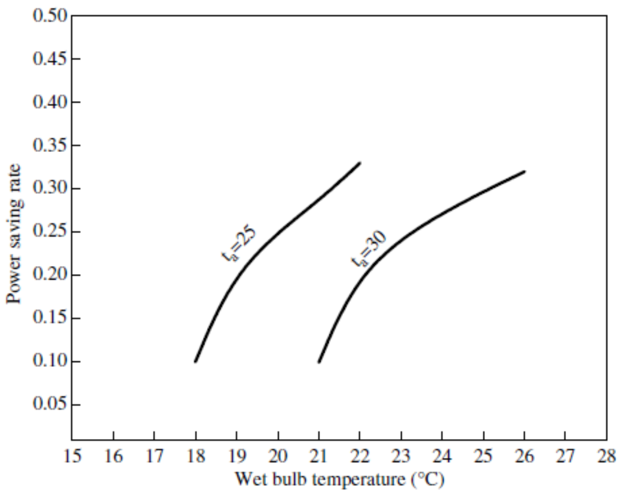


Figure 2-8 Effect of inlet air condition on electricity saving rate [51]

Mandegari and Pahlavanzadeh [52] experimentally tested a desiccant wheel integrated with VCS at hot-humid and hot-dry climates and compare it to conventional VCCS to investigate the coefficient of performance (COP) and electrical coefficient of performance (ECOP) of the two in both climates and the results are shown in Figure 2-9 and Figure 2-10 respectively. The results show that COP of VCCS system is far beyond of in dry climates but in humid climates this difference is smaller. From ECOP point of view, electricity consumption of VCCS is smaller than hybrid system in dry climates when ambient temperature is below 40^{o}C approximately but in conditions with high humidity, ECOP is greater in hybrid system in all ambient temperatures. One important thing regarding the COP of hybrid system is that lot of consumed thermal energy could be obtained from solar source which is not evaluated here, thus if we consider it as free-clean thermal source higher magnitudes would be obtained of a useful parameter we can call it "Clean COP".

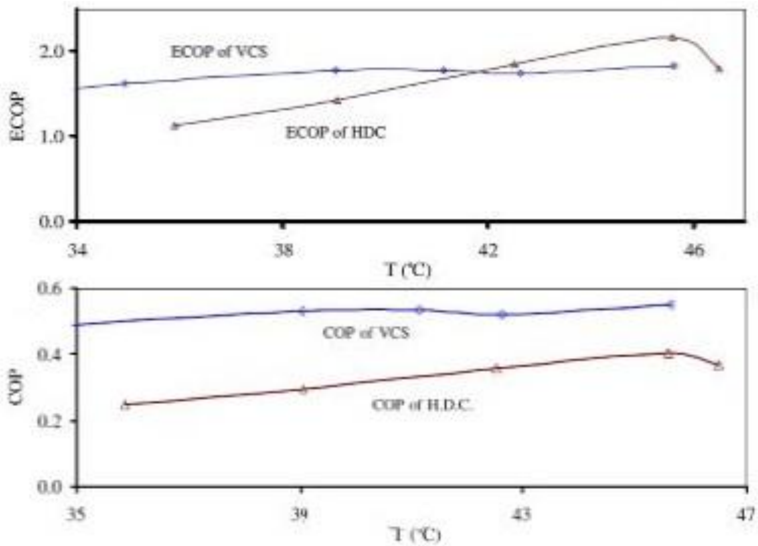


Figure 2-9 COP and ECOP of hybrid and conventional VCS in hot and dry climates [52]

Dhar and Singh [53] investigated four different cycles of DCS integrated with VCS cycles, ventilation-condenser, ventilation-heat exchanger, modified ventilation-heat exchanger and recirculation-condenser in hot-dry and hot-humid climates and compare them to VCCS from energy consumption point of view. Note that to calculate total energy, which is the sum of electricity and heat, electricity is divided by equivalent coefficient of electric energy and thermal energy, assumed to be equal to 0.3 according to the thermal efficiency of power plants (for cycle information refer to reference 63).

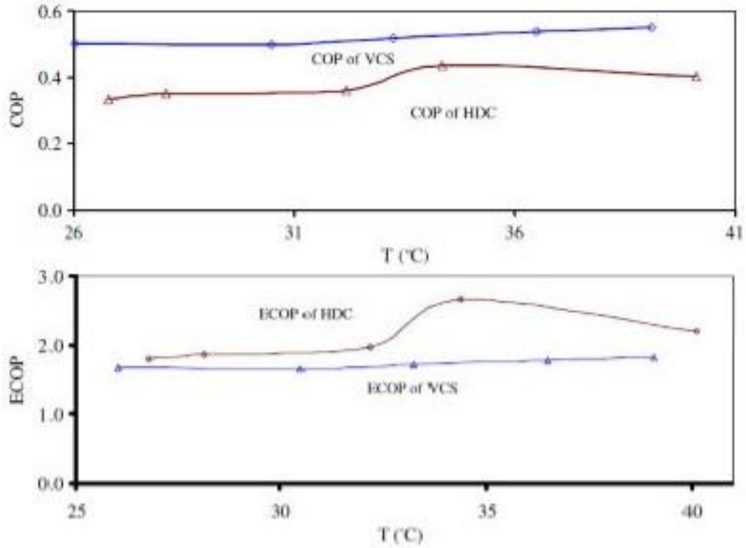


Figure 2-10 COP and ECOP of hybrid and conventional VCS in hot and humid climate[52]

During investigations conditioned space set temperature and humidity are considered 24°C and 9.3gr/kg respectively.Ambient condition for dry climate is 43.2°C and 7.26 gr/kg and for humid condition is considered as 38.6 °C and 16.14 gr/kg and regeneration temperature of 135 °C. Two values are considered as ratio of fresh air to return air (r), 0.1 and 0.2, and also studies are done for both high and low magnitudes of LLR. It should be mentioned that humidity of air entering dehumidifier for the recirculation-condenser cycle is lower compared two all other cycles and consequently regeneration temperature for this cycle is lower but the dehumidifier should be larger in volume due to all volume of supply air flows through it. According to the results, two cycles are more of interest, recirculation and ventilation-condenser. The results of hot-dry climate are not mentioned but for hot-humid climates.

For hot and humid climate, the results are:

1) Except Recirculation cycle which needs one dehumidifier, all other desiccant systems requires two due to high humidity ratio of supply air.
2) Evaporator temperature of VCCS is the lowest and for Ventilation-Condenser is the highest
3) Required Power for each system is shown in Figure 2-11 and Figure 2-12 for low and high LLR respectively.

4) Flow rate of air which passes through dehumidifier and required flow rate to conditioned space is shown in Figure 2-13 and Figure 2-14 respectively for configurations.

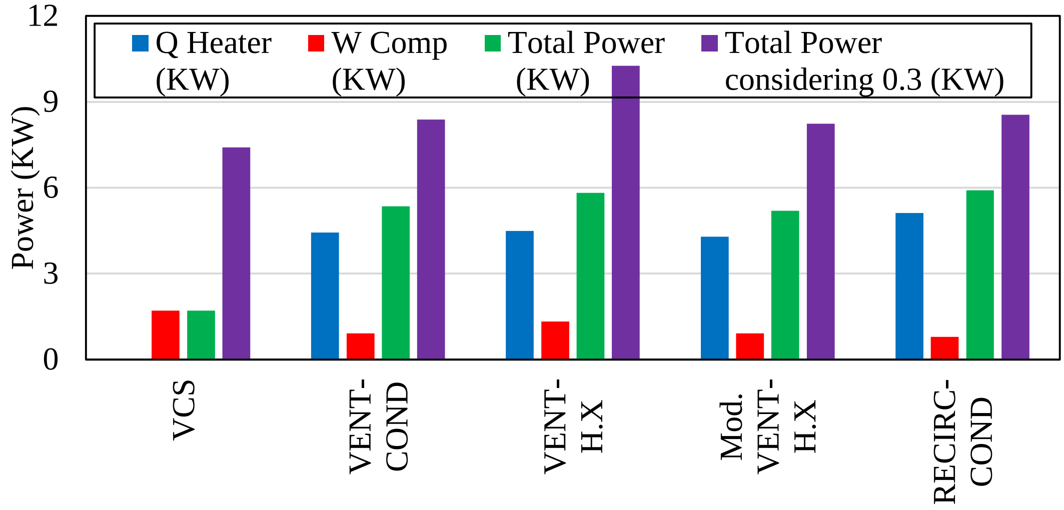


Figure 2-11 Power requirement for different cycles for low LLR

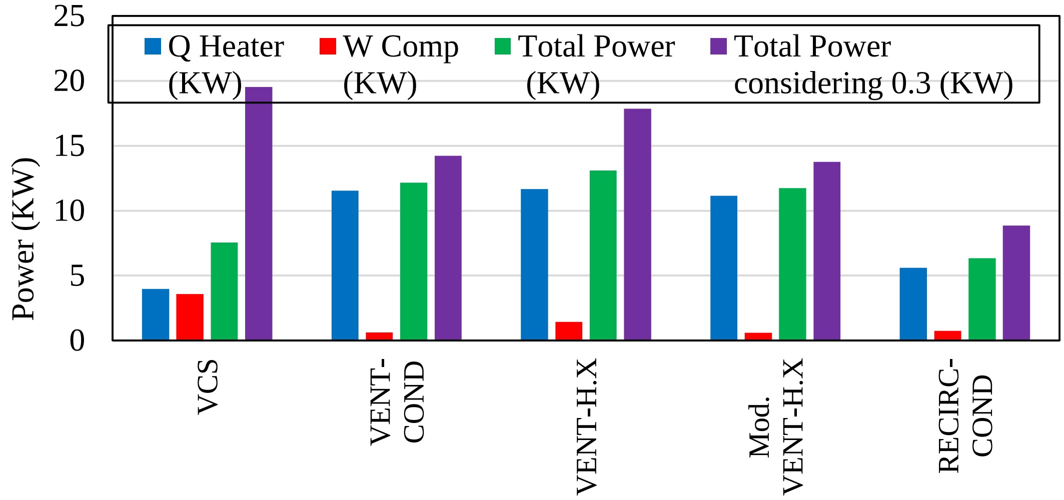


Figure 2-12 Power requirement for different cycles for high LLR

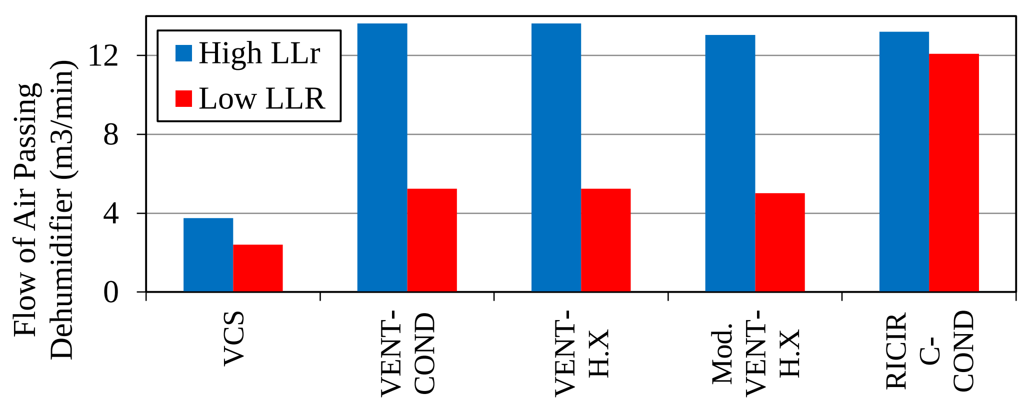


Figure 2-13 Flow rate of air passing dehumidifier

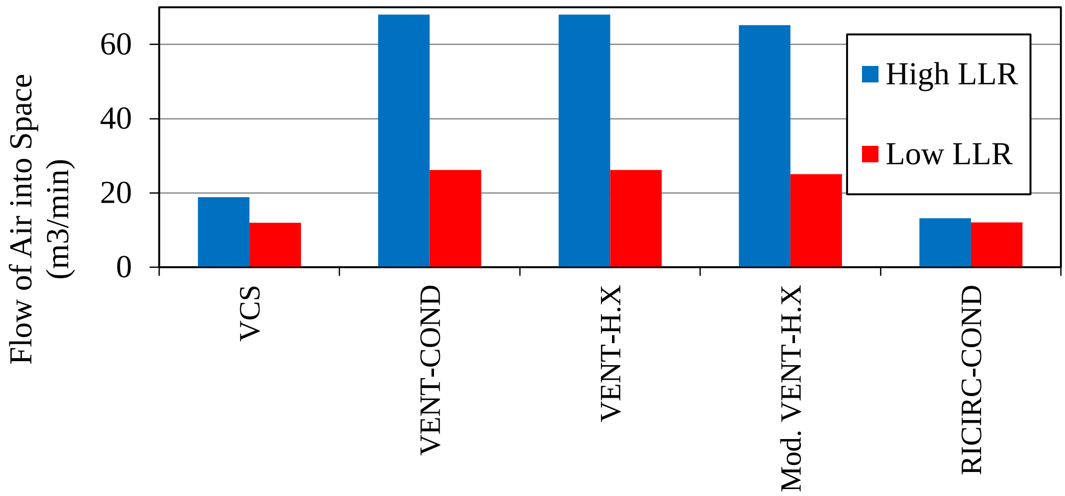


Figure 2-14 Required flow rate to conditioned space

Selecting recirculation cycle as the most beneficial specially by using solar energy as thermal source, investigations are done on the impact of regeneration temperature on cycle energy consumption and the results are shown in Figure 2-15 and Figure 2-16 for low and high LLR respectively:

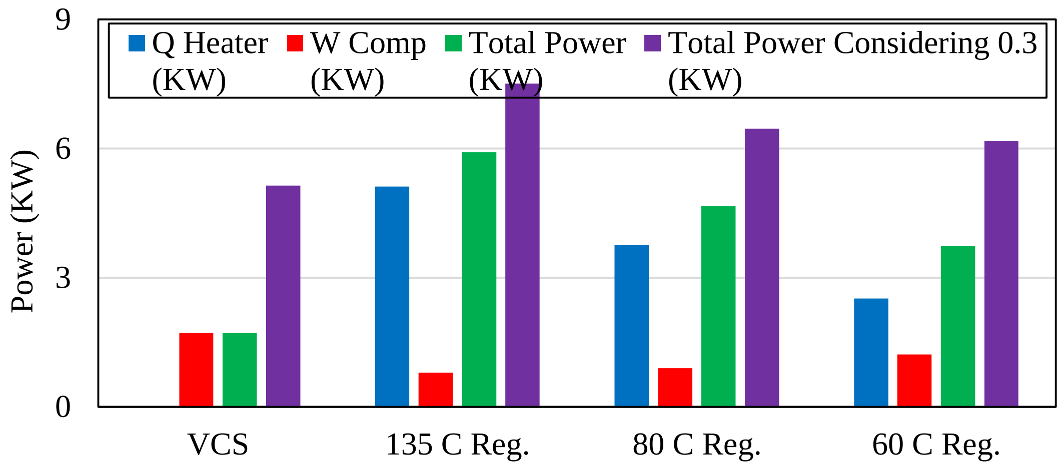


Figure 2-15 Effect of regeneration temperature on recirculation cycle at low LLR

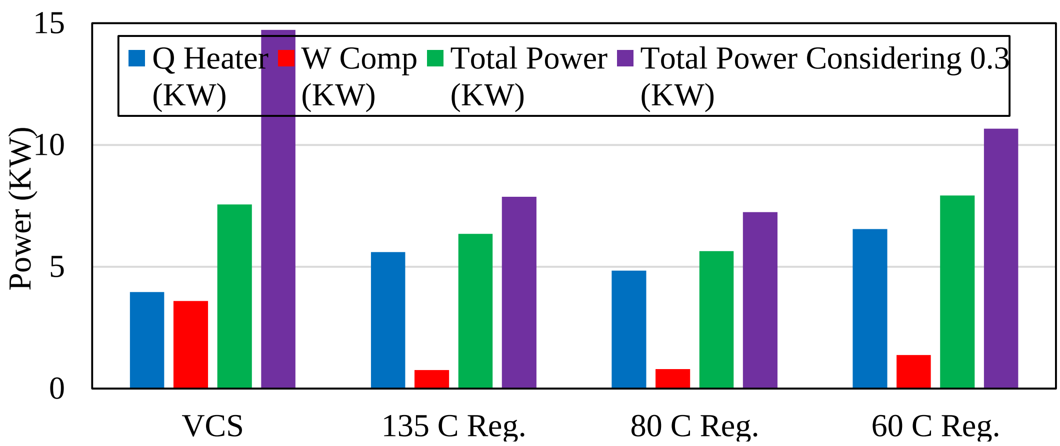


Figure 2-16 Effect of regeneration temperature on recirculation cycle at high LLR

Dai et al [54] have developed a model of desiccant-assisted VCS mathematically with liquid contactors and validated by comparing to experimental results. Using validated model, they investigates three systems, namely VCCS, VCS + Desiccant and VCS + Desiccant + heat exchanger + evaporative cooler and have compare them assuming some parameters for all of them are constant, including inlet and outlet condition of air, supply air mass flow rate and liquid desiccant mass flow rate. The investigations are done for ventilation and recirculation configurations of three cycles mentioned above at ARI conditions, T_{amb} = 35 °C and RH_{amb} = 40 %, and key parameters which are not assumed to be constant are discussed. The interesting assumption at this exploration is considering dehumidification process of supply air as isotherm due to relatively lower temperature of inlet desiccant into absorber compared to inlet air temperature and also using an isenthalpic evaporator cooler. Inlet conditions of supply air is fixed at ARI conditions for ventilation mode and 26.7°C and 50% of relative humidity for recirculation mode but outlet conditions are fixed at 18°C and 60% of RH in both of them and the results are summarized at Figure 2-17.

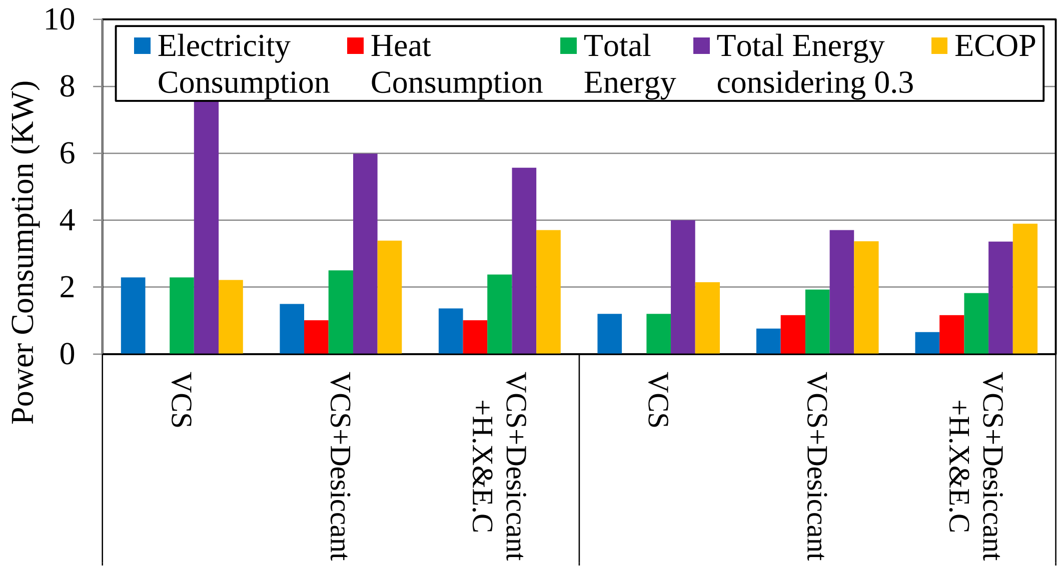


Figure 2-17 Comparison between VCCS, VCS+DCS and VCS+DCS+H.X+E.C

Also parametric studies show that for the cycle with heat exchanger and evaporative cooler, increasing inlet humidity ratio, effectiveness of dehumidifier and effectiveness of evaporator of VCS and also reducing inlet temperature results in higher ECOP but humidity ratio impact is the strongest one. In addition to that, increasing inlet temperature and humidity results in higher demand of condensation air while all other parameters are constant but by effectiveness rising of condenser of VCS, required condensation air flow drops significantly in addition to soaring of ECOP.

Kinsara et al. [55] have proposed a complex hybrid system with $LiCl_2$ run by electricity and independence of thermal energy source. It includes two packed bed contactor as dehumidifier and regenerator, cooling supply air by utilizing a VCS and an evaporative cooler, also another VCS for decreasing the temperature of liquid desiccant flowing to dehumidifier for better dehumidification. The heat produced by condenser of the two VCS is used to heating the secondary air and diluted desiccant on the way of regenerator with the help of 4 heat exchangers. The first one is an air-to-air one for pre-heating of inlet regeneration air by using of outlet-regeneration air, the second one is air-to-liquid for heating diluted desiccant to its maximum temperature using a hot air, the third one is liquid-to-liquid to provide heat transfer between hot strong desiccant and temperate diluted desiccant and the last one is air-to-liquid for pre heating of diluted desiccant by the out-going regeneration air. Following that, Impact of some parameters on CORP, defined as COP of the proposed system to COP of conventional VCCS, is investigated assuming that concentration and of strong solution is remained constant at 40% and providing a range of condition in which the operation of system can provide all requirements. The results show that increasing ambient temperature raises the CORP but humidity ratio has a strong reverse impact. It also proved that effect of ambient temperature and the secondary air flow rate on regeneration temperature is negligible but as ambient humidity goes up gradually, regeneration temperature soars significantly, 5^{o}C approximately for each 5 gr/kg increment. While the impact of liquid desiccant flow rate to supply flow rate is negligible, but temperature of liquid desiccant at dehumidifier is noticeable, by reducing that, load of the VCS using for cooling desiccant increases and for the other one it decreases but in general COPR descends but there exist a maximum temperature for liquid desiccant because the system has to response to demands entirely. In addition to that, at high and low temperature of desiccant, increasing and decreasing of LLR respectively improves the COPR. These results are useful especially for application in hot and humid climates which state that in such conditions, liquid desiccant temperature should be at its maximum magnitude in order to obtain better efficiency. Finally the effect of heat exchangers are investigated and shows that the 1st and the 4th ones have very low impact on COPR compared to the 2nd at the 3rd ones. That is because of the fact that the 1st H.X is air-to-air and the temperature difference between air and liquid in the 4th H.X is not as much as it is for the 2nd and the 3rd.

Hirunlabh et al. [56] is done an experimental investigation on feasibility of hybrid DCS in Thailand during a typical day. Their system consists of a desiccant fixed bed using silica gel and a 1.5 ton split cooling unit. The configuration of system is different from what mentioned yet, return air mixes with fresh air before entering dehumidifier and after that indoor air is combined with the dehumidifier-produced dry air to pass through the evaporator of split unit. They study system behavior for different ventilation ratio, dry air and indoor air combination ratios, but not the effect of return and fresh air ratio and find that there exist an optimum dry and indoor air combination ratio in which electrical energy saving reaches the maximum. Increasing dry air ratio results in higher temperature of air entering evaporator generally but humidity of the air reduces by increasing dry air ratio up to a specific point but after that humidity rises. Since magnitude of humidity is more effective than temperature in the passing-evaporator air, the lowest electricity consumption is occur at the point where humidity is the bottommost. Also by doing economic analysis, it shows payback time of hybrid system is three times lower for large central air-conditioning systems.

Fong et al [57] have done a study on feasibility of six different configurations of solar assisted hybrid desiccant cooling using TRNSYS16 during typical year in Hong Kong and the results are illustrated in Figure 2-18 and Figure 2-19. Five configurations of the study is done by utilizing VCS and one of them VSS to provide chilled water at cooling coil, adopting a desiccant wheel coated with silica gel. Three of five VCS are ventilation cycles of air adopting evacuated tubes, solar photo voltaic panels and also PV/T panels. In addition, two recirculation cycle of air, employing PV and PV/T are also investigated. Finally a recirculation cycle with absorption chiller and evacuated tube is studied. At the end, results are compared to standalone DCS and also conventional VCS in both ventilation and recirculation mode. The study is done on an 8 to 18 working-hour office building at Hong Kong. For comparison, COP, energy consumption and SF for each section and the entire system is adopted. Solar Fraction, SF, shows the proportion of solar energy contribution upon the total energy required to drive cooling system. Also it proves that COP is applicable to compare one system at different situations but not for comparing different systems due to definitions variation of COP at different systems. One thing which is not considered at the study is employing released heat from condenser of VCS or VSS for purpose of regeneration, could reduce the demand to auxiliary thermal energy.

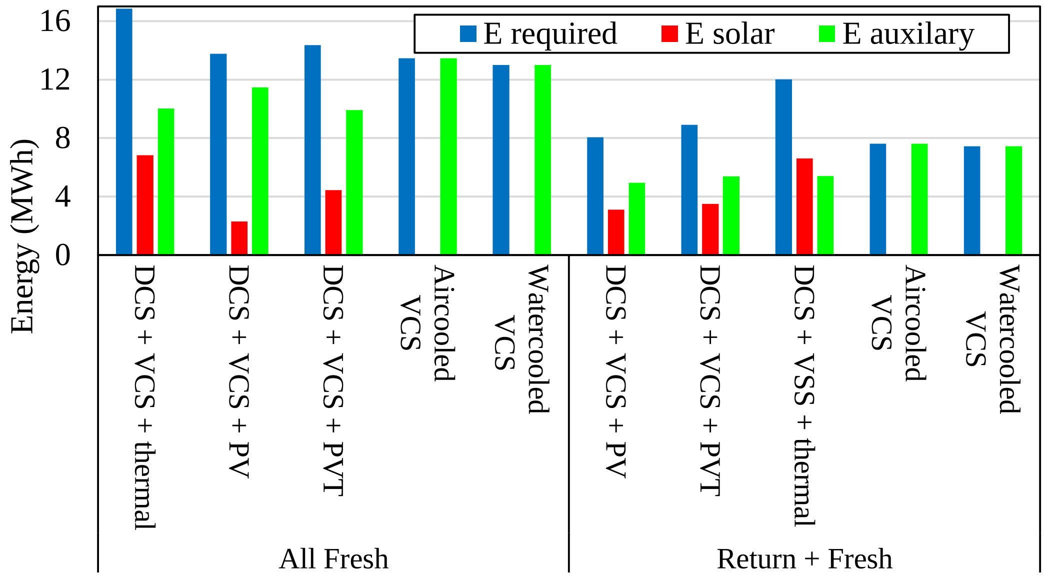


Figure 2-18 Total, solar and auxiliary energy consumption of systems

Finally it should be mentioned that pros and cons of VCS and VSS must be considered when selecting each of them. While VCS consumes electricity, with high CO_2 coefficient, VSS is whole thermal driven that could be obtained from waste source. But there exist some weakness points for VSS such as more working fluid demand, higher regeneration load and impossibility of waste heat utilization produced by refrigeration cycle [58].

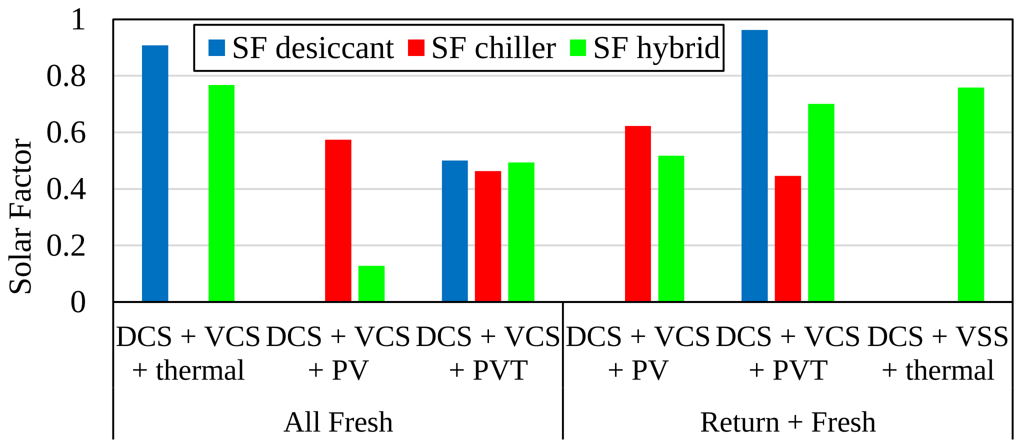


Figure 2-19 Sectional and total solar factor of systems

2.3.3. Desiccant-Assisted Hydronic Cooling

There are several hydronic cooling systems such as chilled ceiling, chilled floor and etc. but among them chilled ceiling design are more of interest. Chilled-ceiling cooling method, first investigated in early 1900s, feature interesting advantageous. Compared to all-air systems it provides better cooling distribution and also forcing water by pumps requires lower energy than fans for air but the most important one is lower energy consumption thanks to smaller ventilation requirement. But in normal chilled ceiling, condensation of water in conditioned space cease utilization of it in hot and humid climates because no dehumidification occurs at this system, thus a DCS can improve the drawbacks. In this design, DCS is only responsible for latent load of space and ventilation air and bring its temperature down to an acceptable level while the total sensible load of space is removed by chilled water, meaning decoupled energy transfer mechanism, and since volume of supply air is reduced, it could be a solution for DCS in tropical climate. In addition to these benefits, possibility of lower temperature of condensation coils, mostly neglected, results in higher COP of chiller. Chilled ceiling, as nonconventional and high efficient method, for air conditioning is combined with DCS and studied by authors. Halliday et al. [59] discussed the feasibility of using solar energy at three different city in U.K using a validated computer model. Their simple desiccant system use a solid desiccant wheel, a heat exchanger, an indirect solar heater and a supplementary cooling coils at supply side and an evaporative cooling chamber at return side in order to dehumidify fresh air and sensibly cooling it down when required but the bulk of the sensible cooling is performed by separate water based system. The results show that using solar energy source decreases total energy consumption, energy cost and CO_2 production by 45% during atypical summer day. Single stage solid desiccant cooling is used by Mavroudaki et al. [60] for air ventilation and removing latent load of it and a water based cooling system (e.g. ceiled chilling) for removing sensible load. The results shows that for locations with high humidity, air flow must be high for satisfying the most harsh condition at that location (since maximum regeneration temperature is 95° C, thus flow rate must be increased for good regeneration). This high flow rate during a year, results in very high latent load. Also solar power is not enough at these locations for sufficient regeneration (although these regions have higher solar irradiation compared to zones with low humidity such as northern and central Europe).It shows that humidity is more important rather than magnitude of solar power. For instance, this system decreases gas consumption in generator by 93% in northern Europe while solar power is very low compared to southern Europe and Mediterranean. Thus a VAV system should be considered (to reduce air flow when latent load is not high and thus lower dehumidification and lower regeneration) or multistep regeneration (to reduce regeneration temperature which does not required high flow because flow rate is increased to maintain the temperature of regeneration below 95^o C. As well as standalone systems, hybrid systems are studied a lot, especially radiant cooling system which is a special design of hybrid systems. Indoor humidity behavior in a DCS integrated with chilled ceiling is investigated by Zhang and Niu [61] and compared to all-air system. The results show that the system consumes less energy compared to all-air system and Air dehumidification and ventilation system should be operated at least 1 hour earlier than the operation of the system is feasible and energy saver.

CHAPTER 3

OBJECTIVE

According to the literatures, desiccant cooling systems are considered as the alternative method to conventional vapor compression systems or a supplement to it. The standalone design is suitable for applications in temperate climates where inlet air humidity is not high, thus deep dehumidification could be applied or utilizing an indirect evaporator and ignoring deep dehumidification. On the other hand, in hot and humid climates removing enough moisture to utilize evaporative cooling requires highly concentrated desiccants and consequently high regeneration temperature, stops employing those systems, thus hybrid systems are recommended which requires lower dehumidification level and could be in all-air or radiant cooling design. Even though lots of investigations are done on all-air systems with especial focus on combination with vapor compression cooling systems, there exist a lack of research and experiment related with investigating the systems operation behavior under variable conditions. In fact, most of studies are spent on specific ambient condition, temperature and humidity, parallel to specific design of system optimized for that condition and exploration of feasibility is done in this way. Actually, for this type of investigations it is better to explore system behavior in a period of year when system operates because of the fact that in practical application the full-load operation hour of system is too few and most of the times it works in partial load, thus the status of cooling system in part-load condition is not the same with full-load condition. On the other hand, when specific inputs are considered for investigation and the system is optimized for that situation, it can be considered as the full-load operation of and the behavior is unclear when inputs are altered. Thus the objectives for this research can be summarized as:

- Characteristic investigation of desiccant cooling cycle behavior by changing ambient and working conditions
- Magnitude and type of energy demand of the cycle at operation working hours and compare it to conventional cooling cycle
- Comparing the magnitude and type of energy demand of both cycles in a specific period of time, June, July and August

To fulfill that, an one-floor sport center building in Middle East Technical University Northern Cyprus Campus (METU NCC) [62], located at the heart of Mediterranean Sea, is designed by Google SketchUp [63] and modified by TRNBuild to be used in TRNSYS®17 [64] platform (provided by GÜNAM [65]) where an ordinary vapor cooling cycle next to a desiccant cooling cycle is model for energy analysis. According to introduction and literature review, LiCl-H_2O solution could be the best material in conjunction with counter-flow structured packed bed contactors in solar-assisted system. Comparing to Hong Kong conditions where a feasibility study during typical year is done by Fong et al [57], Cyprus has higher solar irradiation during cooling time, Figure 3-1, and also lower ambient humidity, Figure 3-2. Note that in the figures, each month radiation is the sum of all radiations during that month but temperature and humidity is the average for that month between hours 8 to 19. Thus, this study extends the current literature by applying the methodology of Fong et al. [54] to a Mediterranean Island. The schematic design of system is shown in Figure 3-3. Considered dehumidifier and regenerator models at here are adiabatic ones but non-adiabatic design also needs to be explored in a separate work The analysis are done for different ratios of indoor, return and fresh air flow rates.

