Mugla Journal of Science and Technology

THERMODYNAMICS AND PSYCHROMETRIC ANALYSIS OF INDOOR SWIMMINGPOOL DEHUMIDIFICATION UNITS

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Received:29.11.2015, Accepted: 16.02.2016
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Abstract

Indoor swimming pool dehumidification units are central air conditioning units that designed for control the relative humidity, prevent buildings, mechanical systems and human healthy from negative impacts of exceed relative humidity.

In this work, dehumidification unit design and working conditions are worked for an indoor swimming pool. Detailed analysis of thermodynamics and psychrometric are represented. Evaporation of pool surface is calculated and air flow rate of a dehumidifier is determined with respect to design conditions. Analysis of thermodynamics and psychrometric are done with respect to application of heat pump systems and desiccant systems. Thermodynamics and psychrometric analysis with respect to working conditions due to seasonal variations of the designed heat pump dehumidifier are done and working conditions and areas are determined for obtain energy efficient.

Keywords: Indoor swimming pool, Air Conditioning, Thermodynamics, Psychrometric, Dehumidification.

KAPALI YÜZME HAVUZU NEM ALMA SANTRALLERİNİN TERMODİNAMİK VE PSİKROMETRİK İNCELENMESİ

Özet

Havuz nem alma santralleri, kapalı yüzme havuzlarında bağıl nemi kontrol etmek, yüksek bağıl nemin binaya, mekanik sisteme ve insan sağlığına olumsuz etkilerinden korumak için tasarlanan merkezi iklimlendirme üniteleridir.

Bu çalışmada kapalı yüzme havuzları için nem alma santrali tasarımı ve çalışma koşulları ele alınmıştır. Psikrometrik ve termodinamik analizleri ayrıntılı biçimde sunulmuştur. Belirlenen havuz mahali tasarım şartlarına göre, havuz yüzeyindeki buharlaşma oranı hesaplanmış ve santral hava debisi bu orana göre belirlenmiştir. Isı geri kazanımlı soğutma çevrimli ve kurutmalı (desiccant) nem alma santrali uygulamalarının termodinamik analizleri ve psikrometrik incelemeleri yapılmıştır. Tasarımı yapılan soğutma çevrimli nem alma santralinin mevsimsel değişimlere göre çalışma koşulları üzerine termodinamik ve psikrometrik incelemeler gerçekleştirilmiş ve enerji verimliliği sağlayan çalışma koşulları ve bölgeleri belirlenmiştir.

Anahtar Kelimeler: Kapalı yüzme havuzu, İklimlendirme, Termodinamik, Psikrometri, Nem alma.

1 Introduction

Moisture control in indoor swimming pools is an important issue in terms of human healthy, construction and mechanical system.

The most important problem in indoor swimming pools is exceed relative humidity. Exceed relative humidity (>65%) cause, condensation on the walls and windows, constructive damage in buildings, rusting and defect in mechanical and electrical systems.

2 Calculation Of Evaporation On Pool Surface And Air Flow Rate

First of all, latent heat gain of pool ambient must be calculated to control relative humidity of swimming pool area. Latent heat gain of a pool space is occurred two main sections. First one is evaporation on pool surface second one is human resourced latent heat. In this study, effect of human into pool to evaporation is including to calculation however, latent heat from spectators is not consider. therefore, this analysis is for evaporation at indoor swimming pool without tribune.

Evaporation occurred difference between partial vapor pressure of ambient and saturated vapor pressure on water surface. This difference is a main reason of evaporation. The other effects of evaporation are humans into the pool and air velocity on pool surface. Humans into pool can cause waving pool water, so it increase pool surface area and evaporation effect. If air velocity on pool surface is above 0,25 m/s, it effects evaporation positively . In this study, velocity on pool surface is assumed below 0,25 m/s.

