

Feasibility Study & Design of a Seawater Air-Conditioning System for USP Tuvalu Campus



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Summary

The cost of electricity is very high in most of the Pacific Island Countries. Also, being tropical countries, they require a considerable amount of energy to condition hot and humid climate conditions to the required human comfort conditions. Increase in the number of buildings with development makes it necessary to find new air conditioning techniques.

Seawater air-conditioning (SWAC) is a relatively new concept that utilizes water from the deep ocean where the temperature is considerably lower (typically 4-7°C) to provide cooling to buildings. Moreover, deep seawater has vast minerals and resources that can be used for aquaculture, desalination, and energy production.

This project focuses on feasibility study and design of a seawater air-conditioning system for USP Tuvalu campus.

The first phase of the project included a feasibility check to see if the necessary resources are available, which included the measurement of seawater temperature at different depths, estimation of the total air-conditioning load for the USP Tuvalu campus using CAMEL software, and sizing the major components of the system.

The second phase of the project included sizing the chilled water loop components for lab testing of a scaled-down model of the campus, the calculation of ductwork and the air handling unit, dimensional analysis of model and construction and testing of the model and its cooling, and finally a study of the economic viability of seawater air conditioning of the USP Tuvalu Campus.

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Nomenclature

AHU – Air Handling Unit

SWAC – Sea Water Air-Conditioning

USP – The University of the South Pacific

TOR – Ton of Refrigeration

kW – kiloWatt

HVAC – Heating, Ventilation and Air-Conditioning

PLC – Programmable Logic Controller

TRNSYS – Transient System Simulation Tool

BMS – Building Management System

DCS – District Cooling System

AC – Air-Conditioning

ΔT – Change in Temperature

S/A – Supply Air

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1.0 Introduction

There are 328 million cubic miles of seawater covering two thirds of the earth's surface, considered a delicate yet fundamental element of the earth's ecosystem and are substantially vital to sustaining life on the planet [1].

Connection between ocean and people has been one of the essential factors in human civilization. Among many uses, the oceans provide food to human life, are a source of essential minerals, absorb greenhouse gases, generate oxygen, minimize climate change, provide means to transport, and influence temperature and weather patterns. To date, less than five percent of the oceans have been explored; however, continuous development in tools and techniques has opened new windows to exploring the oceans to a far greater extent, enabling discovery of new sources for energy, identification of novel life forms, better understanding of humanity's present and past, and most importantly understanding the impact of human activities on the oceans [1].

There are five oceans in the world, out of which Pacific Ocean is considered to be the largest and the deepest, containing fifty nine million square miles of water [2]. Located within the Pacific Ocean between latitudes five degrees and eleven degrees south and longitudes hundred seventy six degrees and hundred eighty degrees east, Tuvalu, formerly known as Ellice Islands, consists of nine atolls totaling a land area of ten square miles [3]. Being one of the smallest countries in the world, Tuvalu has a population of slightly over eleven thousand, according to the most recent household survey done by the Secretariat of the Pacific Community [3].

Tuvalu is facing the threat of a dangerous rise in sea level due to climate change. As predicted by the National Research Council, the global sea level would rise by fifty five inches by the end of the year 2100 which would greatly cause damage to the country's coral roots wiping out the entire country [4]. This makes it absolutely necessary for people of Tuvalu to engage in utilization of green and renewable energy sources. Engagement to green sources by such a small nation would not only help in minimizing damage to the ozone layer but would also set example for other nations around the world. Surrounded by the Ocean, Tuvalu has the possibility to explore, reach and extract minerals and resources for a variety of uses.

One of the unique and essential properties of oceans is that as the depth increases the water gets cooler. According to Arnold Gordon, the temperature of water at a depth of one thousand meters varies typically from one to five degrees Celsius [5].

The University of the South Pacific can play its part in helping minimize the damages caused to the ozone layer by utilizing renewable energy sources such as solar, waves and

wind. One of the ways to support this initiative is to commence the use of deep sea cold water to air condition the Campuses of the University.

A seawater air conditioning (SWAC) system utilizes a water pump to suck water from deep sea for cooling. Seawater is pumped inland and circulated through a heat exchanger. The heat exchanger has another loop of fresh water circulating through it as well. Upon arriving to the heat exchanger the seawater gains heat from the fresh water and is discharged to the ocean at a lesser depth. Upon giving off heat to the seawater, the fresh water becomes cool and is passed to the Air Handling Units (AHU) installed in the buildings. The AHU contains a fan coil unit which operates similar to the internal unit of a conventional air conditioning unit. The fan draws air past the coils and fins of the unit where heat from the air is transferred to the water inside the coil, in turn making the air cool. This air is then supplied to the building and the cycle continues.

2.0 Objectives

- To check the feasibility of implementing a SWAC system in Tuvalu.
- Estimation of the total cooling load for the USP Tuvalu campus using CAMEL software.
- Determine the sea depth at which sea water is cold enough for sufficient cooling applications.
- To choose a suitable SWAC system for the cooling of the above-mentioned campus.
- Use of dimensional analysis to scale down the necessary components for modelling.
- To construct a scaled-down model of the campus and construct it in the Mechanical Engineering Laboratory of USP.
- To estimate the sizing requirements for all the components of the SWAC system.

3.0 Literature Review

Some of the highly relevant papers published in this area are discussed here.

3.1 Technical and Economic Assessment of Seawater Air Conditioning in Hotels

This article discusses the technical and economic aspects of seawater air conditioning for large buildings such as hotels located adjacent to the ocean having cooling loads greater than 1,000 TOR. Comparison was made between three different cooling systems which include a vapor compression chiller system, a seawater air conditioning system and a seawater air conditioning system integrated with a chiller system. Thorough research was done on all of the systems and a methodology to compare all the systems was generated. Firstly, the cooling load for a space was calculated, after which appropriate mathematical expressions were used to calculate the number of components that would be required in each of the systems. Power consumption for each of the systems was tabulated and comparisons made. Finally, an economic assessment was done for a direct seawater air conditioning system and a seawater air conditioning system with chillers. The payback period for both the systems was calculated and comparison was made. The paper stated that seawater air conditioning is one of the emerging techniques which provides cooling effect comparable to conventional cooling methods and reduces use of fossil fuel. It also states that conventional systems would consume 3,500 kW of energy whereas hybrid systems would use 1,800 kW and direct seawater air conditioning systems would only use 400 kW energy for the same cooling load. For the economic assessment, the paper stated that the seawater air conditioning systems would require an initial capital investment of USD 245 million whereas hybrid system would require USD 76 million and a conventional system would require only USD 20 million. The paper concluded that seawater air conditioning enables reductions in power consumption and greenhouse gas emissions and that direct systems can be used for buildings located on the beaches, whereas hybrid systems can be installed in buildings on the beaches as well as buildings near the coast where sea water can be pumped [6].

3.2 Sea Water Air Conditioning [SWAC]; A Cost Effective Alternative

The paper reports the results obtained from economical and technical assessment of the use of seawater air conditioning (SWAC) systems over conventional vapor compression refrigeration systems. The paper states that the ocean contains an enormous amount of natural resources which can be utilized for agriculture, aquaculture, desalination, and energy production and cooling. The paper states that deep seawater, being cold, makes it economically viable to be used in air

conditioning through a SWAC system. The paper presents a case study on a resort called “Sahl-Hasheesh”, located 18 km south of Hurghada in Egypt. A HVAC Load explorer program was used to calculate the cooling load for the hotel. Site maps along with Bathymetry charts were then used to design a pipeline schematic for seawater suction. Appropriate mathematical expressions were then used to size all the necessary components for the system. Numerical programs were then used to determine the optimum seawater piping network considering the capital expenditure, electricity usage for suction, and head loss. An economic analysis using simple payback method and net present value technique was also done to calculate the payback period for the SWAC system. Finally, an economic assessment was also done on a hybrid system which would include a SWAC system integrated with auxiliary chillers. The auxiliary chillers would be controlled by a programmable logic controller (PLC) and would only operate if the SWAC system fails to provide adequate cooling. The paper concluded that a SWAC system is feasible for cooling loads exceeding 5,000 TOR, hence appropriate for “Sahl-Hasheesh” [7].

3.3 Study of Water Cooling Schemes for Commercial Air-Conditioning Applications

This paper reviews the performance of an evaporative air conditioning system operating under the weather conditions of Hong Kong and the economic impact of modifying an existing evaporative air conditioning unit to a fresh water cooling tower system in a commercial building. The paper also sheds light on implementation of seawater district cooling schemes. Firstly, the necessary data for one year for an existing building were used to simulate operation of an evaporative air conditioning system using TRNSYS program. Necessary data from a Building Management System (BMS), which records operation data for air conditioning systems, were used to simulate electricity consumption for all the plants. All of this data were then used with appropriate mathematical expressions to calculate the life cycle cost and the payback period. Finally, the paper outlines the necessary process path for implementation of a district cooling system (DCS) which includes grouping nearby buildings together, calculating the total load for individual districts through the use of cooling load simulation tools such as CARRIER software, and using appropriate mathematical expressions to size necessary components. The paper also states that air conditioning in commercial buildings consumes 50-60% of the total electrical load, hence it is absolutely necessary to consider application of water-cooled technologies. The paper concludes that modification of an existing air-cooled air conditioning system to water-cooled AC system is technically as well as economically practical. [8].

4.0 Methodology

Figure 1 shows the flow chart of the methodology undertaken to complete this project. The feasibility study, load calculation and system sizing were completed in the first phase while system scaling, seawater temperature measurement, model construction, testing, and economic analysis were completed in the second phase, and details of each are provided in sections 6.0, 7.0, 8.0, and 9.0 of this report.

