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Ro-Ro Kargo Gemisi İçin Tanımlanan Kurutuculu Buharlaşmalı Soğutma Sistemlerinin Termodinamik Analizi

Thermodynamic Analysis Of Desiccant Evaporative Cooling Systems Defined For Ro-Ro Cargo Vessel

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Abstract

This study presents the results of the thermodynamic analysis of the new desiccant- evaporative-air-cooling cycles can be applied in ships. The desiccant- evaporative- cooling system is characterized by energy efficiency and low environmental impact. In this work, thermodynamic possibility to install desiccant cooling system (DCS) has been studied for a M/V ASSTAR Trabzon Ro-Ro cargo vessel, by using fraction of the heat rejected by existing on-board engine. The baseline system is incorporated a desiccant dehumidifier, a heat exchanger, an indirect evaporative cooler, and a direct evaporative cooler. The system offered sufficient sensible and latent cooling capacities for a wide range of climatic, while allowing in flux of outside air in excess of what is typically required for Ro-Ro cargo vessel. The present work aims at identifying the parameters of the system cycle and investigates their effect on the performance of the waste-heat driven cooling systems. And, the effect of different return air flow rates usage on the system performance is another aspect of the study. Two ways are considered for mixing process of the return and outside air streams; one is consists of two recirculation cycles, other one is a ventilation cycle, all of them have been examined and demonstrated. The maximum coefficient of thermal performance (COP) of a waste-heat driven cooling cycles were determined by assuming that the cycles are totally reversible.

***Keywords**: Dessicant, dehumification, indirect evoparative cooling, heat recovery IC engine*

Öz

Bu çalışma gemilerde uygulanabilecek yeni nem almalı ve buharlaşmalı soğutma çevrimlerinin, termodinamik analiz sonuçlarını sunmaktadır. Nem almalı buharlaşmalı soğutma sistemi, enerji verimliliği ve düşük çevresel etkisi ile karakterize edilir. Bu çalışmada, gemide mevcut motor tarafından atılan atık ısının bir kısmı kullanılarak, M/V ASSTAR Trabzon Ro-Ro kargo gemisi için nem almalı ve buharlaşmalı soğutma sisteminin (DCS) termodinamik açıdan kullanılma olasılığı incelenmiştir. Temel sistem nem alma cihazı, ısı eşanjörü, dolaylı buharlaştırıcı soğutucudan ve doğrudan buharlaştırıcı soğutucudan oluşmaktadır. Sistem, çok çeşitli iklim koşulları için yeterli derecede duyarlı ve gizli soğutma kapasitesi sağlarken, Ro-Ro kargo gemisi için gerekli dış hava akışından daha fazlasını kullanımına imkan vermektedir. Bu çalışmada, sistem çevrim parametrelerini tanımlamayı ve bunların atık ısıyla çalışan soğutma sistemlerinin performansı üzerindeki etkilerini araştırmayı amaçlanmaktadır. Farklı dönüş hava debisi kullanımının sistem performansı üzerindeki etkisi de çalışmanın diğer bir özelliğidir. Geri dönüş havasının ve dış havanın karıştırılması için iki yaklaşım tanıtılmıştır; bunlardan biri iki farklı sirkülasyon çevrimlerinden oluşmakta iken, diğeri ise egzos çevrimi olmaktadır, hepsi de ayrı ayrı incelenmiş ve gösterilmiştir. Atık ısı ile çalışan soğutma çevrimlerinin maksimum ısıl performans katsayısı (COP), tanıtılan çevrimlerin tamamen tersinir çevrimler olduğu varsayılarak belirlenmiştir.

Anahtar Kelimeler: *Nem alıcı, nem alma, dolaylı buharlaşmalı soğutma, ısı geri kazanımlı IC motor*

1. Introduction

The main contributor to increasing atmospheric carbon dioxide, concentration is the combustion of fossil fuels from the marine propulsion. The energy consumed heating, ventilating and air conditioning (HVAC) application in vessel accounts for considerable amount of the total energy consumption. The effective use of energy is important in vessels. DCS uses recovered energy from the waste energy produced by the engine for air conditioning in marine application [1]. The performance evaluation on DCS helps the judgement of the improvement of the use of energy and air quality. In general HVAC application, DCS may be utilized to reduce energy consumption or to replace conventional refrigeration system [2 to 5].