There are several formulas to calculation of evaporation. The earliest formula is determined by Willis H. Carrier in 1918. Nowadays, most commonly used formula is determined by M. Mohammed Shah and published by Heating- Piping and Air-Conditioning journal in 1990[1].This formula is as following:

$$\dot{m}_b = A \times [35 \cdot F_a \cdot \rho_a \cdot (\rho_a - \rho_w)^{1/3} \times (\omega_w - \omega_a)] \quad (1)$$

In this Formula:

A Pool surface area [m^2]
$\dot{m}_b$ Evaporation at Pool Surface [kg/h]
ρ_a Average Density of Ambient Air [kg/m^3]
ρ_w Density of Saturated air at Pool Surface [kg/m^3]
ω_a Average Humidity ratio of Ambient Air [kg/kg$_a$]

ω_w Humidity ratio of Saturated Air at Pool Surface [kg/kg$_a$]
F_a A Value, depends on human inside pool.
$F_a=1+1{,}4N/A$ (2)

N Human quantity at the same time in the pool
If $F_a < 1{,}25$ so, $F_a=1{,}25$
Design conditions of pool ambient are below:
Pool Dimensions
height: 25m width: 12,5 m depth: 2m

Air Design Conditions	30 °C KT, %50 - 60 RH
Pool Water Design Conditions	28°C
Air velocity on pool surface	0,2 m/s
Human quantity in the pool	10

2.1 Calculation of Evaporation

A= 312,5 m^2
$\rho_a = {}^1/_{0,8813} = 1{,}1347$ kg/m^3
$\rho_w = {}^1/_{0,8861} = 1{,}1285$ kg/m^3
$\omega_a = 0{,}0163$ kg/kg$_a$
$\omega_w = 0{,}0240$ kg/kg$_a$
$F_a=1+1{,}4$ N/A
N= Human quantity in the pool
N =10,
$F_a = 1 + (1{,}4 \cdot 10)/312{,}5 = 1{,}0448$
If $F_a<1.25 \rightarrow F_a=1{,}25$

$\dot{m}_b=312{,}5[35\times1.25\times1{,}1347(1{,}1347-1{,}1285)^{1/3}(0{,}0240-0{,}0163)]$
$\dot{m}_b = 21{,}94$ kg/h Evaporation on pool surface. This calculation will be used for design conditions at next chapters.

2.2 Calculation of Air Flow Rate

$\Delta\omega=(0.016\text{-}0{,}0105) = 0{,}0055$ kg/kg$_a$
$\frac{\sum \dot{m}b}{\Delta\omega} = \frac{21.94}{0{,}0055} = 3989$ kg/h
$\Delta V=\frac{3989\ kg/h}{(1/0.88)} = 3510$ m^3/h
Air flow rate of dehumidification unit is 3510 m^3/h.

3 Design Of Dehumidification Unit With Refrigeration Cycle

Dehumidification by refrigeration is a prevalent method for dehumidification. This involves dehumidifying by cooling the air sucked from the spot further than the saturation point and condensation of the moisture within.
In terms of providing energy efficiency, heat recovery exchangers are used in these units. Most prevalent of these are the heat-piped and plated heat recovery heat exchangers.
In this study, examination of both without heat recovery and heat recovery (plated and heat-piped heat exchangers) done separately.
Preconditions of designed refrigeration cycled dehumidification unit are as following;

7 °C evaporation,
50 °C condensation,
5 °C super heating,
3 °C subcooling.

3.1 Pool Dehumidification Unit without Heat Recovery

First, Dehumidification device without heat recovery is examined (Figure 1). In psychrometric chart, point 1 symbolizes air at pool ambient and point 2 symbolizes the outlet of evaporator. Point 3 is defined as supply air. The moisture receiving capacity calculation depends on the rate of moisture between points 1 and 3. Cooling capacity depends on the difference of enthalpy between 1 and 2. As in Figure 2, humidity ratio on point 1 is 16g/kg, and on point 3 is 10g/kg. Enthalpy value of point 1 is 71,8 kJ/kg, and of point 2 is 40,2 kJ/kg. Examination dehumidification Unit without heat recovery is below [2,4]. Compressor capacity is calculated by taking the average scroll compressor COP value as 3,8.