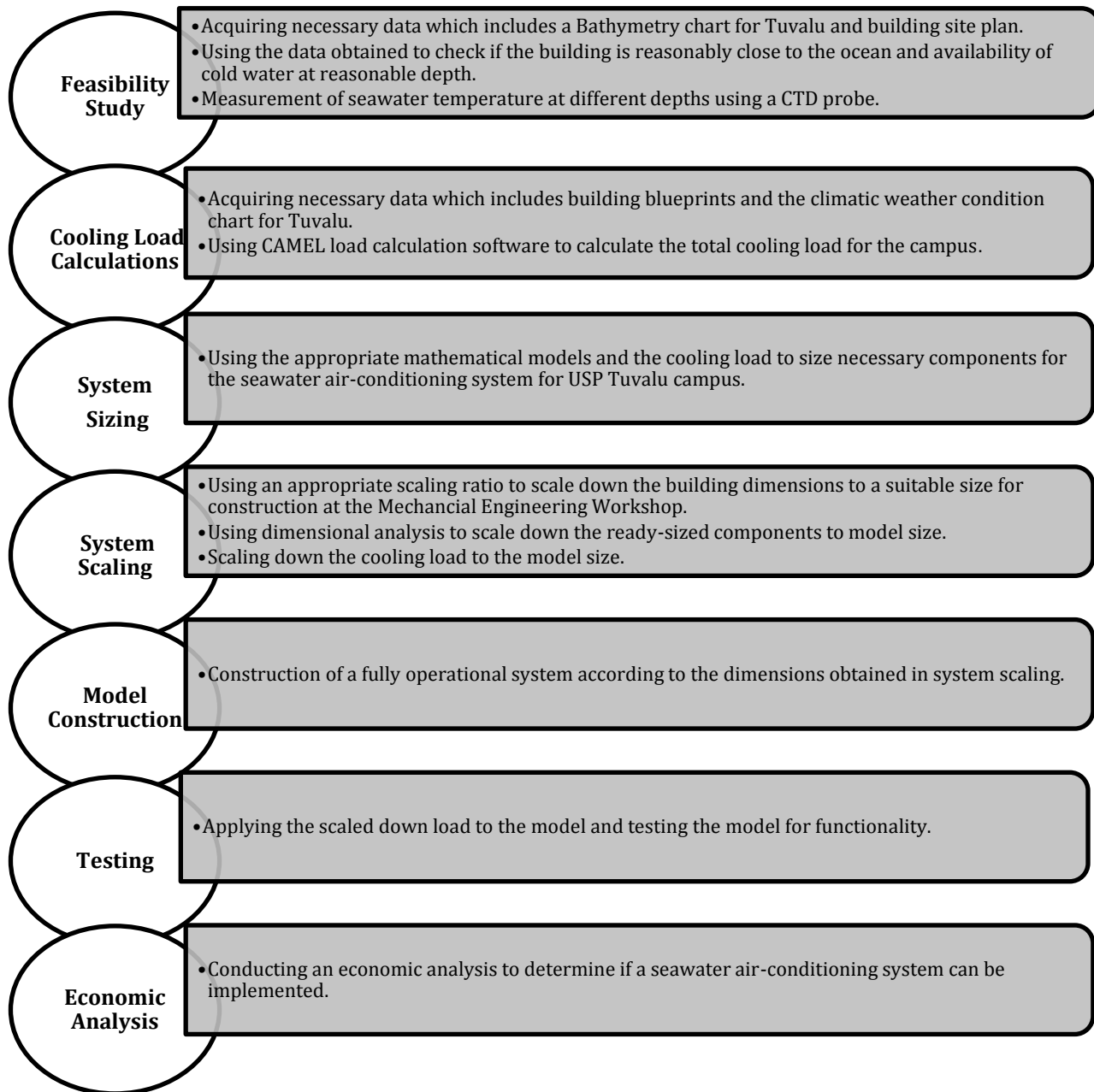


Figure 1: Steps undertaken to complete this project.

5.0 Cooling Load Estimation and System Sizing

This chapter presents the first phase of the project. The work in the first phase included thorough research on SWAC systems and their components. After research, the next phase of the project was data collection. Necessary project data such as the Bathymetry chart for Tuvalu, building blue-prints, and weather data for Tuvalu were collected. The seawater temperatures at different depths were measured using a CTD probe. After successful data collection, the load calculation software, CAMEL, was used to calculate the total cooling load for USP Tuvalu Campus. Finally, appropriate mathematical expressions along with the calculated cooling load were used to size the major components of a SWAC system for USP Tuvalu Campus.

5.1 Load Calculation using CAMEL

For calculation of the total cooling load for the campus the following steps were followed:

5.1.1 CAMEL Software

The CAMEL software allows cooling load calculation when all the geometric details and year-round ambient conditions are fed in to it.

5.1.2 Data Collection

Data collection involved obtaining the following:

- **Building Blue-print** – These contained necessary details and dimensions for the building which were to be fed in to the program. Later, the actual measurements of the dimensions were performed during the visit to Tuvalu.
- **Google map** – To identify the building's north direction since it was not available on the floor plan.
- **Weather chart for Tuvalu** – To input climatic design conditions for Tuvalu.

5.1.3 CAMEL Project Screens

CAMEL Program contains two series of tabs which require data input. The Project tab has a total of six screens in it which require general project information and information which could be cross-referenced while inputting data for individual zones in the AHU screen tabs. All the necessary details in the project screen were fed in according to the data obtained. Figure 2 shows details of one of the tab's in-project screens.

Project Title:

Comments (empty):

Location:

Title: Tuvalu

	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC
3pm *CDB	32.2	32	32.2	32.2	32	32.1	31.7	31.6	31.7	31.7	32.1	32.2
3pm *CWB	27.9	27.7	27.9	28.1	28.1	28.1	27.7	27.7	27.4	27.6	27.7	27.9
Daily Range	3.5	3.8	3.9	3.9	3.8	3.4	3.4	3.4	3.5	4	4	3.9

Latitude: ☒ South ☐ North

Elevation (m):

Winter Design *CDB:

Winter Design %RH: Winter Warm Up %:

Min. leaving coil temp °C:

Ambient Design condition for Desiccant HCU's

Design WB: Design %RH: Defaults:

Default Plant Operating Time

Start:

Finish:

Building Rotation

North Wall

PLAN

Shading Effectiveness

☐ Shading Effectiveness

☐ Equivalent Overhang

☐ Include Adjacent Shading

☐ Disable Load Calcs

Figure 2: Project tab for user-defined general project details.

5.1.4 CAMEL AHU Screens

Since the building contains a total of eight rooms which require cooling, each air handling unit was dedicated to one room so that individual rooms would have their own control system, since some rooms would require cooling for only certain hours whereas rooms like the satellite room and the computer lab may require longer cooling hours. Details for each room were filled in the individual AHU tab and necessary information from the project screen tabs were cross referenced. Figure 3 shows details of all the AHU tabs that required information input.

AHU No	AHU Title	AHU	Coil	Zone No	Room Title	Zone Room	External	Partitions	Internal
1	AHU 1			1	Classroom				
2	AHU 2			1	Admin office				
3	AHU 3			1	Account office				
4	AHU 4			1	Library				
5	AHU 5			1	Computer lab				
6	AHU 6			1	Satellite room				
7	AHU 7			1	Director's room				
8	AHU 8			1	Courtyard				

Figure 3: General AHU tab showing all the sections which required information input.

5.1.5 View Screen

The view screen shows details of the results from the program. After validation of all the details, the program was run and the total cooling load for the building was estimated. The primary plant results tab, given below as Figure 4, show the total cooling load for the building. Details for individual AHU units were also available on the view screens which include the breakdown of loads – for example, the total sensible and latent heat that will be generated in each room along with details of load due to people giving-off heat, external heat, and the heat from equipment and lighting. Graphs for the entire plant were also available on the view screen which showed the grand total *heat versus time* plot.

PRIMARY PLANT RESULTS (Excludes unitary plant)											
CHILLER PLANT											
Operating hours 1am to midn											
AIR HANDLING LOAD				3pm Dec=		114					
PROVISION FOR CHILLER PUMP(S) (1%)						=		1.14			
PROVISION FOR CHILLER PIPE GAINS (2%)						=		2.28			
CHILLER GRAND TOTAL HEAT (kW)						=		117			
CIRCUITS											
----- Cooling -----> <- Heating -----											
No	Type	Off	Description	GTH kW	GTSens kW	Peak Time	Amb DB	Room avWB	GTH kW	Amb WB	Room avDB
1	CWFC	1	Chiller 1	114	55.4	3pm Dec	32.2				
Legend: CWL - Condenser Water Loop											
VRF - Variable Refrigerant Flow											
WLHP - Water Loop Heat Pump											
CWFC - Chilled/Hot Water Fan Coil											

Figure 4: Primary plant results tab.

5.1.6 Result

The total cooling load was estimated to be 117 kW, as can be seen from the Fig. 4.

5.2 System Sizing Calculations

Once the total cooling load of the USP Tuvalu campus was calculated, a safety factor of 12% was applied, which was rounded off to 132 kW. All the components sized hereafter take into consideration this new load. A simple SWAC system, shown in Figure 45, having a seawater pump, a heat exchanger, and a chilled water pump was selected for the campus.

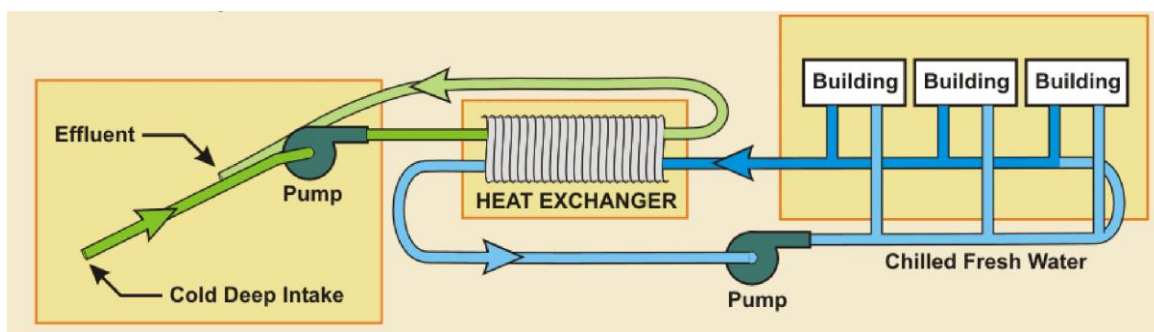


Figure 5: Layout of the recommended SWAC system

5.2.1 Energy balance

There were two energy balances that were applied. Firstly an energy balance was done between the chilled water supply and the total cooling load of the building. Then an energy balance was done between the chilled water supply and seawater supply in the heat exchanger. The purpose of doing these energy balances was to calculate the mass flow rate of each of the supply pipelines. Since the heat transfer equations aside from the mass flow rate variable contained another variable, which was the temperature difference before and after heat absorption, different scenarios were considered where the temperature difference was altered and its effect on mass flow rate was seen.

Table 1: Effect of chilled water temperature difference on the required mass flow rate.

