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Optimizing Cooling System Design: A Case Study for Innovation Building

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Abstract: This paper presents the design and optimization of a cooling system tailored for innovation building, focusing on maximizing energy efficiency while ensuring optimal thermal comfort for occupants. The proposed cooling system integrates several key components, including chillers, air handling units, distribution systems, and control mechanisms. The system utilizes a combination of centralized and decentralized cooling strategies to efficiently manage thermal loads across different zones within the building. Additionally, advanced control algorithms are employed to dynamically adjust cooling operations based on real-time data, occupant preferences, and external environmental conditions. The optimization process involves comprehensive simulation studies and performance evaluations using building energy modeling software and an Hourly Analysis Program (HAP). Various design parameters such as equipment sizing, layout configurations, air distribution patterns, and control strategies are iteratively refined to achieve the desired balance between energy efficiency, cost-effectiveness, and occupant comfort. The design and optimization of a cooling system for an Innovation building represents a critical step toward achieving sustainable and energy-efficient building operations. By leveraging advanced technologies and innovative design approaches, this cooling system offers a viable solution for addressing the cooling needs of Innovation buildings while mitigating the environmental impact associated with traditional HVAC systems. The proposed design was applied in a case study of an innovation building during the cooling seasons and the results of the optimization demonstrate significant improvements in energy consumption, with substantial reductions in both electricity demand and greenhouse gas emissions compared to conventional cooling systems. Furthermore, the proposed cooling system maintains consistent thermal conditions throughout the building, ensuring a comfortable indoor environment for occupants while minimizing the risk of thermal discomfort and overheating.

Keywords: Energy efficiency, Energy engineering, Cooling system, HVAC systems

Introduction

Studying heat load via the HAP program (Hourly Analysis Program) and air conditioning absorption cycle using chilled water (6°m -12°m) for the Innovation building at Near East University in Nicosia, an area of 2555.7 m^2, which consists of three floors including lecture halls, laboratories, teacher's offices, and Cafe. The building is located in Nicosia with a 35.2 Deg, longitude of 33.3 degrees and elevation of 224m. The summer design dry bulb is 37.2 ℃, the summer coincident wet bulb is 20.6℃ and the summer daily range is 14.4K.

Absorption systems utilize heat energy to generate refrigeration or heating, and sometimes to increase the temperature of waste heat. Aqueous lithium bromide (LiBr) is commonly used to absorb refrigerant and water vapor, resulting in a higher performance coefficient. The modern absorption systems have been categorized as follows: absorption chillers, which use heat energy for refrigeration; absorption chiller heaters, which provide both cooling and heating; absorption heat pumps, which extract heat energy from the evaporator through the absorber, and add it to the heat input in the generator, and release it as hot water in the condenser for heating;

and absorption heat transformers, which elevate the temperature of waste heat to a level higher than any other input fluid stream supplied to the absorption heat transformer (Alefeld & Ziegler, 1985). Absorption chillers have heat exchangers to conserve heat energy input and increase heat transfer efficiency (Gommed & Grossman, 1990). This ensures the transformer is always in the best possible condition for maximum energy efficiency. The level difference also helps to prevent overheating and other potential issues. The heat input for absorption chillers can be derived from other sources than direct-fired burners, such as industrial waste heat or exhaust heat from steam turbines or engines (Grossman, 1982).

Background

In the 1950s and 1960s, central refrigeration plants commonly used centrifugal chillers powered by electric motors and absorption chillers that used steam for summer cooling. Steam was a popular choice because many central plants, which utilized steam for winter heating, had an excess supply during the summer, and energy costs were not a significant concern at that time. However, following the 1973 energy crisis, the natural gas and oil prices needed to fuel steam boilers rose dramatically. The coefficient of performance (COP) in the early model of single-stage, indirect-fired steam absorption chillers was only between the interval of 0.6 to 0.7, making them less energy-efficient and unable to compete with electric centrifugal chillers. Consequently, many absorption chillers were replaced by centrifugal chillers during the late 1970s and 1980s. Moreover, in recent years, absorption chillers have seen a resurgence in popularity due to increased energy efficiency and cost savings. Many modern absorption chillers are now able to achieve COPs of 0.8 or higher, making them competitive with electric centrifugal chillers.

Due to the high costs of building new power plants, electric utility companies began imposing high demand charges and increasing cost-per-unit charges during peak usage periods. Recently, double-effect, direct-fired absorption chillers have been developed in Japan and the United States, achieving a COP of around 1. These chillers are more efficient and have lower operating costs than traditional chillers, making them an attractive option for many businesses. They can also be used to generate electricity during peak usage periods, reducing energy costs even further.