At presented work, all system components except contactors are adopted from the software library. Also parameters of these components are not based on detailed design values, such as coil contact factor or pumps and fans motors efficiency, and actually this is out of the aim of this work but are based on suitable assumed values. For dehumidifier and regenerator, no new model is developed but existed algebraic effectiveness equations are adopted from open literature to be used in TRNSYS next to other available components model.

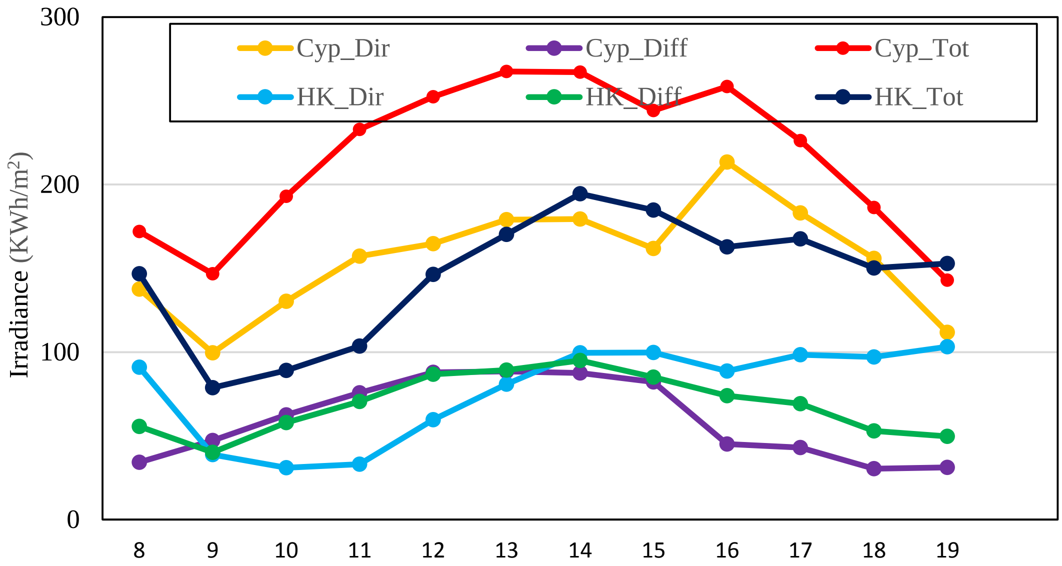


Figure 3-1 Monthly Solar Irradiation per square meter for Hon Kong and Northern Cyprus

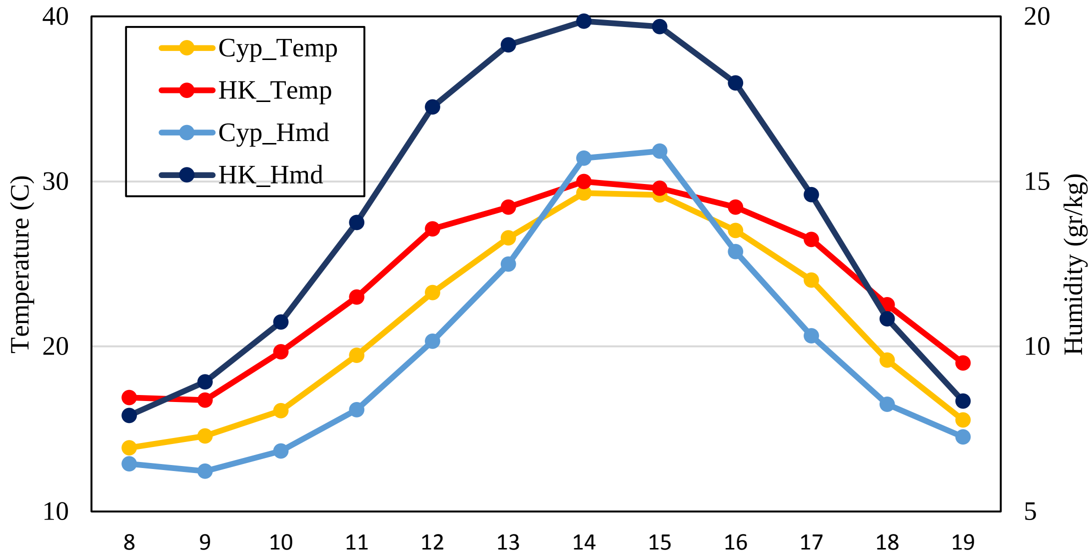


Figure 3-2 Monthly temperature and humidity average

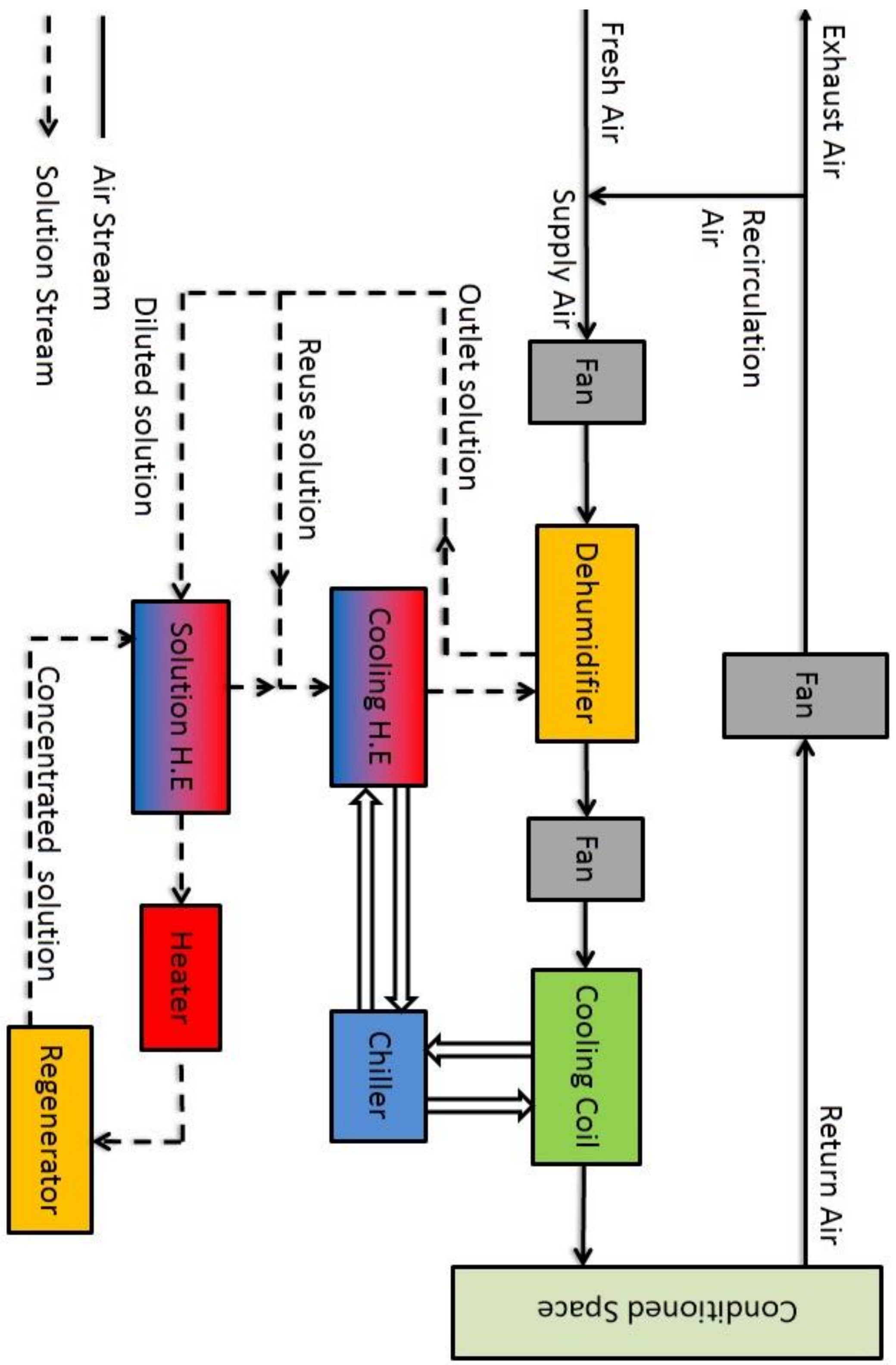


Figure 3-3 Schematic diagram of the DCS Cycle

CHAPTER 4

BUILDING MODEL

This chapter is consists of two major section. The first is building Google Sketchup model which is only the geometrical model. The second is TRNBuild model which includes thermal properties and details. Building geometry is defined in Google Sketchup software (with SKB and SKP file formats) and then imported to TRNBuild through TRNSYS Plugin as IDF file format. TRNBuild is a part of TRNSYS platform to create or develop a thermal space/s and its detailed properties. At the environment, thermal properties of built material of the building is specified by adopting its vast library. Also, ventilation and infiltration air flow rates, all internal thermal loads next to indoor design temperature and humidity can be specified. All these data are saved in b file format further will be read and used by TRNSYS through Unit56 to calculate heating/cooling loads or indoor temperature and humidity.

4.1. Google Sketchup model

Considered conditioned structure is a sport center with 33 meters length, 24 meters width and 3.5 meters height, Figure 4-1. Also there exist an unconditioned part of building at its eastern side, hence all surfaces of conditioned space are exposed to ambient except eastern wall which is shaded whole days. In addition, all walls except the eastern wall are equipped with a 0.6 meter height window for day lightning and the length of that is 23 meters for south and north wall but 32 meters for western wall. All details are exposed in Table 4-1.

4.2. TRNBuild model

Generated model in Google Sketchup is saved in SKB and SKP files and then converted to IDF file by TRNSYS plugin on Google Sketchup to be imported into TRNBuild to specify thermal and load properties. Even though it consists of several parts, Figure 4-2, only some of them are modified while the rest are remained as default. Affected section are zone window (Figure 4-3), Inputs (Figure 4-9) and Outputs (Figure 4-10). It should be noted that during import procedure, it is considered that moveable shadings will be opened and closed if total radiation on facade are more than 140 $W.m^{-2}$ and less than 120 $W.m^{-2}$ respectively, obtained from TRNSYS default values.

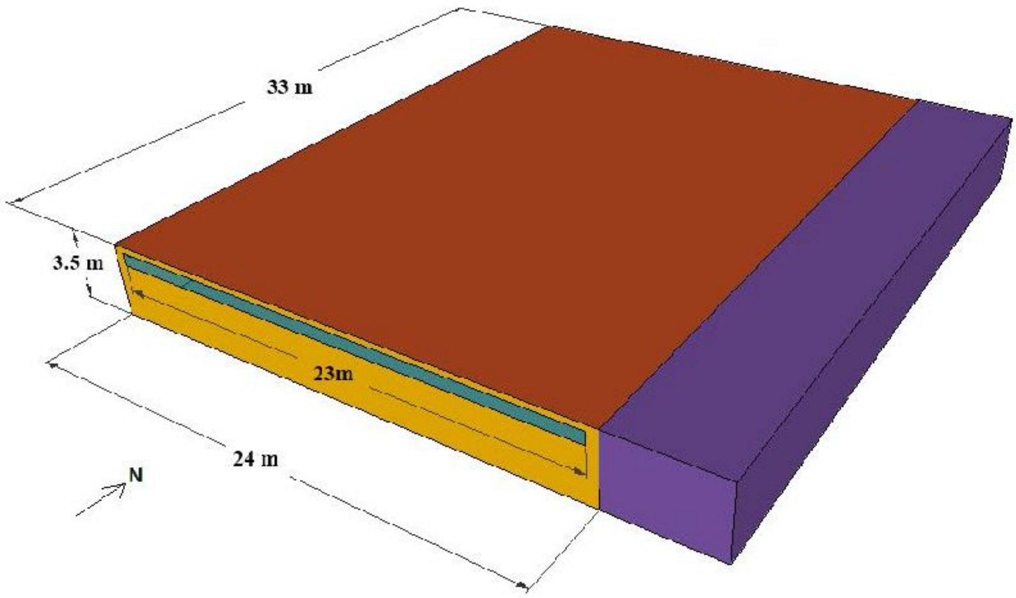


Figure 4-1Building and environment model in Google Sketchup

Table 4-1 Walls and windows dimensions

Orientation	Wall			Window		
	Length (m)	Height or Width (m)	Area (m2)	Length (m)	Height (m)	Area (m)
East	33	3.5	115.5	-	-	-
West	33	3.5	115.5	32	0.6	19.2
North and south	24	3.5	84	23	0.6	13.8
Horizontal	24	33	792	-	-	-

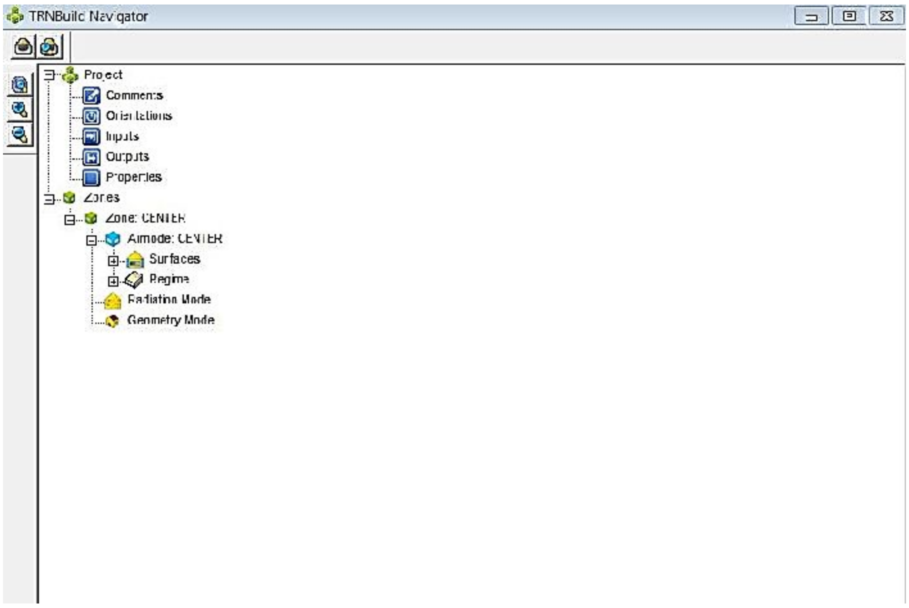


Figure 4-2TRNBuild main window

4.2.1. Zone window

Zone window, Figure 4-3, includes envelop thermal properties, type manager, regime data and some other parameters.

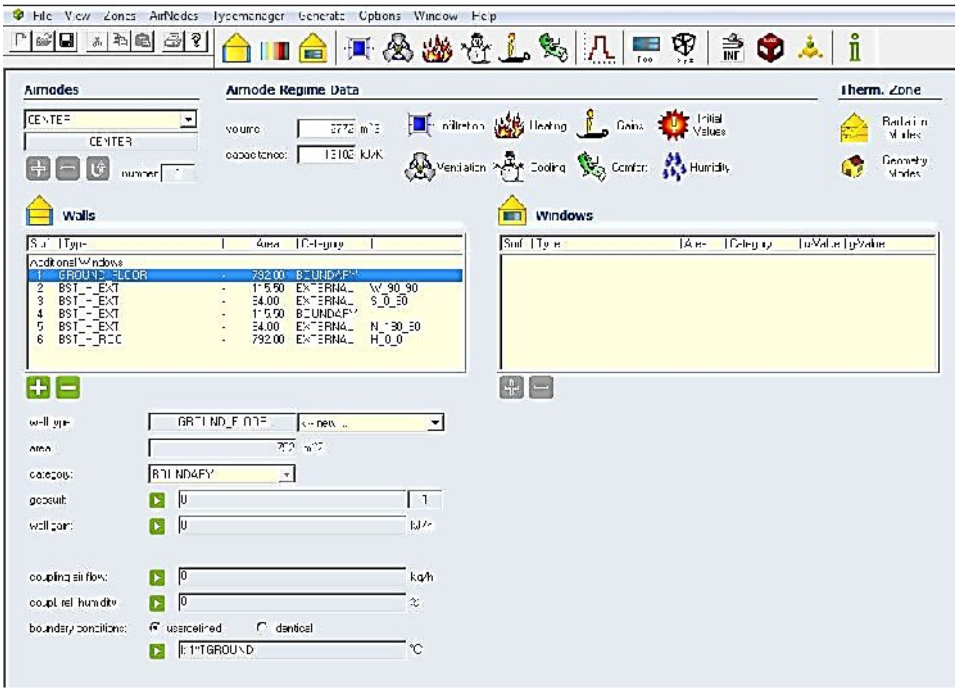


Figure 4-3 Zone window

4.2.1.1. Envelop thermal Properties

At zone window, walls, roof, floor and windows construction material properties can be specified manually or from TRNBuild vast library and the latter is adopted at here. Walls and windows areas, exposed conditions and materials are summarized in Table 4-2.

Table 4-2 Building envelop thermal properties

Orientation	Wall			Window	
	Area	Bound. Cond.	Const. Type	Area	Const. Type
East	115.5 m^2	Identical	BST_H_EXT	-	-
North and South	84 m^2	External	BST_H_EXT	13.8 m^2	INS2_Ar_2
West	115.5 m^2	External	BST_H_EXT	19.2 m^2	INS2_Ar_2
Roof	792 m^2	External	BST_H_ROO	-	-
Floor	792 m^2	Input	Ground	-	-

At here, BST_H_EXT and BST_H_ROO stand for Bestest Heavyweight External wall and roof respectively. Also INS2_Ar_2 is two-layer window filled with Argon gas between them. Detailed properties of walls and windows are available at “Wall type manager” and “Window type manager” parts. Since there exist an unconditioned space at the right side of the conditioned one, eastern wall boundary condition is set to “Identical” meaning that the temperature at both side of the wall is the same and hence no heat transfer. Other walls and roof boundary condition are set to “External” which means the outer side of wall is ambient condition and that transfer through them depends on ambient conditions. Finally, floor boundary condition is considered as “Input” meaning that its temperature further will be specified in TRNSYS and inputted into building model.

4.2.1.2. Required regime data

4.2.1.2.1. Volume and Thermal capacitance

The first parameter is building volume which is filled automatically, 2772 m^3, because 3D building geometry is imported to TRNBuild at first step.

The second parameter is thermal capacitance which is defined as:

$$C = V_{building} \times \rho_A \times C_{p,A} \times f \qquad (4-1)$$

Indoor condition is going to be set 24^{o}C and 50% relative humidity. Density and specific heat of air at this condition is about 1.19 kg.m^{-3} 1.01 kJ.kg^{-1}.K^{-1} respectively. Full air properties at indoor state is displayed at Table 4-3. Also a factor of 4 is considered for weight of appliance which is a good assumption for a sport center. Thus:

Capacitance = 2772 m^{3} x 1.17 kg.m^{-3} x 1.01 kJ.kg^{-1} x 4 = 13102 kJ.K^{-1}

Table 4-3 Air properties at indoor state

Pressure (atm)	Tempe rature (^{o}C)	Relative humidity (%)	Humidity ratio (gr.kg^{-1})	Specific enthalpy (kJ.kg^{-1})	Specific heat (kJ.kg^{-1}.K^{-1})	Density (kg/m^{-3})
1	24	50	9.3	47.79	1.01	1.17

4.2.1.2.2. Infiltration

Infiltration, also known as air leakage, is the flow of outdoor air into a building through cracks and other unintentional openings, driven usually by natural pressure difference and is undesirable from energy conservation point of view while heating and cooling equipment operate. At current model infiltration considered zero because of a positive pressure inside building, resulted from high-flow ventilation system, hence it is set to off.

4.2.1.2.3. Heating and Cooling

For heating and cooling purposes, not only heating but also cooling regime data is set to off. Because by turning them on, TRNBuild calculates heating/cooling loads according to defined set temperature and humidity and no external system can be connected to structure but in off mode any external heating/cooling system can be linked to building through ventilation air.

4.2.1.2.4. Gains

This section provides data related to indoor heat gains and comprises gains from people, computer, lighting and others resources and is shown in Figure 4-4. In current model, gain from computer and other sections are set off while that from people and lights are described below.

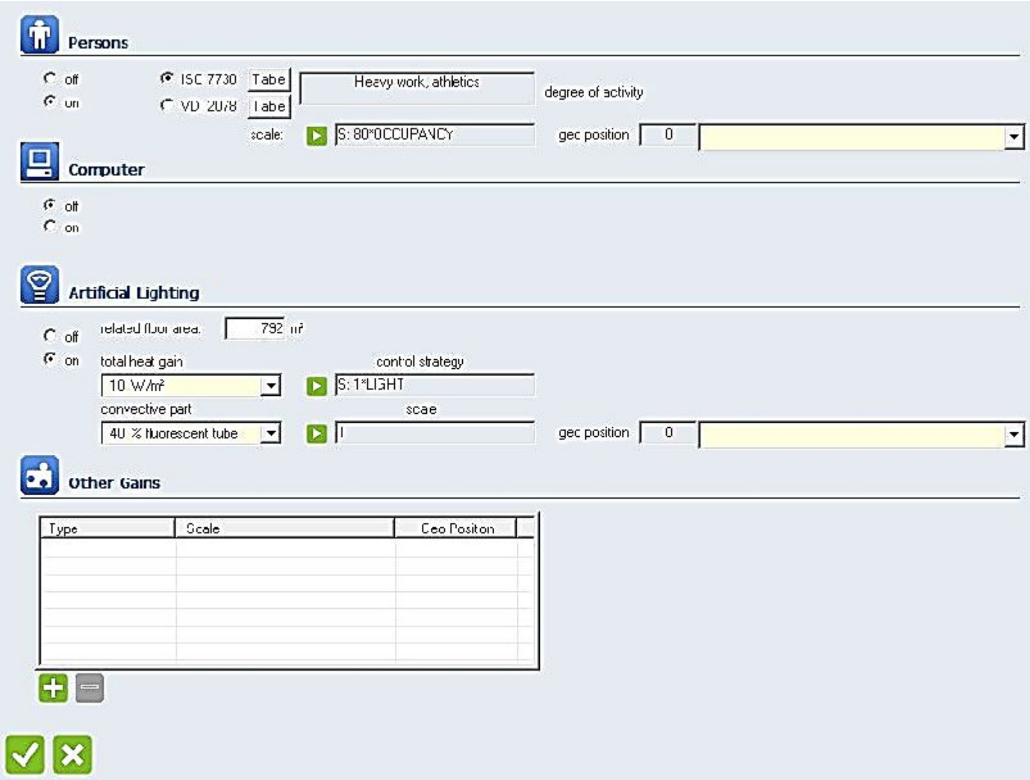


Figure 4-4 Gain window

People

For unknown number of occupants, the reference [66] suggests 10 people per 1000 ft^2 (or 10 people per 100 m^2)of floor area. Thus it is assumed 80 person do sport between 9 A.M. and 8 P.M. To do this, an occupancy schedule is created which is 0 people between hours 0 to 9, 80 people between hours 9 to 20 and again 0 people between hours 20 to 24. For type of activity, “Heavy work, athletic” is the suitable choice from available table, Figure 4-5.

Lighting

Lightning plays important role in determining cooling load and sufficient accuracy is required. One good estimation of lighting power in a space, if it is unknown, is to use ASHRAE Fundamental handbook [4] which provides required lighting power per square meter. According to that, required wattage per square meter is 10 Watt for a sport center hall. Also, type of the adopted light is fluorescent tube with 40% convective heat emission and 60% radiative. In addition, a schedule is created which is turns lights on from 18 to 20. All these inputs are demonstrated in Figure 4-4.

Rates of Heat Gain from Occupants of Cond

No.	Degree of Activity	Typical Application	Total Heat Adjusted		Sensible Heat		Latent Heat	
			Watts	Btu/h	Watts	Btu/h	Watts	Btu/h
01	Seated at rest	Theatre, Movie	100	350	60	210	40	140
02	Seated, very light writing	Office, Hotels, Apts	120	420	65	230	55	190
03	Seated, eating	Restaurant	170	580	75	255	95	325
04	Seated, light work, typing	Office, Hotels, Apts	150	510	75	255	75	255
05	Standing, light work or working slowly	Retail Store, Bank	185	640	90	315	95	325
06	light bench work	Factory	230	780	100	345	130	435
07	walking 1,3 m/s (3 mph) light machine work	Factory	305	1040	100	345	205	695
08	Bowling	Bowling Alley	280	960	100	345	180	615
09	moderate dancing	Dance Hall	375	1280	120	405	255	875
10	Heavy work, lifting Heavy machine work	Factory	470	1600	165	565	300	1035
11	Heavy work, athletics	Gymnasium	525	1800	185	635	340	1165

Figure 4-5 Rates of heat gain from occupants in TRNBuild

4.2.1.2.5. Ventilation

Ventilation air is a combination of outdoor air and recirculation air with the aim of providing suitable temperature and humidity next to required fresh air of structure, Figure 4-6 .

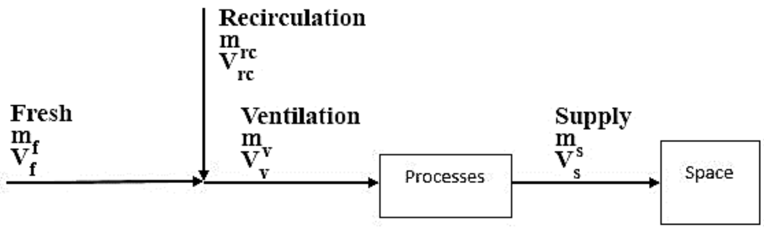


Figure 4-6 Ventilation system diagram

Indoor design temperature and relative humidity for air conditioning are set to 24 ^{o}C and 50% respectively [66]. It results in 9.3 gr/kg humidity ratio if air pressure is considered 1 atm (101325 Pa) because altitude of considered location is low enough to be ignored. Minimum humidity ratio of supply air is considered 7 gr.kg^{-1} because lower values requires chilled water temperature below 5^{o}C which is difficult to obtain with regular cooling units. According to calculated maximum latent cooling load, which is not described here for brevity, and supply air humidity ratio, 7 gr.kg^{-}1, required supply air flow rate is calculated by Eq. (4-2):

$$\dot{v}_s = \frac{q_{lat}}{g_{cs} - g_s} \times \frac{T}{\rho_{stn}(273 + t_{stn})h_{fg}} \qquad (4-2)$$

Inserting values into the equation, supply air flow rate would be 3.72 m^3s^{-1}.

For ventilation system design it is assumed that supply air temperature is 18° C with properties in Table 4-4, so supply air mass flow rate would be:

$$\dot{m}_s = \dot{v}_s \times \rho_s = 4.46$$

Table 4-4 Supply air state

Pressure (atm)	Temperature (°C)	Relative humidity (%)	Humidity ratio ($gr.kg^{-1}$)	Specific enthalpy ($kJ.kg^{-1}$)	Specific heat ($kJ.kg^{-1}.K^{-1}$)	Density (kg/m^{-3})
1	18	55	7	35.93	1.02	1.2

Since mass rate of dehumidified vapor is negligible, it can be assumed that mass flow rate of ventilation air is the same with supply air, so that ventilation mass flow rate is 4.46 $kg.s^{-1}$.

Minimum outdoor fresh air flow rate is calculated according to ASHRAE standard 62.1 [67] as described below.

$$\dot{v}_{bz} = R_p P_z + R_a A_z \qquad (4-3)$$

Where

$\dot{v}_{bz}$ is breathing zone minimum outdoor air flow ($L.s^{-1}$)
R_p is required air flow rate per person ($L.s^{-1}.person^{-1}$)
P_z is number of people (-)
R_a is required air flow rate per square meter ($L.s^{-1}.m^{-2}$)
A_z is floor area (m^2)

Table 4-5 Fresh air requirement table [67]

Occupancy category	People outdoor air rate R_p ($L.s^{-1}.person^{-1}$)	Area outdoor air rate R_a ($L.s^{-1}.m^{-2}$)	Default values	
			Occupancy density (#/100 m^2)	Outdoor air rate (L.s-1.person-1)
Health club / weight room	10	0.3	10	13

According to values of Table 4-5,

$$\dot{v}_{bz} = (10 * 80) + (792 * 0.3) = 1037\left(\frac{L}{s}\right) = 1.037\left(\frac{m^3}{s}\right)$$

Zone outdoor air flow is:

$$\dot{v}_{oz} = \frac{\dot{v}_{bz}}{E_z} \qquad (4-4)$$

Where E_z is air distribution effectiveness. For ceiling supply of cool air, E_z is 1 [67], thus:

$$\dot{v}_{oz} = 1.037\ \frac{m^3}{s}$$

And finally for a single zone structure, outdoor fresh air flow rate is:

$$\dot{v}_f = \dot{v}_{oz} = 1.037\ \frac{m^3}{s}$$

While energy analysis are done according to metrological data sets (in TMY2 format) of desired location, ventilation system design is based on outdoor design condition which is adopted from ASHRAE [4] for the city of Larnaca, which is the most close, available city, Table 4-6.

Table 4-6 Outdoor air properties

Pressure (atm)	Temperature (^{o}C)	Relative humidity (%)	Humidity ratio (gr.kg^{-1})	Specific enthalpy (kJ.kg^{-1})	Specific heat (kJ.kg^{-1}.K^{-1})	Density (kg/m^{-3})
1	33	43	13.5	67.94	1.03	1.13

Thus fresh air mass flow rate is:

$$\dot{m}_f = \dot{v}_f \times \rho_f = 1.037 * 1.13 = 1.17\ \frac{kg}{s}$$

Hence, recirculation mass flow rate is:

$$\dot{m}_s = \dot{m}_f + \dot{m}_{rc} \rightarrow \dot{m}_{rc} = \dot{m}_v - \dot{m}_s = 4.46 - 1.17 = 3.29\ \frac{kg}{s}$$

According to the definition of recirculation air, it comes back from conditioned space and hence adopts states of indoor air. From Table 4-3, density is 1.17 kg.m^3 which results in 2.81 m^3.s^{-1} of volumetric flow rate of recirculation air.

Finally ventilation air volumetric flow rate would be:

$$\dot{v}_v = \dot{v}_{rc} + \dot{v}_f = 2.81 + 1.04 = 3.85\ m^3.s^{-1}$$

This magnitude of air, as a combination of outdoor fresh and recirculation airs, goes toward dehumidifiers to be dehumidified down to 7 g.kg^{-1} of humidity ratio and suitable temperature. Mass and volumetric flow rates of all streams are summarized in Figure 4-7.

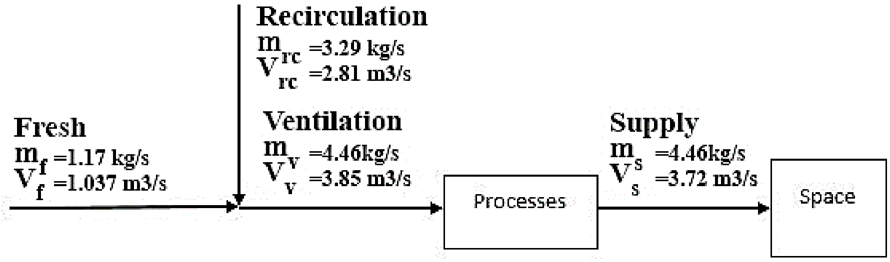


Figure 4-7 Ventilation system design values

While only supply air volumetric flow rate is used in ventilation section of TRNBuild model, its other properties further will be obtained from TRNSYS at both DCS and VCCS cycle models through inputs as will be described in Section 5.3.2.1. Supply flow rate in TRNBuild is specified as air change per hour (ACH), Figure 4-8, defined in Eq. (4-5) and is calculated 4.83 h^{-1}.

$$ACH = \frac{\dot{v}_s \times 3600}{V_{buil}} \qquad (4-5)$$

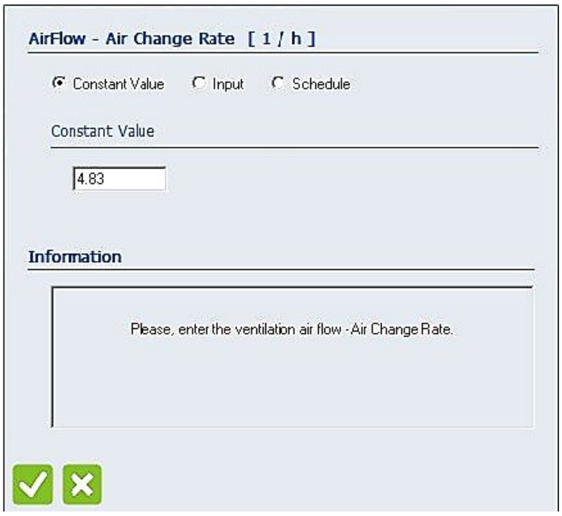


Figure 4-8 Ventilation window

4.2.1.2.6. Initial values and Humidity

Indoor air initial values, used only for calculations start point, are set to 24^{o}C and 50% relative humidity (comfort condition) and it only affects few initial hours of transient model analyses. Humidity model is also set to "Simple humidity model".