C.W inlet (°C)	C.W outlet (°C)	ΔT	mass flow (kg/s)	volume flow (m ³ /hr)
16	14	2	15.76307619	56.74707428
16	13	3	10.50871746	37.83138285
16	12	4	7.881538094	28.37353714
16	11	5	6.305230475	22.69882971
16	10	6	5.254358729	18.91569143
16	9	7	4.503736054	16.21344979

In Table 1, the temperature of chilled water entering the heat exchanger is the same for all scenarios since it is assumed that the chilled water

temperature would be at 16°C while leaving the building (re-entering the heat exchanger).

Table 2: Effect of cold seawater temperature difference on the required mass flow rate.

S.W inlet (°C)	S.W outlet (°C)	ΔT	mass flow (kg/s)	volume flow (m ³ /hr)
7	8	1	34.28571429	120.4181185
7	9	2	17.14285714	60.20905923
7	10	3	11.42857143	40.13937282
7	11	4	8.571428571	30.10452962
7	12	5	6.857142857	24.08362369
7	13	6	5.714285714	20.06968641

In Table 2 above, the temperature of seawater entering the heat exchanger is deemed constant at 7°C since the pumping depth is 800 m where temperature of seawater is around 6°C with an allowance of 1°C for the temperature rise as a result of friction in the pipe and heat gain from the surrounding water. Figure 4 shows the temperature variation of seawater with depth from the CTD measurements that were performed in Tuvalu in December 2017 – about 1.5 km from the shore. The temperature drop is higher compared to that reported in ref. [9].

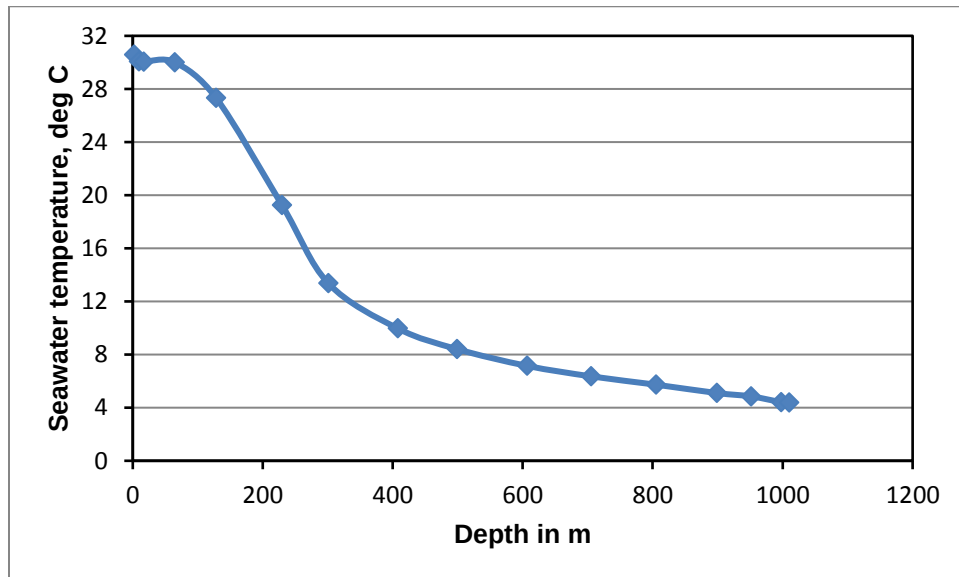


Figure 6: Temperature variation of seawater with depth in Tuvalu.

To choose the optimum scenarios for both loops, a corrugated plate heat exchanger or a shell-and-tube heat exchanger can be used for maximum heat transfer. Option 5 from Table 1 was selected for chilled water loop where the temperature of the water entering the heat exchanger is 16°C and the exiting

temperature is 10°C. For the seawater loop, Option 3 from Table 2 was selected with an inlet temperature of 7°C and an outlet temperature of 10°C. It is assumed that the chilled water would reach equilibrium with the seawater before exiting the heat exchanger. Also the settings selected allow the flow rate to be minimized, therefore low suction pumping power is required.

5.2.2 Suction pipe

The length of the seawater suction pipe required was approximated by the Pythagoras theorem where the depth from which the seawater was to be pumped was 800 m. The horizontal distance from shore to where the depth reached the required depth was found from the bathymetry chart to be 1.5 km. Therefore the total length of suction pipe required was approximately 1.7 km which was taken as 2,000 m, keeping allowance for any possible obstacles from the cold water inlet to the SWAC system.

Since the volume flow rate of the seawater required is calculated in the section 5.2.1, the average velocity of the seawater inside the pipe can be found. Another variable which influences the velocity of the fluid is the cross-sectional area of the pipe. Since the velocity of water inside the pipe was to be maintained below 2.4 m/s to reduce heat gain due to frictional effects, multiple pipe diameters were selected and their influence on velocity of seawater was seen.

Table 3: Effect of internal diameter of pipe on the velocity and pumping power.

Nominal size	Internal diameter (cm)	Velocity (m/s)	Reynolds Number (Re)	Darcy's friction factor (f)	Pressure in pipe (pa)	Head (m)	Velocity head (m)	Total head (m)	Pump power (W)
8	8.062	1.4415727	63507.97311	0.01777	465528.1695	45.417	0.103907	45.521289	5722.67633
10	10.049	0.92784716	50950.47061	0.018618	162103.0931	15.815	0.043045	15.857981	1993.57475
12	11.918	0.65965274	42960.33556	0.019309	71649.86625	6.9902	0.021757	7.0119879	881.507055
13	12.503	0.59936817	40950.27427	0.0195104	56972.79481	5.5583	0.017962	5.5762836	701.018504
14	13.087	0.54706878	39122.89136	0.0197045	45797.04051	4.468	0.014964	4.4829682	563.573141
16	14.956	0.41888154	34233.83787	0.0202885	24190.51776	2.3601	0.008773	2.3688236	297.794967
18	16.826	0.33094846	30429.17385	0.0208248	13776.81806	1.3441	0.005476	1.3495562	169.658488
20	18.695	0.26808419	27387.0703	0.0213219	8330.499864	0.8127	0.003593	0.8163252	102.623733
22	20.565	0.22154641	24896.7313	0.0217867	5284.700639	0.5156	0.002454	0.5180347	65.1243611
24	22.434	0.18616957	22822.55858	0.0222237	3489.436464	0.3404	0.001733	0.3421658	43.0151268

In Table 3, the diameters listed are from a catalogue by Fabco Plastics which is available in **Appendix 1**. It can be seen that all the pipes listed can be used since the velocity is under 2.4 m/s. An interesting phenomenon which can be seen is that as the pipe diameter is increased linearly, the pumping power decreased. Note that the pumping power listed here is due to friction head

and velocity head, but the total pumping power would include the density and the losses in the heat exchanger and will be discussed in the following sections. To keep both the capital cost of the pipe and the pump low, as well as the running cost of the pump i.e. power consumption, the size of pipe chosen had a nominal diameter of 10.

5.2.3 Heat Exchanger

The heat exchanger is chosen before the seawater pump, so the pressure loss in the heat exchanger can be accommodated in the calculations for the pump.

Ideally, in the heat exchanger, the heat gained by the cold fluid is equal to the heat lost by the cold fluid, but this not the case in reality, therefore an average heat transfer was calculated which came to be 132 kW.

Two suppliers of heat exchangers, WEIFENG JIJANTI FACTORY (for shell and tube heat exchanger), and TIANJIN BOTAI HEAT-EXCHANGER EQUIPMENT CO. LTD. (for plate type heat exchanger) were contacted and a detailed list of specifications was sent. The suppliers sent quotations and the details of the heat exchangers with the drawings, that are provided in **Appendix 2** and **Appendix 3**. The maximum pressure loss is 0.1 bar.

5.2.4 Seawater pump

To calculate the seawater pumping power, the determination of the total head required is essential. The total head required is the sum of the friction head, velocity head, density head, elevation head, and the head loss in the heat exchanger. Since the friction head and velocity head have been calculated earlier, only the other three heads need to be calculated. Since the elevation of the heat exchanger above sea level is 3 m, this would be the elevation head. The head loss in the heat exchanger is 0.1 bar which converts to a head of approximately 1m. The head loss in the cold deep sea water pipe can be calculated using the formula:

$$\delta = \frac{\sigma_{CW}D - \int_0^D \sigma(z)dz}{1000 + \sigma_{CW}} \quad (1)$$

where:

- σ_{CW} is the density of seawater at suction depth
- D is the suction depth

- $\int_0^D \sigma(z) dz$ is the integration of the density of seawater variation with respect to depth from sea level=0 to depth D

This results in a density head of about 1m.

Therefore the total head required to overcome by the pump is around 25 m, which results in a pumping power requirement of 6 kW with an assumed efficiency of 60%.

5.2.5 Air Handling unit and duct work calculations

This section contains the necessary information required to size the air handling unit and the ducts that would carry the cold air to the space to be cooled as well as carry the return air back to the AHU.

Firstly, a suitable location was selected to place the AHU outside the building of the USP Tuvalu campus, and then the duct schematic was drawn over the blueprint of the building, ensuring that a duct outlet was given at each of the eight rooms that require cooling. Also, it was ensured that a return duct outlet was present at each of the rooms. Figure 7 shows the designed ductwork for the campus. The blue lines indicate the supply line and the red lines indicate the return lines.