Cost Analysis

A cost analysis is frequently necessary. In the cost analysis, (Aumann,1996) recommended including not only the chiller itself but also the energy and original costs of auxiliaries, such as condenser pumps, cooling towers, and tower fans, given the higher heat rejection in absorption chillers. Auxiliary energy costs can rise to 30% up for absorption chillers compared to ones that use electric power (Aumann, 1996). Additionally, absorption chillers require regular maintenance to ensure that the chiller is performing optimally. This maintenance requires additional labor and materials, which can further increase the overall cost. The absorption chiller has an annual energy cost saving of $20,375 and a simple payback period of about 6 years (ASHRAE,1998). This highlights the importance of considering the total cost of an absorption chiller when analyzing its efficiency. It is also important to consider environmental factors, such as emissions when considering the total cost of an absorption chiller.

Method

Hourly Analysis Program (HAP)

HAP is intended for consulting engineers, design/build contractors, HVAC contractors, facility engineers, and other professionals engaged in designing and analyzing commercial building HVAC systems. The program serves as a robust tool for system design and component sizing. HAP provides users with a comprehensive suite of powerful tools that help them quickly and efficiently model and optimize HVAC systems. It allows users to quickly and accurately analyze system performance and calculate energy consumption. HAP is also compliant with the latest industry standards and regulations.

Carrier's Hourly Analysis Program integrates two powerful tools into a single package: versatile features for designing HVAC systems in commercial buildings and advanced energy analysis capabilities for comparing the energy consumption and operating costs of different design alternatives. The program helps users identify the most cost-effective system design solution, considering both upfront and operational costs. It allows users to optimize system performance and minimize energy consumption. They are integrating both tools in one package results in significant time savings, as the input data and results from system design calculations can be directly utilized in energy studies. The program helps designers create energy-efficient systems while optimizing system performance. It also helps to identify potential saving opportunities and develop strategies that optimize the energy efficiency of a building.

A. Tools of the Program:

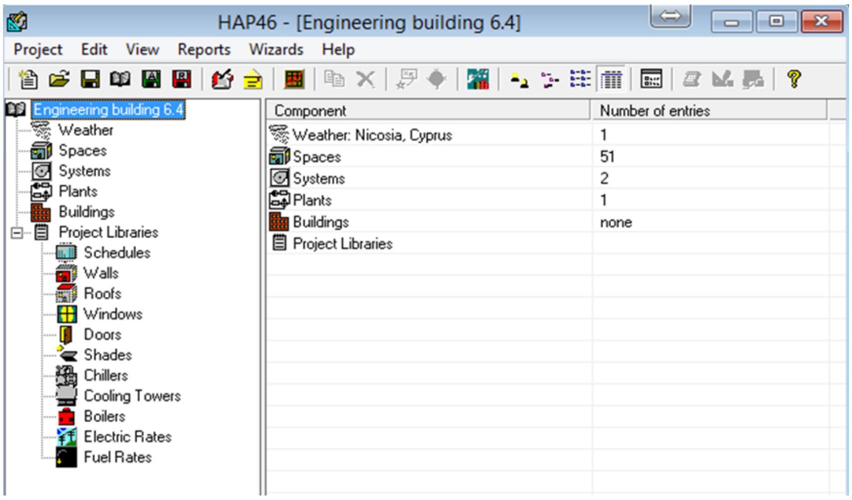

Figure 1. User layout of tools for HAP

B. Weaher:

"Weather Data" encompasses the temperature, humidity, and solar radiation conditions that affect the building and its HVAC equipment. In HAP, this term also includes information about the building's geographical location, local time characteristics, and soil properties. Weather data significantly impacts building loads and equipment operation, making it crucial for load calculations and system performance assessments. Weather data can also be used to assess the energy efficiency of buildings and HVAC systems. İt can also be used to forecast the future performance of a building or system. Finally, weather data can be used to optimize building operations.