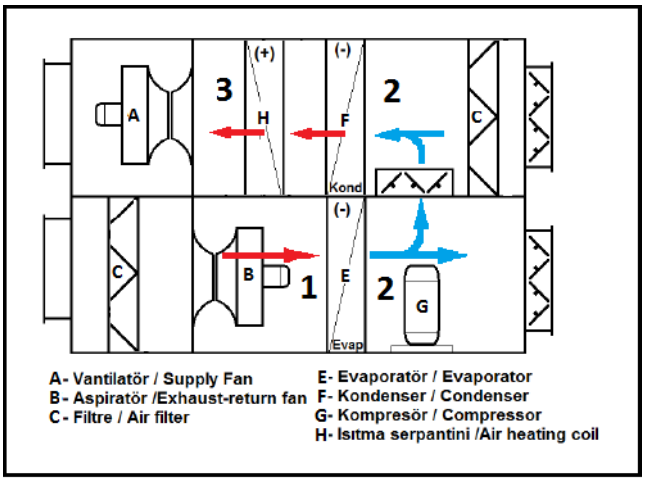

Figure 1. Schema of dehumidification unit without heat recovery

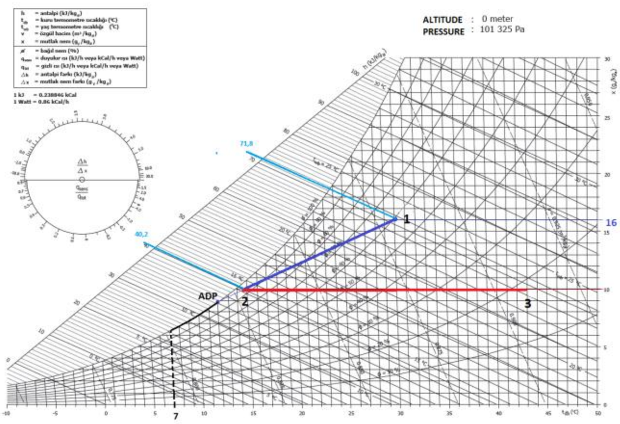

Figure 2. Examination of dehumidification unit without heat recovery

$Q_{EVAP} = \dot{m} \times \Delta h$
$\dot{m}_w = 3989 \times (0{,}016-0{,}010) = 23{,}9$ kg/h dehumidification capacity[6].
$Q_{EVAP} = \dot{m}_a \times \Delta h - \dot{m}_w \times h_w$ [6]
$Q_{EVAP} = 3989 \times (71{,}8-40{,}2) - 23{,}9 \times 63{,}0 = 124546{,}7$ kJ/h $=34{,}6$ kW cooling capacity.
$W_{KOMP.} = 34{,}6 / 3{,}8 = 9{,}1$ kW.

3.2 Pool Dehumidification Unit with Heat Pipe Heat Recovery

Second, dehumidification unit with heat pipe heat recovery is examined (Figure 3). In psychrometric chart, point 1 symbolizes the air in pool ambient and point 5 symbolizes the supply air. Point 2 is defined as heat pipe outlet, evaporator inlet, Point 3 as evaporator outlet, heat pipe inlet; Point 4 as plated heat pipe outlet, condenser inlet. As in previous calculations, the dehumidification capacity calculation depends on the rate of moisture between points 1 and 5. Cooling capacity depends on the difference of enthalpy between 2 and 3. As in Figure 4, humidity ratio on point 1 is 16g/kg, and on point 5 is 10g/kg. Enthalpy value of point 2 is 67,3 kJ/kg, and of point 3 is 39,8 kJ/kg. Outlet temperature from heat pipe unit is calculated according to EN 308 standards, and the value of efficiency is taken from producer companies. Examination of plated heat regaining dehumidification unit is below [2,4]. Compressor capacity is calculated by taking the average scroll compressor COP value as 3,8.the average scroll compressor COP value as 3,8.