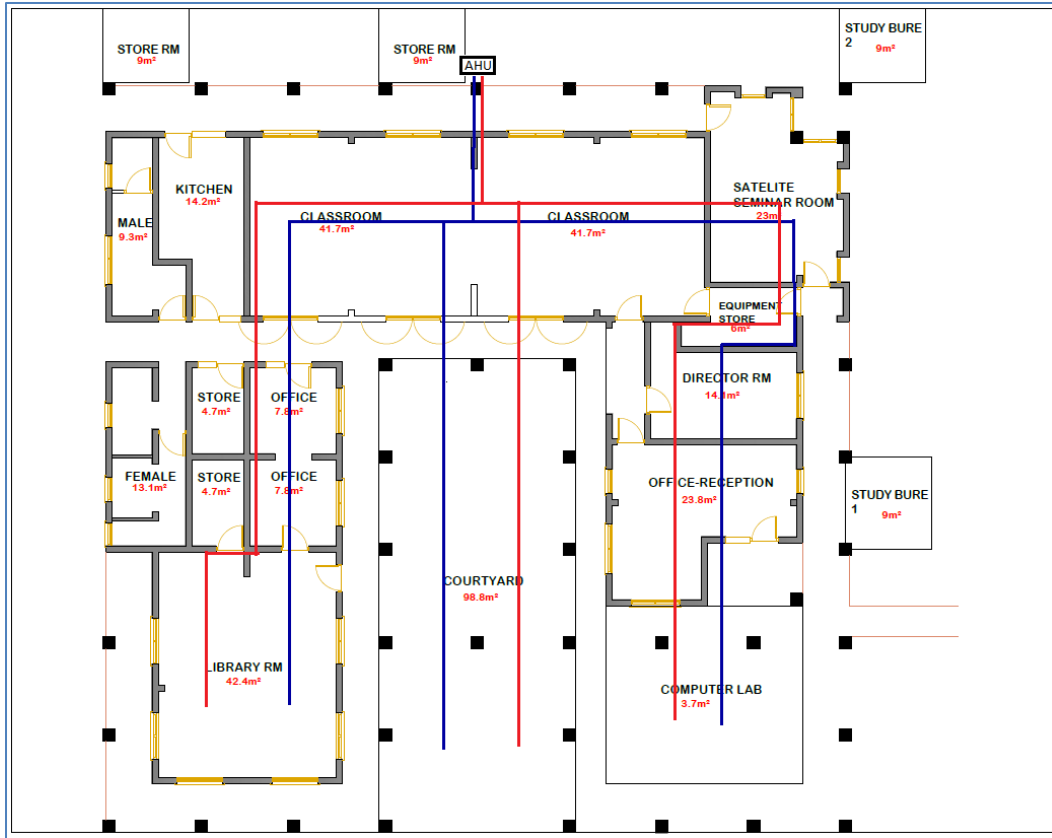


Figure 7: Duct route on the campus plan.

In the design process of the duct lines, it was ensured that minimum bends are used to minimize pressure losses. The AHU is placed in the chosen location so that air can be circulated to both halves of the building simultaneously rather than the air entering each room from one side.

Once the design was finalized, the total length of the duct was determined. The total supply line length was calculated to be 70 m while the total return line length was calculated to be 75 m.

5.2.6 Volume Flow rate of air

The volume flow rate of air required depends on the cooling load per room. This data was acquired from the CAMEL software. Figure 8 shows the total supply air required by the rooms. The duct has to handle both supply air (S/A) and outside air. The total supply air is 3,326 l/s, which converts to approximately 12,000 m³/hr.

AHU SUMMARY - COOLING											
At Time of Peak Grand Total Heat (GTH)											
TITLE	NO. OFF	S/A l/s	OUT/AIR l/s %		GTH kW	GTSH kW	COIL ENT CDB CWB	COIL LVG CDB CWB	PRECON NO.		
AHU 1	1	535	358	67	29.9	13.0	29.2 24.8	9.1 9.3			
AHU 2	1	214	53	24	6.02	3.54	25.3 19.6	11.6 11.2			
AHU 3	1	33	15	46	1.41	0.68	27.2 22.4	10.1 10.0			
AHU 4	1	637	190	30	19.9	11.0	25.7 20.4	11.5 11.2			
AHU 5	1	380	88	23	10.3	6.10	25.0 19.4	11.7 11.3			
AHU 6	1	160	70	43	6.54	3.24	27.0 22.0	10.3 10.1			
AHU 7	1	106	24	23	2.88	1.72	25.1 19.4	11.7 11.3			
AHU 8	1	620	417	67	34.2	15.0	29.2 24.7	9.1 9.3			

Figure 8: Snapshot from CAMEL showing total air supply required per room.

The velocity of the air inside the ducts also has to be maintained at specific levels. For the flow rate mentioned above, the appropriate velocity of air is around 7 m/s [10], the selection table is available in **Appendix 4**. From the flow rate and the recommended velocity the area of the duct was calculated to be 0.476 m², which gave the diameter of the duct to be 0.8 m. This is the size of the duct exiting the AHU. Since the duct is split in two directions, separate flow rate was calculated in each direction and the duct diameter was reduced.

The flow rate required for the right side of the building was 7,000 m³/hr. whereas for the left side of the building it was 5,000 m³/hr. The required diameters of the ducts were 0.6 m and 0.5 m respectively. After the reduction at the T-junction at the exit of the AHU, the duct diameter was kept constant for simplicity in calculations.

5.2.7 Fan power required

The fan power required inside the AHU can approximated using the equation [10]:

$$P_{fan} = \Delta P \times Q \quad (2)$$

where:

- P_{fan} is the fan power in Watts
- ΔP is the pressure loss to be overcome by fan
- Q is the flowrate of air through the pump

Since the flow rate is already determined in the earlier section, the only other requirement is the estimation of the total pressure loss.

The total pressure loss in the duct system can be categorized into two components: major losses and minor losses. The major losses are primarily due to friction, while the minor losses are due to the components installed (e.g. 90° bends and T junctions).

To calculate the friction losses in the duct, the following formula was used [11]:

$$\Delta P_L = f \frac{L}{D} \frac{\rho V^2}{2} \quad (3)$$

where

- ΔP_L is the pressure loss inside the duct
- L is the total length of duct
- f is the Darcy's Friction factor
- D is the internal diameter of the duct
- ρ is the density of air
- V is the velocity of air in the duct

This is the same equation which was used to calculate the pressure loss in the pipes sized in the previous sections. The length of the duct was taken as the full length of supply and return ducts which totalled 145 m.

Since the friction factor is also required by the pressure loss equation, the Reynolds number was calculated to be 324,288.6, considering the kinematic viscosity of air as $15.11 \times 10^{-6} \text{ m}^2/\text{s}$ at an average temperature (inside the ducts) of 20°C. The material which was chosen for duct construction was standard galvanized sheet metal; hence its relative roughness was used to find the friction factor, which came to 0.0162.

Finally, the pressure loss in the pipe was calculated to be 74.6 Pa. The pressure loss is low since the density of the air is much lower compared to the seawater.

To calculate the pressure loss due to minor losses, the following formula was used [11]:

$$\Delta P_L = \frac{\rho V^2}{2} \sum K_r \quad (4)$$

where:

$\sum K_r$ is the loss due to components present

In the design of the duct in Figure 7, there are ten 90° elbows present, four T-junctions, and two contractions in the supply duct and two enlargements in the return duct. The loss coefficients were selected from a standard text book and are presented in **Appendix 5** and **Appendix 6**.

The total K_r value was calculated to be 6.22 which resulted in a pressure loss of 183.63 Pa.

The total of both pressure losses came to be 260 Pa; this, multiplied with the max flow rate inside the ducts of 12,000 m³/hr (which converts to 3.33 m³/s), resulted in a total fan power of 0.9 kW.

Therefore an AHU needs to be selected which can supply 3.33 m³/s of air with a total fan power of 0.9 kW or greater.

5.2.8 Chilled water supply pipe line

The pumping station is located near the shore but the building to be cooled is located about 80 m from the shore therefore the chilled water needs to be pumped from the shore to the AHU near the building. Figure 9 shows the proposed chilled water pipe lines. It was ensured that the pipe lines do not interfere with any of the existing buildings. The blue line indicates the supply of chilled water while the red line indicates the return. With the scale given in the bottom right hand corner it can be estimated that a total of 100 m of supply and 100 m of return pipe line is required. Since the water is cold, to prevent heat gain proper insulation would be needed. A more preferable way is to keep the supply pipe line underground.

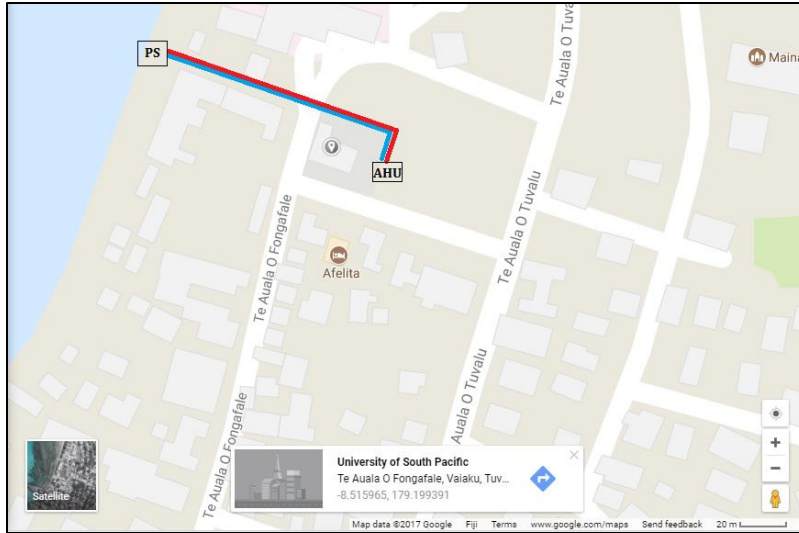


Figure 9: Proposed seawater pipeline.

5.2.9 Chilled water pump

To size the chilled water pump, similar steps to those which were used to calculate the seawater pumping power could be used. The total head required would include the friction head, head loss in the heat exchanger, head loss in the AHU coils, and the velocity head. Table 4 shows the relationship between the pump power and the internal diameter of pipe. The overall efficiency of the pump was taken as 60%.

Table 4: Pump power required for different internal diameters of pipe.

Nominal size	Internal diameter (cm)	Velocity (m/s)	Reynolds Number (Re)	Darcy's friction factor (f)	Pressure in pipe (pa)	Head (m)	Velocity head (m)	Total head (m)	Pump power (W)
2	5.25	2.66995485	140172.6295	0.0167834	113945.8415	11.395	0.356433	11.751017	1160.27378
3	7.729	1.23190129	95213.65051	0.0181747	17842.91334	1.7843	0.075879	1.8601704	183.669796
4	10.24	0.70181494	71865.85008	0.0192942	4640.257107	0.464	0.024627	0.4886529	48.2486893
5	12.82	0.44776122	57402.9879	0.0202615	1584.333188	0.1584	0.010025	0.1684578	16.6332152
6	15.41	0.30989691	47755.11388	0.0211065	657.6852171	0.0658	0.004802	0.0705703	6.96798403

The 3" pipe is chosen for supplying the chilled cold water to the AHU, since it gives a reasonably high velocity head without requiring a very high pumping power.

The head loss in the heat exchanger was calculated in the previous section to be 1 m. The head loss in the AHU coils depends on a lot of parameters which include:

- The number of coils

- The length of coil
- The bend radius
- The step down from the 3" supply pipe to the copper tubes

Therefore, before the determination of the pressure loss of chilled water in the AHU, a suitable AHU needs to be selected. A reputable company, SHANDONG GRAD GROUP CO., LTD was contacted and the details of an AHU that met the air flow rate and the fan power were obtained along with a quotation. The quotation and the specifications of the AHU are provided in **Appendix 7**. Since the details did not include the pressure loss in the coils, the supplier was contacted again and the pressure loss details were requested. The head loss in the cooling coils was a maximum of 4 m (0.4 bars).