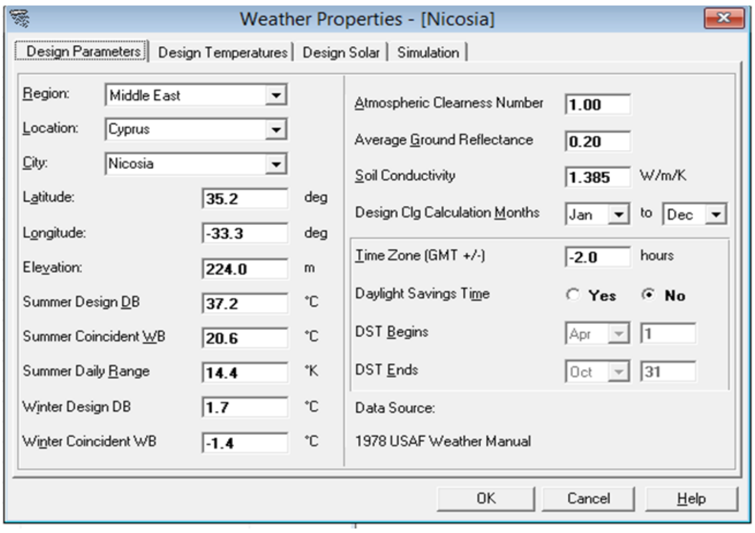

Figure 2. Weather tools layout of the HAP

Calculation

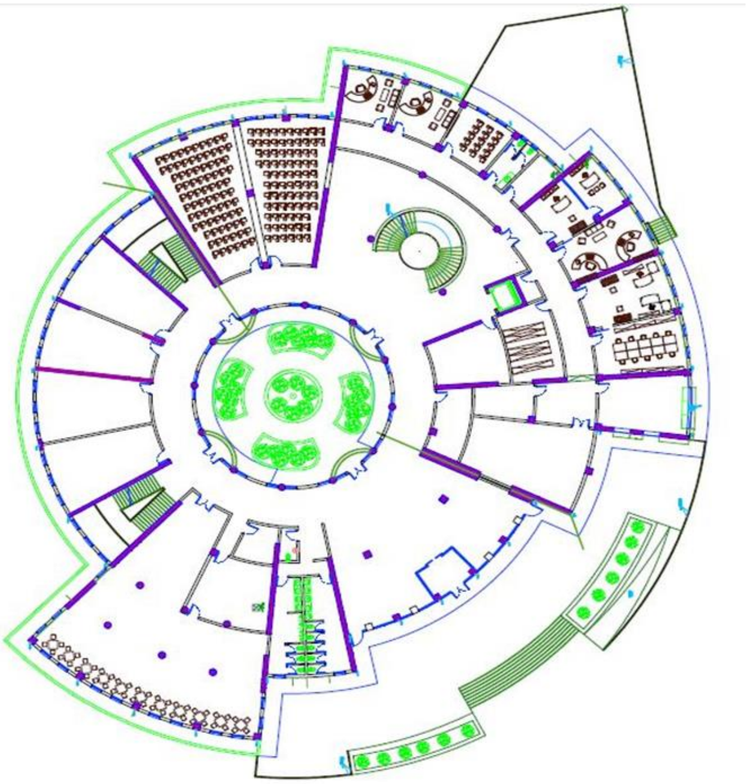

Figure 3. Schematic layout of the innovation building

A. Building load

The building loan entails the data essential for the calculations, the weather inputs, the walls, windows, and roof are all essential load pointers for the calculation required in the design.

Weather Inputs

In the beginning, we selected weather properties of the design area by choosing Region, Location, and City

Table 1. Design parameter of weather inputs

City Name	Nicosia
Location	Cyprus
Latitude	35.2°C
Longitude	-33.3°C
Elevation	224m
Summer design dry-bulb	37.2°C
Summer daily wet bulb	20.6°C
Summer daily range	14.4K
Winter design dry-bulb	1.7°C
Winter design wet bulb	-1.4°C
Atmospheric clearness number	1.00
Average ground reflectance	0.2
Soil conductivity	1.385 W/(m-°K)
Local time zone (GMT +/- N hours)	-2 hours
Consider daylight savings time	No
Simulation weather data	N/A
Current data	1978 USAF Weather Manual
Design Cooling Months	January to December

B. Project Properties

Wall Input

Table 2. Property details of the external wall

Month	N	NNE	NE	ENE	E	ESE	SE	SSE	S
January	68.8	68.8	68.8	307.1	529.0	687.9	784.1	801.0	785.9
February	81.6	81.6	197.7	429.1	640.5	750.1	788.9	754.2	718.4
March	95.8	95.8	345.8	556.3	691.6	762.8	720.1	641.5	591.2
April	109.9	240.4	453.0	629.0	709.4	692.8	610.0	480.6	412.2
May	119.4	340.4	517.3	655.5	696.6	632.7	514.7	354.4	278.9
June	153.3	373.5	542.0	660.4	681.1	600.0	469.1	300.2	229.2
July	121.9	333.6	520.3	647.7	679.1	620.0	499.6	341.0	268.5
August	115.0	232.4	447.1	608.8	679.4	672.2	586.3	462.0	397.1
September	99.7	99.7	315.5	530.4	658.7	724.4	696.2	620.7	578.9
October	84.9	84.9	182.8	426.2	594.4	730.1	753.0	726.4	704.4
November	70.5	70.5	79.0	278.2	522.8	682.5	769.3	783.7	780.6
December	63.5	63.5	63.5	245.4	480.6	647.5	766.7	801.6	799.9
Month	**SSW**	**SW**	**WSW**	**W**	**WNW**	**NW**	**NNW**	**HOR**	**Mult**
January	800.7	782.6	691.4	524.8	309.7	68.8	68.8	496.6	1.00
February	753.9	788.5	751.9	640.0	426.3	199.0	81.6	632.1	1.00
March	644.6	726.1	762.2	701.5	542.4	347.7	95.8	754.9	1.00
April	483.2	608.2	701.7	705.4	617.2	465.4	231.1	830.8	1.00
May	354.7	509.3	641.8	686.9	653.6	534.9	331.6	859.4	1.00
June	300.2	461.8	609.4	668.2	661.2	554.8	363.2	860.4	1.00
July	341.8	493.9	625.8	668.6	646.2	527.0	324.4	846.0	1.00
August	465.9	586.2	676.5	680.2	596.9	452.4	229.5	813.5	1.00
September	625.4	701.4	713.2	670.9	525.7	307.6	99.7	734.4	1.00
October	727.2	754.4	729.5	592.6	427.1	180.8	84.9	627.7	1.00
November	780.1	765.5	686.0	518.0	292.7	76.8	70.5	500.5	1.00
December	795.5	756.9	661.6	472.1	249.1	63.5	63.5	441.8	1.00