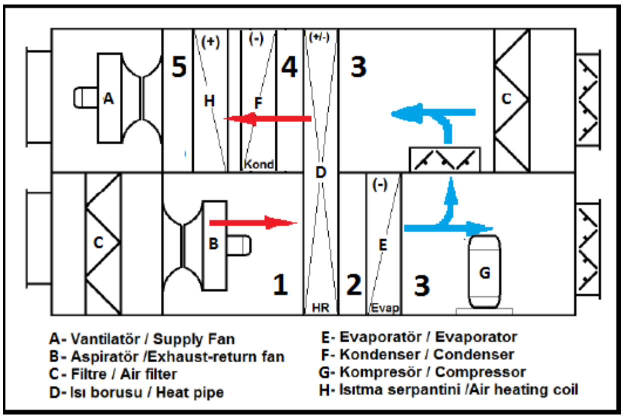

Figure 3. Schema of dehumidification unit with heat pipe heat recovery

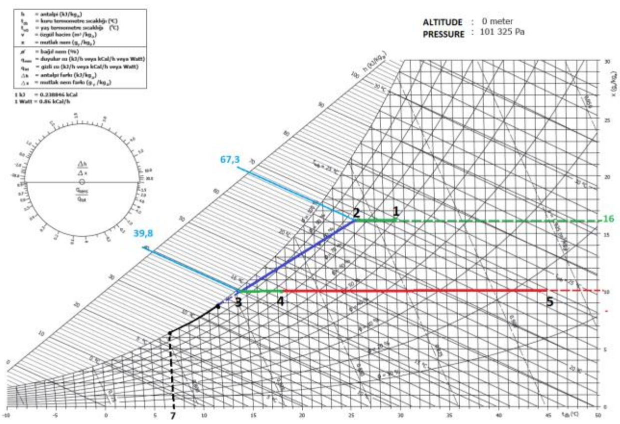
Figure 4. Examination of dehumidification unit with plate type heat recovery

$\eta_{HEAT\ PIPE}=\frac{Tx-Tout}{Tin-Tout}$

$\frac{Tx-14}{30-14}=0{,}266$ so, T_x=18 °C

$Q_{EVAP} = \dot{m}_a \times \Delta h - \dot{m}_w \times h_w$ [6]

$\dot{m}_w$ = 3989 x (0,016-0,010) = 23,9 kg/h dehumidification capacity [6].

$_{EVAP}$ =3989 x (67,3-39,8) – 23,9 x 63,0=108191,8 kJ/h =30 kW evaporator capacity.

$W_{KOMP.}$= 30 / 3,8 = 7,9 kW.

3.3 Pool Dehumidification Unit with Plat Type Heat Recovery

Finally, dehumidification device with plate type heat recovery is examined (Figure 5). In psychrometric chart, point 1symbolizes the air in pool ambient and point 5 symbolizes the supply air. Point 2 is defined as heat recovery outlet, evaporator inlet, Point 3 as evaporator outlet, heat recovery inlet; Point 4 as plated heat recovery outlet, condenser inlet. As in previous calculations, the dehumidification capacity calculation depends on the rate of moisture between points 1 and 5. Cooling capacity depends on the difference of enthalpy between 2 and 3. As in Figure 6, humidity ratio on point 1 is 16g/kg, and on point 5 is 10g/kg. Enthalpy value of point 2 is 62 kJ/kg, and of point 3 is 39,3 kJ/kg. Outlet temperature from plated heat recovery unit is calculated according to EN 308 standards, and the value of efficiency is taken from producer companies. Examination of plated heat regaining dehumidification unit is below [2,4]. Compressor capacity is calculated by taking the average scroll compressor COP value as 3,8.