In order to to create comfortable cooled air conditions for the buildings with lowest energy consumption and highest efficiency, a novel indirect evaporative air cooler was developed and was patented as "Maisotsenko cycle". A novel heat and mass exchanger (HMX) introduced for the Maisotsenko cycle which has wet and dry sides of a plate like indirect evaporative air cooler components but using different air flow configurations creates different thermodynamic cycle [6].

Evaporative cooling is a technology that can substantially reduce the cooling energy requirement in the vessel. There are three types of evaporative cooling process: Direct, indirect and indirect/direct. In direct evaporative cooling processes, the air is bought into direct contact with water in the direct evaporative cooler. Indirect evaporative cooling is achieved by sensibly cooling a primary air stream through heat exchanger. Heat is transferred to a secondary fluid on the cold side of the heat exchanger, which ultimately rejects heat to atmosphere the evaporation effect, the cold side fluid may be air, water or in the case of heat pipe refrigerant. The performance results of a thermally activated desiccant cooling system in a combined heat and power application incorporating a reciprocating internal combustion engine was presented by [7]. In the Eastern Black Sea, DEC systems were handled by to use the waste heat from the furnaces of a tea factory [8]. The study presents the results of the thermodynamic analysis of the introduced cooling systems.

It is the objective of this paper to demonstrate the thermodynamic advantage of a desiccant evaporative cooling system in the context of marine engine. In marine engine is implemented in the proposed system which provides a heat recovery opportunity for regeneration of the desiccant dehumidifier. The specific objectives to be achieved in this study are as follows:

- Demonstrate the cooling and thermodynamic performance of the system and its capability to provide comfort for a marine course climatic condition.
- Thermal performance analyses for assessment of different operating system configuration.

2. Material and Method

2.1. Power system definition of the M/V ASSTAR Trabzon Ro-Ro cargo vessel

The ship called M/V ASSTAR TRABZON considered in the study is a RO-RO cargo ship with a diesel engine. A Controllable Pitch Propeller is used as propeller in the RO-RO cargo ship. The ship has 12-cylinder, turbocharged ship diesel engine of the V-type used as a single propulsion machine. Schematic layout of the ship power system and energy flows are shown in Fig.1 and Fig.2, respectively and also technical specification of the diesel engine is presented in Table 1. [9].

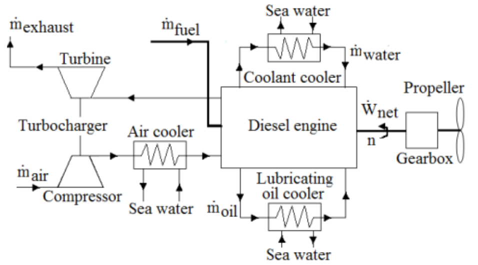

Figure 1. Schematic layout of the ship power system [9].

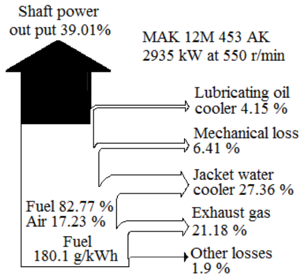

Figure 2. Energy flows of MAK 12M 453 AK turbo-diesel engine used for a M/V ASSTAR Trabzon Ro-Ro cargo vessel.

Table 1. Technical specification of the diesel engine used in the vessel [9].

MAK 12M 453 AK	
Type	Four stroke
Configuration	V type
Number of cylinders	12
Cylinder bore	320 mm
Stroke high	420 mm
Nominal engine speed	550 rpm
Nominal power	2935kW
Number of valves	24 intake+24exhaust
Flywheel direction	Counterclockwise
Injection order	A1-B6-A4-B3-A2-B5 A6-B1-A3-B4-A5-B2

The engine speed of the diesel engine having turbocharger was kept constant at 550 rpm and different engine loads were obtained by changing the propeller blade angles. Engine load of 70%, was created at 18^0 propeller blade angle. Measured performance value of the diesel engine system load ratio is given in Table.2 [9]. The exhaust gas discharged from the gas turbine was used to heat the regeneration air. Other data given in Table 2 are given to show the other waste heat potential of the ship. The variety of waste heat is important to show that different regeneration can be used as heating sources for the air conditioning method.

Table 2. Performance values of the diesel engine system on 70% load ratio [9].