Wall Details

Outside Surface Color: Light
Absorptivity: 0.450
Overall U-Value: 1.958W/(m²-°K)

Table 3. Wall layer details (inside to outside)

Layers	Thickness mm	Density kg/m³	Specific Ht. kJ / (kg - °K)	R-Value (m²-°K)/W	Weight kg/m²
Inside surface resistance	0.000	0.0	0.00	0.12064	0.0
20 mm plaster	19.990	1858.1	0.84	0.02599	37.1
203mm common brick	203.200	1922.2	0.84	0.27954	390.6
20 mm plaster	20.000	1858.1	0.84	0.02599	37.2
Outside surface resistance	0.000	0.0	0.00	0.05864	0.0
Totals	**243.190**	**-**	**-**	**0.51079**	**464.9**

Roof Input

Outside Surface Color: Dark
Absorptivity: 0.900
Overall U-Value: 0.601W/(m²-°K)

Table 4. Roof layer details (inside to outside)

Layers	Thickness mm	Density kg/m³	Specific Ht. kJ / (kg - °K)	R-Value (m²-°K)/W	Weight kg/m²
Inside surface resistance	0.000	0.0	0.00	0.12064	0.0
200 concretes	250.000	2242.6	0.84	0.11579	448.5
Leveling concrete 70mm	70.000	400.0	0.84	0.12000	28.0
RSI-1.2 board insulation	25.400	32.0	0.92	1.22299	0.8
Asphalt roll	1.588	1121.3	1.51	0.02698	1.8
Outside surface resistance	0.000	0.0	0.00	0.05864	0.0
Totals	**346.988**	**-**	**-**	**1.66503**	**479.1**

Windows Input

Glass window
Window Details
Detailed Input: Yes
Height: 2.40m
Width: 1.00m
Frame Type: Aluminum without thermal breaks
Internal Shade Type: Roller Shades - Light - Translucent
Overall U-Value: 3.353W/(m²-°K)
Overall, Shade Coefficient: 0.334

Table 5. Glass details (gap type 13mm air space)

Glazing	Glass Type	Transmissivity	Reflectivity	Absorptivity
Outer Glazing	6mm blue-green reflective	0.282	0.295	0.423
Glazing #2	6mm clear	0.792	0.079	0.129
Glazing #3	not used	1.000	0.000	0.000

Air System Input

1. General Details

Air System Name: Default System
Equipment Type: Chilled Water AHU
Air System Type: Single Zone CAV
Number of zones: 1

2. System Components

Ventilation Air Data

Airflow Control: Constant Ventilation Airflow
Ventilation Sizing Method: Sum of Space OA Airflows
Unocc. Damper Position: Closed
Damper Leak Rate: 0%
Outdoor Air CO2 Level: 400ppm

3. Dehumidification Data

Maximum RH Setpoint: 50%
Heating Source: Hot Water

4. Central Cooling Data

Supply Air Temperature: 14.4°C

Coil Bypass Factor:	0.100
Cooling Source:	Chilled Water
Schedule:	JFMAMJJASOND
Capacity Control:	Cycled or Staged Capacity - Fan On

5. Supply Fan Data

Fan Type :	Forward Curved
Configuration:	Draw-thru
Fan Performance:	0Pa
Overall Efficiency:	54%

6. Duct System Data

Supply Duct Data

Duct Heat Gain:	0%
Duct Leakage:	0%

7. Return Duct or Plenum Data

Return Air via *Ducted Return*

8. Thermostats and Zone Data

Zone:	All
Cooling T-stat: Occ:	23.0°C
Cooling T-stat: Unocc:.	25.0°C
Heating T-stat: Occ:	21.1°C
Heating T-stat: Unocc:.	18.3°C
T-stat Throttling Range:	0.83°K
Diversity Factor:	100%
Direct Exhaust Airflow:	0.0L/s
Direct Exhaust Fan kW:	0.0kW
Thermostat Schedule:	fan
Unoccupied Cooling is:	Available

9. Supply Terminals Data

Zone:	All
Terminal Type:	Diffuser
Minimum Airflow:	0.00L/s/person

10. Zone Heating Units

Zone:	All
Zone Heating Unit Type:	None
Zone Unit Heat Source:	Hot Water
Zone Heating Unit Schedule:	JFMAMJJASOND

Systems Design Report

Air System Information

Air System Name:	Default System	Number of zones:	1
Equipment Class:	CW AHU	Floor Area:	2545.5 m²
Air System Type:	SZCAV	Location Nicosia:	Cyprus