Finally, since all the heads were calculated, the total power requirement of the chilled water pump was determined to be 1.5 kW.

6.0 Dimensional Analysis

Before model construction, dimensional analysis is crucial. Dimensional analysis is done to scale all the system components according to the scale to which the model is to be constructed.

For the model construction, only the chilled water loop together with the USP Tuvalu campus model were made.

6.1 Model scale down

A scale down factor of 10 was chosen to reduce the building size from the original to the model i.e. all dimensions of the original building would be divided by 10 and these would be the corresponding dimensions on the model. Since the original building's overall length and width were approximately 24m x 24m, the model building had overall dimensions of 2.4m x 2.4m. The ceiling height was 2.7 m for the original and 0.27 m for the model.

6.2 Cooling Load scale down

The cooling load was scaled down using the CAMEL software. All the dimensions of the original building entered were reduced by a factor of 10 and the cooling load was recalculated. The cooling load for the model came to be 1.17 kW which was one hundredth of the original cooling load.

6.3 Pump selection

The pump for supplying chilled water to the building model was selected off the shelf with the smallest power consumption and the lowest cost. The pump power was 1 hp (750 W). This pump was of higher specifications than what are required for driving the flow in to the building model. Because of the unavailability of the correct-sized pump, this pump was used in the project.

When scaling down from a prototype to a model it is important that the power, discharge, and head coefficients of both the model and prototype remain the same. These parameters can be calculated using the following relations [11]:

$$\text{Power coefficient} = \frac{P}{\rho N^3 D^5}$$

$$\text{Head coefficient} = \frac{g H_m}{N^2 D^2}$$

$$\text{Discharge coefficient} = \frac{Q}{ND^3}$$

For the model, the flow rate was measured manually after installation, which was found to be 8 L/s (2.9 m³/hr). At this flow rate, the head is higher than what is required to drive the flow through the building model. The performance charts of both the pumps are provided in **Appendix 8** and **Appendix 9**. The impeller diameters and the rpm of the motors driving the pumps were specified by the manufacturer in the specifications of the pump.

7.0 Model Construction

After successfully completing the dimensional analysis, the next phase of the project involved construction of the building model and the AHU. Since a seawater pump was not locally available for testing and demonstration, a single loop system for cooling was used; hence the seawater loop was not part of the construction phase. This meant that a single pump would be used to supply water to the air handling units hence the heat exchanger was not constructed. With the advancement in technology of heat exchangers for OTEC applications, the pressure loss in the heat exchanger now requires less than 10% additional pumping power.

7.1 Materials used

A detailed list of all the parts and materials needed for the construction phase was prepared. Local retailers were contacted to obtain quotations of all the parts and materials.

Since 1/10th scale was appropriate for construction, an Excel spreadsheet was developed where dimensions of the building which included details such as length and width of the building were entered and the spreadsheet gave the scaled down values. This minimized the need to regularly refer to the building blueprints for dimensions.

To construct the base of the model, plyboards and 2" x 3" wood were used.

After completion of base, the structure for the building frame was laid. For building frame construction, 25mm x 25mm dressed timber was used since walls were to be of Perspex, which would have collapsed if joined on their own. Figure 10 shows the structure of the building laid out with the use of 25mm x 25mm timber.



Figure 10: The layout of the building frame.

After fastening the layout of the building to the base, the next step was to cut and join the studs for the wall. Dimensions from the spreadsheet were used to make frames of each wall at a time. Figure 11 shows the constructed base of the project model along with the wall studs constructed and ready for the structure.

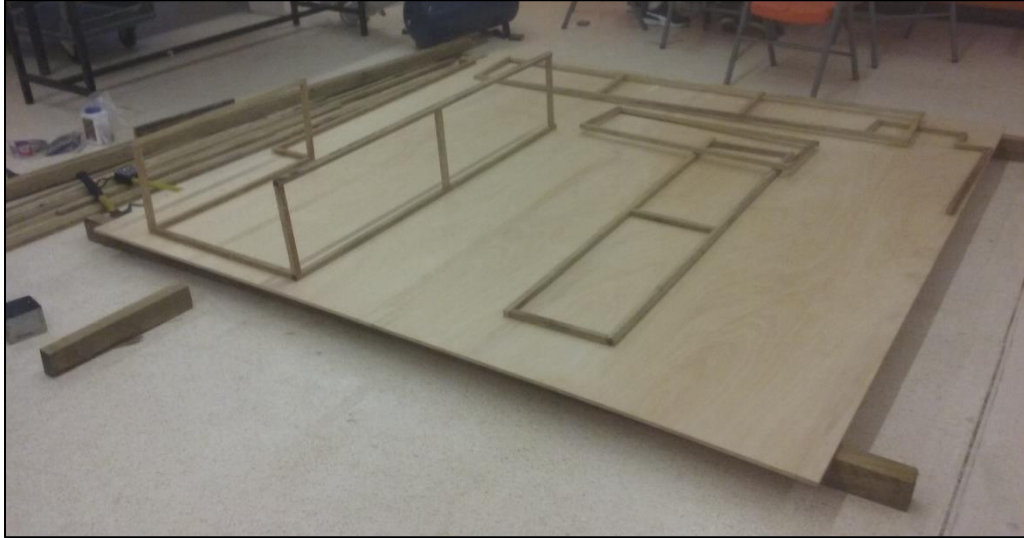


Figure 11: Base of the building model with wall studs attached.

Upon successfully completing the frames for the wall, the next task was to construct the structure for the roof. The roofing for the original campus was configured in a gabled shape with gable angle of 20 degrees, hence the model was also to be constructed as such because the geometry of the roof would have an effect on the overall building volume and cooling requirements. Since a gabled roof has a complex geometry, the roofing frames were constructed in five separate parts and then assembled. Figure 12 shows three of the roofing structures completed and attached to the model.

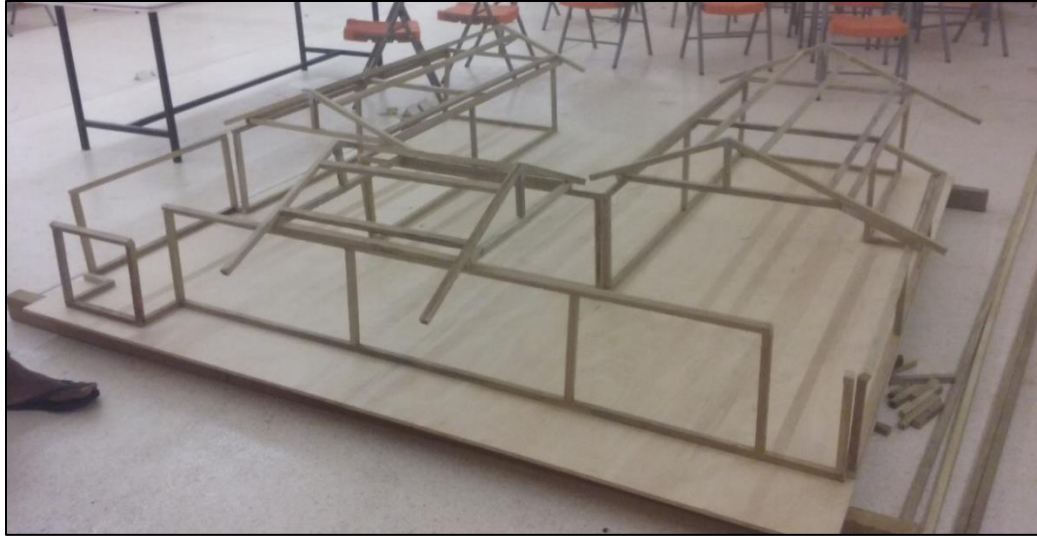


Figure 12: Separate structures with roofings.

Upon completion of the five frames, they were joined to the studs of the wall hence completing the frame-work as shown in Figure 13.



Figure 13: Model with complete frame work.

The next phase of the modeling work involved attaching the roofing irons to the frames. In order to match the measure of heat transmission through the roof, sheet metal was used for roofing. Figure 14 shows installation of the sheet metals to the frame. Sheet metal was cut along the length to avoid joints in the middle section of the model.



Figure 14: Installation of sheet metal.

Upon securing the sheet metal to the frame, ridge caps were bent out of sheet metals and were secured to the roof. V-drains were also bent out of flat irons and were attached to the roof. Figure 15 given below shows how the ridge caps and V-drains were bent out of sheet metals.



Figure 15: Installation of ridge caps and V-drains.

Upon completion of the roof work, the next step was to attach the walls to the stud. Since the windows for the building were made out of glass and the U-values between glass and Perspex were similar, the windows were not constructed separately. Hence all the walls were sealed with Perspex. Figure 16 shows the building model after attachment of Perspex to the wall frames. Screws were used to fasten the Perspex to the frames.

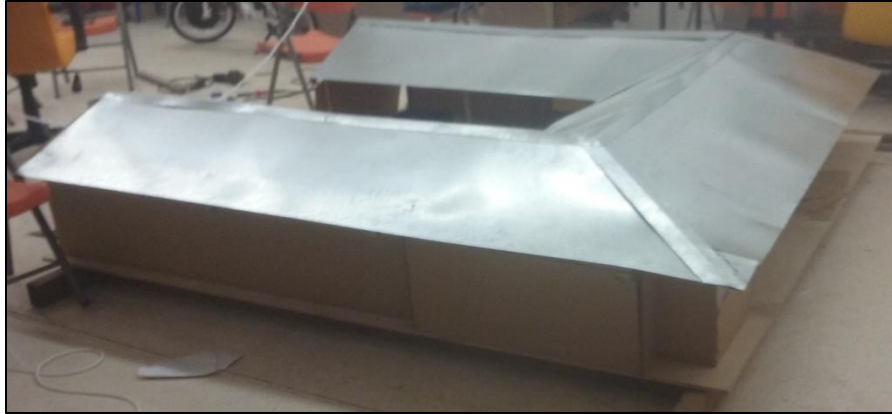


Figure 16: Perspex attached to the frames to seal the model.

Finally, the gaps and edges around the building model were sealed with silicone to avoid leakage of air so that effective cooling takes place.

7.2 AHU Construction

Research on a suitable fan coil unit which would circulate air inside the model as well as be an effective heat transfer medium between the cold water and the air was conducted.