For more information about humidity model readers can refer to TRNSYS documentation [68].

4.2.1.3. Radiation and Geometry mode

All options of Radiation window are set to “Standard model” because “Detailed model” results in very high amounts of calculation, followed by running time increment. For more information about detailed and standard models and their difference readers can refer to the reference [68]. Geometry model is set to “3D data” because initial geometry model is created in 3D by Google Sketchup and imported to TRNBuild. If no 3D model is adopted, other options should be opted.

4.2.2. Inputs

Defined input at here are used at different places of TRNBuild model but their value will be specified further in TRNSYS model. In fact, their purpose is to import those transient, required input data into TRNBuild model which are not currently available and will be calculated in TRNSYS environment at run time such as solar irradiance angle or ground temperature which depends on ambient conditions and time of year. At the inputs window, Figure 4-9, there some standard inputs which are used by TRNBuild and cannot be removed. Also, new, required inputs such as ventilation air temperature and humidity can be defined.

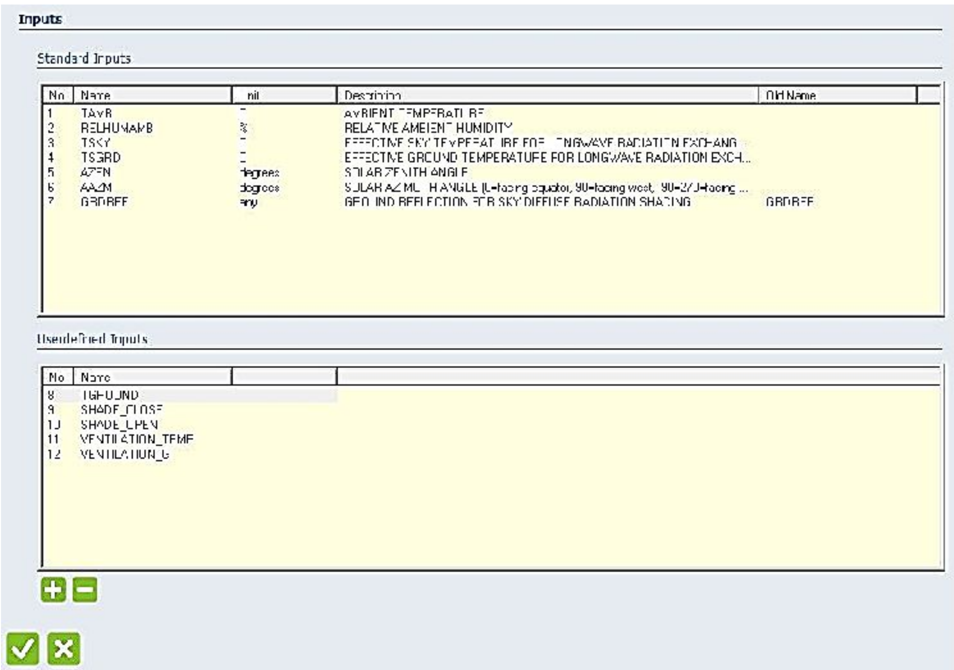


Figure 4-9 Inputs window

At used defined section, TGROUND is the ground temperature at each hour which is calculated in TRNSYS environment. SHADE_CLOSE and SHADE_OPEN are feedbacks from TRNSYS for opening or closing of shades over windows. VENTILATION_TEMP and VENTILATION_G are ventilation temperature and humidity ratio for each hour depending on ambient conditions.

4.2.3. Outputs

At the outputs window, Figure 4-10, required outputs can be selected according to demand to be shown or used in TRNSYS. To do this, desired outputs have to be selected from possible outputs menu, Figure 4-11.

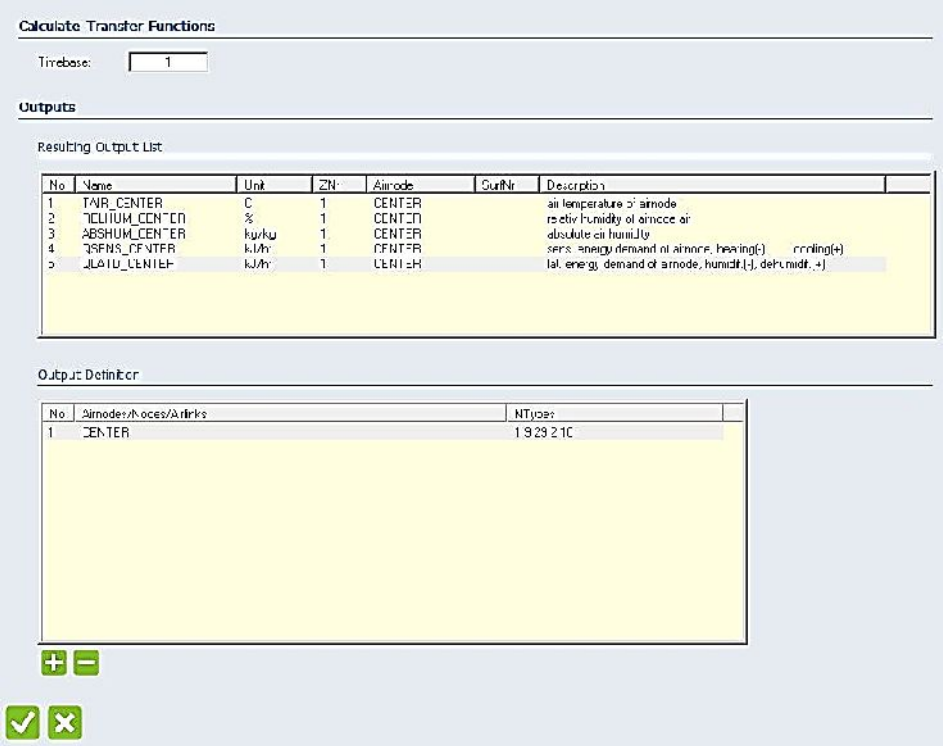


Figure 4-10 Outputs window

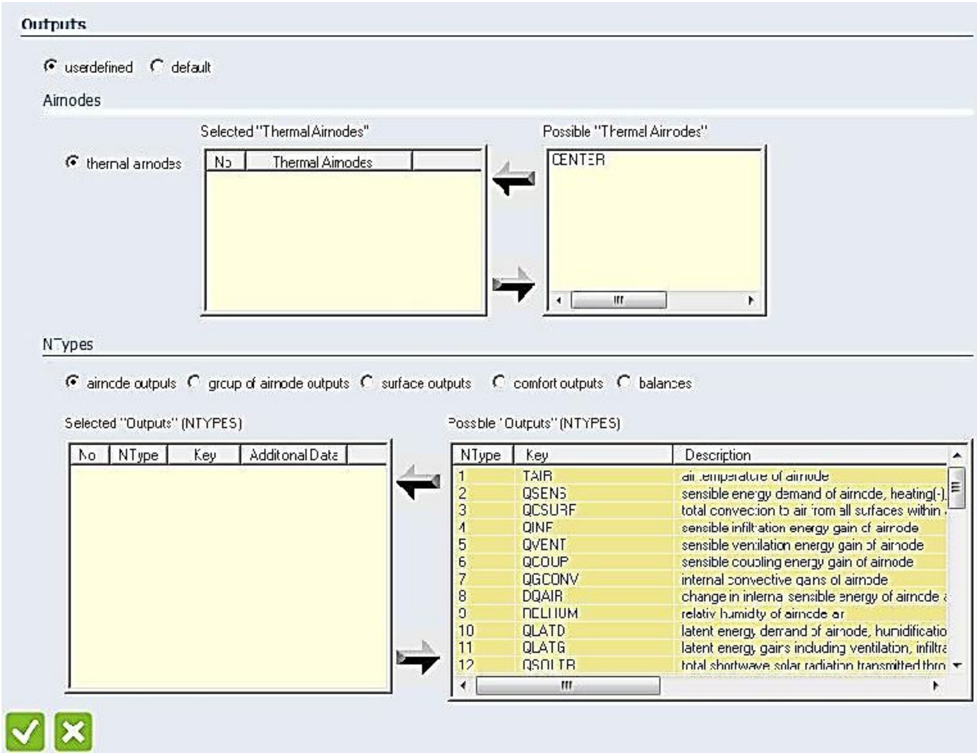


Figure 4-11 Available outputs menu window

CHAPTER 5

CONVENTIONAL COOLING CYCLE MODEL

At this section, conventional cooling cycle is modeled in TRNSYS environment and connected the structure described in Chapter 4: Building model with the aim of providing suitable temperature and humidity. The schematic diagrams of conventional cooling cycle is shown in Figure 5-1. In the cycle, specific amount of return and outdoor fresh air are mixed and brought into contact with chilled water of chiller through at cooling coil where air temperature drops significantly with the results of moisture condensation (mechanical dehumidification). Then dehumidified air is reheated in reheater up to desired temperature, depending on conditions and sensible cooling load and finally is fed into space.

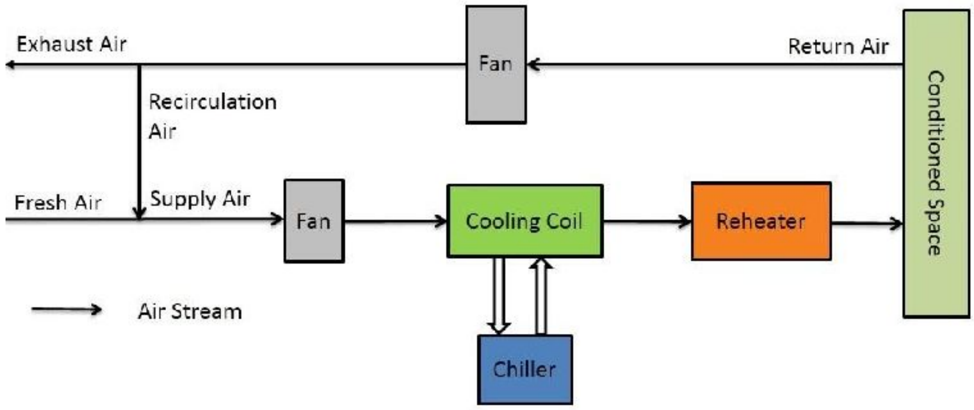


Figure 5-1 Schematic diagram of conventional cooling cycle

5.1. Intro into TRNSYS

A TRNSYS model is made out of several individual component, called Types. These models of individual components are then connected for information flow between them in a larger system run. Each Type (component) is described by a mathematical model in the TRNSYS simulation engine.

Each Type consists of three main windows, “Parameters”, “Inputs” and “Outputs”. Parameter window of each Type provides users to specify the constant physical characteristic of that component. As an example, Figure 5-2 shows the parameter window of a fan (Type 744) with editable constant properties such as motor efficiency, motor heat loss to air and etc. While this sample is shown in picture mode, parameter window of all other Types are shown in tables like Table 5-1 which shows Figure 5-2 in table mode.

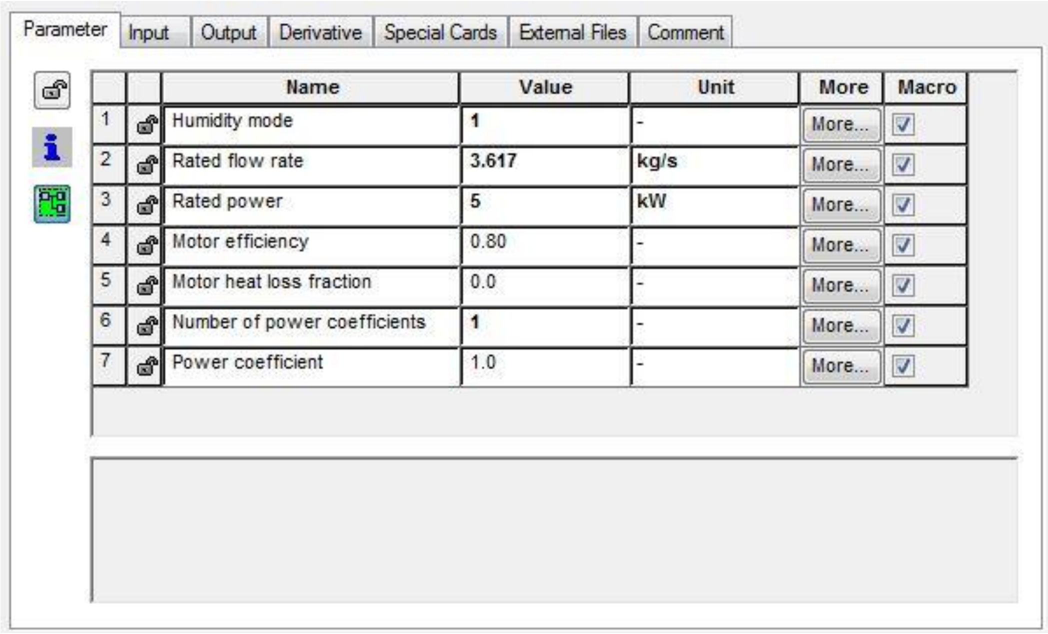
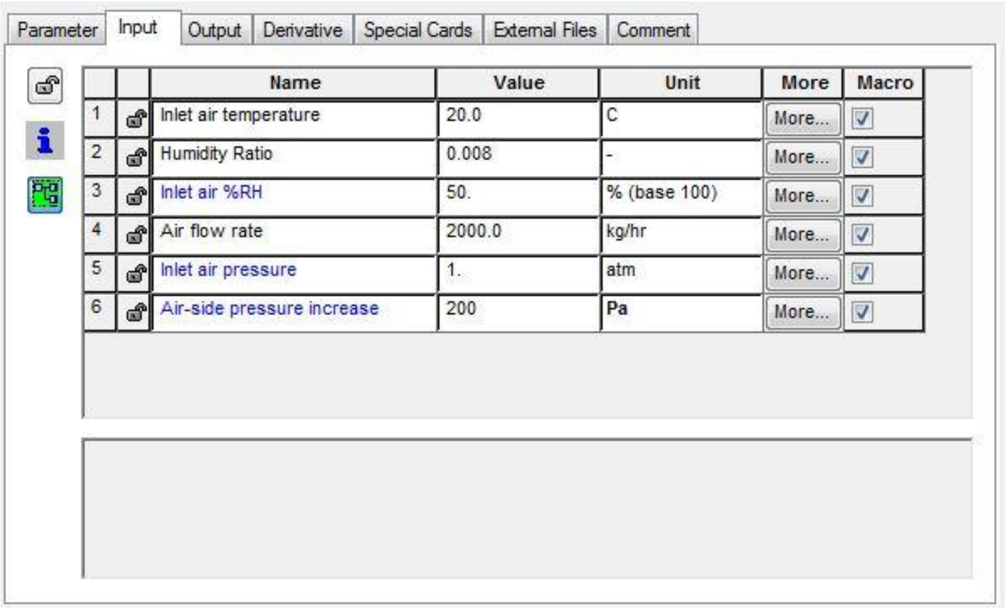


Figure 5-2 Parameter window sample

Table 5-1 Table of parameter window sample

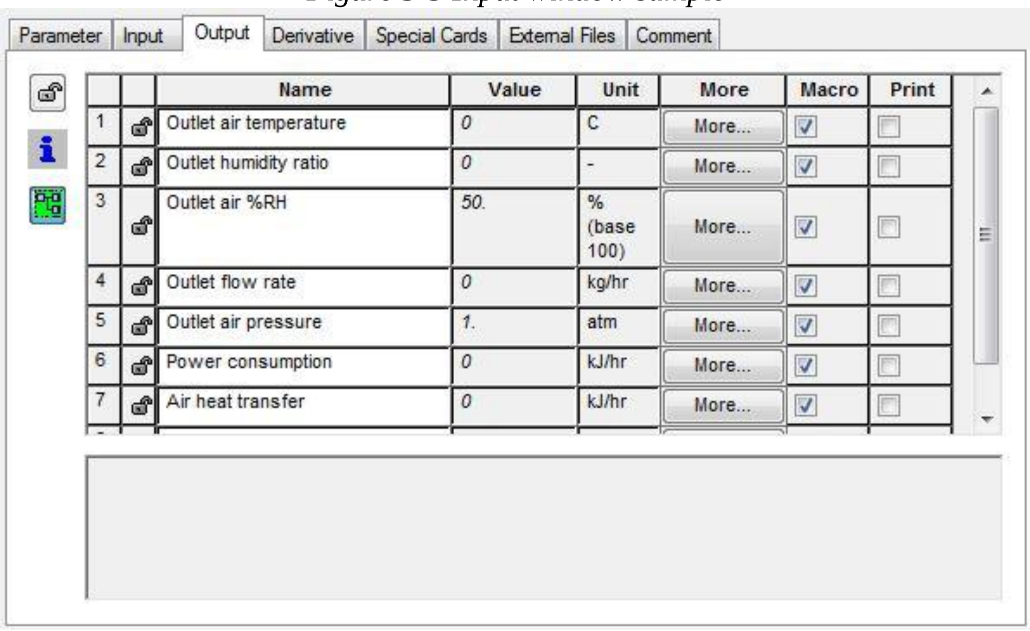
Name	Value	Unit
Humidity mode	1	-
Rated flow rate	3.617	kg/s
Rated power	5	kW
Motor efficiency	0.8	-
Motor heat loss fraction	0	-
Number of power coefficients	1	-
Power coefficient	1	-

Inputs window shows the required input data for a Type which are the output of other Types. Figure 5-3 displays the input window for a fan component (Type 744). Unconnected inputs (blue names at input window) remain constant at specified values. Similarly, Output window displays available output from a Type which could be connected to another Type as their inputs. The output window for Type 744 is shown in Figure 5-4.



	Name	Value	Unit	More	Macro
1	Inlet air temperature	20.0	C	More...	
2	Humidity Ratio	0.008	-	More...	
3	Inlet air %RH	50.	% (base 100)	More...	
4	Air flow rate	2000.0	kg/hr	More...	
5	Inlet air pressure	1.	atm	More...	
6	Air-side pressure increase	200	Pa	More...	

Figure 5-3 Input window sample



	Name	Value	Unit	More	Macro	Print
1	Outlet air temperature	0	C	More...		
2	Outlet humidity ratio	0	-	More...		
3	Outlet air %RH	50.	% (base 100)	More...		
4	Outlet flow rate	0	kg/hr	More...		
5	Outlet air pressure	1.	atm	More...		
6	Power consumption	0	kJ/hr	More...		
7	Air heat transfer	0	kJ/hr	More...		

Figure 5-4 Output window sample Model

An important step in simulating the operation is specifying the values of transient variables between the output(s) of one component to input(s) of another one. This is done by using a link between two components, so that information flow can occur. To specify the details of the link between two components, the user should use the graphical user interface (GUI) opened by double-clicking on the link, shown in

Figure 5-5. To specify the connection between two components, the user connects the outputs of the first component (left-side) to the required inputs of the second component (right-side).

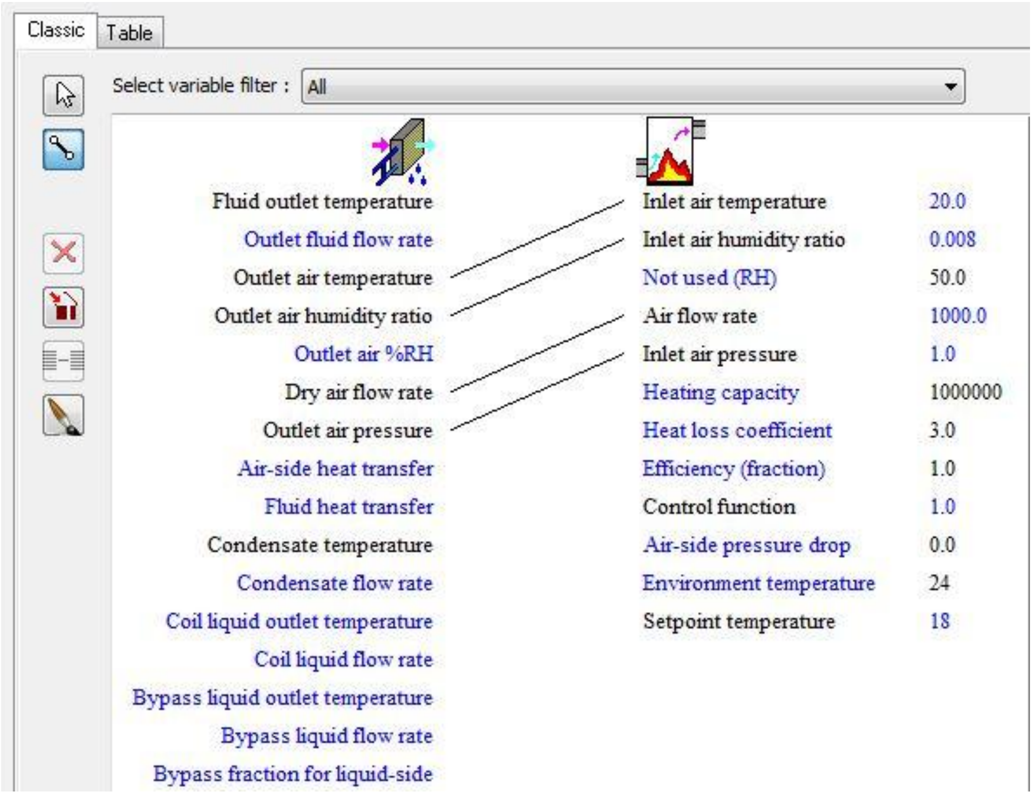


Figure 5-5 Sample connection window between cooling coil and reheater

5.2. Model's working principle

Figure 5-6 demonstrates current work TRNSYS model where Types and information flows are demonstrated by icons and links between them respectively. Information flow could be real mass cycle path (continuous lines) or non-real information flow between two components (dotted lines). Connections in Figure 5-6 are categorized as below:

- Black continuous lines: air flow path
- Blue continuous lines: chilled water flow path
- Black dotted lines: information flow between components
- Purple dotted lines: Iteration cycle for Type 22
- Grey dotted lines: Results of iterations from Type 22
- Light blue dotted lines: Schedule data for the components
- Red dotted line: For unit conversion
- Light green dotted lines: Outputs to be graphed or saved in Excel format

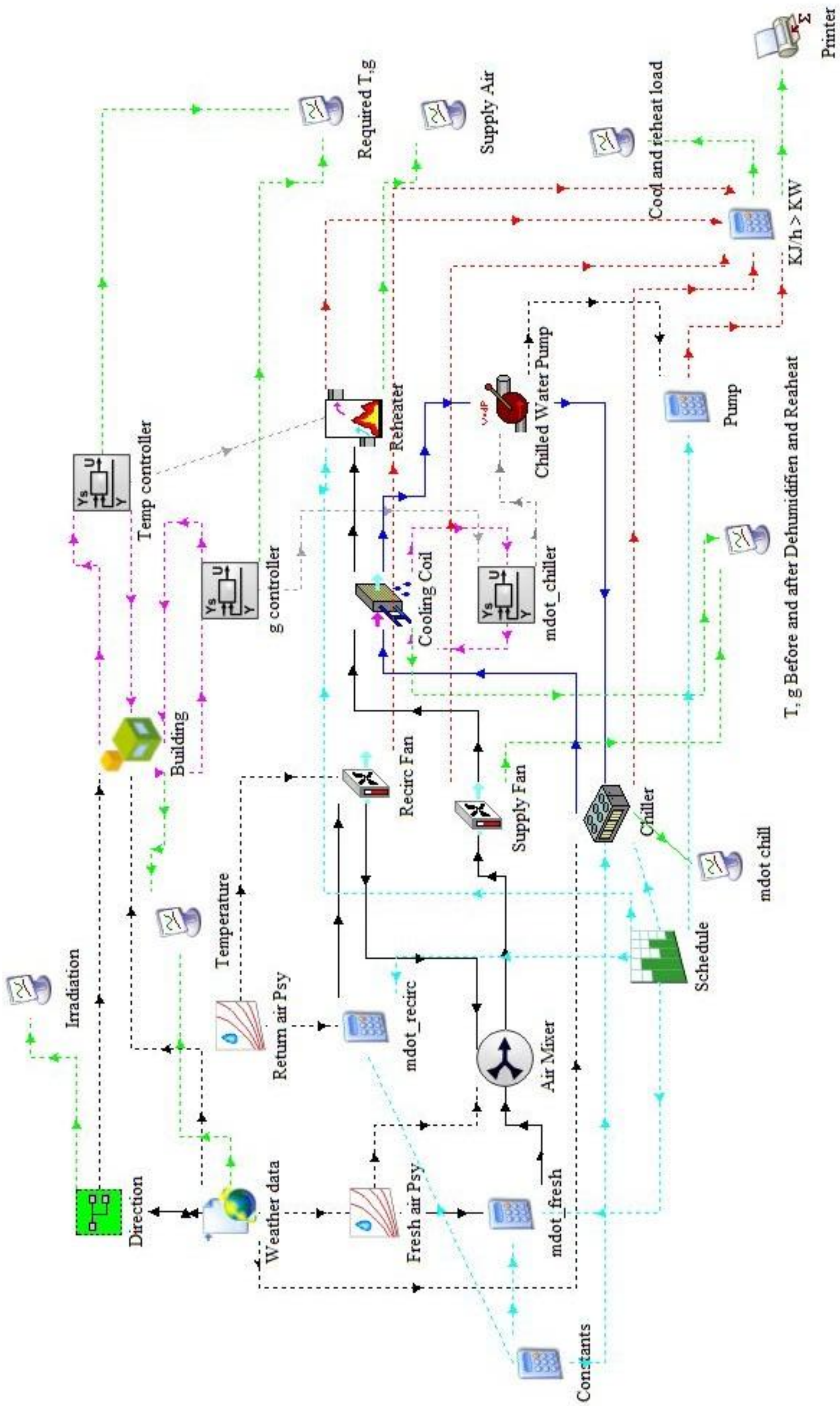


Figure 5-6 TRNSYS diagram of conventional cooling cycle

Recirculation and fresh air enters the ventilation cycle from “mdot_recirc” and “mdot_fresh” components respectively and based on air mass flow rate calculations in Building modelWhile the recirculation air temperature and humidity is fixed at indoor design condition, those of fresh air are obtained from “Weather data” component. However other psychometric properties of both streams are obtained by utilizing two psychometric components, Type 33c, named “Return air Psy” and “Frsh air Psy”. After passing recirculation fan, Type 744, recirculation air is mixed with fresh air at “Air mixer”, Type 11g, which gives mixture flow rate next to its state. After that, the mixture is passed “Supply Fan” to move toward “Cooling coil”, Type 508d, where it got cold and dehumidified down to required magnitude of humidity ratio. Then, air continuous its way to “Reheater”, Type 121a, to warm up to desired temperature. Demanded values for cooling coil and reheater are determined by two iterative feedback controllers, Type 22, are described further.

The “Building” icon, Type 56, is the building model described in previous chapter and in fact this is a connection between TRNBuild and TRNSYS. The Type only calculates average indoor air temperature, Eq. (5-3), and humidity ratio, Eq. (5-6), by the use of energy and mass balance phenomena at each time step but not required ventilation air states. Utilized Types are listed in Table 5-2.

Table 5-2 List of conventional cooling cycle components in TRNSYS model

Name	Type	Note
Building	56	Designed building in TRNBuild
Ground Temperature	77	Determining ground temperature
Temp Controller	22	Calculating supply air temperature
g controller	22	Calculating supply air humidity
Direction	Macro	Sun irradiance on different directions of building
Weather data	15-2	Importing TMY2 weather data
Return air Psy	33c	Recirculation air properties
Fresh air Psy	33c	Fresh air properties
mdot_recirc	Equation	Calculating recirculation air mass flow rate
mdot_fresh	Equation	Calculating fresh air mass flow rate
Air Mixer	11g	Mixing fresh and recirculation air
Recirc Fan	744	Fan for recirculation air

Table 5-2 (Continued)

Supply Fan	744	Fan for supply air
Cooling coil	508d	Cooling and dehumidifying of air to required humidity ratio
mdot chilling	22	Calculating required chilled water mass flow rate
Reheater	121a	Reheating of supply air
Chilled Water Pump	742	Water pump between chiller and cooling coil
Chiller	655	Providing chilled water
Schedule	14h	Providing schedule for the units
Pump	Equation	Applying Pump schedule
kJ/h>kW	Equation	Converting kJ per hour to kW
Constant	Equation	Providing constant values for parametric studies
Printer	46b	Saving output data in excel format
Required T,g	65d	Graphing required temperature and humidity ratio of supply air
Supply air	65d	Graphing temperature and humidity ratio of supply air
Cool and reheat load	65d	Graphing consumed power of each component
T,g before and after cooling coil	65d	Graphing States of air before and after cooling coil

Indoor air temperature at the end of any time step is:

$$T_\tau = T_{\tau-\Delta\tau} + \frac{Q_{sen,\Delta\tau}}{C} \qquad (5-1)$$

Since indoor air temperature variation is linear, the average temperature during time period would be:

$$T_{avg} = \frac{T_\tau + T_{\tau-\Delta\tau}}{2} \qquad (5-2)$$

Combining Eq. (5-1) and Eq. (5-2) results in:

$$T_{avg} = \frac{2T_{\tau-\Delta\tau} + Q_{sen,\Delta\tau}}{2C} \qquad (5-3)$$

Indoor air humidity ratio at the end of any time step is:

$$g_{\tau} = g_{\tau-\Delta\tau} + \frac{Q_{lat,\Delta\tau}}{h_{evap}} \qquad (5-4)$$

In similar way:

$$g_{avg} = \frac{g_{\tau} + g_{\tau-\Delta\tau}}{2} \qquad (5-5)$$

Thus:

$$g_{avg} = \frac{2g_{\tau-\Delta\tau} + Q_{lat,\Delta\tau}}{2h_{evap}} \qquad (5-6)$$

$Q_{sens,\Delta\tau}$ and $Q_{lat,\Delta\tau}$ include ventilation system, infiltration, internal gains and gains through walls and windows. Thus average indoor air temperature and humidity ratio could be set to their design values (25°C and 9.3 $gr.kg^{-1}$) by controlling ventilation air temperature and humidity ratio. To do this, two iterative feedback controllers, Type 22, are adopted and named "Temp controller" and "g controller". Obtained values are then forwarded to cooling coil and reheater in order to preparing desired supply air.

5.3. Types and connections

Types and connections between them are divided into three man groups, namely Non-cycle, air cycle and chilled water cycle.

5.3.1. Non-cycle components

This section involves some components that are not enumerate any cycle but are used a lot in accordance with other components.

5.3.1.1. Building

The first component is simulated building model and all the parameter window values are left to default, Table 5-3.

Table 5-3 Building parameter window values

Name	Value	Unit
Logical unit for building description file	31	-
Star network calculation switch	1	-
Weighting factor for operative temperature	0.5	-

Several data from variety of components are inputted into this Type, as demonstrated in Table 5-4. Ambient air temperature and humidity, effective sky temperature and ground reflectance are inputted data from “Weather”. “Direction” macro provides solar azimuth angles to be used by building. Also ventilation air temperature and humidity ratio are imported from “Temp controller” and “g controller” respectively. All these data next to some other, provided by TRNBuild, are used at each time step to calculate indoor air and humidity which are outputs of the Type, Table 5-5.

Table 5-4 Building input

Output		**Building Input**
From	**parameter**	
Weather	Dry bulb temperature	AMB
Weather	PERCENT RELATIVE HUMIDITY	RELHUMAMB
Weather	EFFECTIVE SKY TEMPERATURE	TSKY
Weather	Dry bulb temperature	TSGRD
Direction	Azimuth angel->AAZM_TYPE 56	AAZM
Weather	Ground reflectance	GRDREF
Temp controller	Control signal	VEMTILATION_TEMP
g controller	Control signal	VENTILATION_G

Table 5-5 Building output

Building output	**Input**	
	To	**parameter**
TAIR_CENTER	Temp controller	controlled variable
ABSHUM_CENTER	g controller	controlled variable

As mentioned above, one iterative feedback controller is used for specifying required temperature, named “Temp controller” and one for humidity ratio, called ”g controller” while both are connected to Type 56 as shown in Table 5-4 and Table 5-5. Building temperature output (TAIR_CENTER), defined in Chapter 4: Building model as output, is connected to “control variable” input of “Temp controller” and “control signal” output from “Temp controller” is connected to “VENTILATION_ TEMP” input of building, defined in Chapter 4: Building model as input. Thus, at each time step Type 22 iterates “control signal” toward building to get different “control variable” values from building till the latter become equivalent to “Setpoint value”, hence “Setpoint value” of Type 22 is set to indoor design condition which is 24^{o}C. Minimum and maximum control signals are also set to 5 and 25 respectively. The same connections exists between building and the other Type 22, “g controller”, but set point is set to 9.3 and minimum and maximum control signals to 0 and 0.02 respectively.