Sizing Calculation Information

Zone and Space Sizing Method

Zone L/s	Sum *of space airflow rates*	Calculation Months:	Jan to Dec
Space L/s	Individual *peak space loads*	Sizing Data:	Calculate

Central Cooling Coil Sizing Data

Total coil load:	554.1kW	Load occurs at	Aug 1500	
Sensible coil load:	469.8kW	OA DB / WB:	37.2 / 20.6°C	
Coil L/s at Aug 1500:	26424L/s	Entering DB / WB:	27.1 / 17.8°C	
Max block L/s:	26424L/s	Leaving DB / WB:	11.9 / 11.1°C	
The sum of peak zone L/s:	26424L/s	Coil ADP:	10.3°C	
Sensible heat ratio:	0.848	Bypass Factor:	0.100	
m²/kW:	4.6	Resulting RH:	50%	
W/m² :	217.7	Design supply temp.	14.4°C	
Water flow @ 6.0 °K rise:	22.10L/s	Zone T-stat Check	1 of 1	OK
		Max zone temperature deviation:	0.0°K	

Central Heating Coil Sizing Data

Max coil load:	214.3Kw	Load occurs at	Dec 0800
Coil L/s at Dec 0800:	26424L/s	W/m²:	84.2
Max coil L/s :	26424L/s	Ent. DB / Lvg DB:	11.5 / 18.4°K
Water flow @ 11.1 °K drop:	4.62L/s		

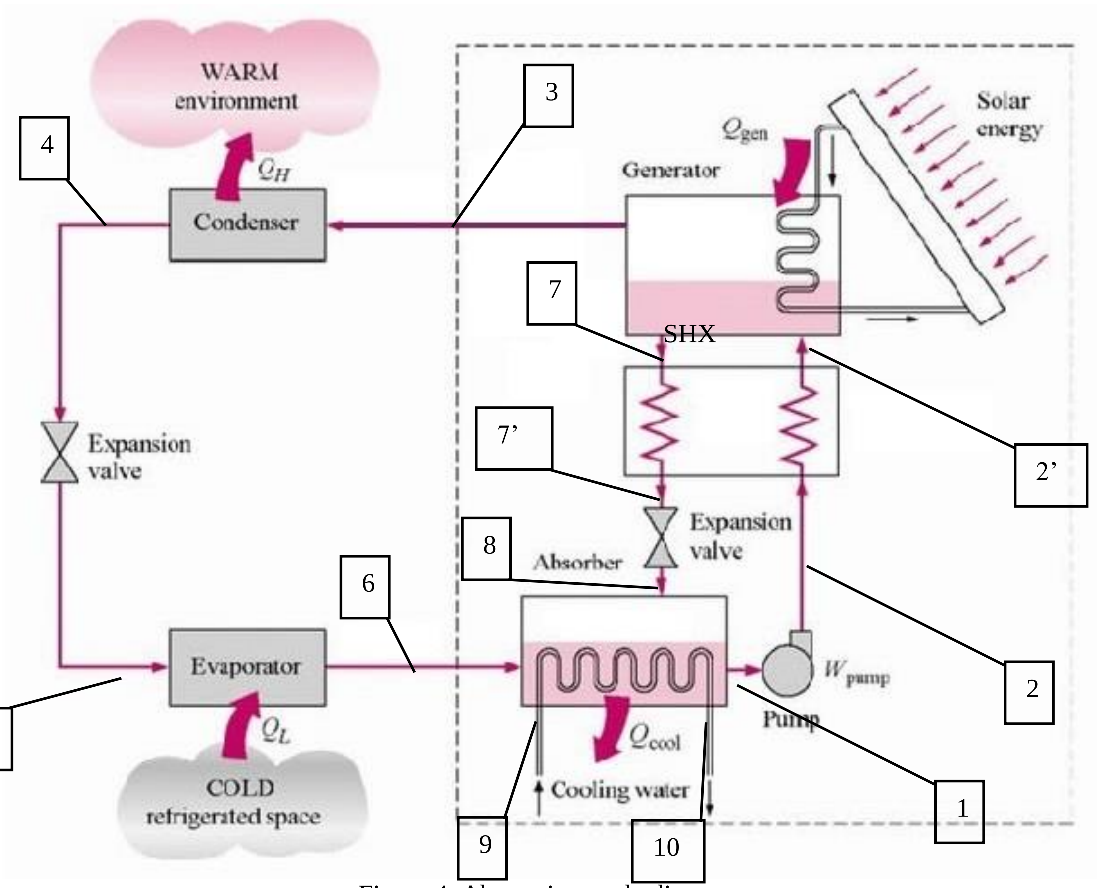

Figure 4. Absorption cycle diagram

Supply Fan Sizing Data

Actual max L/s: 26424L/s
Standard L/s: 25730L/s
Actual max L/(s-m²): 10.38L/(s-m²)

Outdoor Ventilation Air Data

Design airflow L/s: 6568L/s
L/(s-m²): 2.58L/(s-m²)