A radial fan coil unit similar to the one used in an internal unit of a split type air conditioning unit was considered. Due to the unavailability of a small size AHU locally, it was decided that a large AHU from a split type unit would be used by cutting it to the required size. An old AHU was obtained from the Properties department. With the help of the Technicians, the AHU was dismantled. Figure 17 shows the dismantled AHU.

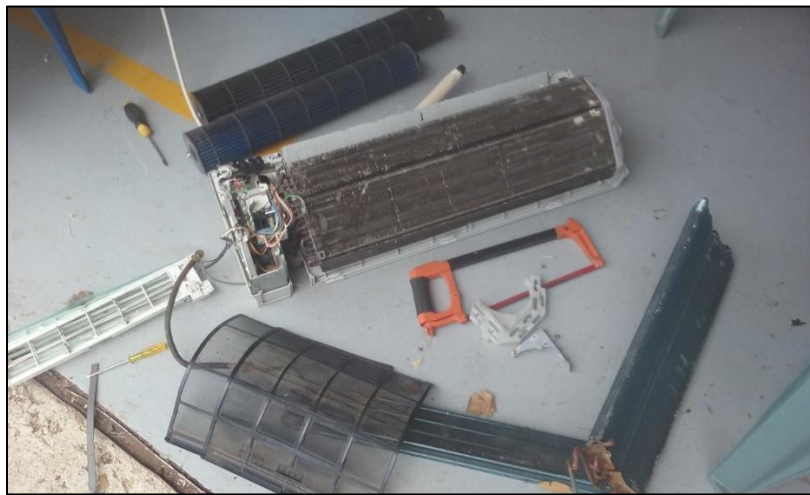


Figure 17: The dismantled AHU.

After dismantling, it was noted the fan along with the coil could be cut in half to make two separate units. Hence the unit was split into two and the housings for both the units were made from Perspex and Plyboard. The coil unit contained copper tubings running in parallel, hence upon cutting required U-shaped copper pipes to be cut and joined to the cut ends to complete the circuit from inlet to outlet. Each coil required eight U-shaped copper pipes to be connected to the coil to allow the water to flow from inlet to outlet without any leakage. However, the leakage from the two coils could not be stopped completely and hence, it was decided to search for coils of the required size in the market. Fortunately, two used coils of the required size were available with a refrigeration company, which were purchased and used for the project. This coil was assembled with the fan and the housing to complete the construction of the AHUs. Figure 18 shows the final assembled AHU.



Figure 18: Final assembled AHU.

7.3 Pump Testing

Upon completion of the model parts, the next phase was to connect the AHUs to the pump for water circulation. Prior to this, the functionality of the water pump was tested. Appropriate wires and lugs were connected to the pump and the pump was mounted on a workbench. A drum was filled with water and the inlet of the pump was connected to the drum. The pump required manual priming, after which the pump was turned on. Since the maximum suction head for the pump was relatively low, the pump was placed at almost the same height as the suction head.

7.4 Water circuit

After successfully testing the pump, the AHUs were connected to the water pump through which the chilled water would circulate. For this, plastic pipes were used so that the AHUs can be moved around within the model during testing. Binding wires

were used to firmly tie the pipes to both the AHUs and the pump. Figure 19 shows the AHU installed inside the building model along with pipe connections between the AHU and the water pump.



Figure 19: The Model with complete water circuit.

8.0 Model Testing

This section discusses how the constructed model was tested for its effectiveness and its functionality

8.1 Leak testing of the of the pipe works and the AHU coils

The leak test was necessary to find and fix any water leakages. Presence of any leakage would cause chilled water loss.

Initially, since the AHU was made with copper tubes connected by flexible plastic pipes, the joints were very weak and after each connection the AHU was immersed in water and blown through the inlet pipe to see any bubbles escaping to indicate leakage. After fixing the leaks, this was again tested under pressure and then attached to the AHUs.

Once all the flexible plastic pipes were connected from the pump to the AHU and vice versa, the pumps were run and if any leaks were present the joints were tightened until the model was free of any leaks.

8.2 Heating the model

The cooling load calculated for the model in section 6.2 was depicted in the model through the use of hot plates. Two 750 W plates at half their rated power were placed at each side of the building, which represented a total cooling load of 0.76 kW. The heat plates were left on until the temperature inside the model stabilized at 32°C.

Once it was ascertained that the desired temperature was attainable inside the building, further testing commenced.

8.3 Positioning of the two AHU's

The AHUs were positioned as shown in Figure 20 below, with the flow of air as indicated by the blue arrows. Initially it was aimed to move the AHUs around the building and see the effect of cooling, but since the building need to be sealed off prior to testing, it was not advisable to remove the silicon seal and reseal. Therefore all the testing was based on this initial configuration.

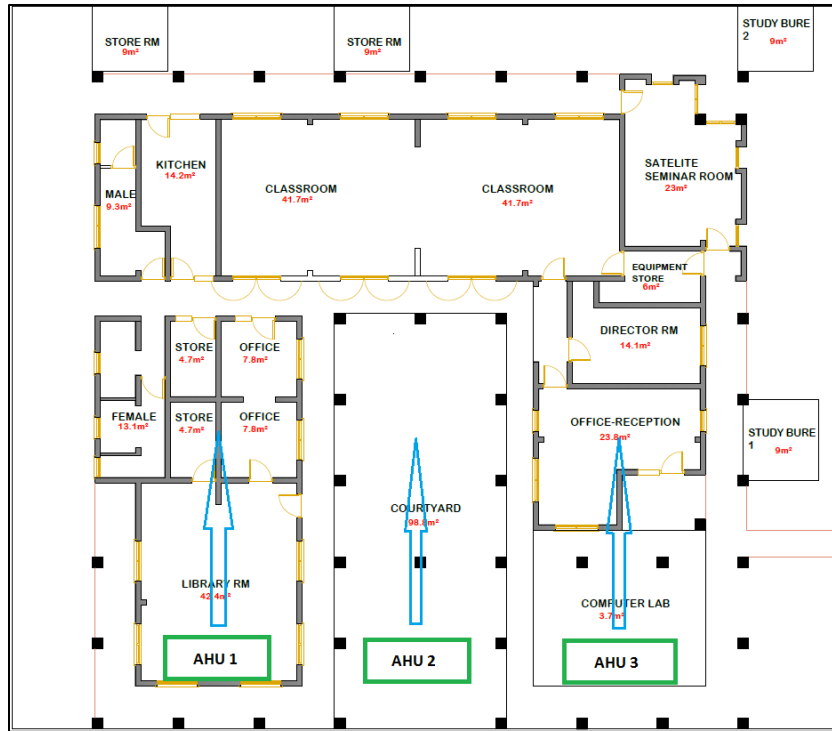


Figure 20: The Location of AHUs and the direction of air flow.

The load was increased slowly to 2 kW and its effect on the increase in temperature inside the building from an ambient of 28°C on that day was observed. Figure 21 shows the trend observed. The temperature inside the building was taken after 10 mins the load was applied. It can be seen that the temperature inside the building rose to 35°C when a load of 2 kW was applied.

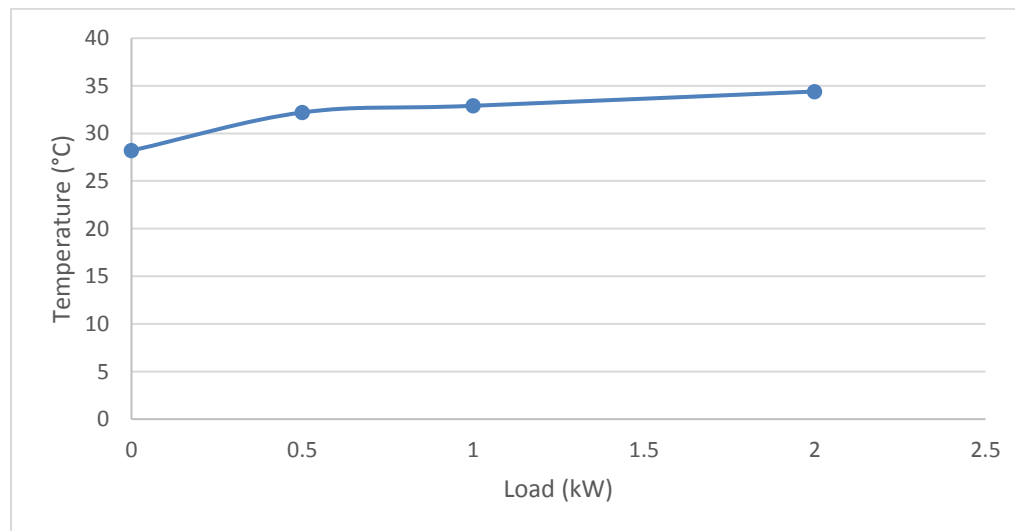


Figure 21: The Effect of cooling load on the temperature inside the building.

8.4 Effect of cooling load on the time to reach 22°C inside the building

The time taken for the temperature inside the building to drop to 22°C was monitored after the cold water was supplied to cool the building model. As the cooling load increased, the time to cool increased exponentially as seen in Figure 22. Even though the temperature did not increase by a drastic amount; the amount of heat added to the air mass inside the building was large. Therefore the air had to be passed through the AHUs multiple times to remove all the heat and to reach the desired temperature, hence requiring a greater cooling time.

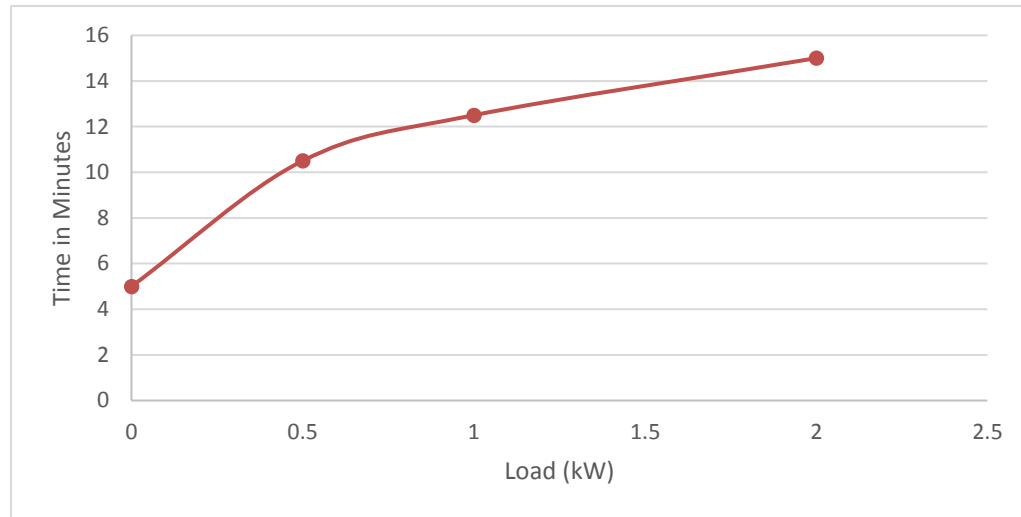


Figure 22: The Effect of cooling load on the time taken for the air temperature to drop to 22°C.