MAK 12M 453 AK	
Mass flow of fuel	0,109 kg/s
Mass flow of air	3,43 kg/s
Air temperature	25 °C
Mass flow of air	3,43 kg/s
Air temperature	96 °C
Mass flow of air	3,00 kg/s
Air temperature	42 °C
Mass flow of the sea water	12,80 kg/s
Intake temperature of seawater	25 °C
Mass flow of the sea water	12,80 kg/s
Output temperature of seawater	28 °C
Exhaust gass temperature	358°C
Exhaust gass temperature	325 °C
Mass flow of coolant	34,12 kg/s
Coolant intake temperature	58 °C
Coolant output temperature	68 °C
Mass flow of lubricating oil	18,43 kg/s
Lubricating oil intake temperature	48 °C
Lubricating oil output temperature	54°C

2.2. Description of Desiccant Evaporative Cooling System

It can be stated, based on existing studies, that there is a major drawback in wider application of the desiccant-evaporative air conditioning systems, since the coefficient of thermal performance (COP) of the systems (defined as obtained cooling capacity divided by required thermal energy for regeneration of the desiccant) are generally lower than 1.0.

The purpose of the desiccant evaporative air cooling system is to use waste heat in some way for air cooling. In this system, the waste heat is used in a heat exchanger for heating the regeneration air. The regeneration air is used for drying the desiccant. After the humid and hot air from the atmosphere passes through the dehumidifier, the specific moisture decreases and the temperature rises. The temperature is then reduced by passing through a heat exchanger. This air is then passed through an evaporative refrigeration unit to bring the space to cooling conditions, whereby the specific humidity remains constant, the temperature is reduced to the space temperature, and the relative humidity adjustment is provided by a direct evaporation unit. Detailed descriptions are given for the system described in Section 2.3. The use of fans in the system and the use of additional electric heaters in general causes the oefficient of thermal performance (COP) to drop. In the literature for the method is indicated that Coefficient of thermal performance (COP) is below 1 [10,11,12,13]. In this study, the electrical energy used in the fans is not considered. In addition, additional electric heaters are not used.

2.3. System Configurations

The waste heat from the exhaust gas of MAK 12M 453 AK turbo-diesel engine is treated as a thermal energy source for the proposed desiccant cooling systems which can be applied for the supply air entering the indoor space. As shown in Figure 3., the system under consideration consists of a power generation

MAK 12M 453 AK turbo-diesel engine, a thermally activated rotary desiccant wheel (DW), an air-to-air heat exchanger (HX), an indirect evaporative cooler (IEC), and a direct evaporative cooler (DEC) as a model 1. The psychometric cycle of mode 1 is shown in Fig.4. In operation (see in Fig.3 and Fig.4), the outdoor air station-1 is divided for two streams, one for IEC station 7 and other one for process air [1]. The air stream for IEC station-7 mixes with return air station-6 in the ratio of 1:2. The process air station-1 is heated up to condition of station-2 by means of the desiccant wheel (DW) while the specific enthalpy of the process air remains constant until it reaches 0.03% humidity. This condition can be set up in accordance with the type of the desiccant to be used. Then dehumidified and hot process air is cooled by the air to air heat exchanger (HX). In the HX, the sensible cooling is achieved for the process air by using exhaust air from the IEC station-8 having 100% humidity. The process air at the exit of the IEC station-4 is cooled and humidified until it reaches the supply air condition station-5.

Then conditioned air enters the indoor space station-6. The regeneration air station-10 is heated up to station-11 by means of the hot exhaust gases come from diesel engine using a gas-to- air heat exchanger. This air station-11, can be finally be used to regenerate the (DW). After the regeneration, the air is exhausted and can be released at station-12. Other two subsystems models of the desiccant cooling system (model 2 and model 3) are defined to satisfy the required conditions of the supply air entering the indoor space, can be seen in Figure. 4 to through Figure. 8.