5.3.1.2. Direction

Direction is a macro with the purpose of calculating solar azimuth angle at each time step for building load. It is made out of three components, Figure 5-7, which are created by TRNSYS automatically during building import procedure and all values are default. Table 5-6 and Table 5-7 show import and output connections.

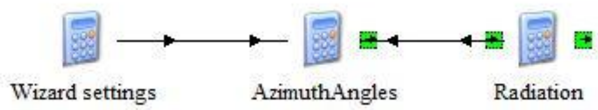


Figure 5-7 Direction macro components

Table 5-6 Direction input

Output		Direction Input
From	parameter	
Weather data	Solar zenith angle	Radiation->Input1
Weather data	Solar azimuth angle	Radiation->Input2
Weather data	Total tilted surface radiation for surface-1	Radiation->Input3
Weather data	Beam radiation for surface-1	Radiation->Input4
Weather data	Angle of incidence for surface-1	Radiation->Input5
Weather data	Total tilted surface radiation for surface-2	Radiation->Input6
Weather data	Beam radiation for surface-2	Radiation->Input7
Weather data	Angle of incidence for surface-2	Radiation->Input8
Weather data	Total tilted surface radiation for surface-3	Radiation->Input9
Weather data	Beam radiation for surface-3	Radiation->Input10
Weather data	Angle of incidence for surface-3	Radiation->Input11
Weather data	Total tilted surface radiation for surface-4	Radiation->Input12
Weather data	Beam radiation for surface-4	Radiation->Input13
Weather data	Angle of incidence for surface-4	Radiation->Input14

Table 5-7 Direction output

Direction output	Input	
	To	parameter
Azimuth angle->AAZM_TYP56	Building	AAZM

5.3.1.3. Weather Data

Type 15 is a component with external file input, meaning no input window is available for that but it reads Typical Meteorological Year (TMY2) formatted external data files. This file was obtained by METEONORM® software [69] for Guzelyurt/ Northern Cyprus. All parameter window values are left to default so is not presented here but outputs in Table 5-8.

Table 5-8 Weather data output

Weather data output	Input	
	To	parameter
Dry bulb temperature	Chiller	Ambient temperature
Dry bulb temperature	Fresh air Psy	Dry bulb temp.
Humidity ratio	Fresh air Psy	Absolute humidity ratio
Dry bulb temperature	Building	AMB
PERCENT RELATIVE HUMIDITY	Building	RELHUMAMB
EFFECTIVE SKY TEMPERATURE	Building	TSKY
Dry bulb temperature	Building	TSGRD
Ground reflectance	Building	GRDREF
Solar zenith angle	Direction	Radiation->Input1
Solar azimuth angle	Direction	Radiation->Input2
Total tilted surface radiation for surface-1	Direction	Radiation->Input3
Beam radiation for surface-1	Direction	Radiation->Input4
Angle of incidence for surface-1	Direction	Radiation->Input5
Total tilted surface radiation for surface-2	Direction	Radiation->Input6
Beam radiation for surface-2	Direction	Radiation->Input7
Angle of incidence for surface-2	Direction	Radiation->Input8
Total tilted surface radiation for surface-3	Direction	Radiation->Input9
Beam radiation for surface-3	Direction	Radiation->Input10
Angle of incidence for surface-3	Direction	Radiation->Input11

Table 5-8 (Continued)

Total tilted surface radiation for surface-4	Direction	Radiation->Input12
Beam radiation for surface-4	Direction	Radiation->Input13
Angle of incidence for surface-4	Direction	Radiation->Input14

5.3.1.4. Schedule

"Schedule" is Type 14h with the aim of providing working hour schedule for required components. At current model, just one schedule is created as Figure 5-8 and applied to all desired Types. While there is no input for this Type, output window is shown in Table 5-9.

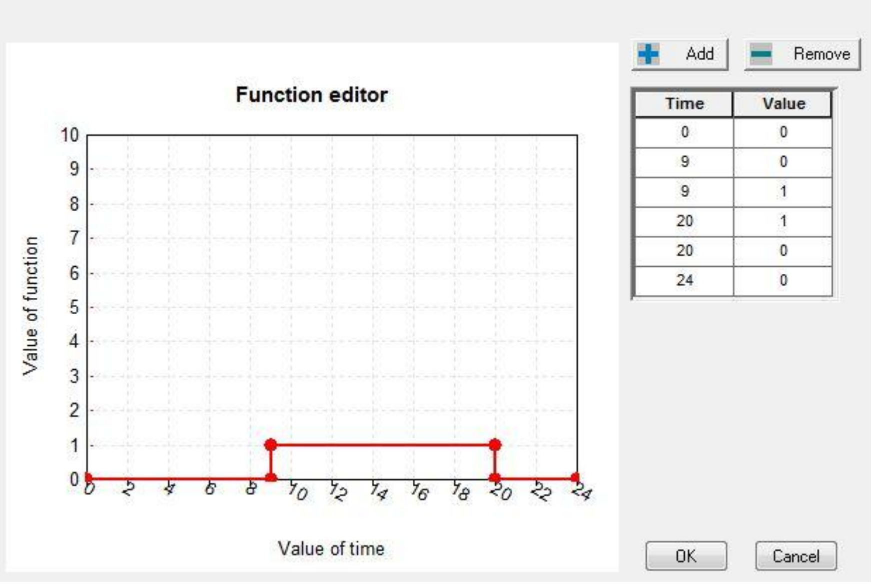


Figure 5-8 Working hour schedule graph

Table 5-9 Schedule output

Schedule output	Input	
	To	parameter
Instantaneous value of function over the time step	mdot_fresh	Schedule
Instantaneous value of function over the time step	mdot_recirc	Schedule
Instantaneous value of function over the time step	Reheater	Control function
Instantaneous value of function over the time step	Chiller	Chiller control signal
Instantaneous value of function over the time step	Pump	Schedule

5.3.1.5. Constant Equation

Since, some values are constant at model, they are defined in output section of "Constant" Equation so could be changed easily at model for parametric investigations. Constant values are as below and output window is presented in Table 5-10.

T_Chiller_set is the chilled water temperature produced by chiller and is set to 5 ^{o}C
q_fresh is the fresh air volumetric flow and is set to 1.037 $m^3.s^{-1}$
q_recirc is the recirculation air volumetric flow rate and is set to 2.81 $m^3.s^{-1}$

Table 5-10 Constant Equation output

Constant output	**Input**	
	To	**parameter**
q_fresh	mdot_fresh	q_fresh
q_recirc	mdot_recirc	q_recirc
T_chiller_set	Chiller	Set point temperature

5.3.1.6. Pump and unit conversion

Since there is no control function for "Chilled water pump", an Equation is adopted for applying schedule which multiplies pump work and schedule value, Figure 5-9.

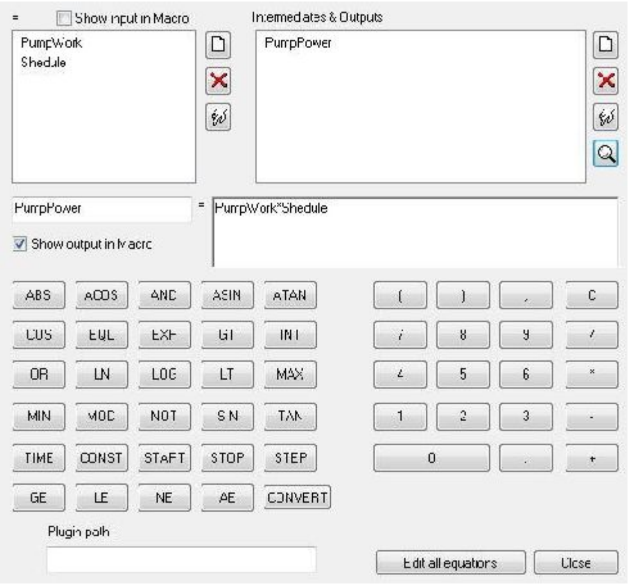


Figure 5-9 Pump Equation

Also an Equation is created for converting consumed power of components which are all $kJ.hr^{-1}$ to kW. This is done by dividing them by 3600 and is shown in Figure 5-10.

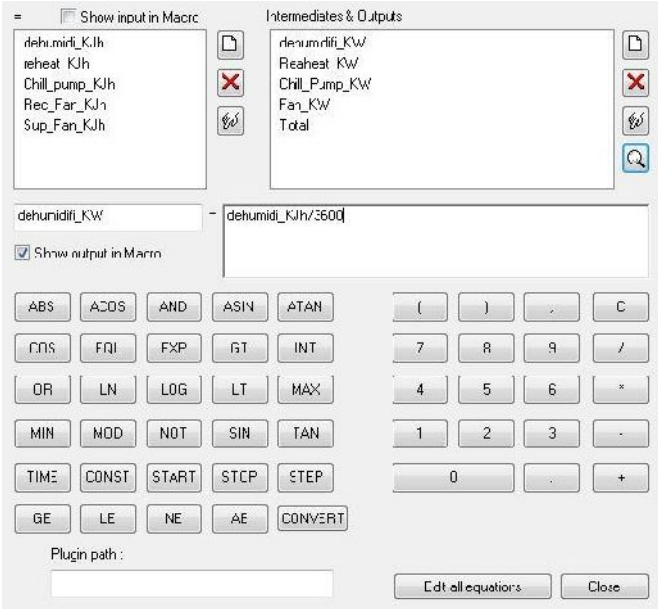


Figure 5-10 Unit conversion Equation

5.3.2. Air cycle components

5.3.2.1. Fresh and recirculation air mass flow rates

Since recirculation and fresh air volumetric flow rates are constant during system working hours, as calculated in, two Equation Types are adopted and named “mdot_recirc”, Figure 5-11, and “mdot_fresh”, Figure 5-12, to calculate recirculation and fresh air mass flow rates based on air properties. At the figures, the left column is the input list of Equation while the right column is output list, defined according to the equation(s) mentioned below of that.

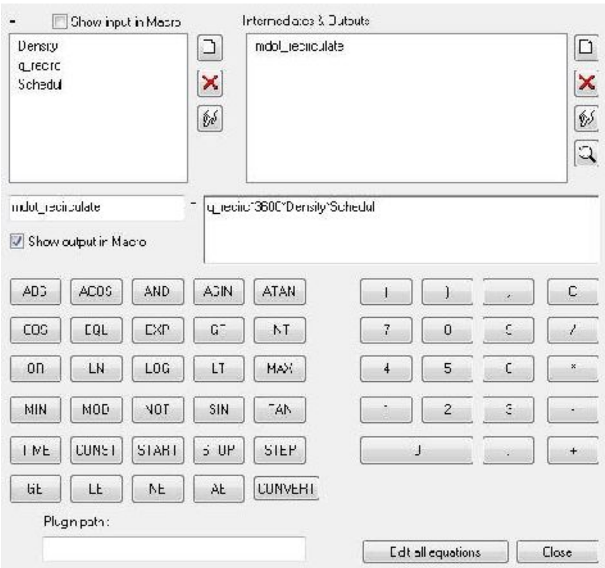


Figure 5-11 "mdot_recirc" Equation

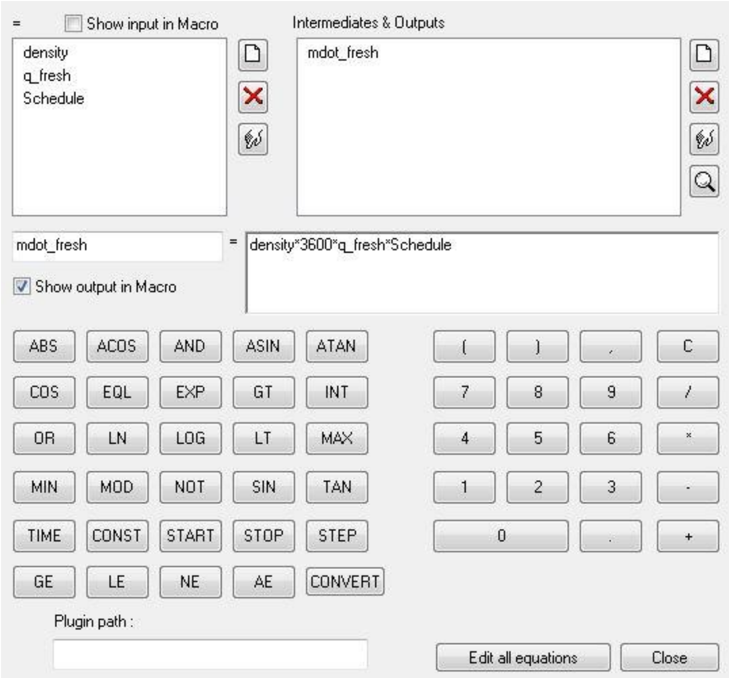


Figure 5-12 "mdot_fresh" Equation

At here, in both Equations, air mass flow rate is defined as

$$\dot{M}_A = \rho_A \times \dot{v}_A \times 3600 \times Working\ hour\ schedule \qquad (5-7)$$

All these values are input to both Equations and are listed in Table 5-11 and Table 5-12.

Table 5-11 mdot_recirc Equation inputs

Output		mdot_recirc
From	**parameter**	**Input**
Return air Psy	Density of mixture	Density
Constant	q_recirc	q_recirc
Schedule	Instantaneous value of function over the time step	Schedule

Table 5-12 mdot_fresh Equation inputs

Output		mdot_fresh
From	**parameter**	**Input**
Fresh air Psy	Density of mixture	Density
Constant	q_fresh	q_fresh
Schedule	Instantaneous value of function over the time step	Schedule

"Fresh air Psy" and "Return air Psy" both are Type 33c which calculate wet air psychometric properties based on its temperature and humidity ratio. For "Return air

Psy”, since return air temperature and humidity are known and constant, design condition, no input connection is required but input values are entered as constant. Parameter input and output windows values are listed in Table 5-13 to Table 5-15.

Table 5-13 "Return air Psy" and "Fresh air Psy" parameter window

Name	Value	Unit
Psychometric mode	4	-
Wet bulb mode	1	-
Error mode	2	-

Table 5-14 "Return air Psy" input window

Parameter	Value	Unit
Dry bulb temperature	24	C
Absolute humidity ratio	0.0093	kg/kg
Pressure	1	atm

Table 5-15 "Return airPsy" output window

Building output	Input	
	To	parameter
Density of mixture	mdot_recirc	Density
Humidity ratio	Recirc fan	Humidity ratio
Dry bulb temperature	Recirc fan	Inlet air temperature

However, for “Fresh air Psy” inputs are not constant but depends on ambient conditions, thus a connection between “Weather data” and the component is required, Table 5-16. Output window is also demonstrated in Table 5-17.

Table 5-16 "Fresh air Psy" input window

Output		Fresh air Psy Input
From	parameter	
Weather data	Dry bulb temperature	Dry bulb temp.
Weather data	Humidity ratio	Absolute humidity ratio

Table 5-17 "Fresh air Psy" output window

Fresh air output	Input	
	To	parameter
Density of mixture	mdot_fresh	Density

Table 5-17 (Continued)

Humidity ratio	Air mixer	Fresh g
Dry bulb temperature	Air mixer	Fresh T

5.3.2.2. Recirculation fan

While fresh air goes to air mixer directly, recirculation air has to be passed variable speed recirculation fan, Type744, which is named “Recirc fan”. Recirculation fan parameter, input and output windows are presented in Table 5-18 to Table 5-20. For inputs, recirculation air mass flow rates is transferred from “mdot_recirc” and its properties from “Return air Psy”.

Table 5-18 “Recirc fan” parameters

Name	Value	Unit
Humidity mode	1	-
Rated flow rate	3.617	kg/s
Rated power	5	kW
Motor efficiency	0.8	-
Motor heat loss fraction	0	-
Number of power coefficients	1	-
Power coefficient	1	-

Table 5-19 “Recirc fan” inputs

Output		Recirc fan Input
From	parameter	
mdot_recirc	mdot_recirculate	Air flow rate
Return air Psy	Humidity ratio	Humidity ratio
Return air Psy	Dry bulb temperature	Inlet air temperature

Table 5-20 “Recirc fan” outputs

Recirc fan output	Input	
	To	parameter
Outlet air temperature	Air mixer	Recirc T
Outlet humidity ratio	Air mixer	Recirc g
Outlet flow rate	Air mixer	Recirc m

At input window, “air-side pressure increase” is set to 200 Pa and “inlet air pressure” to 1 atm. Work of fan and outlet air temperature are calculated according to Eq. (5-8) and Eq. (5-9) respectively.

$$W = W_{rated} \times \frac{\dot{M}}{\dot{M}_{rated}} \quad (5-8)$$

$$T_{out} = T_{in} + \frac{Q}{\dot{M}C_{p,A}} \quad (5-9)$$

Where

$$Q = W \times (1 - \eta_{motor}) \times \eta_{heatfraction} \quad (5-10)$$

5.3.2.3. Air Mixer

Recirculation air (after passing recirculation fan) is mixed with outdoor fresh air at “Air mixer” adiabatically which is actually a tee piece, Type 11g. At parameter window, mode value is set to 6 and Figure 5-13, Eq. (5-11), Eq. (5-12) and Eq. (5-13) demonstrates the working principle of that. Input and out of air mixer is displayed in Table 5-21 and Table 5-22.

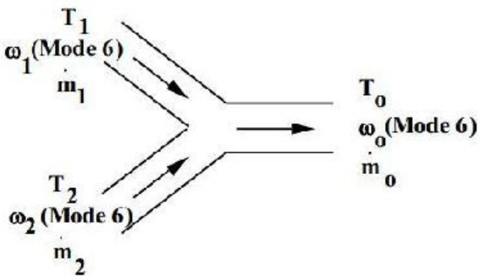


Figure 5-13 Type 11g, mode 6 working principle

$$T_{out} = \frac{\dot{M}_1 T_1 + \dot{M}_2 T_2}{\dot{M}_1 + \dot{M}_2} \quad (5-11)$$

$$g_{out} = \frac{\dot{M}_1 g_1 + M_2 g_2}{\dot{M}_1 + \dot{M}_2} \quad (5-12)$$

$$\dot{m}_{out} = \dot{M}_1 + \dot{M}_2 \quad (5-13)$$

Table 5-21"Air mixer" input

Output		Air mixer Input
From	**parameter**	
Fresh air Psy	Humidity ratio	Fresh g

Table 5-21 (Continued)

Fresh air Psy	Dry bulb temp	Fresh T
mdot_fresh	mdot_fresh	Fresh m
Recirc fan	Outlet air temperature	Recirc T
Recirc fan	Outlet humidity ratio	Recirc g
Recirc fan	outlet flow rate	Recirc m

Table 5-22 "Air mixer" output

Air mixer output	**Input**	
	To	**parameter**
Outlet temperature	Supply fan	Inlet air temperature
Outlet humidity ratio	Supply fan	Humidity ratio
Outlet flow rate	Supply fan	Air flow rate

5.3.2.4. Supply fan

After combination, mixed air passes “Supply fan” to overcome cooling coil pressure drop. Adopted type is the same as recirculation fan, thus working details are not repeated here again but “air-side pressure increase” and “inlet air pressure” are set to 1000 Pa and 1 atm respectively. Parameters, inputs and outputs are available in Table 5-23 to Table 5-25.

Table 5-23 Supply fan parameters

Name	**Value**	**Unit**
Humidity mode	1	-
Rated flow rate	6.3	kg/s
Rated power	7.5	kW
Motor efficiency	0.8	-
Motor heat loss fraction	0	-
Number of power coefficients	1	-
Power coefficient	1	-

Table 5-24 Supply fan input

Output		**Supply fan Input**
From	**parameter**	
Air mixer	Outlet temperature	Inlet air temperature

Table 5-24 (Continued)

Air mixer	Outlet humidity ratio	Humidity ratio
Air mixer	Outlet flow rate	Air flow rate

Table 5-25 Supply fan output

Supply fan output	**Input**	
	To	**parameter**
Outlet air temperature	Cooling coil	Air inlet temperature
Outlet humidity ratio	Cooling coil	Air inlet humidity ratio
Outlet flow rate	Cooling coil	Air flow rate

5.3.2.5. Cooling coil

A simple cooling coil for air humidity control is used, Type 508d, which is based on bypass fraction approach. Knowing air and chilled water properties and flow rates, the Type gives air and water outlet states. Since the aim of coil at here is to control air humidity ratio, an iterative feedback controller is used next to coil to specify chilled water flow rate so desired outlet air humidity ratio is obtained and is discusses at Water cycle. Cooling coil parameter window and input and output connections are presented in Table 5-26 to Table 5-28.

Table 5-26 Cooling coil parameter

Name	**Value**	**Unit**
control mode	2	-
Humidity mode	1	-
Liquid specific heat	4.19	kJ/kg.K

Table 5-27 Cooling coil input

Output		**Cooling coil Input**
From	**parameter**	
Supply fan	Outlet air temperature	Air inlet temperature
Supply fan	Outlet humidity ratio	Air inlet humidity ratio
Supply fan	Outlet flow rate	Air flow rate
Chiller	Outlet fluid temperature	Fluid inlet temperature
mdot_chiller	control signal	Fluid flow rate

Table 5-28 Cooling coil output

Cooling coil output	Input	
	To	**parameter**
Outlet air temperature	Reheater	Inlet air temperature
Outlet air humidity ratio	Reheater	Inlet air humidity ratio
Dry air flow rate	Reheater	Air flow rate
Outlet air pressure	Reheater	Inlet air pressure
Fluid outlet temperature	Chilled water pump	Input fluid temperature
Outlet air humidity ratio	mdot_chiller	Controlled variable

However some inputs are entered constant as below:

- Air pressure = 1000 Pa
- Air-side pressure drop 500 Pa
- Coil bypass fraction = 0.07
- Setpoint: outlet air humidity = 0.0001 $kg.kg^{-1}$

5.3.2.6. Reheater

Type 121a is utilized for heating dry, cold air coming from cooling coil to required temperature. Parameters are set as Table 5-29.

Table 5-29 Reheater parameter

Name	Value	Unit
Humidity mode	1	-
Surface area	1	m^2

While some inputs are considered constant, Table 5-30, renaming connections are shown in Table 5-31. Hating capacity is considered high enough so the device is able to provide air with desired temperature under any circumstances.

Table 5-30 Reheater constant inputs

Name	Value	Unit
Heating capacity	1000000	kJ/hr
Heat loss coefficient	3	$kJ/hr.m^2.K$
Efficiency	100	%
Air-side pressure drop	500	Pa

Table 5-31 Reheater input

Output		Reheater Input
From	**parameter**	
Cooling coil	Outlet air temperature	Inlet air temperature
Cooling coil	Outlet humidity ratio	Inlet humidity ratio
Cooling coil	Dry air flow rate	Air flow rate
Cooling coil	Outlet air pressure	Inlet air pressure
Weather data	Dry bulb temperature	Ambient temperature
Temp controller	Control signal	Setpoint temperature
Schedule	Instantaneous value of function over the time step	Control Function

As mentioned before, set point temperature is determined by “Temp controller” iterative feedback controller at each time step. Also since this component is the last one at air cycle before entering space, no output connection is defined except for those for output data from model which are described at 5.3.4.

5.3.3. Chilled water cycle components

This cycle is for chilled water from chiller to cooling coil and vice versa.

5.3.3.1. Water flow determiner

To obtain required chilled water flow rate at cooling coil, an iterative feedback controller, Type 22, is adopted and named “mdot_chiller”. Outlet air humidity ratio from cooling coil is the control variable for this Type and control signal of that is fluid flow rat of cooling coil. Hence, the components alters water flow rate until outlet humidity ratio is set to set point which is defined by “g controller” iterative feedback controller. After determination, result is sent to “Chilled water pump”. Parameters are the same with two previous iterative feedback controllers so are not mentioned here. Input and Output are connected according to Table 5-32 and Table 5-33.

Table 5-32 “mdot_chiller” input

Output		mdot_chiller
From	**parameter**	**Input**
Cooling coil	Outlet air humidity ratio	Controlled variable
g Controller	Control signal	Set point

Table 5-33 "mdot_chiller" output

mdot_chiller output	Input	
	To	**parameter**
Control signal	Cooling coil	Fluid flow rate
Control signal	Chilled water pump	Inlet fluid flow rate

5.3.3.2. Water pump

Type 742 is a variable speed pump which is used in the model as “Chilled water pump” and is responsible for circulating required water between cooling coil and chiller. Parameter window values are available in Table 5-34.

Table 5-34 Chilled water pump parameter

Name	Value	Unit
Fluid specific heat	4.19	kJ/kg.K
Fluid density	1000	kg/m^3
motor heat loss fraction	1	-

Since some input data are constant during period of time, they are not connected to any component and inputted as constant, Table 5-35 and other are connected as Table 5-36.

Table 5-35 Chilled water pump constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	10	kPa

Table 5-36 Chilled water pump input

Output		Pump Input
From	**parameter**	
Cooling oil	Fluid outlet temperature	Inlet fluid temperature
mdot_chiller	Control signal	Inlet fluid flow rate

Output of pump is connected to chiller, Table 5-37.

Table 5-37 Chilled water pump output

Pump output	Input	
	To	**parameter**
Outlet fluid temperature	Chiller	Chilled water inlet temperature
Outlet flow rate	Chiller	Chilled water flow rate

5.3.3.3. Chiller

Chilled water from cooling coil goes to chiller through "Chilled water pump" and then it returned back after cooling in Chiller. A vapor compression chiller model, Type 655, is used for providing chilled water. Chiller data are provided by two external files, "Chiller data" and "Chiller part-load data" and are editable, however these files are not changed. All chiller parameter are default values because they should fit chiller data files. Table 5-38 shows parameter values.

Table 5-38 Chiller parameter

Name	Value	Unit
Rated C.O.P	3	kg/m^3
Logical unit for performance data	32	-
Logical unit for PLR data file	33	-
Fluid specific heat	4.19	kJ/kg.K
Number of ambient temperatures	5	-
Number of CHW set points	6	-
Number of part load ratios	5	-

Input and Output connections are displayed at Table 5-39 and Table 5-40.

Table 5-39 Chiller input

Output		Chiller Input
From	**parameter**	
Chilled water pump	Outlet fluid temperature	Chilled water inlet temperature
Chilled water pump	Outlet flow rate	Chilled water flow rate
Weather data	Dry bulb temperature	Ambient temperature
Constant	T_chiller_set	Setpoint temperature
Schedule	Instantaneous value of function over the time step	Chiller control signal

Table 5-40 Chiller output

Chiller output	**Input**	
	To	**parameter**
Outlet fluid temperature	Cooling coil	Fluid inlet temperature

5.3.4. Outputs

After developing cycle models, online graphs and printers are adopted for demonstrating and saving results, Type 65d and 46b respectively. It should be noted that power consumption outputs from the components are in kJ hr^{-1}, hence are first converted to kW, as described in 5.3.1.6, and then to graphs and printers. Table 5-41 shows monitored outputs of entire model.

Table 5-41 Monitored outputs

Type	**Output parameter**	**Unit**
Building	TAIR_CENTER	^{o}C
Building	ABSHUM_CENTER	$gr.kg^{-1}$
Direction	Radiation->IT_H_0_0	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IB_H_0_0	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IT_S_0_90	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IB_S_0_90	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IT_W_90_90	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IB_W_90_90	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IT_N_180_90	$kJ.(hr.m^{2})^{-1}$
Direction	Radiation->IB_N_180_90	$kJ.(hr.m^{2})^{-1}$
Chiller	Fluid flow rate	$kg.hr^{-1}$
Chiller	COP	-
Chiller	Chiller power	kW
Supply fan	Outlet humidity ratio	$gr.kg^{-1}$
Supply fan	Power consumption	kW
Recirculation fan	Power consumption	kW
Cooling coil	Outlet air temperature	^{o}C
Cooling coil	Outlet air humidity ratio	$gr.kg^{-1}$
Cooling coil	Condensate temperature	^{o}C
Weather data	Dry bulb temperature	^{o}C

Weather data	Humidity ratio	$gr.kg^{-1}$
Temp Controller	Control signal	-
g controller	Control signal	-
Reheater	Outlet air temperature	^{o}C
Reheater	Outlet air humidity ratio	$gr.kg^{-1}$
Reheater	Required heating	kW
Pump	Power consumption	kW

CHAPTER 6

DEHUMIDIFIER AND REGENERATOR MODELS

Since liquid desiccant dehumidifier and regenerator mathematical models are not available in TRNSYS nor TESS library [70], their models are adopted from open literature and are explained at this chapter before describing desiccant cooling cycle model in TRNSYS which is depicted in DESICCANT COOLING CYCLE MODEL.

As described in section 2.1.2.1, there are three main mathematical modeling methods for contactors: 1) computational fluid dynamics model (CFD) which is the most exact one; 2) number of transfer units (NTU); and 3) fitting algebraic equation, the least exact model. However the last one is adopted here because analysis of a finite difference model is very complex and iterative numerical solutions are required. Hence, algebraic methods is adopted which are simple equations used for quick prediction of moisture effectiveness (ε) with the aim of determining air outlet properties using Eq. (6-1) and Eq. (6-2) which are obtained by Eq. (2-1) and Eq. (2-2). Note these two equations explicitly show that the contactor effectiveness is a function of the humidity ratio (g) and enthalpy (h) of the inlet air (subscript A,i) and its equilibrium state (subscript A,equ) streams.

$$g_{A,o} = \mathrm{g_{A,i}} - \varepsilon_{\mathrm{g}}\left(\mathrm{g_{A,i}} - \mathrm{g_{A,equ}}\right) \qquad (6-1)$$

$$h_{A,o} = \mathrm{h_{A,i}} - \varepsilon_{\mathrm{h}}\left(\mathrm{h_{A,i}} - \mathrm{h_{A,equ}}\right) \qquad (6-2)$$

$g_{A,equ}$ is the humidity ratio of air when it is in equilibrium with inlet solution, meaning that water partial pressure of air is equal to vapor pressure of inlet solution. Similarly, $h_{A,equ}$ is obtained when air temperature and humidity ratio is equal to inlet solution temperature and vapor pressure respectively. Using this set of equations, air outlet humidity ratio and enthalpy are obtained if dehumidification and enthalpy effectiveness of dehumidifier are known and subsequently, the air outlet temperature. While the accuracy of the method is slightly lower compared to finite difference, it is still acceptable for some applications such as air-conditioning and in fact their simplicity is beneficial as it enables hourly system analysis in a reasonable amount of time. Algebraic equations are obtained through curve fitting of the available input and output data of dehumidifiers and regenerators, either using experimental tests or finite difference results. Unfortunately, this type of information is rare and only a handful of such studies are available in the open literature.

6.1. Dehumidifier

Martin and Goswami [32] developed correlations for dehumidification, Eq. (6-3,) and enthalpy effectiveness, Eq. (6-4), of a counter-flow random packed bed dehumidifier for lithium chloride (LiCl) obtained through curve fittings.

$$\varepsilon_{\mathrm{g}} = 1 - 48.3\left(\frac{\mathrm{L}}{\mathrm{A}}\right)^{0.396\frac{\gamma}{0.029}-1.57}\left(\frac{\mathrm{h_{A,i}}}{\mathrm{h_{L,i}}}\right)^{-0.751}(\mathrm{a_t Z})^{0.0331\frac{\gamma}{0.029}-0.906} \qquad (6-3)$$

$$\varepsilon_{\mathrm{H}} = 1 - 3.77\left(\frac{\mathrm{L}}{\mathrm{A}}\right)^{0.289\frac{\gamma}{0.029}-1.12}\left(\frac{\mathrm{h_{A,i}}}{\mathrm{h_{L,i}}}\right)^{-0.528}(\mathrm{a_t Z})^{-0.0044\frac{\gamma}{0.029}-0.365} \qquad (6-4)$$

Equations are based on experiments of Chung et al. [19], hence conditions of experiments have to be considered while using them. During experiments, solution concentration and flow rate remain constant at 30% and 40% and 3 GPM (11.37 liters per minute) respectively while other parameters vary.

For Chung et al.'s experiments [19], the air and solution inlet flow rates are 0.028 and 0.00019 $m^3.s^{-1}$ respectively with a 0.01826 m^2 dehumidifier cross sectional area, resulting in 0.767 $m^3\ m^{-2}\ s^{-1}$ and 0.010405 $m^3\ m^{-2}\ s^{-1}$ of air and solution superficial flow rates respectively, defined in Eq. (6-5). Since current ventilation air flow rate is 3.85 $m^3\ s^{-1}$, the required dehumidifier cross sectional area is 2.48 m^2 in order to have the same air superficial flow rate and consequently the solution flow rate is 0.026 $m^3 s^{-1}$.

$$\dot{v}_{superficial} = \frac{\dot{v}_{stream}}{A_{dehumidifier}} \qquad (6-5)$$

According to the previous work of Ge, Xiao, and Niu [71], the dehumidification rate is easier to control by controlling the solution temperature than the concentration. Hence, at current work liquid desiccant enters the dehumidifier with a constant flow rate, 0.026 $m^3\ s^{-1}$, and concentration, as mentioned above, and at the temperature required to absorb desired amount of moisture, Eq. (6-6), and producing dry air and a dilute and warm desiccant solution.

$$\dot{M}_{cond} = \dot{M}_{A,i}\left(\frac{g_{A,i} - g_{A,o}}{1 - g_{A,o}}\right) \qquad (6-6)$$

After knowing the air outlet humidity, enthalpy and following them temperature, other outputs can be obtained through mass and energy balance equations at the dehumidifier, Eq. (6-7) to Eq. (6-10).