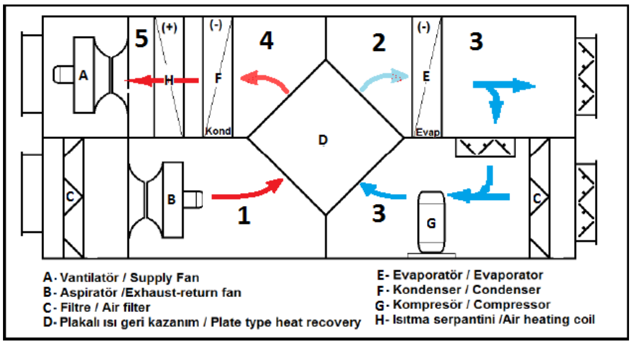

Figure 5. Schema of dehumidification unit with plate type heat recovery

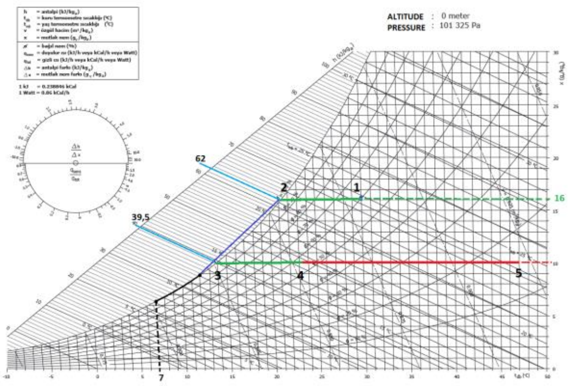
Figure 6. Examination of dehumidification unit with plate type heat recovery

$\eta_{HEAT\ PIPE}=\frac{Tx-Tout}{Tin-Tout}$

$\frac{Tx-14}{30-14}=0.60$ so, T_x=23,6 °C

$\dot{m}_w$ = 3989 x (0,016-0,010) = 23,9 kg/h dehumidification capacity[6].

$Q_{EVAP} = \dot{m}_a \times \Delta h - \dot{m}_w \times h_w$ [6]

Q_{EVAP} =3989 x (62-39,5)-23,9 x 63,0 =88246,8 kJ/h =24,5 kW evaporator capacity.

$W_{KOMP.}$= 24,5 / 3,8 = 6,4 kW.

4 Design Of Desiccant Dehumidification Unit

Desiccant dehumidification unit (Figure 7) are better to use when the desired level of moisture is below of what dehumidification unit with refrigeration cycles can provide as minimum. In units with refrigeration cycles, dehumidification capacity is practically unable to go below under condensation point since the device is limited to that point. Therefore, desiccant dehumidification systems are developed. In this systems, rotating drums are used, dehumidification (desiccant) main material of which is silica gel.

In this application, dehumidification unit that works with 100% internal air is used. 3/4 of air that passes over the drum is used in dehumidification, remaining air is used in throwing out the moisture that is absorbed by the drum. In the drying unit, design and examination of which was done, a rotor with 200mm thickness and with silica gel was chosen to be used in the air velocity of 4m/s. In this unit, 3/4 of the rotor will be used for dehumidification and air with a temperature of 30 °C and with a relative humidity of %60 will be blown over. Another reaction air with temperature of 120 °C will be blown over from 1/3 of it in order to throw out the moisture [7,11].