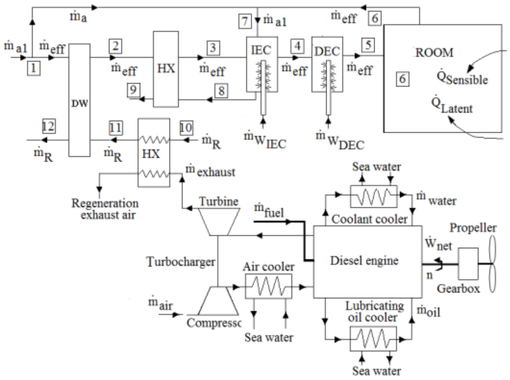

Figure 3. System schematics (Exhaust Heat Recovery and model-1)

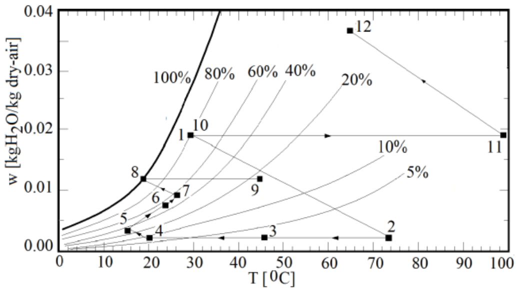

Figure 4. The psychometric cycle of model-1

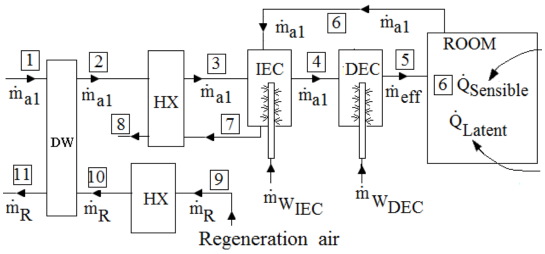

Figure 5. System schematics for model-2

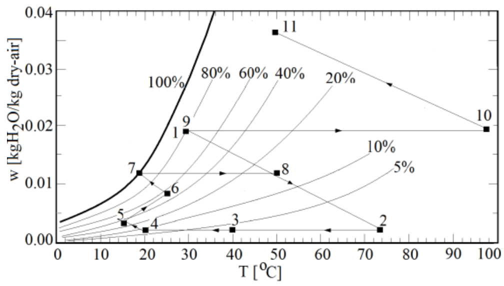

Figure 6. The psychometric cycle of the model-2

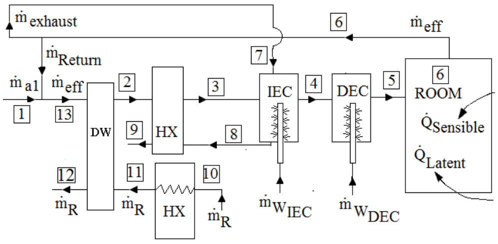

Figure 7.System schematics for model-3

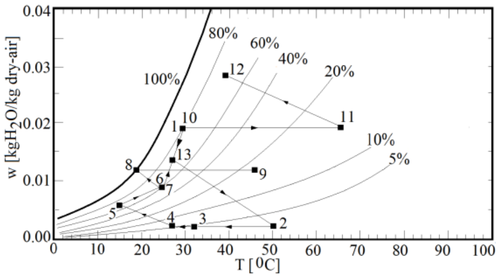

Figure 8. The psychometric cycle of the mode-3

2.4.Methodology: Computational application

The HVAC calculations are done by using mass continuity and the steady flow energy equations with the help of the psychometric diagram. The waste heat used for the proposed systems is the lowest thermal energy that is exhausted from the diesel engine by exhaust gases. The exhaust air volume flow rate and the temperature limits to be used for the process air volume flow rate to be conditioned can be ensured by the operating conditions of the diesel engine. Using a gas-to-air heat exchanger, regeneration air temperature can be reached 100 ^{o}C or higher by means of exhaust gases provided by diesel engine. The mass flow rate of the process air and the mass flow rate of the regeneration air are selected to be equal. Model 2 is defined as the use of the recirculated air all over the IEC. Model 3 is defined as the use of only a part of the recirculation air in (IEC).Different return air configuration effects on system performance has been aimed at.

The operation modes of three systems are determined based on the outdoor air conditions, where the outdoor air is hot/warm and very humid. The peak thermal loads of the cooled space were estimated using the calculation method recommended by ISO-7547, which utilizes the load calculation procedure and basic assumptions that are critical for designing HVAC systems serving passenger cabins in ships [5]. In this study, the outdoor air condition with at 38◦C dry bulb and 80%relative humidity was considered for the summer design condition. The design room air conditions used at 24◦C dry bulb and 50% relative humidity for the summer. The design conditions assumed for summer in this study should only be used in normal climate regions and not in extremely hot climates.