For air:

$$\dot{M}_{A,o} = \frac{\dot{M}_{A,i}(1 - g_{A,i})}{1 - g_{A,o}} \qquad (6-7)$$

And for the solution:

$$\dot{M}_{L,o} = \dot{M}_{L,i} + \dot{M}_{cond} \qquad (6-8)$$

$$X_{L,o} = \frac{M_{L,i}X_{L,i}}{\dot{M}_{L,o}} \qquad (6-9)$$

$$h_{L,o} = \frac{\dot{M}_{L,i}h_{L,i} + \dot{M}_{A,i}h_{A,i} - M_{A,o}h_{A,o}}{\dot{M}_{L,o}} \qquad (6-10)$$

6.2. Regenerator

Since there is no algebraic model for regenerator effectiveness at open literature, the regenerator is modeled according to Gandhidasan [34] theoretical study through Eq. (6-11) which calculates superficial water evaporation rate from solution in dehumidifier. In this general model, which could be applied to any regenerator with overlooking its type and operating conditions, outlet air temperature of regenerator is required, to obtain β value, thus it has to be used in accordance with experiments data which are fit to adopted dehumidifier working conditions, particularly diluted and regenerated solutions concentration to and from regenerator. Among experimental studies, the work by Fumo and Goswami [24] is the most appropriate one when concentrations are taken into account. Table 6-1 shows the adopted experiment data from set of experiments.

$$\dot{M}_{evap} = \frac{1}{h_{fg}}\left[Q - \dot{M}_A C_{p_A} \beta \left(T_{L,i} - T_{A,i}\right)\right] \qquad (6-11)$$

Where:

$$\beta = \frac{T_{A,o} - T_{A,i}}{T_{L,i} - T_{A,i}}$$

Table 6-1 Utilized experiment data [24]

Inlet						Outlet				
A_i (kg m^{-2} s^{-1})	$T_{A,i}$ (°C)	$g_{A,i}$ (kg kg^{-1})	L_i (kg m^{-2} s^{-1})	$T_{L,i}$ (°C)	X_i (%)	$T_{A,o}$ (°C)	$g_{A,o}$ (gr kg^{-1})	$T_{L,o}$ (°C)	X_o (%)	$\dot{m}_{evap}$ (gr m^{-2} s^{-1})
1.438	29.8	0.0177	6.479	65.1	34.5	57.5	48.8	56.6	35.2	2.10

According to Table 6-1, diluted and concentrated solution to and from regenerator is 34.5% and 35.2% respectively, not far from dehumidifier concentration. Similar to the dehumidifier, data are available as a superficial flow rate, thus it is possible to change the regenerator cross sectional area in accordance with current solution flow rate to have the same superficial flow rate. So it is possible to control solution outlet concentration by controlling evaporation rate which is function of solution and air inlet conditions. Inlet air temperature is constant, according to Table 6-1, and other parameters are known but solution inlet temperature. Hence required solution concentration can be obtained by varying its inlet temperature.

6.3. Concentration adjustment

Aforementioned, Chung's experiments are done for desiccant solution inlet concentrations of 30% and 40%, meaning that Eq. (6-3) and Eq. (6-4) are validated for these two concentrations. On the other hand, available data that are used for regeneration shows that regenerator produces solution with 35.2% which is different from dehumidifier inlet concentration. Also, concentration of diluted solution toward regenerator has to be 34.5%, hence outlet solution from dehumidifier must be at this value but it is unknown. To overcome these, two adjustments are done as below.

6.3.1. Dehumidifier inlet solution adjustment

As mentioned in LITERATURE REVIEW (Koronaki et al. [23]), while the solution inlet concentration affects the amount of condensed moisture in the dehumidifier, it does not have an effect on the moisture effectiveness, Table 6-1, and the reason for that is when the inlet solution concentration increases, its humidity ratio decreases and consequently the air outlet concentration decreases. As a result and according to Eq. (6-1), the moisture effectiveness remains approximately constant.

$$X_{L,i} \uparrow \rightarrow \left\{ \begin{array}{l} g_{A,o} \downarrow \\ \\ g_{L,i} \downarrow \end{array} \right\} \rightarrow \varepsilon_y \cong contstant$$

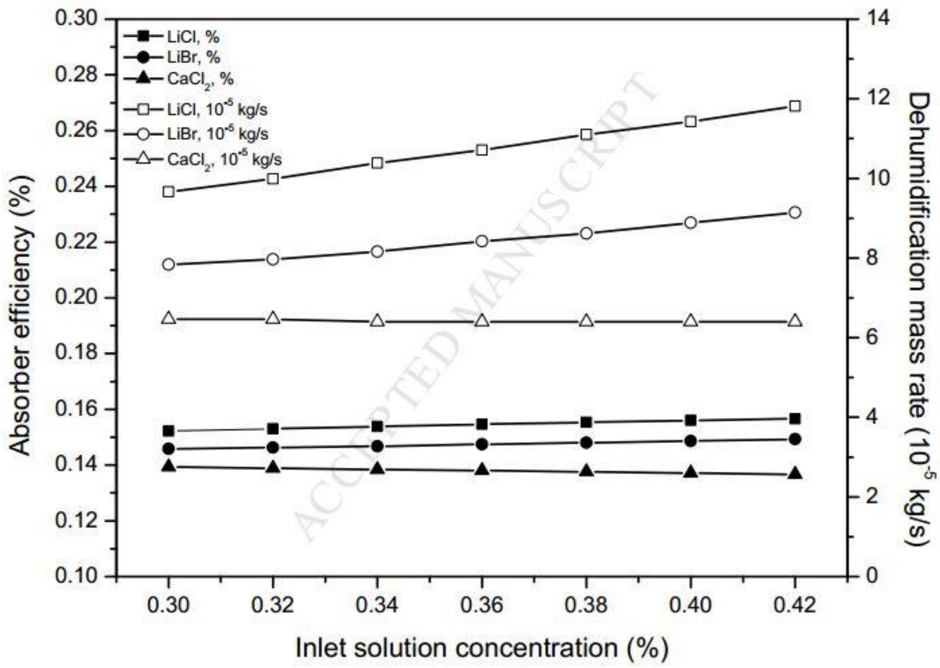


Figure 6-1 Effect of inlet solution concentration [23]

While enthalpy effectiveness is not investigated in the paper by Koronaki et al. [23], similar conclusions can be drawn. As $X_{L,i}$ increases, $g_{A,o}$ decreases, meaning that lower $h_{A,o}$'s will be obtained. From the solution point of view, higher $X_{L,i}$'s result in lower $h_{L,i}$'s because the enthalpy of pure water is much higher than that for pure LiCl. Finally, according to Eq. (6-2), enthalpy effectiveness can remain approximately constant and in fact, this is an assumption in this study.

$$X_{L,i} \uparrow \rightarrow \left\{ \begin{array}{l} h_{A,o} \downarrow \\ \\ h_{L,i} \downarrow \end{array} \right\} \rightarrow \varepsilon_h \cong contstant$$

In conclusion, calculating the enthalpy and moisture effectiveness through Eq. (6-3) and Eq. (6-4), inlet concentration is assumed as 30% but when trying to obtain air outlet properties by Eq. (6-1) and Eq. (6-2), the exact inlet solution concentration, 35.2%, is applied. It means that in real application, dehumidifier concentration must be 35.2%.

6.3.2. Regenerator inlet solution adjustment

The ratio of solution to air flow rate must be very high to achieve perfect wetting of dehumidifier [19]. As a result, a very small change in solution concentration occurs enabling the outlet solution to be used again if it is only cooled and without any regeneration. To model this, the outlet solution flow rate is split into two flows, one flow is reused again only by cooling it from the dehumidifier outlet temperature to the dehumidifier inlet temperature at constant concentration, 35.2%, and the other flow is sent to the regenerator at the dehumidifier outlet temperature and regenerator inlet concentration which is 34.5%, Figure 6-2, and are calculated by writing mass balance for solution and solved LiCl.

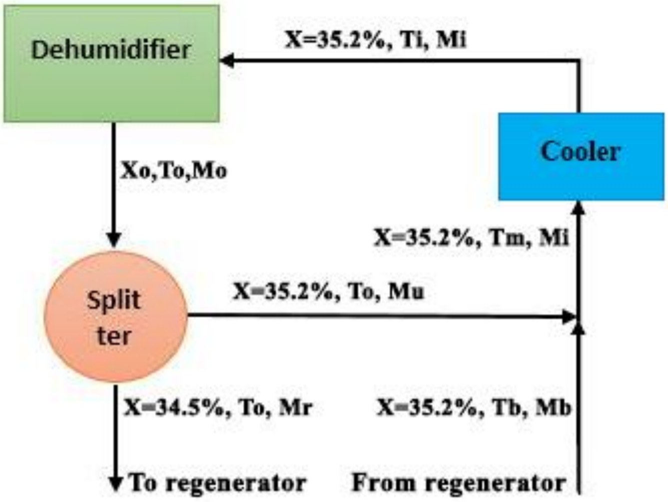


Figure 6-2 Regenerator inlet solution adjustment

$$for\ solution: \qquad \dot{M}_{deh,o} = \dot{M}_r + \dot{M}_u \qquad (6-12)$$

$$for\ LiCl: \qquad \dot{M}_{deh,o} \times X_{deh,o} = \dot{M}_r \times X_{reg,i} + \dot{M}_u \times X_{deh,i} \qquad (6-13)$$

Solving these two equation for $\dot{M}_u$ and $\dot{M}_r$ results in Eq. (6-14) and Eq. (6-15):

$$\dot{\mathrm{M}}_{\mathrm{u}} = \frac{\dot{\mathrm{M}}_{\mathrm{deh,o}}\left(\mathrm{X}_{\mathrm{deh,o}} - \mathrm{X}_{\mathrm{reg,i}}\right)}{\mathrm{X}_{\mathrm{deh,i}} - \mathrm{X}_{\mathrm{reg,i}}} \qquad (6-14)$$

$$\dot{\mathrm{M}}_{\mathrm{r}} = \frac{\dot{\mathrm{M}}_{\mathrm{deh,o}}\left(\mathrm{X}_{\mathrm{deh,i}} - \mathrm{X}_{\mathrm{deh,o}}\right)}{\mathrm{X}_{\mathrm{deh,i}} - \mathrm{X}_{\mathrm{reg,i}}} \qquad (6-15)$$

Also writing mass balance equation for solution and water at regenerator provides required evaporation rate in generator, $\dot{m}_{evap}$, and return solution flow rate from regenerator, $\dot{m}_b$, and is presented in Eq. (6-18) and Eq. (6-19).

$$for\ solution: \qquad \dot{M}_r = \dot{M}_{evap} + \dot{M}_b \qquad (6-16)$$

$$for\ water: \qquad \dot{M}_r \times (1 - X_{reg,i}) = \dot{M}_{evap} + \dot{M}_b \times (1 - X_{reg,o}) \qquad (6-17)$$

Solving for $\dot{M}_{evap}$ and $\dot{m}_b$:

$$\dot{M}_b = \frac{\dot{M}_r \times X_{reg,i}}{X_{reg,o}} \qquad (6-18)$$

$$\dot{M}_{evap} = \dot{M}_r \times \frac{X_{reg,o} - X_{reg,i}}{X_{reg,o}} \qquad (6-19)$$

Thus, concentration of inlet solution to regenerator is solved in spite of unknown and different dehumidifier outlet solution concentration next to required evaporation arte.

6.4. LiCl properties

In addition to contactors modeling, properties of lithium chloride are necessary. Hence vapor pressure and enthalpy equations of that is adopted from Chaudhari and Patil [72] while the rest properties equations are from Conde [73].

6.4.1. Vapor pressure

Vapor pressure of solution in kPa is obtained by Eq. (6-20). At here concentration is in fraction and temperature in Kelvin.

$$\log P = \left(A + \frac{B}{T} + \frac{C}{T^2}\right) \times 0.133322 \qquad (6-20)$$

Where:

$$A = 8.202988 - (0.1353801 \times b) + (0.0179222 \times b^2) - (0.0005292 \times b^3)$$

$$B = -1727.8 + (583845 \times b) - (10.208 \times b^2) + (0.3125 \times b^3)$$

$$C = -95014 - (4701 \times b) + (929.081 \times b^2) - (31.766 \times b^3)$$

$$b = \frac{X}{0.042394 \times (1 - X)}$$

6.4.2. Enthalpy

Eq. (6-21) shows enthalpy of lithium chloride solution in kJ per kilogram of solution. But not like vapor procure, concentration is in percentage.

$$h = A + (B \times t) + (C \times t^2) \qquad (6-21)$$

Where:

$$A = -66.2324 + (11.2711 \times \mathrm{X}) - (0.79853 \times X^2) + (0.021534 \times X^3) - (10^{-4} 1.66352 \times X^4)$$

$$B = 4.5751 - (0.146924 \times X) + (0.006307226 \times X^2) - (10^{-4} \times 1.38054 \times X^3) + (10^{-6} \times 1.0669 \times X^4)$$

$$C = (-10^{-4} \times 8.09689) + (10^{-4} \times 2.18145 \times X) - (10^{-5} \times 1.36194 \times X^2) + (10^{-7} \times 3.20998 \times X^3) - (10^{-9} \times 2.64266 \times X^4)$$

6.4.3. Density

Solution relative density (to water at same temperature) with fractional concentration of X is shown in Eq. (6-22).

$$\rho_{relative} = 1 + \left(5.54 \times \frac{X}{1 - X}\right) - \left(0.3 \times \left(\frac{X}{1 - X}\right)^2\right) + \left(0.1 \times \left(\frac{X}{1 - X}\right)^3\right) \qquad (6-22)$$

6.4.4. Surface tension

Solution relative surface tension (to water at same temperature) with fractional concentration of X and absolute temperature of T is displayed in Eq. (6-23)

$$\gamma_{relative} = 1 + (2.7 \times X \times \theta) + (14.75 \times X \times \theta^2) + (2.44 \times X^2 \times \theta^3) \qquad (6-23)$$

Where:

$$\theta = \frac{T}{647.1}$$

6.4.5. Specific thermal capacity

Solution relative specific thermal capacity (to water at same temperature) with fractional concentration of X is displayed in Eq. (6-24). Note that at this equation is valid for concentration higher than 3.1% of solution.

$$C_{relative} = 1 - \big[[(58.52 \times \lambda^{0.02}) - (105.63 \times \lambda^{0.04}) + (47.79 \times \lambda^{0.06})] \times [0.13 + 0.63 \times X]\big] \qquad (6-24)$$

Where:

$$\lambda = \frac{T}{228} - 1$$

All these properties equations are adopted in model to obtain desiccant solution thermal properties at required locations and at each time step in desiccant cooling cycle model in TRNSYS which is provided in DESICCANT COOLING CYCLE MODEL.

CHAPTER 7

DESICCANT COOLING CYCLE MODEL

In this chapter, hybrid desiccant cooling cycle is modeled in TRNSYS environment and connected to the building energy model described in Chapter 4: Building model with the aim of providing suitable temperature and humidity. The schematic diagram of hybrid Desiccant cooling cycle is shown in Figure 7-1. In the cycle, specific amount of return and outdoor fresh air are mixed and brought into contact with desiccant solution in dehumidifier which results in chemical dehumidification of ventilation air. Then dry air is cooled to desired temperature in cooling coil, depending on conditions and sensible cooling load and finally is fed into space. Unlike the conventional cooling cycle, no overcooling occurs at here so no reheating is required.

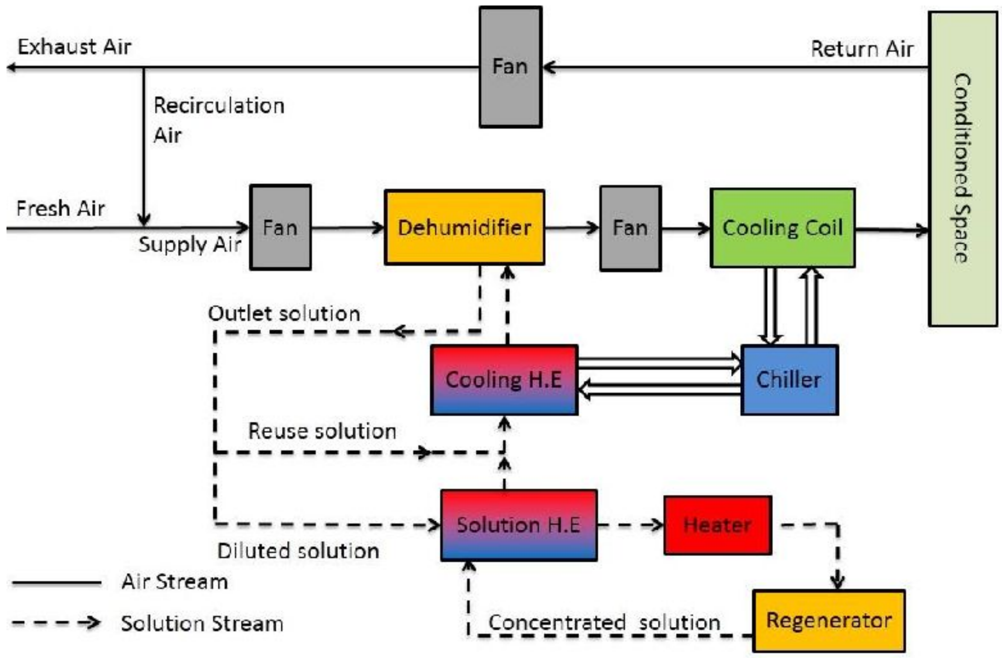


Figure 7-1 Schematic diagram of Desiccant cooling cycle

7.1. Working Principle

Figure 7-2 demonstrates TRNSYS model of the Desiccant cooling cycle. Similar to CONVENTIONAL COOLING CYCLE MODEL, icons and connection between them represent components and information flow among them, could be real (continuous lines) or non-real (dotted lines). Connections in Figure 5-6 are categorized like CONVENTIONAL COOLING CYCLE MODEL except that one desiccant solution (LiCl) cycle is added here and shown by bold, continuous, cyan line. Utilized components are listed in Table 7-1. Note that “Constant” and “kJ/h->kW” Equations and plotters and printers are not shown in Figure 7-2 due to complexity of the model.

The principle of model at here is the same with CONVENTIONAL COOLING CYCLE MODEL, hence is mentioned briefly. Recirculation and fresh air enters the cycle from “Recirculation air” and “Fresh air” components respectively with the first at indoor design condition and the latter at ambient state, obtained from “Weather data” component. The same two psychometric components are adopted here “Return air Psy” and “Fresh air Psy” with the aim of providing both air steams properties. Coming from recirculation fan, recirculation air is mixed with fresh air to make ventilation air which flows through “Supply Fan 1” before reaching dehumidifier where its humidity drops down to required magnitude. Since air pressure drop is relatively high at dehumidifier, air continues its way toward cooling coil by help of “Supply fan 2” to overcome pressure losses at coil. Two iterative feedback controllers. “Temperature controller” and “Humidity Controller” are adopted to obtain required temperature and humidity ratio of supply air at each time step and sending results to dehumidifier and cooling coil respectively.

7.2. Types and Connections

Types and connections between them are divided into four man groups, namely Non-cycle, air cycle, desiccant solution cycle and chilled water cycle.

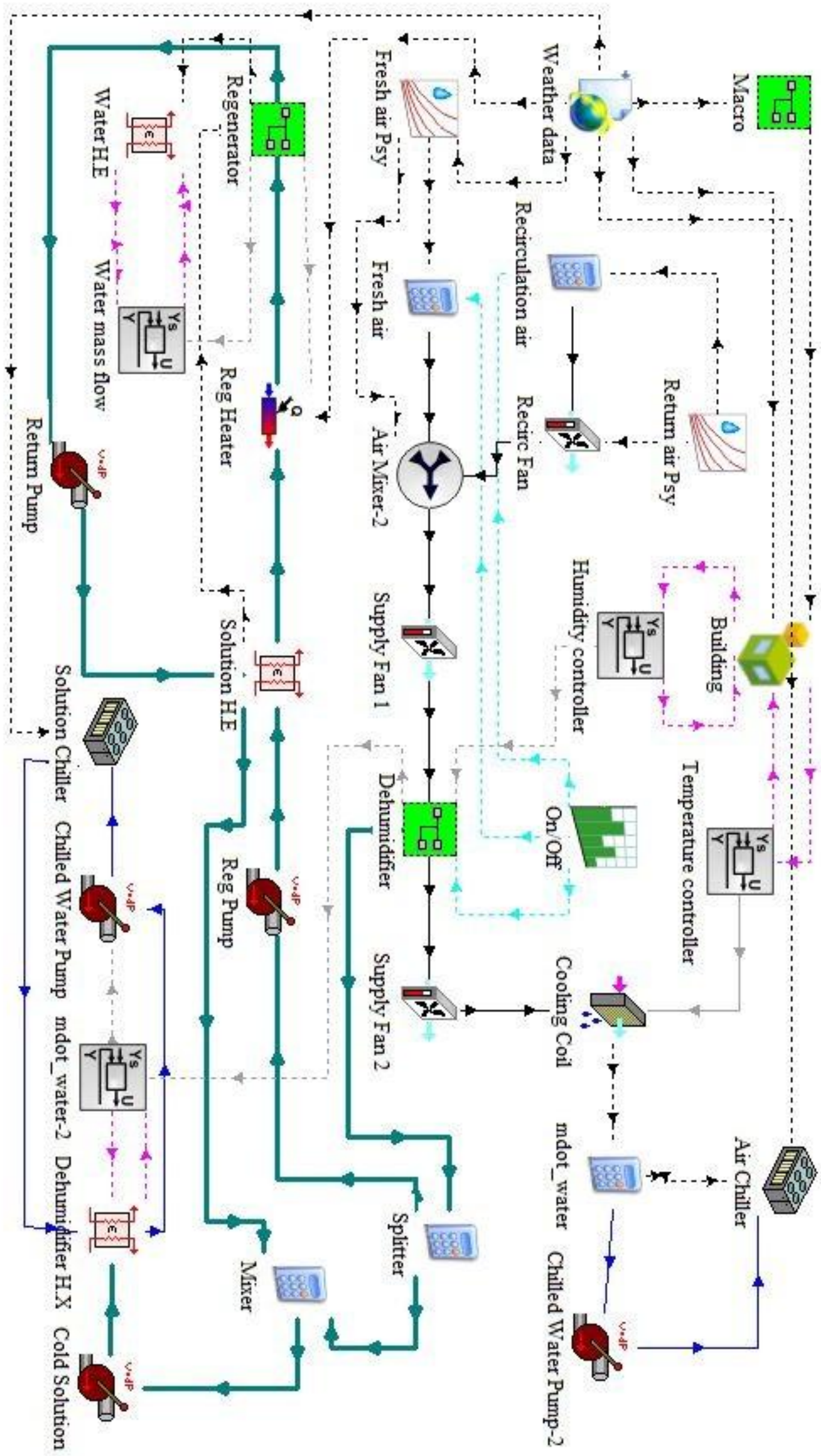


Figure 7-2 TRNSYS diagram of hybrid desiccant cooling cycle

Table 7-1 List of desiccant cooling cycle components in TRNSYS model

Name	Type	Note
Building	56	Designed building in TRNBuild
Temperature Controller	22	Calculating supply air temperature
Humidity controller	22	Calculating supply air humidity
Direction	Macro	Sun irradiance on different directions of building
Weather data	15-2	Inputting weather data
Return air Psy	33c	Recirculation air properties
Fresh air Psy	33c	Fresh air properties
Recirculation air	Equation	Recirculation air flow rate
Fresh air	Equation	Fresh air flow rate
Air Mixer	11g	Mixing fresh and recirculation air as ventilation air
Recirc Fan	744	Fan for recirculation air
Supply Fan 1	744	Fan for supply air
Dehumidifier	Macro	Dehumidifier model
Supply Fan 2	744	Fan for supply air
Cooling coil	752f	Cooling and dehumidifying of air to required humidity ratio
Chilled water pump 2	742	Pump between cooling coil and its chiller
mdot water	Equation	Calculating required chilled water mass flow rate for cooling coil
Air chiller	655	Providing chilled water for cooling coil
Splitter	Equation	Splitting solution from dehumidifier to two streams
Reg pump	742	Solution pump for that part of solution to be regenerated
Solution H.E	91	Heat exchanger with aim of precooling and preheating solution
Reg Heater	6	Heating solution to desired temperature for regeneration
Regenerator	Macro	Regenerator model
Return pump	742	Solution pump between regenerator and mixer
Mixer	Equation	Combining reuse and regenerated solution toward dehumidifier

Table 7-1 (Continued)

Cold solution	742	Solution pump between mixer and dehumidifier
Dehumidifier H.X	91	Heat exchanger with aim of cooling solution
mdot_water-2	22	Calculating required chilled water for solution heat exchanger
Chilled Water Pump	742	Water pump between chiller and solution heat exchanger
Solution chiller	655	Providing chilled water for cooling solution
Solar H.E	91	Heat exchanger for heating solution by solar collectors
solar mass flow	22	Calculating required hot water flow rate for solution heating from solar collector
On/Off	14h	Providing schedule for required units
Constant	Equation	Defining constant values for parameter studies
kJ/h>kW	Equation	Converting kJ per hour to kW
Printer	46b	Saving output data in excel file
Online plotters	65d	Graphing output data

7.2.1. Non-cycle components

These components, their parameters and connections are exactly the same as described in CONVENTIONAL COOLING CYCLE MODEL so is not repeated here. For detailed information refer to section 5.3.1.

7.2.2. Air cycle components

Components of this cycle is the same with air cycle components of CONVENTIONAL COOLING CYCLE MODEL until point of dehumidifier, hence prior components of that are not described here and are available in section 5.3.2. However, remain of this cycle is different from that of conventional cycle and is described here.

7.2.2.1. Dehumidifier

As mentioned before, after passing “Supply fan 1”, ventilation air brought into contact with concentrated, cold desiccant solution with the aim of drying. While dehumidifier is demonstrated by an icon in TRNSYS model, “Dehumidifier” macro, it is developed by adopting several units which are located inside the macro, Figure 7-3.

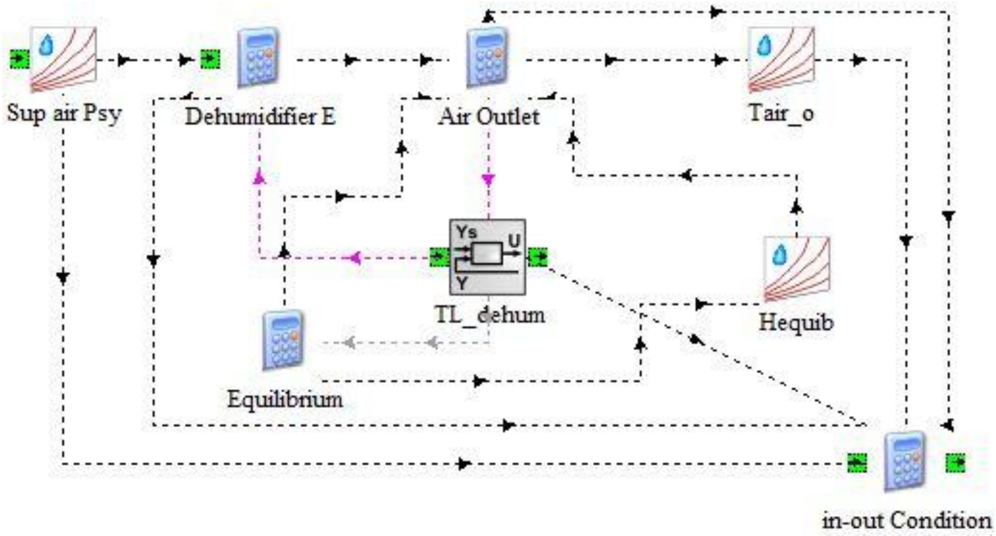


Figure 7-3 Inside of Dehumidifier macro

“Dehumidifier E” Equation, calculates dehumidifier moisture and enthalpy effectiveness according to air and solution inlet states by defining Eq. (6-3) and Eq. (6-4) inside the Equation. According to Eq. (6-3), Eq. (6-4), dehumidifier moisture and enthalpy effectiveness depends on solution and air inlet states. Hence one Type 33c, named “Sup air Psy”, is adopted and located between to “Supply fan 1” and “Dehumidifier E” to provide inlet air thermal properties from the first to the latter. Table 7-2, Table 7-3 and Table 7-4 show “Sup air Psy”'s parameters, input and output data respectively.

Table 7-2 “Sup air Psy” parameter

Name	Value	Unit
Psychometric mode	4	-
Wet bulb mode	1	-
Error mode	2	-

Table 7-3 "Sup air Psy" input

Output		Sup air Psy input
From	**parameter**	
Supply Fan 1	Outlet air temperature	Dry bulb temperature
Supply Fan 1	Outlet humidity ratio	absolute humidity ratio

Table 7-4 "Sup air Psy" output

Sup air Psy output	Input	
	To	**parameter**

Table7-4 (Continued)

Enthalpy	Dehumidifier E	h_ai
Enthalpy	In-out condition	H_ai

For solution inlet conditions, Eq. (6-20) to Eq. (6-24) are also defined at "Dehumidifier E" which are functions of solution inlet concentration and temperature. As mentioned before, inlet solution concentration is considered constant, 40% for calculating ε_h and ε_g but the temperature is unknown and is used as controlling parameter and is described further. Input and output connections of "Dehumidifier E" are shown in Table 7-5 and Table 7-6.

Table 7-5 "Dehumidifier E" input

Output		**Dehumidifier E input**
From	**parameter**	
Supply Fan 1	Outlet air temperature	T_ai
Supply Fan 1	Outlet humidity ratio	g_ai
Sup air Psy	Enthalpy	h_ai
TL_dehum	Control signal	T_Li
On/Off	Instantaneous value of function over the time step	Schedule

Table 7-6 "Dehumidifier E" output

Dehumidifier E output	**Input**	
	To	**parameter**
mdot_sol_i	in-out condition	mdot_sol_i
h_ai	Air outlet	h_ai
Ey	Air outlet	Ey
Eh	Air outlet	Eh
g_ai	Air outlet	g_ai

Air equilibrium humidity ratio and temperature are calculated by "Equilibrium" Equation according to solution inlet concentration, 35.2% and unknown temperature. Also "Equilibrium" Equitation is connected to a Type 33c,"Hequib", for calculating air equilibrium enthalpy. "Equilibrium" and "Hequib" connections are presented here.

Table 7-7"Equilibrium" input

Output		**Equilibrium input**
From	**parameter**	
TL_dehum	Control signal	T_Li

Table 7-8 "Equilibrium" output

Equilibrium output	Input	
	To	**parameter**
g_equib	Air outlet	g_equib
g_equib	Hequib	Dry bulb temperature
T_equib	Hequib	Absolute humidity ratic

Table 7-9 "Hequib" parameter

Name	Value	Unit
Psychometric mode	4	-
Wet bulb mode	1	-
Error mode	2	-

Table 7-10 "Hequib" input

Output		Hequib input
From	**parameter**	
Equilibrium	g_equib	Dry bulb temperature
Equilibrium	T_equib	absolute humidity ratio

Table 7-11 "Hequib" output

Hequib output	Input	
	To	**parameter**
Enthalpy	Air Outlet	h_equib

Knowing dehumidifier moisture and enthalpy effectiveness next to inlet and equilibrium air humidity ratio and enthalpy, outlet air humidity ratio and enthalpy is calculated by “Air outlet” according to Eq. (6-1) and Eq. (6-2) with connections in Table 7-12 and Table 7-13.

Table 7-12 "Air Outlet" input

Output		Air Outlet input
From	**parameter**	
Dehumidifier E	Ey	Ey
Dehumidifier E	Eg	Eg
Dehumidifier E	g_ai	g_ai
Dehumidifier E	h_ai	h_ai
Equilibrium	g_equib	g_equib
Hequib	h_equib	h_equib

Table 7-13 "Air Outlet" output

Air Outlet output	Input	
	To	**parameter**
g_ao	TL_dehum	controlled variable
g_ao	Tair_o	absolute humidity ratio
h_ao	Tair_o	Enthalpy of air

Since solution inlet temperature is controlled in order to obtain specific air humidity ratio from dehumidifier, an iterative feedback controller with the name of "TL_dehum" is used to calculate required solution temperature, so desired humidity ratio of air can be reached at "Air Outlet". To do this, calculated outlet air humidity ratio from "Air outlet" is connected to "TL_dehum" and the latter is connected to "Dehumidifier E" and "Equilibrium" to provide solution inlet temperature. Table 7-14 and Table 7-15 demonstrate iterative feedback controller's input and output. Parameter values are left to default so are not mentioned here.