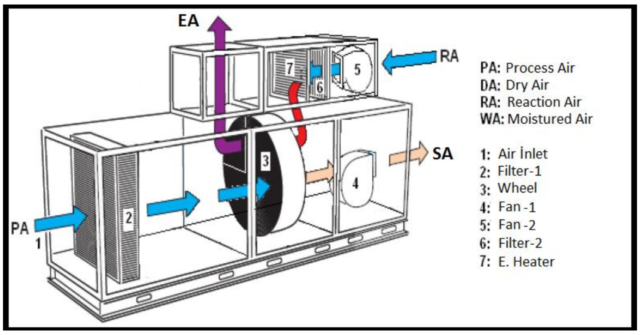

Figure 7. Schema of desiccant dehumidification Unit

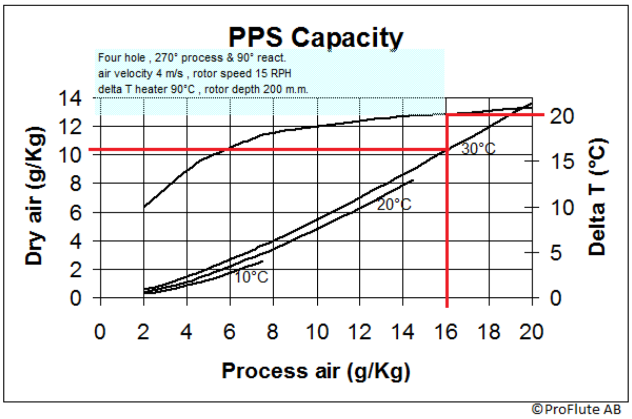

Figure 8. Selection diagram of silica gel wheel

Condition of accepted spot was determined as 16 g/kg moisture ratio. By using the diagram of choice within Figure 8, dry air moisture ratio and dry thermometer temperature that comes out of the rotor could be found.

Temperature of air outlet of the rotor 20+30=50 °C, humidity ratio 10,2 g/kg. These values are marked on the Figure 5, psychrometric diagram and were examined. SA is the supply air to the spot and EA is the exhaust air. 30 °C 60% relative humidity air is put through the rotor, dehumidificated and blown to the spot (SA). Sucked air is heated to 120 °C, put through the rotor and the rotor was dried after sucking the moisture [2,4,5].

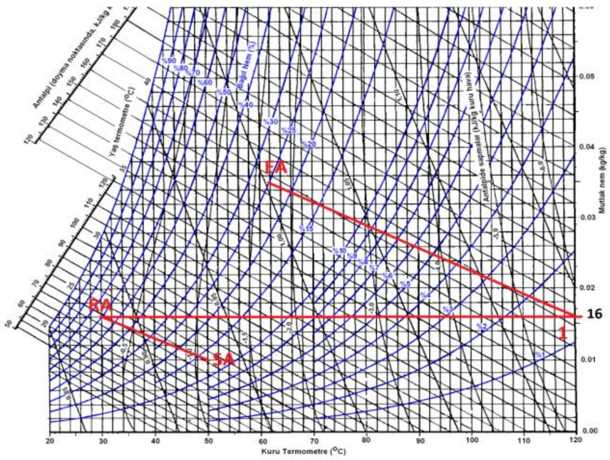

Figure 9. Examination of desiccant dehumidifier unit

Air flow rate is as below. 1/3 of dried air flow rate is taken as reaction air flow rate. With this air flow rate, electricity heater capacity is determined. Specific weight and specific heat values in the formula are considered by taking the average of entrance and exhaust temperature values.

Dehumidification capacity is 21,94 kg/h.
Air flow rate
$V_{proces\ air}$=21,94 x 1000/[(16-10,2) x 1,18]=3205 m^3/h [2,4,5].

$Q_{Electircity\ heater}= V_{proces\ air}/3 \times [(\rho_{RA}+\rho_1)/2] \times [(c_{pRA}+c_{p1})/2] \times \Delta T$
$Q_{Electircity\ heater}=3205/3 \times [(1{,}136+0{,}877)/2] \times [(1{,}007+1{,}011)/2] \times (120-30) = 97645{,}9$ kJ/h $=27{,}1$ kW

5 Control Of A Dehumidification Unit With Refrigeration Cycle

Control of dehumidification units are examined in two parts. First one is the principle of working according to external air condition, second one is the principle of working according to pool ambient air condition.