The transmission heat gains and losses for each marine space surface can be calculated using Equation (1) [14]. The heat transmission between the cabins was neglected based on the assumption that the indoor air temperature in each cabin (i.e., 24 °C) was identical.

$$Q = \Delta T\left[(K_v . A_v) + (K_g . A_g)\right] \quad (1)$$

Where K_v is the heat transfer coefficient of the external surface in (W/(m^{2o}C)) and K_g is the heat transfer coefficient of the external glass window in [W/(m^{2o}C)]. Solar heat gain from windows and the deck are estimated in this calculation. A_v is heat transfer surface area for the deck A_g is the surface area of all windows.

The sensible and latent heat gains from a person exerting medium work were assumed to be 70 W/person and 50 W/person, respectively. For the lighting heat gain, fluorescent lighting equipment was assumed; the lighting equipment yielded a sensible heat emission of 8 W/m^2. Two persons per square meter for the accommodation spaces and passenger cabins were considered in this study. Because of the high level of air-tightness in ship spaces, air infiltration and leakage in the cabin were neglected while calculating the thermal load. Table 3. lists the physical information of the modelled cooling spaces.

Table 3. Physical parameters of the model cooling space for calculating the thermal load.

Space size	Width is 16 m, Height is 5m, Length is 5 m
Schedule on board	At solar noon time
Space conditions in Summer	24 °C, 50%
Heat transfer surface area	~400 m^2
Window-to-deck ratio	15%
Occupants People	10 (Sensible heat: 70 W/person, latent heat: 50 W/person)

The conventional constant air volume (CAV) systems are considered for three the desiccant-evaporative air conditioning models in this study. The energy consumption of the CAV system for proposed models were calculated based on the defined peak cooling loads of the model space in the ship. The indoor space cooling load is determined by means of the sensible heat factor, SHF, is defined by the following equation:

$$SHF = \frac{\dot{Q}_s}{\dot{Q}_s + \dot{Q}_L} \qquad (2)$$

Where $\dot{Q}_s$ stands for the room sensible cooling load and $\dot{Q}_L$ stands for the room latent cooling load. The typical value of SHF is calculated to be 0.7 for three models.

$$\dot{Q}_s = \dot{m}_a C_p (T_5 - T_4) \qquad (3)$$

$$\dot{Q}_L = \dot{m}_a h_{fg} (w_5 - w_4) \qquad (4)$$

The total cooling load, $\dot{Q}_{Total}$, can be calculated by

$$\dot{Q}_{Total} = \dot{Q}_s + \dot{Q}_L \qquad (5)$$

In order to evaluate the cooling performance of the cooling system, the coefficient of performance (COP), is considered on the based on thermal energy supply.

$$COP = \frac{\dot{Q}_T}{\dot{Q}_{DES,R}} \qquad (6)$$

The thermal energy supplied for the system is used for the heating of the regeneration air passed in the desiccant wheel. The thermal energy supplied for the regenerative air flow heating is given by the following equation,

$$\dot{Q}_{DES,regen.} = (\dot{m}_a C_P)_R (\Delta T)_R \qquad (7)$$

3. Results and Discussions

The two proposed air stream mixing process for the return air and out side air as a recirculation cycles model-1 and model-3 and the exhaust air stream considred to use directly in a ventilation cycle model-2 were examined and results presented through Figures 9 to Figure 11. The impact of the ambient air conditions on the system coefficient of thermal performance (COP) of three models, for a wide range of ambient relative humidity at different dry bulb temperatures are presented in Figure 9 to Figure 11. When Figure 9 and Figure 10 are examined, according to the results of Model 1 and Model 2, it was obtained that the coefficient of thermal performance (COP) values of the systems were reduced when the relative humidity of the atmospheric air got high values. Similarly, it was observed that atmospheric air tended to reach maximum value of coefficient of thermal performance (COP) as dry thermometer temperature increased. In particular, when the relative humidity of the atmospheric air reached a value of 0.8, it was found that the coefficient of thermal performance (COP) reached a maximum value and the coefficient of thermal performance (COP) decreased when the drying temperature of the air was increased. In Model 3, it is seen from Figure 11 that as the drying temperature of the atmosphere air increases and the relative humidity increases, the coefficient of thermal performance (COP) of the system decreases continuously and does not reach a maximum value. For Model 1, examples of typical values obtained according to atmospheric air conditions are; At 50% relative humidity, the coefficient of thermal performance (COP) at 32^0 C was found to be 0.319. In Model 2, coefficient of thermal performance (COP) was calculated as 0.364 for the same atmospheric weather conditions and coefficient of thermal performance (COP) for the Model 3 was 0.178.