Finally, the effect of chilled water temperature rise once the temperature inside the building stabilized at 22°C was seen as a result of load increase (i.e. the temperature of chilled water was observed at the different cooling loads shown in Figure 23). The inlet temperature of water at the start of the experimentation was constant at 7°C. As the cooling load increased, the water heated at a faster rate. At the maximum load of 2 kW, the water reached a temperature of 21.5°C, so it can be said the water was reaching equilibrium with the air temperature inside the building. Therefore the experimentation was stopped here.

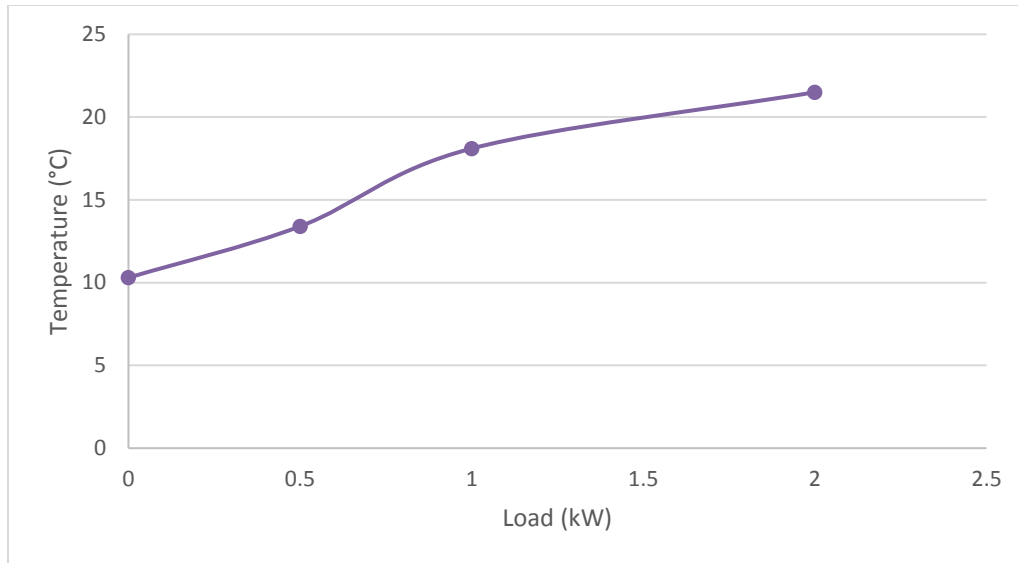


Figure 23: The Effect of cooling load on chilled water exit temperature.

8.5 Conclusions

Cooling to the desired level was achieved at a maximum load of 2 kW which is almost twice the cooling load estimated by the CAMEL software. At the maximum cooling load, the maximum cooling time was 15 mins. Further cooling at higher cooling loads could be achievable if a larger cold water reservoir is used, but is not required for the building under consideration.

9.0 Economic Analysis

The economic analysis is a very important part of any renewable energy project and determines if a project is economically feasible. This section discusses the total initial capital investment required if the USP Tuvalu campus is cooled using cold deep sea water and the associated running cost. Finally, it is compared to a conventional central chiller system and split type air conditioning system and the payback period for each scenario is calculated.

9.1 Initial capital investment

There are two types of cost which make up the initial capital investment. One is the purchasing of the new components and the other is the installation cost. The purchasing cost of the new components includes the shipment cost as well, while the installation cost includes labor and the cost of hiring machinery and equipment. Many times, the installation job is given to a certain company which brings its own machinery and labor and gets the job done, but in this project since work regarding different aspects are involved (e.g. ducting and HDPE pipe installation), one company may not have the capability to carry out different jobs. The installation cost was taken as 8% of the whole project cost. Table 5 shows the purchasing cost of all the components of the SWAC system.

Table 5: Purchasing cost of all the components of the SWAC system.

Component	Cost (USD)
Seawater pump	5,000
HDPE pipe	300,000
Heat exchanger	15,000
chilled water pump	1,000
AHU	2,500
Ductwork	5,000
Effluent pipe	22,000
Chilled water pipe	2,000
TOTAL	352,500

The quotations where obtained were applied (**Appendix 7**). The price of the Heat Exchanger was taken to be the average of the plate type heat exchanger and shell and tube type heat exchanger. The price of the HDPE pipe and effluent pipe includes the installation and the excavation cost and it is calculated using the formula [12]:

$$HDPE \text{ pipe cost} = USD 2850 \times L \times D^{1.4} \quad (8)$$

where:

- L is the total length of HDPE pipe
- D is the nominal diameter of the HDPE pipe

Therefore the total initial cost of the SWAC system including the 8% installation cost is around USD 381,000.

9.2 Running cost

The running cost of the project includes two major components: power consumption cost and maintenance cost. The power consumption of the system includes the electrical demand of the two pumps and the fan power in the air handling unit. Table 6 below shows the electrical cost of the components which require electricity. The price of electricity in Tuvalu is AUD0.56 which converts to USD0.43 (December 2017); this was used to calculate the total electricity cost of USD1,137.60 for a month's operation of the SWAC system.

Table 6: Electrical cost for the components.

component	power rating (kW)	power consumption/month	cost/month (USD)
seawater pump	6	1,440	619.2
chilled water pump	4	960	259.2
AHU	4	960	259.2
TOTAL	14	3,360	1,137.6

The maintenance cost of the system included a number of components which include:

- Pump servicing
- Heat exchanger servicing
- monitoring the condition of HDPE pipes
- monitoring condition of the chilled water pipelines and leak checks
- duct inspection

An estimate of the maintenance cost of the system can be expressed to be USD 500 per month. Therefore the total running cost of the SWAC system can be approximately stated as USD 1,637.60 per month.

9.3 Payback Period

To calculate the payback period, the capital investment and the running cost of the comparative systems need to be calculated.

9.3.1 Scenario 1: Comparison with central water chiller system

A central water chiller system works similar to a SWAC system, but the cold water circulated in the AHU coils is supplied by a chiller. The chiller works on a vapor compression refrigeration system (VCRS) where the evaporator cools the water only. A shell and tube type system is used where the refrigerant flows in the tubes, absorbing heat from the water which flows in the shell side.

A chiller with a 40 TR (ton of refrigeration) capacity was chosen for this application from **Hong Kong Naser Machinery Co., Ltd.** The quotation and specifications of the chiller are available in **Appendix 10**. The duct and AHU would remain the same as for the SWAC system. Table 7 below shows the capital and running cost of the chiller AC system.

Table 7: Capital and running cost of the chiller AC system.

Capital Cost		running cost			
Component	Cost	component	power rating (kW)	kWh/mnth	cost/mnth (USD)
Chiller	15000	chiller	30	7200	3096
Ductwork	5000	AHU	4	960	412.8
AHU	2500	maintenance			300
TOTAL	22500	TOTAL			3808.8

It should be noted that the compressor and pump are parts of the chiller unit and their power rating is given in the specifications.

The payback period can be calculated using the following equation [13]:

$$\text{payback period} = \frac{\text{excess in initial cost}}{\text{savings in running cost}} = \frac{SWAC_{ic} - CCAC_{ic}}{CCAC_{rc} - SWAC_{rc}} \quad (9)$$

where:

- $SWAC_{ic}$ is the initial capital cost of SWAC system
- $CCAC_{ic}$ is the initial capital cost of Central chiller AC system
- $CCAC_{rc}$ is the running cost of the Central chiller AC system per month
- $SWAC_{rc}$ is the running cost of the SWAC system per month

This results in a payback period of 12.67 yrs.

9.3.2 Scenario 2: Comparison with split type AC system

To estimate the costs associated with this system, firstly the cooling capacity of the split type system was estimated for each room based on the cooling load since all split type AC units do not have the same cooling capacity. The

cooling load for each room was extracted from CAMEL as shown in Figure 8. Since the cooling capacities of AC units are normally given in BTU/hr, the cooling load was converted to these units. Then the catalogues of the AC unit sellers were seen for the typical capacities of AC units and the number of units which would meet the cooling load was estimated. The catalogues are present in **Appendix 11**. Since the large BTU/hr (30000) AC machine was not present in the first catalogue a second catalogue was looked up and its details are also mentioned in the Appendix 12.

Once the suitable AC units were found, the power consumption of each of the unit was listed and its running hours were determined to calculate the total electrical cost of the system. Also, the price per unit was extracted from the relevant sources and the initial capital cost was calculated. All of the details are listed in Table 8 below.

Table 8: Running and Capital cost for conventional split type air-conditioning system.

Room Number	cooling required (kW)	Cooling required (Btu/hr)	capacity of unit (BTU/hr)	qty of units required	consumption of unit (kW)	total consumption (kW)	price/unit (USD)	total price (USD)	hours of operation	kWh/day	kWh/mnth	electricity cost/mnth
1	29.9	102023.0458	30000	3	3.25	9.75	2500	7500	8	78	2340	631.8
2	6.02	20541.09484	30000	1	3.25	3.25	2500	2500	8	26	780	210.6
3	1.41	4811.12022	7000	1	0.76	0.76	800	800	8	6.08	182.4	49.248
4	19.9	67901.6258	30000	2	3.25	6.5	2500	5000	8	52	1560	421.2
5	10.3	35145.0626	18000	2	1.82	3.64	1750	3500	24	87.36	2620.8	707.616
6	6.54	22315.40868	12000	2	1.29	2.58	1500	3000	8	20.64	619.2	167.184
7	2.88	9826.96896	12000	1	1.29	1.29	1500	1500	8	10.32	309.6	83.592
8	34.2	116695.2564	30000	4	3.25	13	2500	10000	8	104	3120	842.4
TOTAL	111.15	379259.5833		16		40.77		33800		384.4	11532	3113.64

The maintenance cost of the system is estimated to USD100 per unit per month, which includes inspection and cleaning of the AC units. Since the total number of units is 16 it accumulates a total of USD1,600 per month in maintenance. Therefore the total running cost is approximately USD4,700 per month.