Evaporator Load of Absorption Cycle

Load [Kw]: 555
Chilled Water Flow: (6°C - 12°C) [L/s] 22.1

Absorption Cycle Calculations

$\dot{m}$=22.134 kg/sec

1. at $T_e = 5\ °C \rightarrow P_e = 0.9432\ [kpa]$ low pressure

2- at $T_c = 30\ °C \rightarrow P_c = 4.246\ [kpa]$ high pressure

From chart:

3- $P_7 = 4.246[kpa], T_7 = 85\ °C \rightarrow x_7 = 65\%\ rich\ solution\ consentrate$

4- $P_1 = 0.9432[kpa], T_1 = 30\ °C \rightarrow x_1 = 51.5\%\ poor\ solution\ consentrate$

5- Find enthalpy for outlet point of consider # 4

$T_4 = T_c = 30\ °C\ , \rightarrow x_4 = 0 \rightarrow$ h chart $h_4 = 129\ \frac{kJ}{kg}$

6- Find enthalpy for the outlet point of the evaporator # 6

$T_6 = 6\ °C\ , \rightarrow h_6 = 2487.6\ \frac{kJ}{kg}$

7- calculate the mass flow rate of the refrigerant:

- $Q_t = \dot{m}.(h_6 - h_5) \rightarrow \dot{m}_3 = \frac{\dot{Q}}{h_6 - h_5} = \frac{555}{2487.6-129}$

Where $h_5 = h_4$

$\dot{m}_6 = \dot{m}_5 = \dot{m}_4 = \dot{m}_3 = 0.235$ kg/sec

8- calculate the mass flow rate of the poor solution at point # 7 :

$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_{\acute{2}} = \dot{m}_3 + \dot{m}_7 ------ (1)$

From the mass balance equation of LiBr

$x_{\acute{2}}\ .\dot{m}_{\acute{2}} = x_3\ .\dot{m}_3 + x_7\ .\dot{m}_7\ ,\ x_3 = 0 ------ (2)$

$\rightarrow x_{\acute{2}}\ .(\dot{m}_3 + \dot{m}_7) = x_7\ .\dot{m}_7$

$\rightarrow \dot{m}_7 = \frac{x_{\acute{2}}\ .\dot{m}_3}{x_7 - x_{\acute{2}}} = \frac{0{,}515\ .0{,}235}{0{,}65-0{,}515} = 0{,}896$ kg/sec

9- calculate the mass flow rate of the rich solution at point # $\acute{2}$:

$\dot{m}_{\acute{2}} = \dot{m}_3 + \dot{m}_7 = 0{,}235 + 0{,}896 = 1.131$ kg/sec

10- calculate circulation factor:

Cf = $\frac{\dot{m}_{\acute{2}}}{\dot{m}_3} = \frac{1{,}131}{0{,}235} = 4{,}812$

That means if we want to generate 1 kg of refrigerant, we have to pump 4,812 kg of solution,
Increasing desorbed temperature causes an increasing mass flow rate of refrigerant then Q_E & Cf

Cf = $\frac{x_7}{x_7 - x_{\acute{2}}} = \frac{0{,}65}{0{,}65-0{,}515} = 4{,}8148$

11-find desorbed outlet enthalpy # 3:

at $T_4 = 85\ °C\ \ vapor\ ,$

super-heated vapor chart: h_3 $_{=2642{,}71}$ kJ/ kg

12- find the desorber outlet enthalpy of the liquid solution $\#\,7$:

$T_7 \;= 85\,°C \;\;,\; x_7 \;= 65\;\% \rightarrow (h-x)\; diagram\; of\; LiBr \;\rightarrow h_7 \;= 225$ kJ/ kg

13- find a secondary heat exchanger outlet enthalpy $\#\,7'$:

$T_{\acute{7}} = 60\,°C$ (assuming), $x_{\acute{7}} = 0{,}65 \;\rightarrow (h-x)chart\;\; h_{\acute{7}} \;= 180 \;\;\; kJ/\,kg$

14- find absorber outlet enthalpy $\#\,1$:

$T_1 = 30\,°C \;\;,\;\; x = 0{,}515 \;\;\rightarrow (h-x)chart\;\; h_1 \;= 60 \;\;\; kJ/\,kg$

15- find pump outlet enthalpy $\#\,2$:

$\dot{W}_{pump} = \dot{E}_{mech} + \dot{E}_{loss} = \dot{m}.v\,.(\,p_c\text{-}\,p_E\,) + \dot{m}.(\,u_2 - u_{1)}$

$\eta_{pump} = \frac{\dot{E}_{mech}}{\dot{W}_{pump}}$ = 0,7 (assuming)

$\dot{E}_{mech} \;= 1{,}131\;.10^{-3}$(4,246 -0,9432) = 3,735. $10^{-3}[KW]$

$\dot{W}_{pump} = 5{,}33\,.\,10^{-3}[KW]$

$\dot{W}_{pump} = \dot{m}_{1\,(}\,h_2 - h_{1)} \rightarrow h_2 = h_1 + \frac{\dot{W}_{pump}}{\dot{m}_1}$