Table 7-14 "TL_dehum" input

Output		TL_dehum input
From	**parameter**	
Air Outlet	g_ao	controlled variable
Humidity Controller	Control signal	Setpoint

Table 7-15 "TL_dehum" output

TL_dehum output	Input	
	To	**parameter**
Control signal	Dehumidifier E	T_Li
Control signal	Equilibrium	T_Li
Control signal	in-out condition	T_Li

In this way, feedback controller reads humidity ratio of outlet air from "Air Outlet" and compare it to set point value which is from "Humidity controller" and determines required inlet solution temperature and gives it to "Dehumidifier E" and "Equilibrium" with the aim of correct outlet air humidity ratio from "Air Outlet". After obtaining right value, outlet air humidity ratio and enthalpy is sent to "Tair_o", Type 33a, to obtain air outlet temperature. Parameter, input and output of the Type is displayed in Table 7-16 to Table 7-18.

Table 7-16 "Tair_o" parameter

Name	Value	Unit
Psychometric mode	6	-
Wet bulb mode	1	-
Error mode	2	-

Table 7-17 "Tair_o" input

Output		Tair_out input
From	**parameter**	
Air Outlet	g_ao	Absolute humidity ratio
Air Outlet	h_ao	Enthalpy of air

Table 7-18 "Tair_o" output

Tair_out output	Input	
	To	parameter
Dry bulb temperature	in-out condition	T_ao

Finally an Equation is created as the output of dehumidifier and named “in-out condition” to collect available air and solution inlet and outlet states and flow rates data to calculate other required information, based on Eq. (6-6) to Eq. (6-10), to be used in cycle. Table 7-19 and demonstrates collected data from dehumidifier and forwarding them to the rest of cycle model.

Table 7-19 "in-out condition" input

Output		Air Outlet input
From	parameter	
Sup fan 1	Outlet flow rate	mdot_ai
Sup fan 1	Outlet air temperature	T_ai
Sup fan 1	Outlet humidity ratio	g_ai
Supply air Psy	Enthalpy	h_ai
Air outlet	g_ao	g_ao
Air outlet	h_ao	h_ao
Tair_o	Dry bulb temperature	T_ao
Dehumidifier E	mdot_sol_i	mdot_sol_i
TL_dehum	Control signal	T_Li

Table 7-20 "in-out condition" output

in-out condition output	Input	
	To	parameter
T_ao	Sup fan 2	Inlet air temperature
g_ao	Sup fan 2	Humidity ratio
mdot_ao	Sup fan 2	air flow rate
X_o	Splitter	X_o
T_Lo	Splitter	Ts_o
mdot_Lo	Splitter	mdot_sol
T_Li	mdot_water 2	set point

7.2.2.2. Supply fan 2

Since pressure drop at dehumidifier is high, a secondary fan is installed so leaving air from dehumidifier would have sufficient pressure to pass the cooling coil. So a variable speed fan, Type 744, is adopted in model which is the same as previously used fan models. Parameter, input and outputs connections of the component is available in Table 7-21, Table 7-22 and Table 7-23.

Table 7-21 "Supply fan 2" parameter

Name	Value	Unit
Humidity mode	1	-
Rated flow rate	6.5	kg/s
Rated power	5.4	kW
Motor efficiency	0.8	-
Motor heat loss fraction	0	-
Number of power coefficients	1	-
Power coefficient	1	-

Table 7-22 "Supply fan 2" input

Output		Supply fan 2 input
From	parameter	
Dehumidifier (in-out condition)	T_ao	Inlet air temperature
Dehumidifier (in-out condition)	g_ao	Humidity ratio
Dehumidifier (in-out condition)	mdot_ao	air flow rate

Table 7-23 "Supply fan 2" output

Supply fan 2 output	Input	
	To	parameter
Outlet air temperature	Cooling coil	Air inlet temperature
Outlet humidity ratio	Cooling coil	Humidity ratio
Outlet flow rate	Cooling coil	Air flow rate
Power consumption	kJ/h -> kW	Fan 2 power

7.2.2.3. Cooling coil

After secondary fan, dry air is brought into contact with chilled water for cooling purpose. Since no condensation occurs at the coil, a different cooling coil is used from what has been adopted in CONVENTIONAL COOLING CYCLE MODEL, a “simple cooling coil using bypass fraction; temperature controlled” with Type number of 752f. Set point temperature for cooling is obtained by “Temperature controller” iterative feedback controller. In fact, this Type only calculates heat transfer rate between air and water according to air flow rate and inlet and outlet temperatures. Tables of this component is presented below.

Table 7-24 "Cooling coil" parameter

Name	Value	Unit
Control mode	1	-
Humidity mode	1	-

Table 7-25 "Cooling coil" input

Output		Cooling coil input
From	**parameter**	
Temperature controller	Control signal	Setpoint: outlet air temperature
Supply fan 2	Outlet air temperature	Air inlet temperature
Supply fan 2	Outlet humidity ratio	Humidity ratio
Supply fan 2	Outlet flow rate	Air flow rate

Table 7-26 "Cooling coil" output

Cooling coil output	Input	
	To	**parameter**
Outlet air temperature	mdot_water	Q_cooling
Air side heat transfer	mdot_water	Ta_o

7.2.3. Desiccant solution cycle components

7.2.3.1. Splitter

Desiccant solution cycle is started from dehumidifier where diluted, warm solution leaves dehumidifier and separated into two streams, one to be regenerated with concentration 34.5% (subscript r) and the other to be used again only by being cold with concentration of 35.2% (subscript u). Two do this an Equation is used and named "Splitter" which calculates mass flow rate of each stream according to Eq. (6-14) and Eq. (6-15). Input and outs of the Equitation is shown in Table 7-27 and Table 7-28.

Table 7-27 "Splitter" input

Output		Splitter input
From	**parameter**	
Dehumidifier (in-out condition)	T_Lo	T_Lo
Dehumidifier (in-out condition)	X_o	X_o

Table 7-28 "Splitter" output

Splitter output	Input	
	To	**parameter**
mdot_u	Mixer	mdot_u
T_Lo	Mixer	T_Lo
mdot_r	Reg pump	mdot_r
T_Lo	Reg pump	T_Lo

7.2.3.2. Regenerator Pump

A ratio of solution with concentration of 34.5% and temperature of dehumidifier outlet temperature is forwarded to regenerator to be regenerated so is passed through a variable speed pump, Type 742, which is called “Reg Pump” which provides sufficient power for solution to overcome drops of the path. Parameter values are demonstrated in Table 7-29. At this point solution concentration is 34.5% and average temperature during a conventional operation is 27^{o} C (after running the model), hence solution specific thermal capacity and density are calculated for this values according to Eq. (6-22) and Eq. (6-24).

Table 7-29 "Reg pump" parameter

Name	Value	Unit
Fluid specific heat	2.822	kJ/kg.K
Fluid density	1213	kg/m^3
Motor het loss fraction	0	-

Also some input data to the component are considered constant, as Table 7-30.

Table 7-30 "Reg pump" constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	30	kPa

Other input and output connections of the component are presented in Table 7-31 and Table 7-32.

Table 7-31 "Reg pump" input

Output		Reg pump input
From	parameter	
Splitter	T_Lo	Inlet fluid temperature
Splitter	mdot_r	Inlet fluid flow rate

Table 7-32 "Reg pump" output

Reg pump output	Input	
	To	parameter
Outlet fluid temperature	Solution H.E	Source side inlet temperature
Outlet fluid flow rate	Solution H.E	Source side flow rate
Power consumption	kJ/hr->kW	Reg pump power

7.2.3.3. Solution heat exchanger

After passing the pump, diluted solution is preheated by the hot, strong solution coming back from regenerator in a heat exchanger which is named “Solution H.E”. A zero capacitance heat exchanger is adopted, Type 91, with the parameters in Table 7-33. Specific heat of diluted solution is the same as “Reg pump” but that of the strong solution, on the other side of heat exchanger, is calculated for a solution with 35.2% concentration and temperature of 57^{o} C.

Table 7-33 "Solution H.E" parameter

Name	Value	Unit
Heat exchanger effectiveness	0.7	-
Specific heat f source side fluid	2.845	kJ/kg.K
Specific heat f load side fluid	2.868	kJ/kg.K

Input and output connection between this component and the others are listed in Table 7-34 and Table 7-35.

Table 7-34 "Solution H.X" input

Output		Solution H.E input
From	**parameter**	
Reg pump	Outlet fluid temperature	Source side inlet temperature
Reg pump	Outlet fluid flow rate	Source side flow rate
Return pump	Outlet fluid temperature	Load side inlet temperature
Return pump	Outlet fluid flow rate	Load side flow rate

Table 7-35 "Solution H.X" output

Solution H.E output	Input	
	To	**parameter**
Source side outlet temperature	Reg heater	Inlet fluid temperature
Source side flow rate	Reg heater	Inlet mass flow rate
Load side outlet temperature	Mixer	T_from_reg
Load side flow rate	Mixer	mdot_from_reg
Source side outlet flow rate	Regenerator (Req-evap)	mdot_solution
Source side outlet temperature	Water H.X.	Load side inlet temperature
Source side flow rate	Solar H.X	Load side flow rate

7.2.3.4. Heater

After preheating and just before regeneration, solution is heated up to desired temperature in a liquid heater which is name “Reg heater”. Type 6 is used for this purpose and it is assumed the component is operated with natural gas, hence efficiency is considered 0.79 [64]. Parameter values are as

Table 7-36 "Reg heater" parameter

Name	Value	Unit
Maximum heating rate	1000000	kJ/hr
Specific heat of fluid	2.917	kJ/kg.K
Overall loss coefficient	0	kH/hr.K
Efficiency of auxiliary heater	0.79	-

At here, maximum heating rate is considered at high magnitude, so the heater will have sufficient power to heat solution to desired temperature under any circumstances. Also specific heat of fluid is calculated according to diluted solution concentration and outlet temperature from heat exchanger, 54^{o} C. Input and outputs are presented in Table 7-37 and Table 7-38.

Table 7-37 "Reg heater" input

Output		Reg Heater input
From	**parameter**	
Weather data	Dry bulb temperature	Temperature of surrounding
Regenerator (TL_reg)	Control signal	Setpoint temperature
Solution H.E	Source side outlet temperature	Inlet fluid temperature
Solution H.E	Source side flow rate	Inlet mass flow rate

Table 7-38 "Reg heater" output

Reg heater output	Input	
	To	**parameter**
Outlet fluid flow rate	Regenerator (Gandhidasan reg)	mdot_solution_i
Required heating rate	kJ/hr->kW	Heater power

Since heater's outlet temperature is defined by regenerator (Table 7-37), one link is applied from regenerator toward heater and consequently the link from heater to regenerator only includes solution flow rate (Table 7-38).

7.2.3.5. Regenerator

Similar to dehumidifier but in a simpler manner, regenerator is developed by utilizing two Equations and an iterative feedback controller. Regenerator macro is illustrated in Figure 7-4.

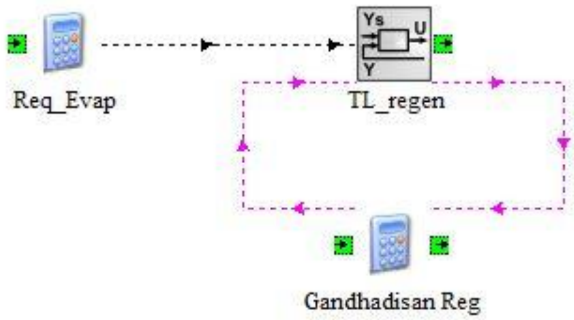


Figure 7-4 Inside of Dehumidifier macro

"Req_Evap" Equation calculates required evaporation rate according to Eq. (6-19) by knowing inlet and outlet solution concentration, 34.5% and 35.2% respectively, and solution flow rate, from "Solution H.E". Calculated evaporation rate then is forwarded to an iterative feedback controller, Type 22, which is named "TL_regen" as set point value. Input and output of "Req_Evap" is shown in Table 7-39 and Table 7-40.

Table 7-39 "Req_Evap" input

Output		Req_Evap input
From	**parameter**	
Solution H.E	Source side flow rate	mdot_solution

Table 7-40 "Req_Evap" output

Req_Evap output	Input	
	To	**parameter**
mdot_req_evap	TL_regen	set point

Duty of "TL_regen" is to determine required solution inlet temperature so desired evaporation occurs in dehumidifier and it is done in accordance with "Gandhidasan Reg". Similar to other Type 22s, parameter values are not changed and left default but input and output are available in Table 7-41 and Table 7-42.

Table 7-41 "TL_Regen" input

Output		TL_Regen input
From	**parameter**	
Req_Evap	mdot_req_evap	Set point
Gandhidasan Reg	mdor_evap	Controlled variable

Table 7-42 "TL_Regen" output

TL_Regen output	Input	
	To	**Parameter**
Control signal	Gandhidasan Reg	Solution inlet temperature
Control signal	Reg heater	Set point

Evaporation rate equation, Eq. (6-11), is defined in "Gandhidasan Reg" Equation. As mentioned before, air flow rate next to its inlet and outlet temperatures are based on those of Table 6-1, so β and other parameters are known except $T_{L,i}$ and q. Thus, evaporation rate from "Gandhidasan Reg" is connected to feedback controller as controlled variable and control signal of the latter is connected to the former as solution inlet temperature. Consequently, by iteration suitable solution inlet temperature is determined and used in "Reg heater" as set point, so required thermal energy for heating solution at heater would be obtained. Note that solution specific heat at dehumidifier is calculated for concentration of 34.5% and 61° C which is the average solution inlet temperature into regenerator. Solution outlet temperature is obtained according to Fumo and Goswami's experiments [24] which is about 6.38° C lower than its inlet temperature. Finally solution outlet flow rate is calculated by Eq. (6-18). Input and output connections of "Gandhidasan Reg" Equation is listed in Table 7-43 and Table 7-44.

Table 7-43 "Gandhidasan Reg" input

Output		Gandhidasan Reg input
From	**parameter**	
TL_Regen	Control signal	Solution inlet temperature
Reg Heater	Outlet fluid flow rate	mdot_sol_i

Table 7-44 "Gandhidasan Reg" output

Gandhidasan Reg output	Input	
	To	**parameter**
mdot_sol_o	Return pump	Inlet fluid temperature
T_sol_o	Return pump	Inlet fluid flow rate
T_sol_i	Solar mass flow	Set point

7.2.3.6. Return pump

A variable speed pump, Type 742, which is called "Return Pump" pumps the hot, strong solution to "Solution H.E" and "Dehumidifier H.X." due to high pressure drop of these units. Parameter values are demonstrated in Table 7-45. At this point solution concentration is 35.2% and average temperature during a conventional operation is 57° C (after running the model), hence solution specific heat and density are calculated for this values according to Eq. (6-22) and Eq. (6-24).

Table 7-45 "Return pump" parameters

Name	Value	Unit
Fluid specific heat	2.911	kJ/kg.K
Fluid density	1204	kg/m^3
Motor het loss fraction	0	-

Also some input data to the component are considered constant, as Table 7-46.

Table 7-46 "Return pump" constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	20	kPa

Other input and output connections of the component is presented in Table 7-47and Table 7-48.

Table 7-47 "Return Pump" input

Output		Return pump input
From	parameter	
Regenerator (Gandhidasan Reg)	mdot_sol_o	Inlet fluid temperature
Regenerator (Gandhidasan Reg	T_sol_o	Inlet fluid flow rate

Table 7-48 "Return pump" output

Return pump output	Input	
	To	parameter
Outlet fluid temperature	Solution H.E	Load side inlet temperature
Outlet fluid flow rate	Solution H.E	Load side flow rate
Power consumption	kJ/hr->kW	Return pump power

After passing “Return pump”, solution is brought into contact with cold, diluted solution from dehumidifier at “Solution H.E” heat exchanger for precooling which was described in 7.2.3.3.

7.2.3.7. Mixer

At mixer, regenerated strong stream with mass flow rate of m_b is combined with the other part of solution from dehumidifier, mass flow rate of m_r, with both at the same concentration but temperatures, Figure 6-2. To model the combination, an Equation with the name of “Mixer” is created. Mass and enthalpy of mixture is calculated by Eq. (7-1) and Eq. (7-2).

$$\dot{M}_o = \dot{M}_u + \dot{M}_b \qquad (7-1)$$

$$h_o = \frac{\dot{M}_u \times h_u + \dot{M}_{b\times} h_b}{\dot{M}_o} \qquad (7-2)$$

Enthalpy of each stream is calculated by Eq. (6-21). Finally, enthalpy of the mixture is converted to temperature because its concentration in known. Connections of the Equation are available in Table 7-49 and Table 7-50.

Table 7-49 "Mixture" input

Output		Mixer input input
From	**parameter**	
Solution H.E	Load side outlet temperature	mdot_from_reg
Solution H.E	Load side flow rate	T_from_reg
Splitter	mdot_u	mdot_from_dehum
Splitter	T_Lo	T_from_dehum

Table 7-50 "Mixture" output

Mixer output	Input	
	To	**parameter**
mdot_mixture	Cold solution pump	Inlet fluid flow rate
T_mixture	Cold solution pump	Inlet fluid temperature

7.2.3.8. Cold solution pump

This pump is used to force solution to pass through “Dehumidifier H.X.”heat exchanger and dehumidifier. Adopted component is a variable speed pump, Type 742, and named “Cold solution” pump. At this point solution specific heat and density are calculated for 35.2% of concentration and temperature of 27^{o}C. Parameter, input and output window data are set according to Table 7-51 to Table 7-54.

Table 7-51 "Cold solution" parameter

Name	Value	Unit
Fluid specific heat	2.805	kJ/kg.K
Fluid density	1218	kg/m^3
Motor het loss fraction	0	-

Table 7-52 "Cold solution" constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	20	kPa

Table 7-53 "Cold solution" input

Output		Air Outlet input
From	**parameter**	
Mixer	T_mixture	Inlet fluid temperature
Mixer	mdot_mixture	Inlet fluid flow rate

Table 7-54 "Cold solution" output

Reg pump output	Input	
	To	**parameter**
Outlet fluid temperature	Dehumidifier H.X.	Source side inlet temperature
Outlet flow rate	Dehumidifier H.X.	Source side flow rate
Power consumption	kJ/hr->kW	Cold solution power

7.2.3.9. Dehumidifier H.X.

“Dehumidifier H.X” is a heat exchanger installed jest before dehumidifier to cool the solution down to desired temperature by chilled water which is described further in 7.2.4.2. Similar to previously used heat exchangers, this is a zero capacitance heat exchanger, Type 91, with parameters and connections in Table 7-55 to Table 7-57. Solution density and heat capacity are calculated for concentration and temperature of 35.2% and 27^{o}C respectively.

Table 7-55 "Dehumidifier H.X." parameter

Name	Value	Unit
Heat exchanger effectiveness	0.7	-
Specific heat f source side fluid	4.19	kJ/kg.K
Specific heat f load side fluid	2.801	kJ/kg.K

Table 7-56 "Dehumidifier H.X." input

Output		Dehumidifier H.X input
From	**parameter**	
Cold solution pump	Outlet fluid temperature	Load side inlet temperature
Cold solution pump	Outlet flow rate	Load side flow rate
Solution chiller	Outlet fluid temperature	Source side inlet temperature
mdot_water-2	Control signal	Source side flow rate

Table 7-57 "Dehumidifier H.X." output

Dehumidifier H.X. output	Input	
	To	**parameter**
Source side outlet temperature	Chilled water pump	Inlet fluid temperature
Load side outlet temperature	mdot_water-2	Controlled variable

According to Table 7-57 outlet solution from heat exchanger is not connected to dehumidifier but solution cycle is finished. However required inlet solution temperature to dehumidifier is calculated by “Dehumidifier” macro and is sent chilled water cycle as set point, to provide solution with that temperature at the exit of heat exchanger, so required power for cooling can be calculated and this is the aim of current model.

7.2.3.10. Hot water application

In addition to solution cycle main components, two extra components are also adopted to this cycle to investigate regeneration process if a hot water flow is used for heating solution obtained from any thermal source such as solar, geothermal, combustion and etc. In fact, these two components calculate hot water demand for solution heating to make hot solution before regenerator. This is done by utilizing a Type 91, “Water H.X.” and a Type 22, “Water mass flow rate”, Figure 7-2, and considering constant temperature of hot source, 75^{o} C.

Inlet of load side of “Water H.X.” is connected to outlet of source side of “Solution H.E” which means one side’s input of “Water H.X.” is preheated solution. On the other hand, the other side of heat exchanger is connected to “Water mass flow rate” to calculate required hot source flow rate while set point is specified by regenerator macro. Parameter, input and output of heat exchanger and controller are displayed in Table 7-58 to Table 7-60 and Table 7-61 to Table 7-63 respectively.

Table 7-58 "Water H.X." parameter

Name	Value	Unit
Heat exchanger effectiveness	0.7	-
Specific heat f source side fluid	4.19	kJ/kg.K
Specific heat f load side fluid	2.92	kJ/kg.K

Table 7-59 "Water H.X." input

Output		Water H.X. input
From	parameter	
Solution H.E	Outlet fluid temperature	Load side inlet temperature
Solution H.E	Outlet flow rate	Load side flow rate
Solar mass flow	Control signal	Source side flow rate

Table 7-60 "Water H.X." output

Water H.X. output	Input	
	To	parameter
Load side outlet temperature	Water mass flow rate	Controlled variable

Table 7-61 "Water mass flow rate" parameter

Name	Value	Unit
Mode	0	-
Maximum number of oscillations	10000	-

Table 7-62 "Water mass flow rate" input

Output		Water mass flow rate Input
From	parameter	
Regenerator (Gandhidasan Reg)	T_Li	Set point
Water H.X.	Load side outlet temperature	controlled variable

Table 7-63 "Water mass flow rate" output

Water mass flow rate output	Input	
	To	parameter
Control signal	Water H.X.	Source side flow rate

7.2.4. Chilled water cycle components

While there exist one chilled water cycle with one chiller in real application for both solution and supply air cooling, at the current model it is divided into two separate cycles with two chiller units due to convergence problem at run.

7.2.4.1. Supply air cooling

In this cycle, produced chilled water from chiller is forwarded to cooling coil to reduce temperature of supply air to demanded value. As mentioned above, adopted cooling coil component only calculates air sided heat transfer with no data for water flow rate and temperature. Hence water inlet and outlet temperature to and from cooling coil next to required flow rate is calculated externally and used in water cycle.

7.2.4.1.1. Water flow rate

To calculate required water flow rate, an Equation with the name of “mdot_water” is created which determine water flow based on air side heat transfer rate from cooling coil and according to Eq. (7-3).

$$\dot{M}_w = \frac{Q_{air}}{C_{p,w} \times (T_{w,o} - T_{w,i})} \qquad (7-3)$$

It is obvious that water inlet and outlet temperature to and from dehumidifier is required to solve this equation. Similar to conventional cycle and with the purpose of comparison, chilled water temperature is set to its maximum values, 15^o C, according to required solution temperature. For coil, water outlet temperature is 3^o C is higher than its inlet value which is a suitable assumption for non-condensation coils [81]. Inouts and outputs are shown in Table 7-64 and Table 7-65.

Table 7-64 "mdot_water" input

Output		mdot_water input
From	**Parameter**	
Cooling coil	Air side heat transfer	q_air

Table 7-65 "mdot_water" output

mdot_water output	Input	
	To	**parameter**
T_wo	Chilled water pump	Inlet fluid temperature
mdot_water	Chilled water pump	Inlet fluid flow rate

7.2.4.1.2. Chilled water pump

Cooling coil's leaving water goes to a pump to be forwarded toward chiller for chilling. Hence a variable speed pump, Type 742, is used after "mdot_water" Equation with parameters and connections in Table 7-66 to Table 7-69.

Table 7-66 "Chilled water pump" parameter

Name	Value	Unit
Fluid specific heat	4.19	kJ/kg.K
Fluid density	1000	kg/m^3
Motor het loss fraction	0	-

Table 7-67 "Chilled water pump" constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	10	kPa

Table 7-68 "Chilled water pump" input

Output		Chilled water pump input
From	**parameter**	
mdot_water	T_wo	Inlet fluid temperature
mdot_water	mdot_water	Inlet fluid flow rate

Table 7-69 "Chilled water pump" output

Chilled water pump output	Input	
	To	parameter
Outlet fluid temperature	Air chiller	Chilled water inlet temperature
Outlet flow rate	Air chiller	Chilled water flow rate
Power consumption	kJ/h->kW	Chilled water pump power

7.2.4.1.3. Air chiller

An air-cooled, vapor compression chiller, Type 655 with the name of "Air chiller", is used for producing 15° C chilled water to be used in cooling coil. Since adopted coil, does not include water properties and flow rate, this cycle will be finished after chiller and only consumed energy by chiller is the output. Chiller data are provided by two external files, "Chiller data" and "Chiller part-load data" and are editable, however these files are not changed. All chiller parameter are default values because they should fit chiller data.

Table 7-70 "Air chiller" parameter

Name	Value	Unit
Rated Capacity	1266000	kJ/hr
Rated C.O.P	3	kg/m^3
Logical unit for performance data	34	
Logical unit for PLR data file	35	
Fluid specific heat	4.19	kJ/kg.K
Number of ambient temperatures	5	
Number of CHW set points	6	
Number of part load ratios	5	-

Table 7-71 "Air chiller" input

Output		Chiller Input
From	parameter	
Chilled water pump	Outlet fluid temperature	Chilled water inlet temperature
Chilled water pump	Outlet flow rate	Chilled water flow rate
Weather data	Dry bulb temperature	Ambient temperature
Constant	Chilled water temp	Setpoint
On/Off	Instantaneous value of function over the time step	Chiller control signal

Table 7-72 "Air chiller" output

Chiller output	Input	
	To	parameter
Chiller power	kJ/hr->kW	air_chiller_power

7.2.4.2. Solution cooling

Chilled water is also used for cooling solution to make it ready to absorb supply air moisture at dehumidifier and this process is done at “Dehumidifier H.X” by the second chilled water cycle. The cycle consists of a chiller, pump, iterative feedback controller and a heat exchanger.

7.2.4.2.1. Iterative feedback controller

An iterative feedback controller, Type 22, is connected to heat exchanger and called “mdot_water2” to determine required chilled water flow rate so that solution outlet temperature at the other side would be at desired magnitude. To do this, solution outlet temperature is connected to controller as controlled variable while control signal of the latter is connected to chilled water flow rate because water inlet temperature is constant at 15^{o} C. Also, required temperature of solution is specified by “Dehumidifier” macro and is the set point for the controller. Parameter, input and output of the component is listed in Table 7-73 to Table 7-75.

Table 7-73 "mdot_water2" parameter

Name	Value	Unit
Mode	0	-
Maximum number of oscillations	10000	-

Table 7-74 "mdot_water2" input

Output		mdot_water2 Input
From	parameter	
Dehumidifier (in-out condition)	T_Li	Set point
Dehumidifier H.X.	Load side outlet temperature	controlled variable

Table 7-75 "mdot_water2" output

mdot_water2 output	Input	
	To	parameter
Control signal	Dehumidifier H.X	Source side flow rate
Control signal	Chilled water pump	Inlet fluid flow rate

7.2.4.2.2. Chilled water pump 2

After obtaining required water flow rate at heat exchanger, its outlet temperature is available from heat exchanger outputs and is connected to a variable speed pump, Type 742, with the name of “Chilled water pump2”. In addition, water flow rate is transferred to pump by feedback controller through control signal value. So the pump sends the water to chiller for cooling to be used gain. Parameter, input and output are set according to Table 7-76 to Table 7-79.

Table 7-76 "Chilled water pump 2" parameter

Name	Value	Unit
Fluid specific heat	4.19	kJ/kg.K
Fluid density	1000	kg/m^3
Motor het loss fraction	0	-

Table 7-77 "Chilled water pump 2" constant inputs

Name	Value	Unit
Overall pump efficiency	0.6	-
Motor efficiency	0.9	-
Pressure drop	20	kPa

Table 7-78 "Chilled water pump 2" input

Output		Chilled water pump 2 input
From	parameter	
Dehumidifier H.X	Source side outlet temperature	Inlet fluid temperature
mdot_water2	Control signal	Inlet fluid flow rate

Table 7-79 "Chilled water pump 2" output

Chilled water pump 2 output	Input	
	To	parameter
Outlet fluid temperature	Solution chiller	Chilled water inlet temperature
Outlet flow rate	Solution chiller	Chilled water flow rate

7.2.4.2.3. Solution chiller

Adopted chiller model is the same as the one used for cooling supply air, Type 655 and is named "Solution chiller". Aim of this chiller is to provide suitable chilled water flow rate at 15^o C to be used at heat exchanger for solution cooling. All parameter values of the Type is the same with "Air chiller" so is available at Table 7-70. Input and output values are connected as below.

Table 7-80 "Solution chiller" input

Output		Chiller Input
From	parameter	
Chilled water pump 2	Outlet fluid temperature	Chilled water inlet temperature
Chilled water pump 2	Outlet flow rate	Chilled water flow rate
Weather data	Dry bulb temperature	Ambient temperature
Constant	Chilled water temp	Setpoint
On/Off	Instantaneous value of function over the time step	Chiller control signal

Table 7-81 "Solution chiller" output

Solution chiller output	Input	
	To	**parameter**
Chiller power	kJ/hr->kW	sol_chiller_power
Outlet fluid temperature	Dehumidifier H.X	Source side inlet temperature

7.2.5. Outputs

Finally, after developing whole cycles, power consumptions of different units are gathered in Equation “kJ/hr->kW” to be used in online plotters and printers. The Equation input connections are listed in Table 7-82.

Table 7-82 "kJ/hr->kW" input

Output		Chiller Input
From	**parameter**	
Air chiller	Chiller power	air_chiller_power
Chilled water pump	Power Consumption	Chilled water pump power
Recirc fan	Power Consumption	Recirc fan power
Supply fan 1	Power Consumption	fan 1 power
Supply fan 2	Power Consumption	fan 2 power
Reg Pump	Power Consumption	Reg pump power
Reg heater	Required heating rate	Heater power
Return pump	Power Consumption	Return pump power
Cold solution pump	Power Consumption	Cold solution power
Chilled water pump 2	Power Consumption	Chilled water pump 2 power
Solution chiller	Chiller power	sol_chiller_power

CHAPTER 8

RESULTS AND DISCUSSION

Results are obtained by connecting desired outputs to online plotters and/or printers to save them in Excel format. Before running the models, some changes are made to the TRNSYS control card window, Figure 8-1. These changes include simulation start and stop time, time step, tolerance integration and convergence.

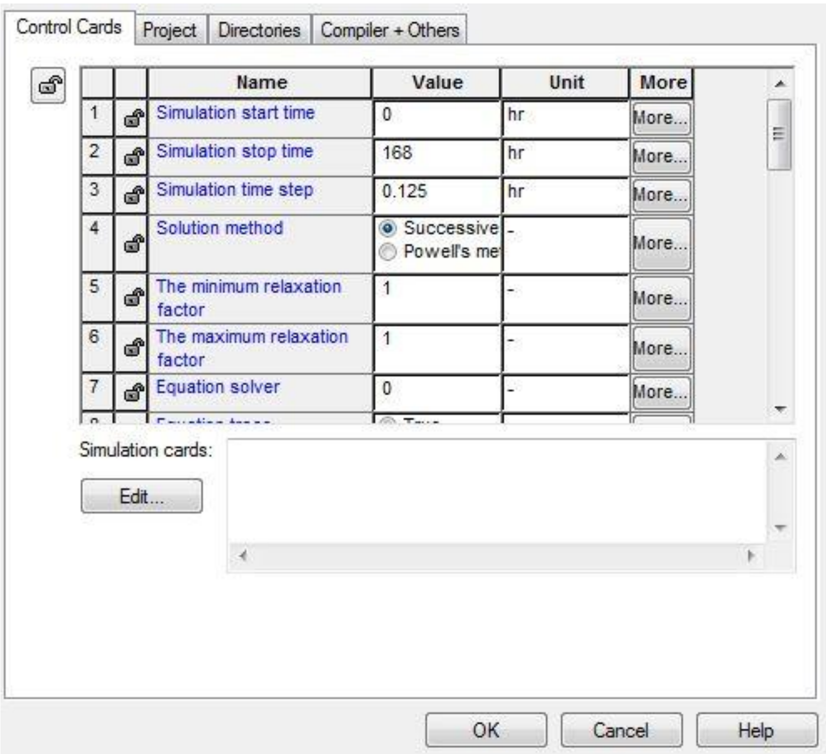


Figure 8-1 TRNSYS setting window

Simulation start and stop time is set according to hours of year for which simulation is run but time step is fixed at 0.125 hour (7.5 minutes). Also, both tolerance integration and convergence are set to 0.01 to reduce running time and avoid convergence problems.

The conventional and desiccant cycle models are run for two periods of times. The first one is for July 21st, as the mid-summer day and to the metrological data set this day is one of the most humid days during the year and for an ordinary summer day, humidity ratio values would be lower to investigate desiccant cooling system behavior for a characteristic day. The second time period is the whole three months of summer, namely June, July and August, to compare the magnitude and type of energy demand of both cycles.