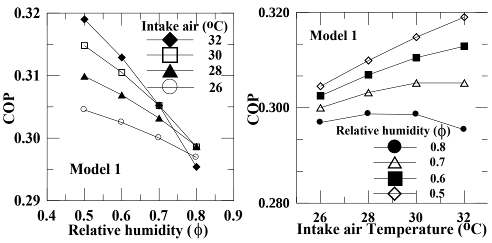

Figure 9. Impact of intake air conditions on the system coefficient of thermal performance (COP) for model-1.

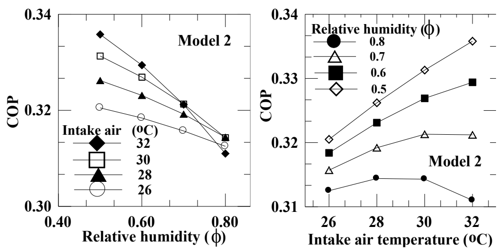

Figure 10. Impact of intake air conditions on the system cooling performance (COP) for model-2.

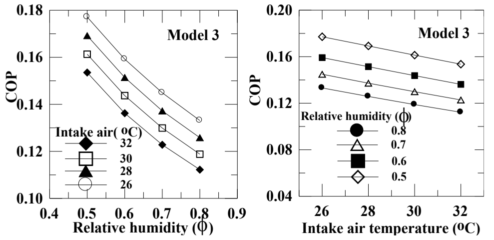

Figure 11. Impact of intake air conditions on the system coefficient of thermal performance (COP) for model-3.

Figure 12. shows the effect of different return air flow rates usage on the system coefficient of thermal performance (COP), for three models where the ambient air dry-bulb temperature changes between 25 to 38°C while relative humidity is kept constant as 0.80. It is seen that Model 2 at given operation condition shows better performance comparing other two model at the same intake air conditions while the cooling load increasing. Model 2 (ventilation cycle) is most promising cycle and operational system for HVAC applications for hot and humid climatic regions.

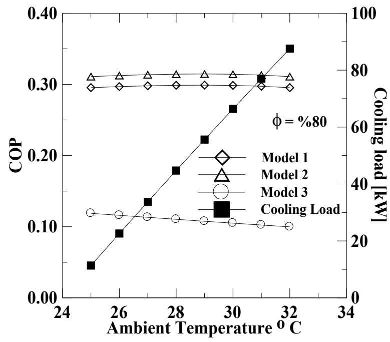

Figure 12.Effect of intake air conditions and operational systems on the coefficient of thermal performance (COP), for three models.

4. Conclusion

Exhaust gases from MAK 12M 453 AK turbo-diesel engine used for a M/V ASSTAR Trabzon Ro-Ro cargo vessel, has a potential of waste heat energy recovery in the field of the desiccant-evaporative air conditioning systems in cargo vessel. Three case studies are introduced in this study that shows the thermodynamic possibilities of the desiccant-evaporative air conditioning systems applications in cargo vessel. Model 2 (ventilation cycle) is most promising cycle and operational system for HVAC applications for hot and humid climatic regions.

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Nomenclature

$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat transfer load (kW)
h	Enthalpy (kJ/kg)
C_p	Specific heat (kJ/kgK)
w	Humidity ratio [kg H_2O/kg dry-air]
T	Temperature (^0C)
DCS	Desiccant cooling system
IEC	Indirect evaporative cooler
COP	Coefficient of thermal performance
HVAC	Heating, ventilating and air conditioning
HMX	Heat and mass exchanger
HX	Heat exchanger
DW	Desiccant wheel
RO-RO	Roll-on/roll-off
a	Air
R	Rejeneration air
eff	Effektive
K	Heat transfer coefficient [W/($m^{2o}C$]
v	External surface
g	External glass window
A	Heat transfer surface area
CAV	Constant air volume
SHF	Sensible heat factor
S	Sensible
L	Latent
DES	Dessicant