In the indoor swimming pools as shown in Figure 10, internal environment comfort condition according to VDI 2089/1 and ASHRAE [3] standards, must be held between 26-30 °C dry bulb temperature and 50-60% relative humidity. In the dehumidification unit designed, pool ambient condition was taken as 30 °C DB and 50% relative humidity. Pool water temperature was designed as 28 °C. Therefore the pool ambient relative humidity must be held between 50% and 60%. In case it goes below 50%, partial vapor pressure of the air will also decrease and evaporation on the pool surface will increase. This situation cause dehumidification unit to work more increase the energy consumption and therefore increase the various chemicals contained within the pool water and threat human health.

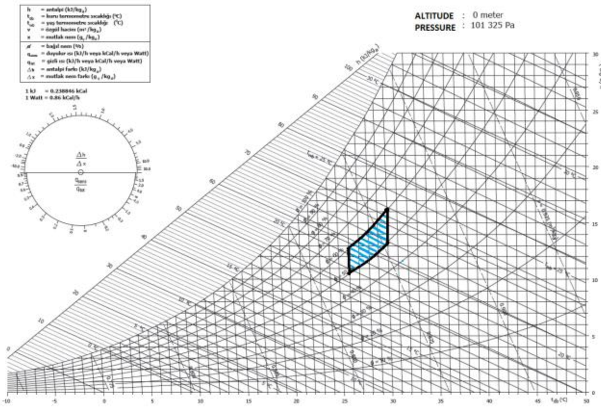

Figure 10. Pool ambient comfort conditions

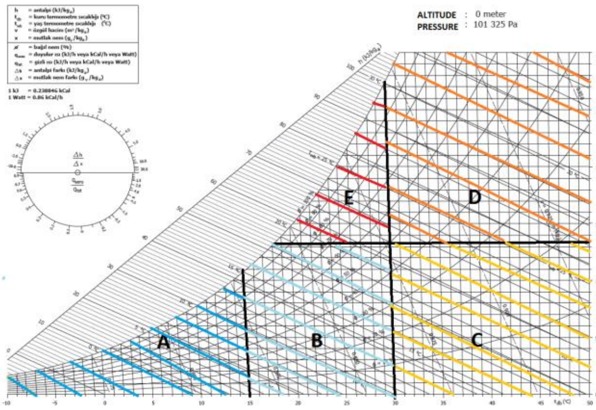

Figure 11. Control sections due to outside air conditions

Pool dehumidification unit control according to external environment condition provides high energy saving. In Figure 11, external environment conditions are divided to five main areas. Borderlines of these areas vary according to the design conditions pool ambient. Design condition must be taken as the origin. The part located on the left of 30 °C dry bulb temperature line is the part that needs heating, and the part on the right does not need heating but could be cooled if wanted to. The area below the 13,3 g/kg humidity ratio is the area without a need for moisture. The part below that line needs moisture. The dry bulb temperature line for 15 °C is the condition for heating with heat pump. In this study, design condition is 30 °C and 50% relative humidity so this point was taken as origin on psychrometric diagram.

Zone A is the one for winter conditions and changeover season weather conditions. In this area, humidity ratio of external air is lesser than the humidity ratio in design conditions, therefore heat pump system does not work and the air sucked from the pool ambient is mixed with proper ratio of fresh air, heated up until the design conditions and blown back to the pool area. In this area, it is not recommended to use 100% fresh air. The reason is, heating costs are higher than mixed air and the blown air with low moisture ratio increases the vaporization on the pool surface.

In case the dry bulb temperature of zone B goes above 15 °C, 100% fresh air could be used and heat pumping is convenient to be used as the heating method. Since the COP (3.5) value in compressor is better at energy efficiency compared to the solid fuel or natural gas systems in order to heat up the water in battery (between 0.70 and 0.80), such applications might be done [8].