8.1. 21st July analyses

In this section, the performance of the developed hybrid desiccant and conventional cooling cycles are investigated on July 21st to see how they rely on load and ambient conditions. Hence, simulation start and stop time is set to hour 4824 and 4848 respectively.

8.1.1. Ambient, building and supply air states

Before presenting the cycle's outputs, hourly ambient and structure conditions at which the cooling cycles are operated is demonstrated to make it possible to comment on their behavior.

Figure 8-2 and Figure 8-3 show irradiation on building external walls and ambient and indoor design conditions respectively.

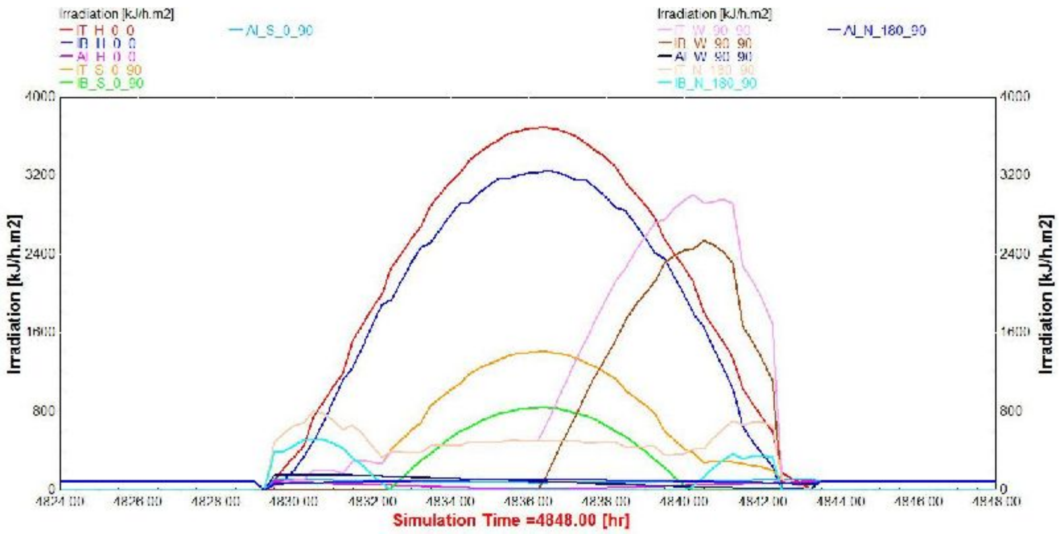


Figure 8-2 Irradiation on July 21st

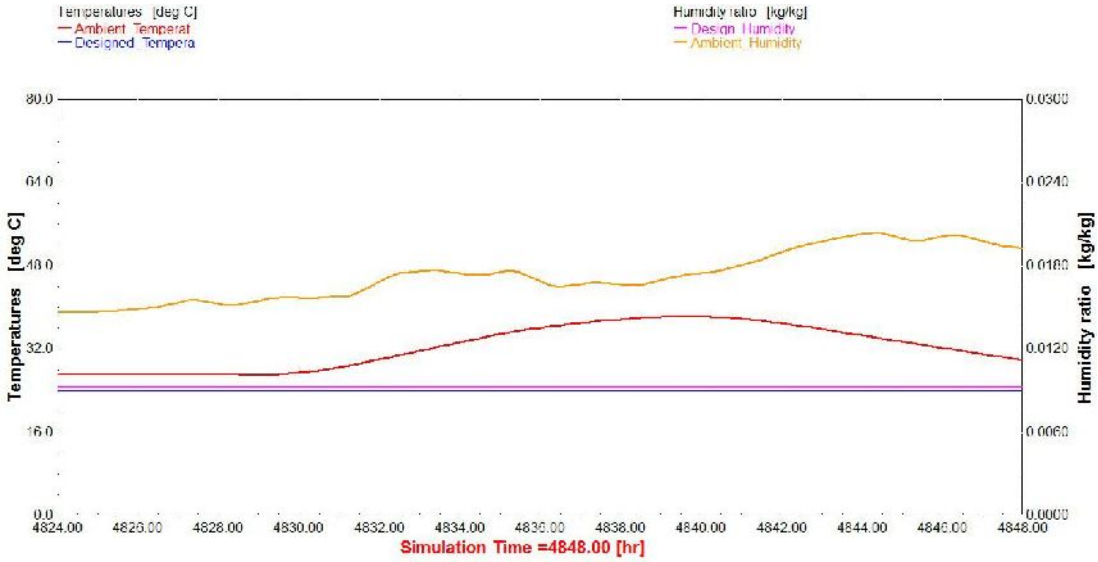


Figure 8-3 Indoor design and ambient temperature and humidity ratio on July 21st

Figure 8-4 shows both sensible and latent loads of building which result from heat transfer through the external walls and internal loads.

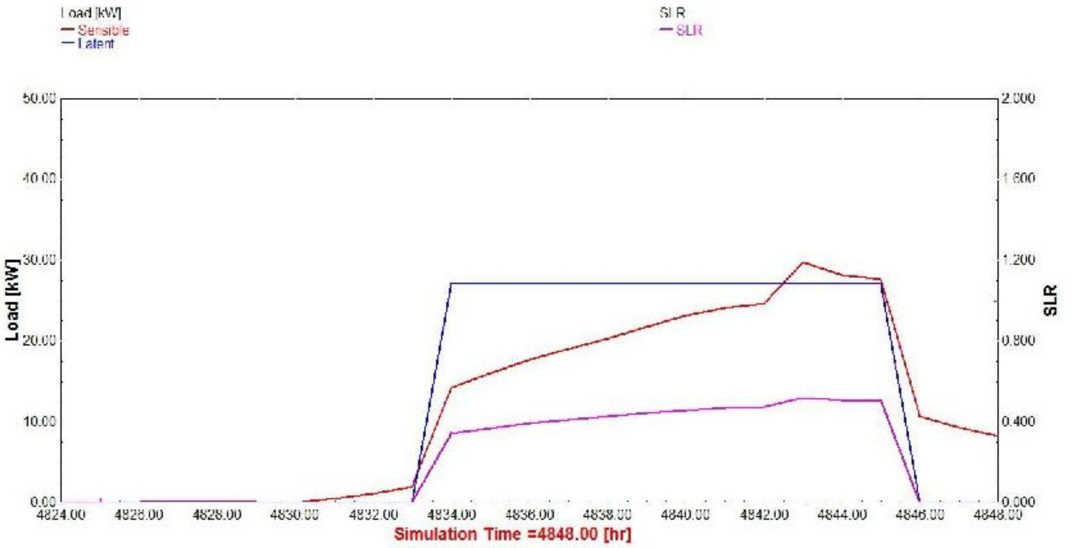


Figure 8-4 Cooling load on July 21st

According to the Figure 8-4, while sensible load varies with time, the latent load is constant. The reason for this is zero infiltration and the only source of moisture generation being occupancy which is considered constant at 80 people. On the other hand, the sensible load strongly depends on ambient conditions, including solar radiation and ambient temperature. The sensible load is lower in the morning because the eastern wall is shaded by a building next to it, Figure 4-1. As the sun altitude gets larger, sensible load increases. A sudden increase occurs at 18:00 due to the lights being turned on.

Sensible and latent load results in increment of indoor temperature and humidity respectively, Figure 8-5. Note that if the relative humidity of the indoor air increases to 100%, any further moisture will be condensed. According to Figure 8-5, two hours of latent load emission is followed by saturated indoor air if no cooling or dehumidifying system is installed.

Preventing indoor air from getting warm and humid requires overcoming sensible and latent loads, done by supply air flow with sufficient low temperature and humidity ratio, as shown in Figure 8-6.

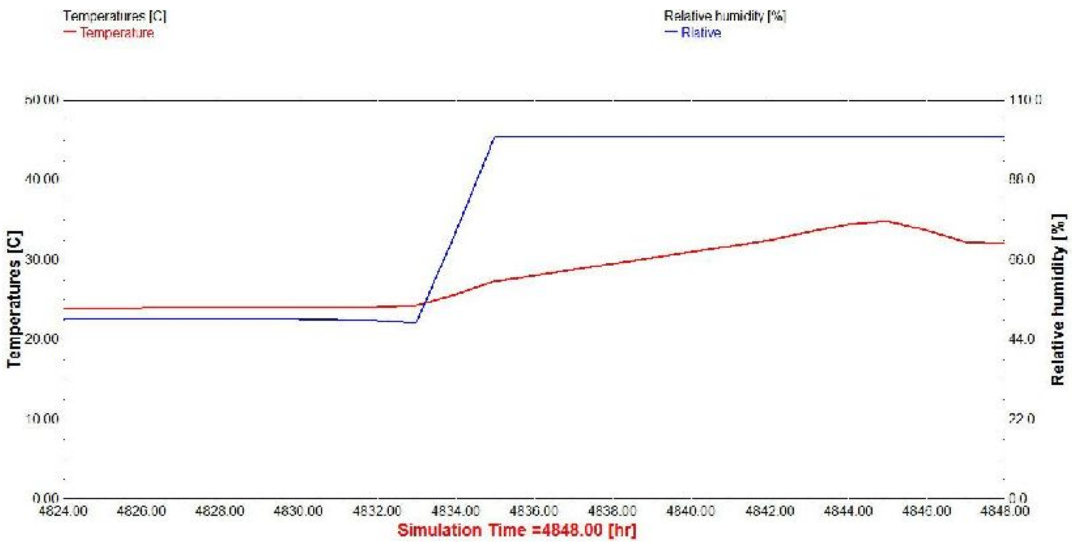


Figure 8-5 Indoor air temperature and relative humidity

As mentioned above, the sensible load in the evening is higher than morning, thus the required temperature of supply air is lower. Also the required humidity ratio is constant, meaning that the latent load does not vary with time.

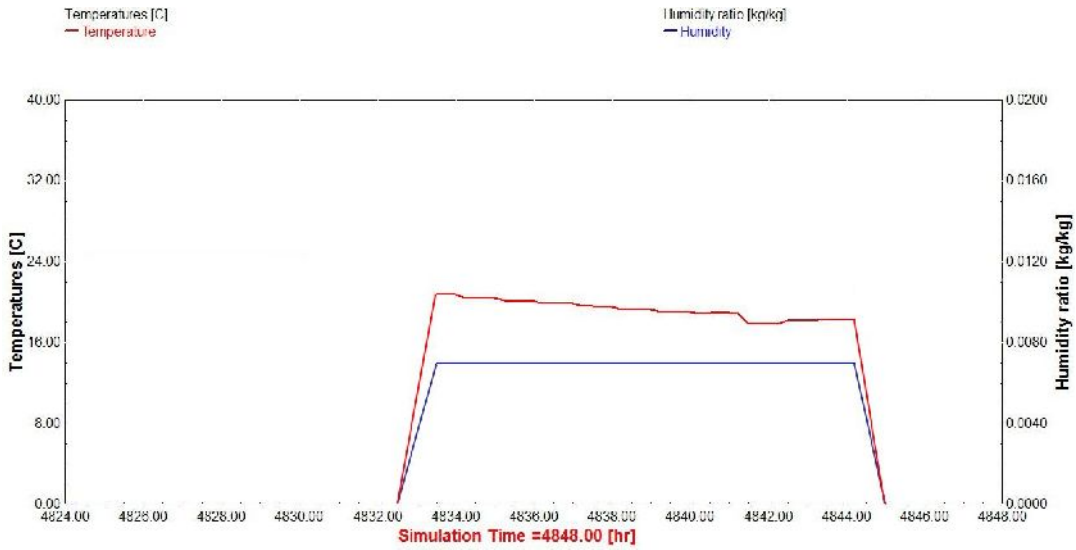


Figure 8-6 Required supply air state

8.1.2. Conventional cooling cycle

Both air cooling and dehumidifying at VCCS occurs at the cooling coil, hence the cooling coil is the most important unit to be investigated. In Figure 8-7 several important outputs of the cooling coil are illustrated.

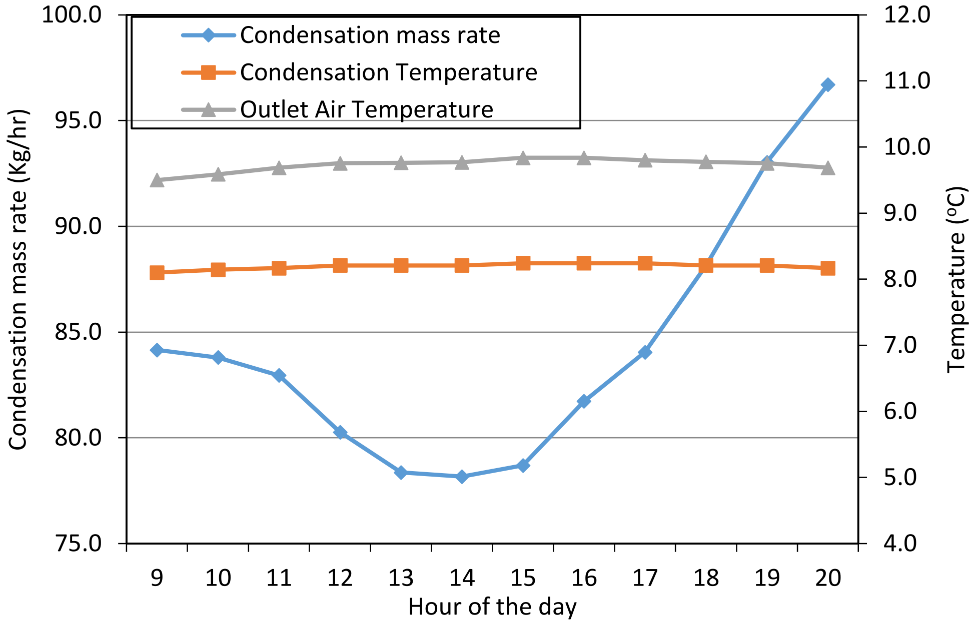


Figure 8-7 Coil outputs of VCCS

Since supply air flow rate and humidity are constant during the operating time, condensation mass flow rate is related to ambient humidity ratio directly. According to Figure 8-3, at midday humidity ratio of the surrounding goes down which is followed by lower condensation rates at that time but higher values are required in the evening due to high ambient humidity. Also condensation temperature is approximately constant at about 8^{o} C all the time because of fixed supply air humidity. Thus 5^{o} C is a suitable inlet chilled water temperature into coil which was considered before. Outlet air temperature is also constant at just below 10^{o} C because inlet chilled water temperature is fixed and it is far below the supply air state, Figure 8-6, thus using reheater for the system is vital. From Figure 8-7, it is obvious there are some fluctuations in condensation and outlet air temperature and it occurs because of coil's bypass fraction but is negligible.

Figure 8-8 shows energy demand of different sectors of VCCS next to COP of chiller. Since supply air temperature is experiencing a downward trend during the day, Figure 8-6, and cooling coil outlet air temperature is constant, Figure 8-7, less reheating is required as time moves forward and consequently lower energy demand. So, required energy of reheater is a function of the sensible load ratio. The higher the sensible load ratio, the lower the reheating demand.

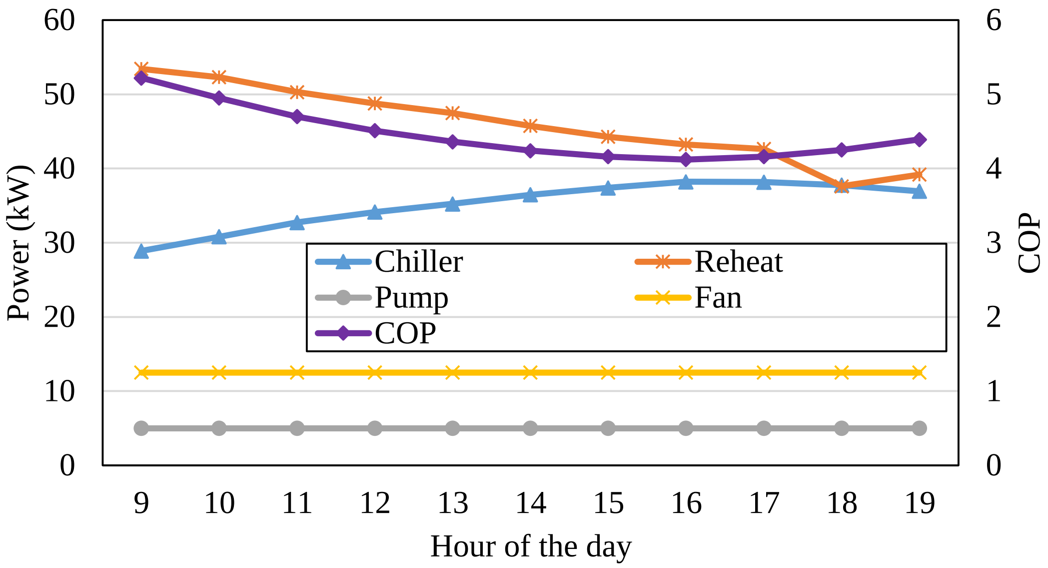


Figure 8-8 VCCS power demand

Being air-cooled, the chiller's condenser temperature depends on ambient temperature, so COP does. Thus, as ambient temperature goes up, COP drops and vice versa, but the COP is independent of ambient humidity. While less condensation occurs at midday than in the morning and evening, energy demand of chiller is higher at that time due to lower COP. Thus, it could be concluded while ambient temperature and sensible load effect chiller and reheater power respectively, ambient humidity does not have any strong impact on cycle energy demand. Other energy consuming units are pumps and fans with constant values at 12.5 and about 5 kW respectively.

8.1.3. Hybrid desiccant cooling cycle

Unlike the VCCS cycle, the dehumidifier is the most important component of the hybrid desiccant cycle, and hence is investigated separately for its characteristics at different circumstances.

Figure 8-9 exhibits temperatures and concentrations at the inlet and outlet of the random packed-bed dehumidifier for the same day, 21 July. Since inlet solution concentration and required humidity ratio of supply air are constant, as the ambient humidity ratio increases, the solution temperature has to be decreased to absorb more humidity, thus its temperature increases at midday when the surrounding humidity ratio is lower compared to morning times.

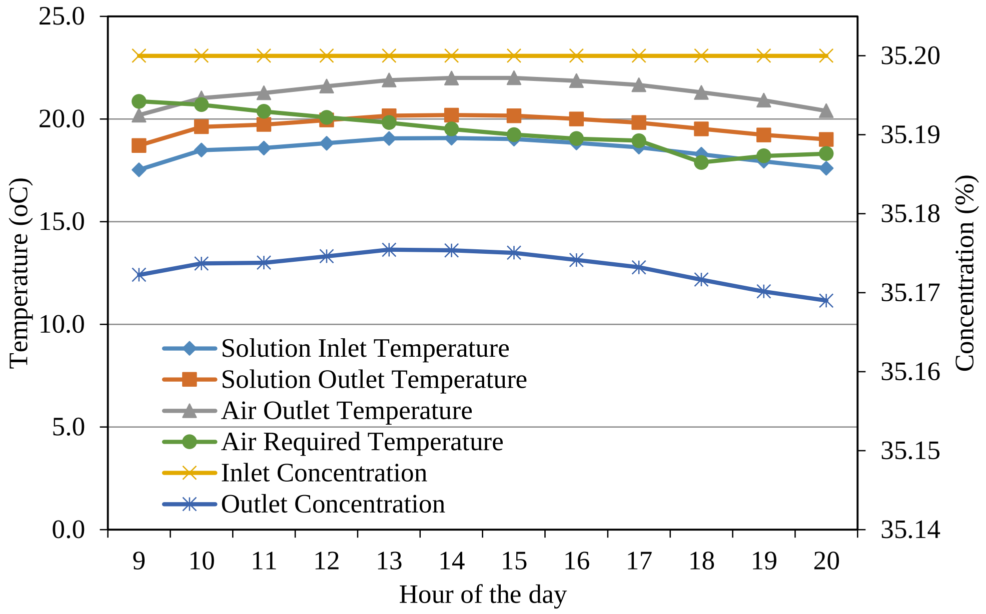


Figure 8-9 Dehumidifier inlet and outlet inputs and outputs

Also it should be noted that the higher ambient temperature at midday does not result in lower solution temperature, hence it could be concluded that the ambient temperature effect on solution inlet temperature is negligible compared to ambient humidity ratio. In the evening, a significant fall in temperature occurs due to the increasing ambient humidity ratio. Following these fluctuations in solution inlet temperature, the solution and air outlet temperatures also fluctuate because their temperatures strongly depend on the solution inlet temperature and this characteristic was also observed in previous studies [24]. From a concentration point of view, since the inlet concentration is constant, as ambient absolute humidity ratio increases, the desiccant solution becomes more diluted, followed by a higher mass flow rate of solution to the regenerator and lower mass flow rate for reuse.

In Figure 8-10 differences between dehumidifier outlet air temperature and that of required supply air is shown for three different summer days, 21st of June, July and August. According to the graph, outlet air temperature from the dehumidifier is higher than the required supply air temperature by only less than 3^{o} C most of the time and is even lower in the early mornings up to 1.5^{o} C, which suggests the the possibility of removing the cooling coil for hybrid desiccant cooling.

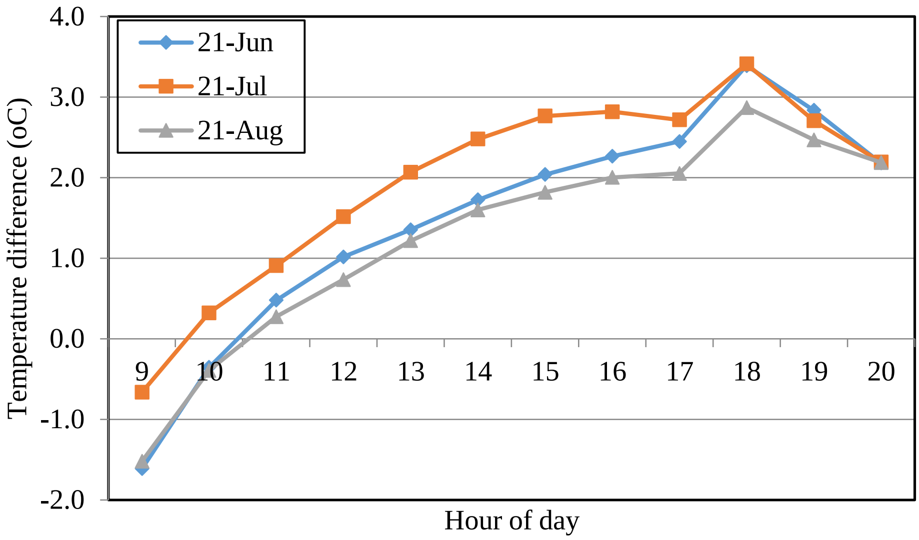


Figure 8-10 Temperature difference

Having lower sensible load, the supply air temperature is higher in morning compared to the rest of day, Figure 8-6, but the ambient humidity ratio is high which is followed by lower inlet desiccant solution and outlet air temperatures as illustrated in Figure 8-9. Differences reach a maximum of 2.8° C at hour 16 where ambient humidity ratio is low and is followed by higher solution temperatures. Also there exists a sudden rise at hour 18 when lights are turned on, resulting in an increment in the sensible load and lower supply air desired temperature.

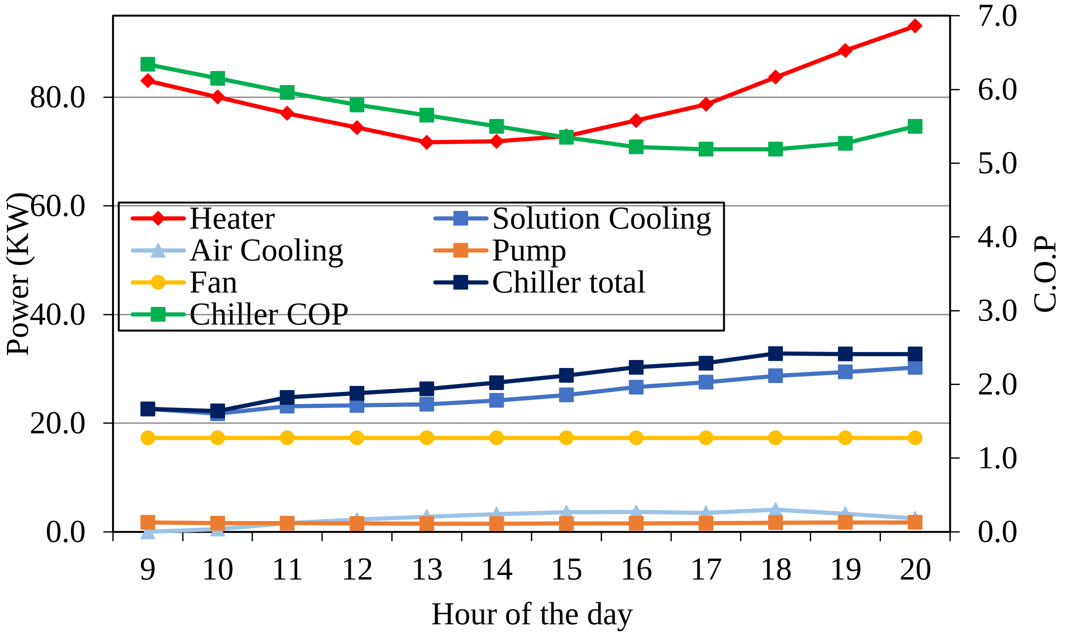


Figure 8-11 DCS power demand

From an energy point of view, the DCS power requirement and chiller COP on July 21st are displayed in Figure 8-11. As ambient temperature increases, the COP of the chiller experiences a large decrement but since the evaporator temperature is set at 15 ^{o}C, its magnitude is still higher than that of the conventional cycle. While the chiller brings the solution temperature to a lower magnitude in the morning compared to midday, its required power for solution cooling is lower thanks to a higher COP in the morning which results from lower ambient temperatures. After midday, the power consumed increases at a higher rate because of a decreasing COP and the need to provide a lower-temperature solution compared to midday. The chiller duty for the cooling supply air is very low compared to that for the solution cooling. Since the air mass flow rate passing over the cooling coil is constant, the temperature difference is the varying parameter in the cooling coil, which is the difference between the required temperature and the dehumidifier outlet air temperature. The chiller duty increases slightly due to decreases in both the COP and required temperature of supply air until hour 18 when it reaches its peak. In addition, the increasing humidity ratio of the surroundings increases the thermal energy required for regeneration due to two facts. Firstly, the inlet solution temperature at the dehumidifier decreases and consequently, that of the outlet solution decreases. Thus more thermal energy is required to bring the solution to the temperature required for regeneration. Secondly, as per Eq. (6-15), the ratio of the solution mass flow rate which is going to be regenerated increases as the outlet concentration decreases. These two facts result in a significant increase in the thermal energy required for regeneration. The required thermal energy is minimum at noon and is maximum in the morning and late evening, which means that if solar-thermal is adopted for solution heating purposes, thermal storage is required. According to Figure 8-11, most of the required energy is consumed for regeneration of the diluted desiccant and it is sensitive to the ambient humidity ratio when compared to Figure 8-3. The sensitivity of performance to ambient humidity ratio indicates that DCS characteristics have to be investigated in transient conditions rather than at specific outside conditions. In this regard, it is important to note that in nearly all the existing literature a constant ambient humidity ratio is investigated while in fact the absolute humidity ratio of the ambient air varies hourly and even a small change in the humidity ratio results in significant variations in the system behavior. Similar to conventional cooling cycle, ventilation air flow rate is constant, thus fan power is also constant but at a higher magnitude due to two supply fans. For pumps, required power is approximately constant because of two facts:

- Because of small variations in dehumidifier inlet solution temperature, chilled water flow rate alters vary slightly.
- While amount of condensation at dehumidifier is higher in the morning and evening than at midday, due to the high flow rate of solution, the variation in outlet concentration is small which is followed by negligible change in diluted solution flow rate according to Eq. (6-15).

Also, it is possible to provide the thermal energy of the DCS from a hot water stream which can be produced by any thermal resource such as solar, geothermal and etc. because the regeneration temperature of the solution is only 65 ^{o}C. Considering hot water temperature of 85 ^{o}C and heat exchanger effectiveness of 0.7, the required hot water flow rate on July 21 from is shown in Figure 8-12. According to the Figure, hot water mass flow rate varies between 1800 to 2500 kg h^{-1} but according to the metrological data set this day is one of the most humid days during the year and for an ordinary summer day, humidity ratio values would be lower. Starting from about 2000 kg h^{-1} in the morning, the required hot water follows a gradual decrease up to hour 13 where a minimum of about 1800 kg h^{-1} occurs and then it undergoes a significant increment to hour 20 when it reaches a maximum of 2500 kg h^{-1}.

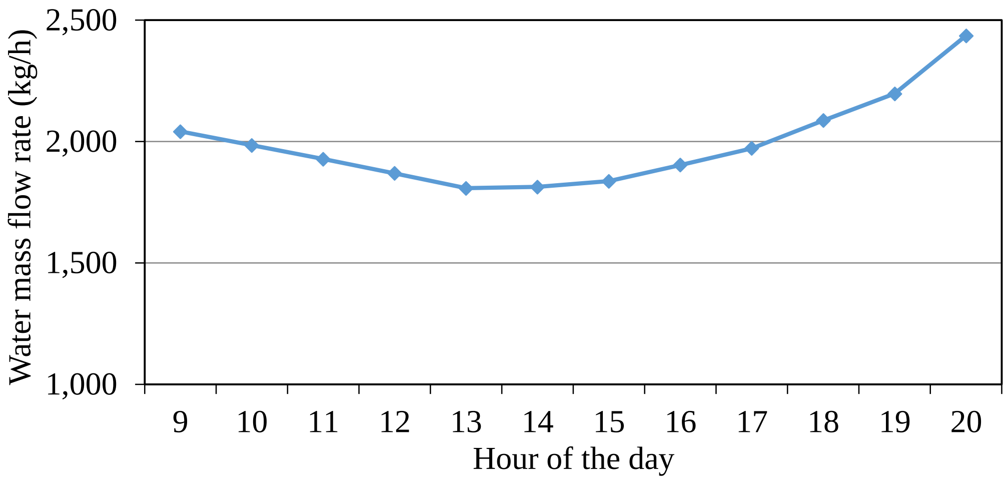


Figure 8-12 Hot water mass flow rate

8.1.4. Comparison of VCCS and DCS

A comparison of VCCS and DCS is done assuming that the reheater of VCCS and regenerator of DCS are non-electrical units. Thus, required power of both cycles are divided into two energy types, electrical and thermal energies, and is demonstrated in Figure 8-13.

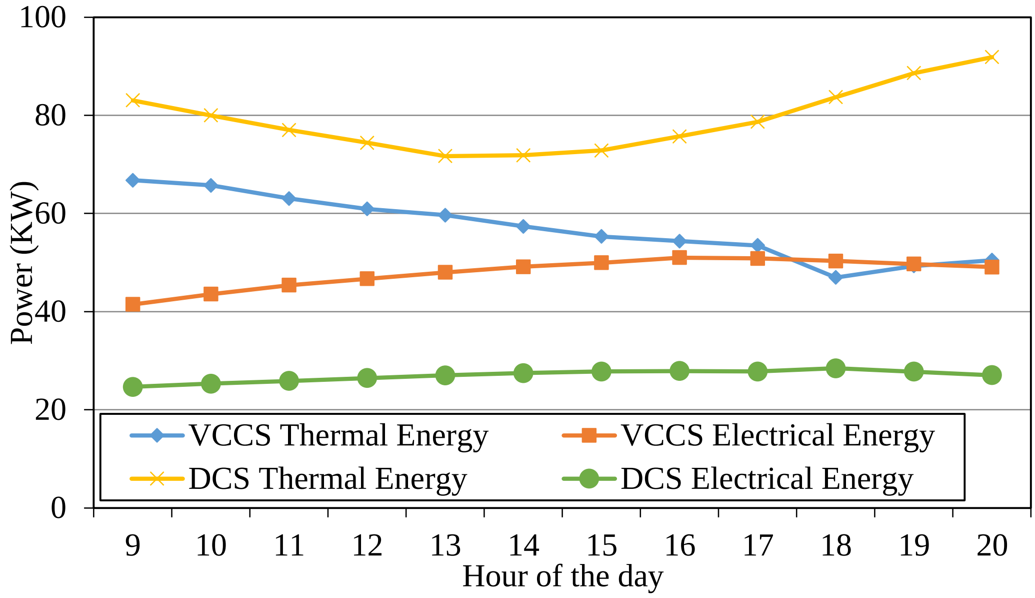


Figure 8-13 Energy demand of both cycles in 21st July

As mentioned previously, the thermal energy demand of VCCS's reheater and DCS's regenerator depend on sensible load ratio and ambient humidity ratio respectively. According to the graph, thermal energy consumption of desiccant cycle is higher for the whole the day and the difference reaches its peak in the evening where sensible load ratio falls and ambient humidity ratio soars. From an electrical energy point of view, VCCS approximately consumes twice as much electricity as DCS which is due to higher chiller COP of the latter and also higher heat transfer effectiveness of liquid-liquid heat exchanger of DCS than gas-liquid heat exchanger (cooling coil) of VCCS.