$h_2 = \;\;\; 60{,}0047 \;\;\; kJ/\,kg \;\;\;\approx h_1$

16- $h_{\grave{2}} =$

$\dot{Q} = \dot{m}_{7\,(}\,h_7 - h_{\acute{7}}\;) = \dot{m}_{2\,(}\,h_{\grave{2}} \;- h_2\;) \rightarrow h_{\grave{2}} = h_2 + \frac{\dot{m}_7}{\dot{m}_2}(h_7 - h_{\acute{7}}\;)$

$h_{\grave{2}} = 60 + \frac{0{,}896}{1{,}131}(225 - \;180\,)$= 95,65 $\;kJ/\,kg$

17- $T_{\grave{2}} \;=??$

$x_{\grave{2}} \;= 51{,}5\;\%\;\;,\; h_{\grave{2}} \;= 95{,}65 \;\rightarrow (h-x) \rightarrow T_{\grave{2}} \;= 45\,°C$

18- calculate the thermal capacity of the desorbed:

$\sum \dot{E}_{in} \;= \sum \dot{E}_{out}$

$\dot{Q}_D + \dot{m}_{\grave{2}}\,.\,h_{\grave{2}} \;= \dot{m}_3\,.\,h_3 \;+\; \dot{m}_7\,.\,h_7 \;\;\rightarrow$

$\dot{Q}_D \;= 0{,}235\;.2642{,}71 + 0{,}896\;.225 - 1{,}131\;.95{,}65 \;\;\;\rightarrow\; \dot{Q}_D \;= 714\;{,}45\;[KW]$

19- thermal capacity of absorber:

$\dot{Q}_A + \dot{m}_2.\; h_2 \;= \dot{m}_6\,.\,h_6 \;+\; \dot{m}_{\acute{7}}\,.\,h_{\acute{7}} \;\;\rightarrow \dot{Q}_A \;\;= 0{,}235\;.2487{,}6 + 0{,}896\;.180 - 1{,}131\;.60 \;\;\rightarrow$

$\dot{Q}_A \;\;= 678\;[KW]$

20- calculate the thermal capacity of the condenser:

$\dot{Q}_C \;\;= \;\dot{m}_3\,.\,h_3 \;+\; \dot{m}_4\,.\,h_4$

$\dot{Q}_C$ $\;=$ 0,235 (2642,71 -129)

$\dot{Q}_C \;\;= 590{,}72\;[KW]$

21- Energy Balance Equation for cycle:

$\sum \dot{E}_{in} \;\;= \sum \dot{E}_{out}$

$\dot{Q}_E \;+ \dot{Q}_D \;\;+ \dot{W}_{pump} \overset{?}{\Leftrightarrow} \dot{Q}_A \;\;+\; \dot{Q}_C$

555 +714,45 +5,33. $10^{-3} \overset{?}{\Leftrightarrow}$ 678 +590,72

$1269{,}45 \;\;\overset{?}{\Leftrightarrow}\;\; 1268{,}72$

Not the difference is quite small, that is acceptable.

22- The coefficient of performance $[COP]$:

$COP = \frac{\dot{Q}_E}{\dot{Q}_D \;+\dot{W}_{pump}} = \frac{\dot{Q}_E}{\dot{Q}_D \;+0} = 0{,}7768$

23-calculate mass flow rate of cooling water for A&C:

$\dot{Q}_A \;+ \dot{Q}_C \;\;= \dot{m}_{10}\;.C_{p10}\,.(\,T_{10} \;- T_9\;)$

$\dot{m}_{10} \;\;= \frac{678+591}{4{,}18(25-20)}:(25-20)\,assuming \rightarrow \dot{m}_{10} \;\;= 60{,}71$ kg/sec

24- $h_{10} \;\;= \;\; C_{p10}\,.T_{10}$ =4,18 (25+273) = 1245,64 kJ/kg

25 - $h_9 \;\;= \;\; C_{p9}\,.T_9$ =4,18 (20+273) = 1224,74 kJ/kg

Table 6. The results of the absorption cycle

#	Temperature [°C]	Concentration %	Enthalpy [Kj/kg]	Mass Flow [kg/sec]
1	30	51.5	60	1.131
2	30	51.5	60	1.131
2'	45	51.5	95.65	1.131
3	85	pure	2642.71	0.235
4	30	pure	129	0.235
5	30	pure	129	0.235
6	6	pure	2487.6	0.235
7	85	56	225	0.896
7'	60	65	180	0.896
8	60	65	180	0.896

Table 7. Fix data for selected collectors in the innovation building

Collector data	A_c [m^2]	$\tau\alpha$	U_L [w/m^{2}K]	T_i [C]	T_a [°C]	F_R
	2.24	0.8645	4	40	10	0.8

By the using of the data from Table 7 and the below equation, we defined the following results in Table 8.