Zone C represents the condition of dry climate. In this zone, heat pump system and heater do not work. 100% fresh air pumped to the pool area. In order to bring the pool ambient to comfort condition, extra cooling with air handling unit or dehumidification unit with cooling serpentine is recommended to be used.

Zone D represents the moist summer condition. Mediterranean coasts in Turkey are included to this zone within specific times of the year. In this zone, device works with 100% fresh air. In order to bring the pool ambient to design condition, a cooling serpentine could be added to dehumidification unit. Another solution is, using a heat pumping system that works in reversible. This way, maintenance costs, extra cooling serpentine and cooling group could be saved.

Zone E represents the changeover and summer season. In this zone, the heat pump works. External air mix is used as the need requires, and is blown to pool area.

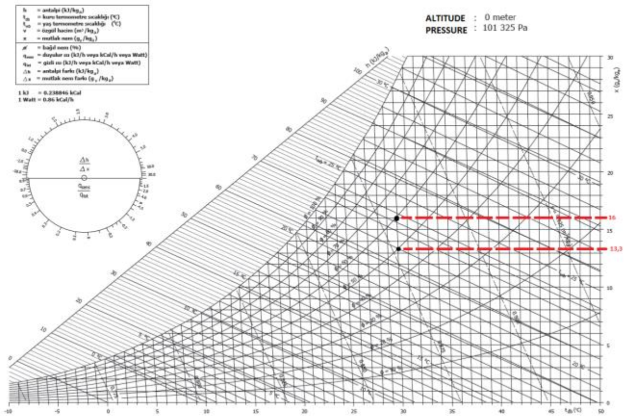

Figure 12. Design conditions for pool ambient

In the design, to hold pool internal environment humidity ratio between 13,3 g/kg and 16 g/kg, a pool dehumidification unit control has been performed. In case it goes below the design condition of 13,3 g/kg, the vaporization on the pool surface increases and therefore it is found convenient to shut down the function of dehumidification of the device, working with %100 internal air or with mixed air as much as the need requires in terms of energy efficiency.

Between 13,3 g/kg and 16 g/kg humidity ratio, the device works on partial load. If the external environment condition requires the compressors to work, compressors work on partial load. If the external air has lesser humidity ratio than the indoor air, the fresh air is blowing directly to the pool ambient. In case of reaching to 16 g/kg humidity ratio or exceeding this limit, the device works on full load. One of the three working option can be selected. these options are 100% internal air and compressors working, 100% fresh air or properly rated mix of fresh air with heating support.

6 Conclusion

In this study, firstly the vaporization rate on the pool surface was calculated. According to the calculation parameters, it is detected that if the difference between pool surface vapor pressure and environment design condition of partial vapor pressure increases, the evaporation rate of the pool surface will also increase in the same way. Additionally to that, it was understood that the difference between pool water temperature and pool ambient design temperature should be 2 degrees as a requirement of comfort condition.

In design of indoor swimming pool dehumidification unit, importance of using heat recovery unit was highlighted. Using heat recovery is proven to provide management and initial cost savings of up to 40%. Also using plate type heat recovery unit was understood to be more energy-efficient than heat pipe recovery unit and is recommended to manufacturers.

Since the limit of dehumidification units with refrigeration cycle is equal to condensation point, going below this point was theoretically possible for units with refrigeration cycles but in practice, it was understood that it causes high maintenance and energy costs. In places with need of very low humidity ratio, desiccant dehumidification systems are found to be more convenient to use and it is also possible to use them in indoor swimming pools with proper choice of drum. However, providing low relative humidity in indoor swimming pool volumes will not be convenient. When compared on equal capacity of dehumidification, it is found that desiccant dehumidification unit spends four times more energy than dehumidification units with cooling cycles.

Lastly, importance of automation in dehumidification units became obvious, a well-controlled device was prevented to work unnecessarily and therefore high operating costs was prevented.

7 Acknowledgment

Editors and Publishing Coordinators of *MJST* thank to all authors and reviewers for their contributions.

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