8.2. Whole summer analysis

8.2.1. Energy Analysis

As stated before, DCS output strongly relies on ambient conditions and needs to be investigated in whole operation time. Thus, both the VCCS and DCS systems are analyzed from an energy point of view for three months, namely June, July and August, and the results are shown in Figure 8-14. Energy demand is divided into two major groups, thermal and electrical energy. Thermal energy is used for heating in DCS and re-heating in VCCS while electrical energy is consumed by all other components in both systems, including fans, chillers and pumps. According to Figure 8-14, the total energy consumption of the two systems are approximately equal in magnitude but different in type and it should be considered that electricity is of a higher quality.

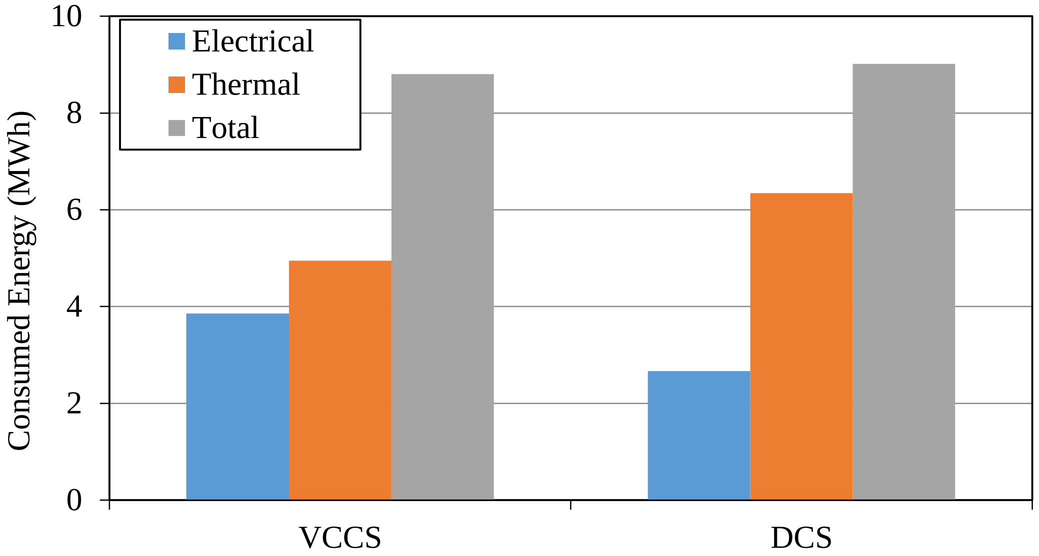


Figure 8-14 Energy demand of both cycles at summer

As a first approximation, the electricity consumption can be multiplied by three when comparing to thermal energy because electricity is usually generated using fossil fuels and the efficiency of the process is about 33% due to plant inefficiencies and grid losses. Therefore for equal total energy consumption, electricity requires 3 times more fossil fuel consumption than thermal energy, as presented in Figure 8-15, and consequently has 3 times higher CO_2 emissions. Thus the DCS is preferable from a pollution point of view and according to the current model, VCCS results in 16.8% more CO_2 emissions if the thermal energy for both systems are provided by the same type of fossil fuel.

Also it is possible to produce thermal energy demand of both cycles from clean, free renewable resource such as solar thermal collectors thanks to low temperature requirement of both cycles. Figure 8-16 demonstrates the results for the circumstance if all the thermal energy is supplied from a clean source by both considering and ignoring electricity generation efficiency. According to the graph, more energy saving occurs at DCS compared to VCCS, at about 30.78% when thermal demand is satisfied by solar thermal collectors and that is because of the fact that more energy in thermal mode is needed. It proves that solar thermal application is a perfect supplement for DCS.

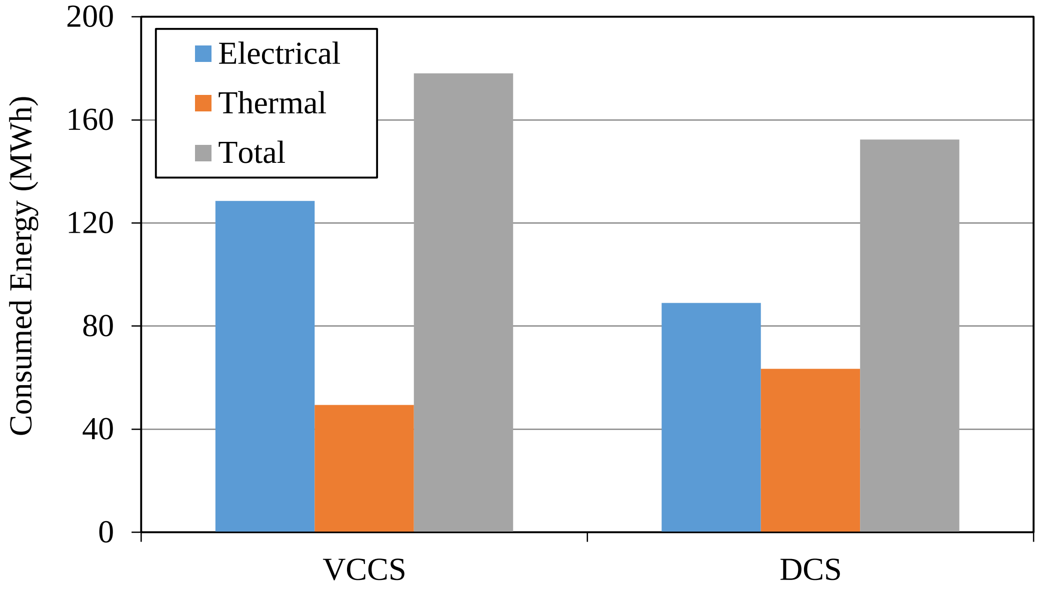


Figure 8-15 Primary energy demand of both cycles at summer

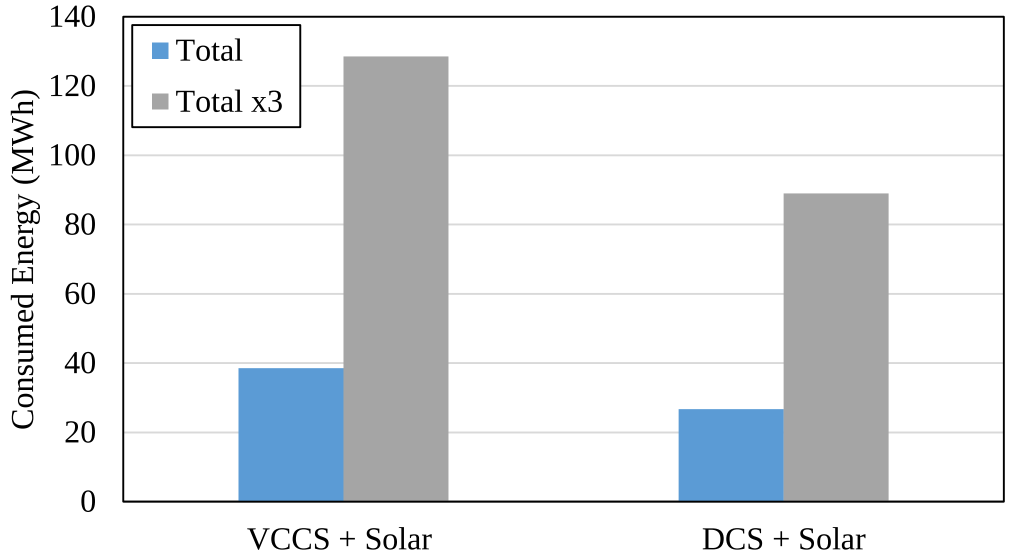


Figure 8-16 Energy demand of both cycles at summer with renewable heating source

8.2.2. Parametric study

Finally, the sum of the consumed energy by the two systems in June, July and August are compared at different supply air flow rates in Figure 8-16. Minimum supply air flow rate is about 4 m^3 s^{-1} because a lower magnitude requires chilled water temperature below 5 ^oC in the VCCS. In DCS, as air flow rate increases, the required temperature of the dehumidifier inlet solution rises and hence the chiller can work at higher temperatures which results in higher C.O.P and vice versa. Different air flow rates with their chilled water temperatures at both cycles are listed in Table 8-1. Also higher and lower supply air flow rates mean increasing and decreasing work of hydronic units respectively, particularly fans. In an opposite manner, air flow increment results in higher temperature of the dehumidifier outlet solution and consequently a lower temperature increment for regeneration because the regeneration temperature is constant, and the reason for this is the constant outlet regenerator concentration. Similarly, a decrement in supply air flow rate is followed by lower dehumidifier solution inlet and outlet temperatures and consequently more thermal energy demand for rising solution temperature up to the regeneration temperature.

However, the energy consumption trend for VCCS is entirely different from the DCS because higher air flow rate results in both higher thermal and electrical energy requirements due to higher fan loads and higher mass of air to be reheated.

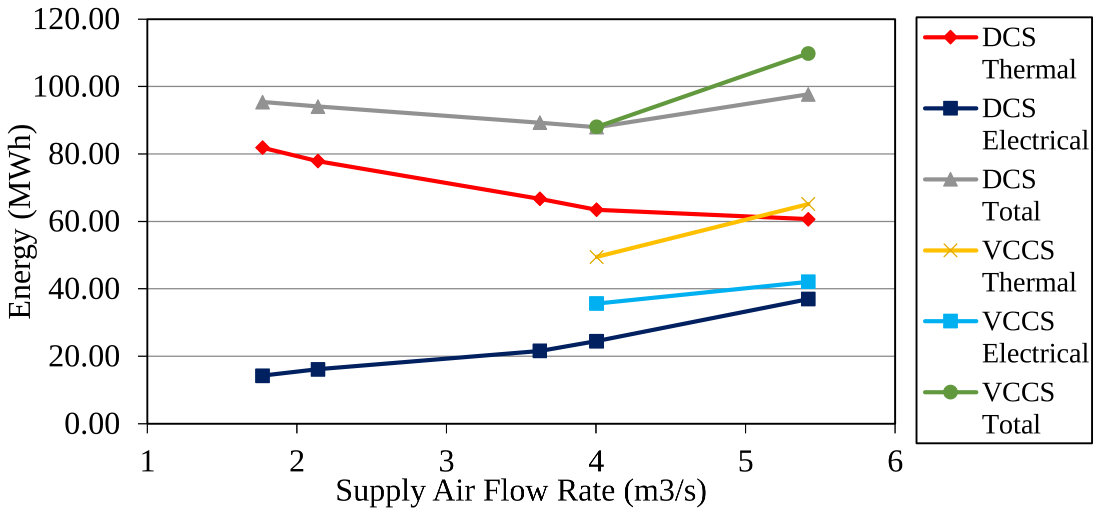


Figure 8-17 Air flow parametric study

Table 8-1 Parametric study's chilled water temperatures

	Supply air flow rate ($m^3\ s^{-1}$)	Chilled water temperature (°C)
DCS	5.42	17
	4.004	15
	3.625	12
	2.14	8
	1.77	5
VCCS	5.42	6.5
	4.004	5

CHAPTER 9

CONCLUSIONS AND FUTURE WORK

Mathematical models for the energetic performance of both hybrid desiccant cooling system (DCS) and conventional all-air vapor compression cooling system (VCCS) are developed. These models are programmed into TRNSYS and simulations are run for a sports center at the METU Northern Cyprus Campus (METU NCC). These two cooling cycles are compared from both operating characteristics and energy consumption point of views using meteorological data for METU NCC. METU NCC is characterized by hot and humid summers. Due to a positive pressure inside the sports center, infiltration is assumed zero and all latent loads are due to occupants.. According to the results, the ambient temperature effect on DCS is very small compared to ambient humidity ratio. The former only affects the COP of the chiller while the latter results in significant changes in both thermal and electrical energy consumption. Thus, to understand the system's performance for a specific location the system should be investigated over a range of operating conditions rather than for specific conditions because the outside humidity ratio, the main parameter, varies hourly during a day and the impact on system characteristic is significant. Herein both systems are investigated over three summer months assuming similar air flow rates. The results show that the total energy demand of VCCS and DCS are approximately the same but differences exist in the type of energy demand. While VCCS requires more electrical energy, the DCS requires more thermal energy which could prove beneficial from pollution and economic points of view. Specifically, for the same required energy, electricity is more expensive than heat and can produce approximately three times more greenhouse gases emission due to inefficiencies in electricity production and losses in the grid. Also it is possible to provide some or all of the required heating energy for DCS using solar energy because the required temperature from the heating source can be as low as 75^{o}C. Hence glazed flat plate collectors could be a suitable choice, since they have a lower initial cost than higher temperature solar thermal collectors. If a solar thermal system is adopted, the best results would be obtained when ambient humidity and solar resources are in the same phase temporally. However in the studied case, they are out of phase diurnally, meaning that when irradiation peaks at midday, the outside humidity ratio has its minimum magnitude and conversely in the morning and evening when solar resources are low, humidity reaches its peak, and thus thermal storage may be required. Finally, the operating characteristics of both systems are investigated for different air flow rates. Results show that as air flow rate increases, the energy demand of VCCS increases significantly and more than that for the DCS. Thus, DCS would save more energy when it is necessary to have an air-conditioning system with high flow rates such as for hospitals or industrial applications. Also, due to limitations in the VCCS's chiller temperature not present for the DCS, running the DCS at lower flow rates than for the VCCS may be possible, and may be appropriate for some applications such as residential applications. Lower air flow rates mean lower electricity demand and higher heat demand. Thus it is better to run the system at as low an air flow as possible to shift more electricity to heat, resulting in lower operating costs and CO_2 emissions. Finally, it is observed that it is possible to remove the cooling coil from DCS cycle because supply air experiences a significant decrease in temperature at the dehumidifier and outlet condition is very close to the supply air state.

While the modeled dehumidifier in the current work was validated for a sufficient range of operating conditions, the dehumidifier model has to be used in accordance with experimental data which are suitable. Hence it is not possible to investigate the DCS system by varying solution regeneration temperature and inlet and outlet concentration. As a result, the dehumidifier and regenerator have to be operated with fixed inlet concentrations which is a barrier in analyzing the cycle. Thus, it is important to provide suitable algebraic model for regeneration of LiCl to overcome this problem. Also, developing the cycle by designing and adding solar collectors and storage system would be beneficial to check the possibility of solar DCS system. As the third improvement, investigating the cycle within a multi-zone building and real control systems would be more close to real applications.

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APPENDIX A

TYPE OF DESICCANT COOLING SYSTEMS

DCS could be classified from two points of view. The first classification is related with whole cooling cycle, standalone, hybrid and desiccant-assisted hydronic cooling which two formers are considered as all-air systems and the latter as water-and-air system. Standalone type benefits the evaporative cooling phenomena at cooling unit, either direct or indirect evaporator, shown in Fig.5, but this application is so limited at humid climate due to over-drying requirement of inlet air resulting into very high thermal energy demand. Thus lying on the same phenomena, the hybrid desiccant system was innovated to be a solution to problem like that. In hybrid design, evaporative cooler is removed and a cooling coil, running by VCCS or VSS is replaced but some other components such as heat exchangers might be added to increase system efficiency beside of utilization of waste heat from cooling unit for regeneration purposes. This is mostly designed for applications in hot and humid climates, in order to avoid deep dehumidification of the process air which is required in standalone design due to the application evaporation for cooling. Desiccant-assisted hydronic cooling is a combination of hybrid DCS and radiant cooling in which they are responsible for removing latent and sensible load of space respectively.

The second classification is related with the state of desiccant material that could be solid or liquid and following that specific types of contactor. The term “contactor” is usually used to mention dehumidifier or regenerator because of contacting of air and desiccant in them. In the case where the desiccant is employed in its solid state, the contactor is usually a rotary wheel with desiccant-coated honeycomb bores and rotates at very slow angular speeds regularly 8~10 revolution per hour [74] as shown in Figure 0A-1.

As the wheel turns, the desiccant passes alternately through the incoming process air where the moisture is adsorbed and through a regenerating zone where the desiccant is dried by return air and the wheel continues to rotate and the process is repeated. Another type of solid desiccant contactor is fixed-bed design uses the packing of solid desiccant to form a sort of adsorbent beds exposed into incoming air stream but the bed cannot dehumidify the process air continuously and needs to be regenerated periodically by regeneration air to be able again attract moisture. The most common solid-state desiccants are silica gel, zeolites, aluminum oxides and LiCl.

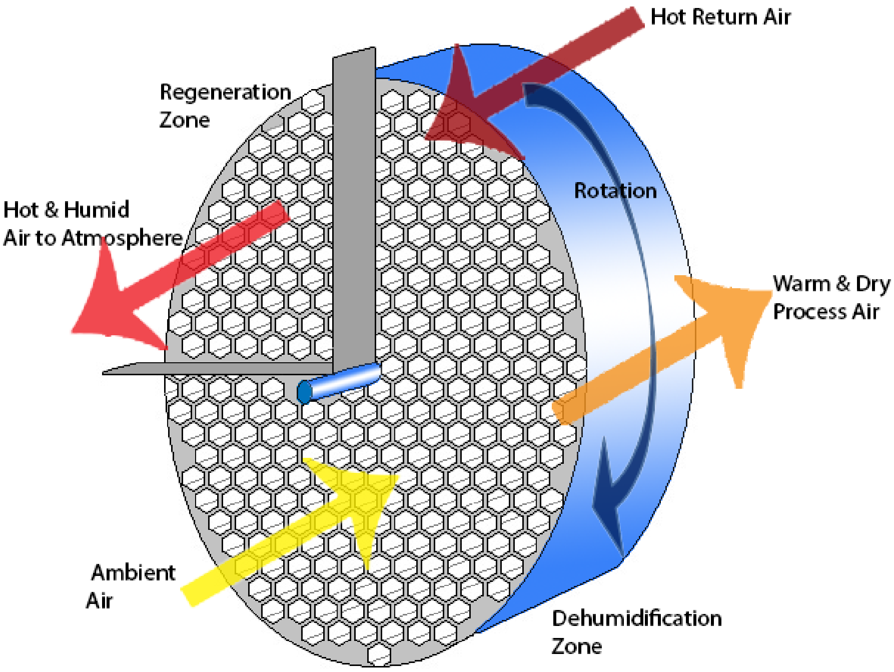

Figure 0A-1 Wheel Type Contactor

In addition to solid contactors, liquid contactors are available with some merits. Lof [75] proposed the first liquid desiccant system by triethylene glycol. Not like to the wheel, using liquid desiccant requires two contactors for separate dehumidifier (absorber) and regenerator (disrober) units in which liquid desiccant is brought into contact with process air and return air respectively but the velocity of air is usually low to unlimber a laminar flow in order to provide better heat and mass transfer [59]. Absorber is a cubic or cylindrical enclosure in which strong solution of desiccant is sprayed at top and moves down by the force of gravity. On the other hand, humid inlet air is brings into contacts by sprinkled desiccant solution. Since the vapor pressure of strong desiccant is lower than surrounding, sorption process occurs and finally temperate diluted desiccant exits at bottom. The configuration of regenerator is just like the dehumidifier and the only difference is the nonexistence of insulating layer and filter layer in the former, provided in latter. Insulating layer to prevent heat transfer from environment and the filter layer for avoiding the carryover of desiccant droplets by process air to conditioned space during dehumidification. Despite of the same configuration, process is completely opposite at regenerator, meaning diluted solution of desiccant is sprayed at top and brought into contact with hot air, higher vapor pressure of desiccant force water vapor to leave desiccant toward hot air and strong desiccant leaves the unit in high temperature. Liquid desiccant cycles through these units for dehumidification and regeneration using a pump but note that using a liquid-liquid heat exchanger between strong and diluted solution is strongly recommended because of the fact that sorption and desorption effectiveness at desiccant material improves by decreasing and increasing the temperature respectively, thus hot strong exiting desiccant from regenerator which is going to go toward dehumidifier needs to be cooled for dehumidification and on the other hand temperate diluted existing desiccant from dehumidifier heading for regeneration must be heated to regeneration temperature, so this regenerator plays a role as pre-heater and pre-cooler for diluted and strong solutions respectively. Beside these units, utilizing a storage tank for energy storage as chemical in strong solution would enhance the potential of solar or waste heat application as thermal sources.

There are several types of liquid contactors but the most regular designs are spray tower, wet wall (falling film) and packed tower (Packed bed), which is most common type. Also a new design of liquid contactor is available which is like the fixed-bed type of solid contactors but utilizing a liquid sorbent and is proposed by Saman and Alizadeh. [76]. Note that both contactors in a system, dehumidifier and regenerator, could adopt different types or the same. Falling film is the simplest design and act like spray chamber of evaporative cooling, instead droplets of desiccant brought into contact with flowing air by utilizing a nozzle but the sorption effectiveness is not high. Wet wall is a vertical cylinder in which desiccant flows over walls. While the effectiveness of this simple system is high with low pressure drop, achieving a thin film of desiccant over walls is the main problem of system [6]. The most common liquid desiccant contactor is the packed bed tower type, in which packing material are installed in a compact column. Packing material is the place where heat and mass transfer occurs between desiccant and air in dehumidifier and regenerator and the aim of that is to increase contacting are between liquid desiccant and air stream but also it results in higher pressure drop [77]. The most important parameters of a packed material are the contacting area per unit volume of packing material (volumetric area), void volume per unit volume of packing materials to measure air flow resistance (void ratio), equivalent diameter, wetting ratio and the intervals between packing material sheets considered 6-8mm usually [6]. Packed tower could be designed in structured or random type as shown in Figure 0A-2 and Figure 0A-3 respectively. Structured packing material consists of several fixed geometry sheets installed parallel to each other while random type where non-regular forms of packing materials are installed randomly but both types are made out of plastic due to corrosiveness of liquid desiccants . In addition to high efficiency because of more provided contacting area and better distribution of descant solution in structured type, pressure drop is also lower compared to random packing type [76] and some researches were conducted on comparison of them.

Figure 0A-2 Structured Packing Material [78]

On the other hand, Cross flow, counter and parallel flow design is possible configuration for both random and structured packed bed liquid contactors and it is demonstrated in Figure A-4 for structured packed bed. In cross flow design liquid desiccant and air flow is from top and side respectively but in counter flow air blows from bottom while in parallel design air and desiccant both come from top.

Figure 0A-3 Random Packing Material [79]

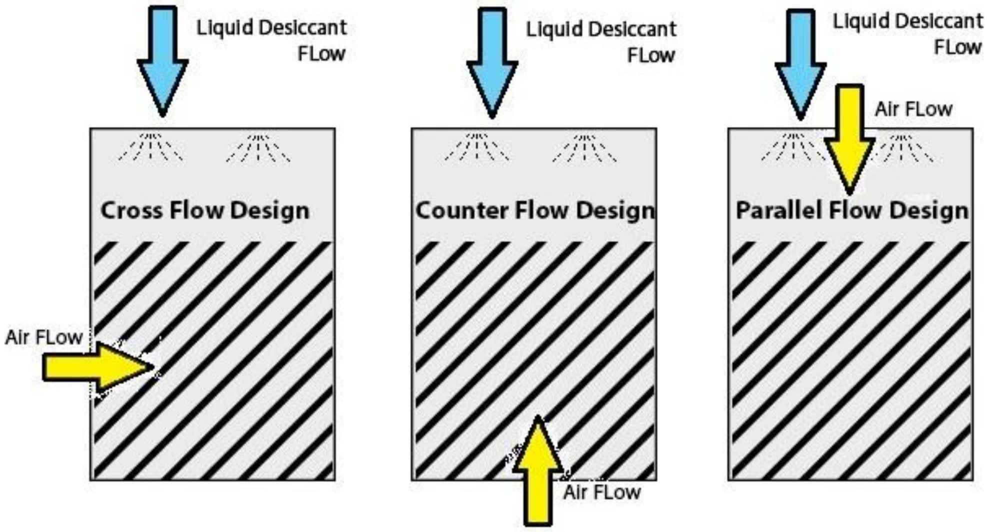

Figure A-4 Patterns of Air Flow in Contactors

There exists other standardization for liquid contactors but only in the application of dehumidifier. Dehumidification in all of liquid-type dehumidifiers could be either adiabatic or isothermal while in solid designs isotherm process is not applicable except fixed-bed design as was mentioned before. Dehumidification process results in releasing heat in dehumidifier, due to water condensation, which is 5% to 25% greater than enthalpy of evaporation [58] and sometimes up to 50% [80]. This temperature increment results in reduction of dehumidification effectiveness and cooling capacity and also less accurate control over temperature and humidity thus, Isotherm feature, as an alternative method, could overcome this problem by avoiding temperature increment and keep it approximately constant or at least lower increment of with the help of an insolation layer to prevent heat. This feature not only increases dehumidification effectiveness but also increases cooling capacity by cooling desiccant and air during dehumidification respectively. It could be done either by utilizing cold coils in layers of packing materials, Figure 0A-5, or replacing packing materials by finned-tube cold coils as contacting area of dehumidifier.

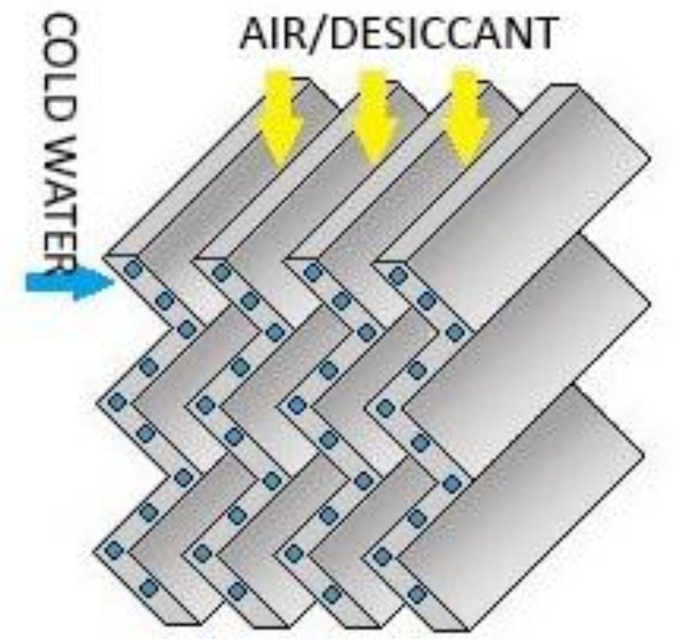

Figure 0A-5 non-adiabatic packing material

Multi-stage dehumidifier is another alternative method of conventional dehumidifiers with prominent advantages. Desiccant temperature increment results in enlarging irreversibility in dehumidifier meaning dropping of sorption effectiveness done by increasing vapor pressure of desiccant. Thus By installing several single-stage dehumidifier in series and pre-cooling inlet desiccant at each stage separately, irreversibility goes down and better sorption will be obtained, Figure 0A-6, because in conventional dehumidifier the mass flow rate of desiccant should be relatively high to fulfill dehumidification demand, thus low difference between inlet and out let concentration of desiccant and high difference between vapor pressure of exiting air and desiccant exist, meaning irreversibility but in a multi-stage design mass flow rate of desiccant solution at each stage is smaller, thus change in concentration is higher and the process air leaves the dehumidifier in an state much closer to equilibrium point with vapor pressure of exiting desiccant solution, consequently irreversibility drop enhancement.

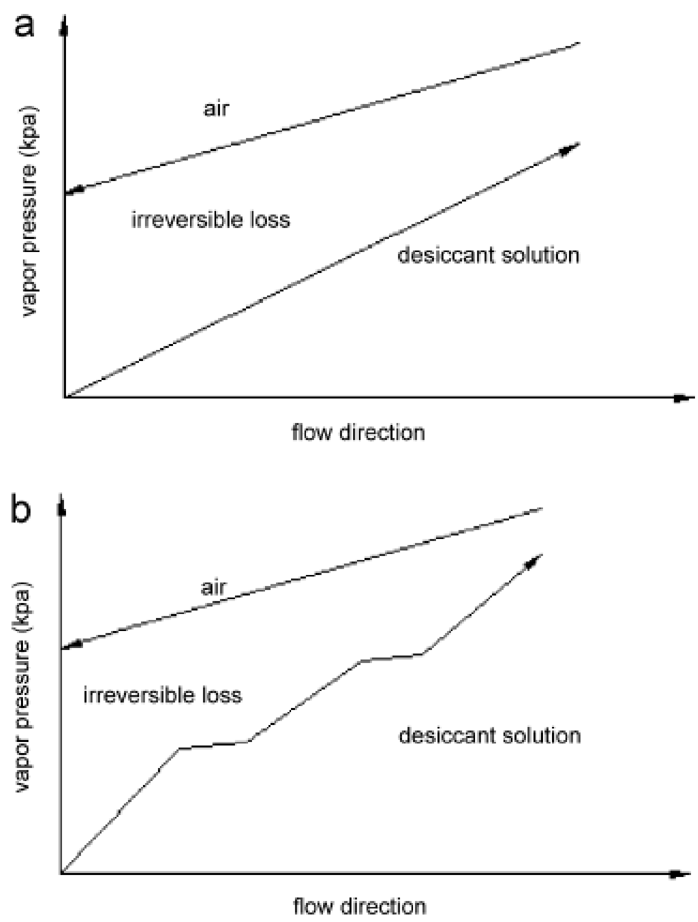

Figure 0A-6 Irreversibility in Single and Multi-stage Dehumidifier [61] a) Single Stage b) Two Stage

As well as dehumidifier specific standardization, there also exist some specific ones for regenerators according to type of regeneration of diluted desiccant by solar collector utilization, since solar energy play an inseparable role in DCS. Direct and indirect types are two general configurations for solar-assisted regenerators but it should be mentioned there exist always a significant need to a backup heater operated by electricity of fossil fuels in case of emergency. In direct configuration, thermal energy from radiation of sun is directed into diluted desiccant directly by passing it through solar collector while in indirect one a secondary fluid, usually water, is heated in collector and by using a fluid-fluid heat exchanger heats the liquid desiccant. Obviously the former one has higher thermal efficiency in addition to remove of contacting chamber but with some disadvantageous. The open configuration is classified related with solar collector consisting of open and close design. Open design is a slopped-flat dark plate exposed to sun irradiation over which weak desiccant flows and is heated and by using ambient air, regeneration process accurse and several theoretical and experimental investigations are done specially about the rate of regeneration. In spite of simplicity and high efficiency, serious disadvantages also exist such as subsiding of hovering dots and harmful particles in the air on the surface of solution or a rain could stop the system completely. On the other hand, close type operation is similar to open design but the top of collector is covered by glass to eliminate the problems of open design and providing better thermal efficiency. The side walls of collector remain open for circulation of air which could be natural or forced but it is difficult to control due to random wind flow affecting evaporation and thermal losses while there exist an optimum air flow rate. In close configuration, water is solar-collecting fluid aiming to heating inlet desiccant of regenerator. While thermal efficiency is lower at this type but overall effectiveness is higher thanks to available large contacting area in chamber and benefiting multi-stage phenomena application described in dehumidifiers but in opposite direction.

The most common liquid desiccants used are Lithium Bromide solution (LiBr), Calcium Chloride solution (CaCl), Lithium Chloride solution (LiCl) and triethylene glycol. Triethylene glycol's too low vapor pressure results in excellent sorption but also evaporation some into process air which is undesired. In addition high viscosity of that, make I difficult to circulate through pipes by the power of pump. Vapor pressure of calcium chloride is higher compared to those others with low stability but the cost is low. Vapor pressure of Lithium Chloride is low enough with high stability but its cost is slightly higher than others. Lithium bromide thermo dynamical features and cost is both intermediate. Pros and cons of liquid dehumidifiers are mentioned below:

Advantages:

Lower regeneration temperature even as 40^{o} C

Higher Sorption Capacity

Lower air temperature increment during dehumidification (Higher cooling capacity)

Capability of storage by chemical energy

Absorbing air contaminants, both organic and inorganic

Bacteria and viruses sterilization by some absorbent such as TiO_2 [17]

Lower pressure drop

Ability for inner cooling of desiccant during dehumidification

Flexibility due to modular structure

Disadvantages:

- More expensive configuration
- Corrosive
- Probability of carryover of desiccant by process air
- Crystallization at high concentration and low temperature

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YAZARIN

Soyadı : KARSHENASS
Adı : Arash
Bölümü : Makina Mühendisliği

TEZİN ADI (İngilizce) :

MODELING AND TRANSIENT ANALYSIS OF A HYBRID LIQUID DESICCANT COOLING SYSTEM

TEZİN TÜRÜ : Yüksek Lisans ☑ Doktora ☐

1. Tezimin tamamı dünya çapında erişime açılsın ve kaynak gösterilmek şartıyla tezimin bir kısmı veya tamamının fotokopisi alınsın. ☐

2. Tezimin tamamı yalnızca Orta Doğu Teknik Üniversitesi kullancılarının erişimine açılsın. (Bu seçenekle tezinizin fotokopisi ya da elektronik kopyası Kütüphane aracılığı ile ODTÜ dışına dağıtılmayacaktır.) ☐

3. Tezim bir (1) yıl süreyle erişime kapalı olsun. (Bu seçenekle tezinizin fotokopisi ya da elektronik kopyası Kütüphane aracılığı ile ODTÜ dışına dağıtılmayacaktır.) ☑

Yazarın imzası .... Tarih 10/09/2014