$$q_u = A_C.F_R.[I_t.\tau\alpha - U_L.(t_i - t_a)]$$

Where:
q_u = useful energy delivered by the collector, W
A_c = total aperture collector area, m^2
I_t = irradiance, total (direct plus diffuse) solar energy incident on the upper surface of sloping collector structure, W/m^2
F_R = correction factor, or collector heat removal efficiency factor, having a value less than 1.0
U_L = overall heat loss coefficient, W/(m^2·K)
τ = transmittance (fraction of incoming solar radiation that reaches absorbing surface), dimensionless
α = absorptance (fraction of solar energy reaching surface that is absorbed), dimensionless
t_i = temperature of fluid entering collector, °C
t_a = atmospheric temperature, °C

Conclusion

In the case of the constant flow rate of water collectors, we must use 678 collectors with the electrical heater to keep the outlet water temperature at 90°C. In the case of the variable flow rate of water collectors, we must use only 603 collectors without any additional equipment's. The negative values q_u in Table 8 mean that the collectors need more energy to overcome the thermal losses in the collectors. The collectors with more collectors require more energy to maintain the same flow rate. This energy loss can be reduced by using a better collector with more efficient heat transfer. Additionally, the collectors should be strategically located to reduce energy losses due to heat loss. If the Generator temperature falls under 85°C the cooling capacity of the evaporator will decrease. It could be expanded in the study to calculate the surface area of heat exchangers (Condenser, Evaporator), and pipe sizes. The surface area of the heat exchangers can be calculated by using the desired temperature difference and the pressure drop across the exchangers. The pipe size should be chosen based on the desired flow rate and the pressure drop across the pipe. The flow rate and pressure drop can be estimated using the pressure drop equation

Scientific Ethics Declaration

The authors declare that the scientific ethical and legal responsibility of this article published in EPSTEM journal belongs to the authors.

Table 8. Useful energy for 680 collectors in the innovation building

							Average solar irradiance during the day	Useful energy delivered by one collector	Mass flow rate to get 90C of hot water			Outlet water Temperature	Heater to keep To = 90C	Useful Energy from 680 collectors	Collector efficiency
Hour	**May**	**June**	**July**	**August**	**September**	**October**	**I Ayr [w/m2]**	**qu [w]**	**m [kg/s]**	**Ti [C]**	**Cp [w/kg.K]**	**To [C]**	**Heater [w]**	**Q [w]**	**η**
5.00	83.70	98.70	55.60	5.10	0.00	0.00	40.52	-152.27	0.01	45.00	4180.00	39.21	1336.46	-103545.12	-1.68
6.00	300.00	309.70	265.90	198.70	117.90	41.50	205.62	103.50				48.93	1080.68	70378.67	0.22
7.00	509.80	514.20	475.20	418.30	342.00	253.90	418.90	433.91				61.49	750.27	295060.96	0.46
8.00	693.70	694.20	660.90	613.40	541.50	449.90	608.93	728.31				72.68	455.87	495250.65	0.53
9.00	838.90	837.80	810.20	769.40	698.10	601.80	759.37	961.36				81.53	222.82	653723.91	0.57
10.00	935.60	935.20	913.20	875.90	800.80	698.00	859.78	1116.92				87.44	67.26	759507.36	0.58
11.00	977.30	979.90	962.90	925.40	842.20	731.50	903.20	1184.18				90.00	0.00	805244.43	0.59
12.00	961.10	969.00	956.10	914.80	819.70	700.00	886.78	1158.75				89.03	25.43	787950.37	0.58
13.00	888.10	903.10	893.30	844.80	734.60	605.60	811.58	1042.25				84.61	141.93	708731.30	0.57
14.00	763.40	786.80	778.50	719.90	593.00	455.10	682.78	842.72				77.02	341.47	573047.57	0.55
15.00	595.20	627.70	619.60	548.80	404.80	260.20	509.38	574.09				66.82	610.09	390380.19	0.50
16.00	395.10	436.60	427.10	343.30	185.10	47.10	305.72	258.57				54.83	925.61	175828.53	0.38
17.00	177.90	226.70	214.50	120.60	0.50	0.00	123.37	-23.92				44.09	1208.11	-16267.19	-0.09

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References

Alefeld, G., & Ziegler, F. (1985). Advanced heat pump and air-conditioning cycles for the working pair H2O/LiBr: domestic and commercial applications. *ASHRAE transactions*, *91*(2B), 2062-2071.

Aumann, D. J. (1996). Selecting chillers in the 90s: accounting for hidden costs. *Heating, Piping, Air Conditioning*, *68*(3), 65-67.

Ashrae. (1998). ASHRAE Handbook, refrigeration, ashrae inc., atlanta, ga. ashrae, ashrae/iesna Standard 90.1-1999, Energy Standard for Buildings Except for New Low-Rise Residential Buildings, Atlanta, 1999.

Gommed, K., & Grossman, G. (1990). *Performance analysis of staged absorption heat pumps: water-lithium bromide systems*. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

Grossman, G. (1982). Adiabatic absorption and desorption for improvement of temperature-boosting absorption heat pumps. *ASHRAE Trans.;(United States)*, *8*(2).

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