The payback period can be calculated similar to Scenario 2, with the initial capital cost of the system to be USD33,800. This results in a payback period of 8.63 years.

10.0 Emission Analysis

This chapter discusses the amount of reduction in harmful pollutant emission, as a result of implementing a seawater air conditioning project. One of the major contributors of emission is the burning of fossil fuels to run electricity generators to produce power. Since this project would decrease the overall electricity consumption as seen in chapter 9, this would also reduce the emissions.

The best way to estimate the reduction in emission is to look at the amount of pollutant produced when 1 kWh of energy is generated with a diesel generator. Then the amount of energy savings can be seen and consequently the emission reduction can be calculated.

The amount of emissions released from burning diesel fuel in a generator to produce 1 kWh of power depends on a lot of factors. Firstly, the type of generator and its efficiency, secondly, the load at which the generator is running, thirdly, the grade of diesel and so forth.

Table 89 below shows an estimate of the emission of a diesel generator [14]. The power savings for the actual project will be about 3372 kWh, therefore this multiplied by the kg of emission per kWh gives the emission reduction per month for the campus.

Table 9: Emission reductions

pollutant name	kg/kW hr	reduction/mnth
carbon dioxide	0.714	2407.608
carbon monoxide	0.000365	1.23078
methane	0.00168	5.66496
nitrogen dioxide	0.00125	4.215
nitrous oxides	0.0000169	0.0569868
sulphur oxides	0.00379	12.77988

It can be seen that the major greenhouse pollutant, carbon dioxide shows maximum reduction. With the reduction in power usage mentioned above the carbon dioxide emission would reduce by almost 2.5 tons for a month.

11.0 Project Budget

The total project budget for the project is shown in a separate Excel sheet. All the parts used in the project were purchased from local enterprises. Funding for some of the components (e.g. pumps) was provided by the School of Engineering.

12.0 Conclusions

The feasibility study and design of a seawater air-conditioning system for The University of the South Pacific's Tuvalu campus was successfully completed.

The work included calculation of the total cooling load for the campus using CAMEL software, use of appropriate mathematical tools to select and size all the major components, using dimensional analysis to scale down the building and the components to an appropriate model size which could be constructed in the Mechanical Engineering workshop of USP, construction of the model, and testing the model for cooling with cold water at 7°C. The maximum cooling load of the USP Tuvalu campus was estimated to be 117 kW. A safety factor of 10% was applied which gave a cooling load of 132 kW; this value was used for all further calculations.

A scaled down model of 1:10 of the actual USP Tuvalu Campus was constructed in the Mechanical Engineering Workshop of USP. Electric heaters were placed inside the building model to simulate the heat gain from all the sources. An appropriate air handling unit was constructed, which was used to cool the air inside the building model. Cooling to a temperature of 22°C was achieved with the help of circulating cold water.

An economic analysis was also conducted in which comparison of a SWAC system was made to a conventional split type system and a conventional chiller type system with reference to the capital costs, maintenance costs and the payback period. A payback period of 8.6 years to 12.6 years was estimated.

An emission analysis showed a reduction of 2.5 tons of carbon dioxide per month with the implementation of this project.

13.0 Acknowledgement

Two students of the Bachelor of Mechanical Engineering programme, Mr. Muzammil Ali and Mr. Reemal Prasad, made significant contribution to this project. Part of the project met their degree requirements while for the additional work they did in January and February 2018, they were paid. Financial support for the trip to Tuvalu for three persons to obtain the details of the USP campus including the temperatures experienced during the peak of summer, for the measurement of sea-water temperature at different depths and also for purchasing parts for the project was provided by IUCN Energy, Ecosystems and Sustainable Livelihoods initiative.

14.0 References

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15.0 Appendices

The red highlighted box indicates the specifications that were used in the project.



HDPE PIPE



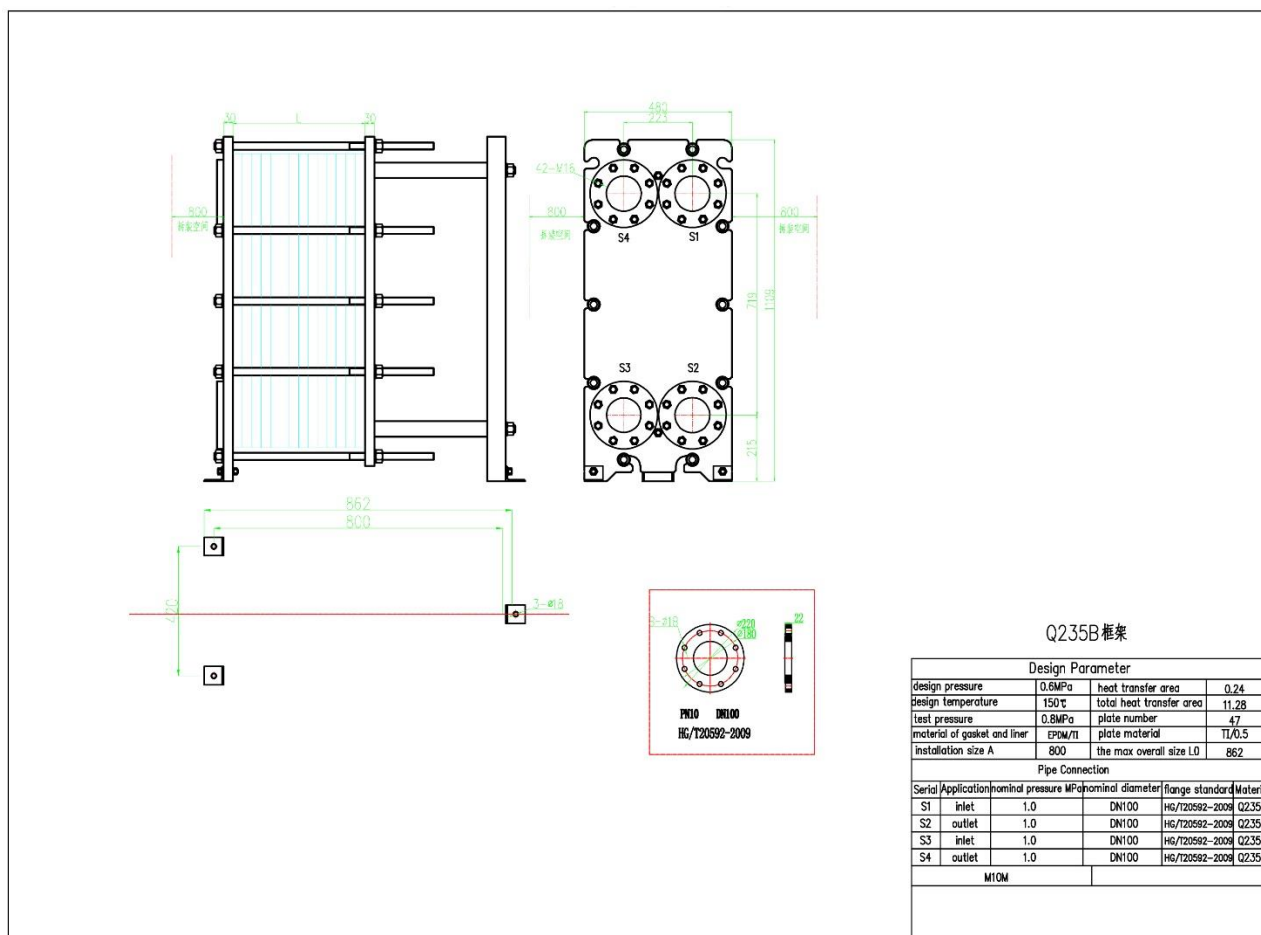
Nominal Pipe Size	DR32.5 (50 PSI)				DR26 (64 PSI)				DR21 (80 PSI)				DR17 (100 PSI)			
	Average Outside Diam- eter	Average Inside Diam- eter	Min. Wall Thick- ness	Average Weight (lbs/ft)	Average Inside Diam- eter	Min. Wall Thick- ness	Average Weight (lbs/ft)	Average Inside Diam- eter	Min. Wall Thick- ness	Average Weight (lbs/ft)	Average Inside Diam- eter	Min. Wall Thick- ness	Average Weight (lbs/ft)			
3	3.50	-	-	-	-	-	-	3.147	0.167	0.76	3.064	0.206	0.93			
4	4.50	4.206	0.138	0.83	4.133	0.173	1.03	4.046	0.214	1.26	3.939	0.265	1.54			
5	5.56	5.200	0.171	1.27	5.109	0.214	1.57	5.001	0.265	1.93	4.869	0.327	2.35			
6	6.63	6.193	0.204	1.80	6.085	0.255	2.23	5.956	0.315	2.73	5.799	0.390	3.33			
7	7.13	6.660	0.219	2.08	6.544	0.274	2.58	6.406	0.339	3.16	6.236	0.418	3.85			
8	8.63	8.062	0.265	3.05	7.922	0.332	3.78	7.754	0.411	4.63	7.549	0.507	5.65			
10	10.75	10.049	0.331	4.73	9.873	0.413	5.87	9.665	0.512	7.19	9.409	0.632	8.77			
12	12.75	11.918	0.392	6.66	11.710	0.490	8.25	11.463	0.607	10.11	11.160	0.750	12.34			
13	13.38	12.503	0.412	7.33	12.284	0.514	9.08	12.025	0.637	11.13	11.707	0.787	13.58			
14	14.00	13.087	0.431	8.03	12.858	0.538	9.95	12.587	0.667	12.19	12.254	0.824	14.87			
16	16.00	14.956	0.492	10.48	14.695	0.615	12.99	14.385	0.762	15.93	14.005	0.941	19.43			
18	18.00	16.826	0.554	13.27	16.532	0.692	16.45	16.183	0.857	20.16	15.755	1.059	24.59			
20	20.00	18.695	0.615	16.38	18.369	0.769	20.30	17.981	0.952	24.88	17.506	1.176	30.35			
22	22.00	20.565	0.677	19.82	20.206	0.846	24.57	19.779	1.048	30.11	19.256	1.294	36.73			
24	24.00	22.434	0.738	23.59	22.043	0.923	29.24	21.577	1.143	35.83	21.007	1.412	43.71			
26	26.00	24.304	0.800	27.68	23.880	1.000	34.31	23.375	1.238	42.05	22.758	1.529	51.30			

1

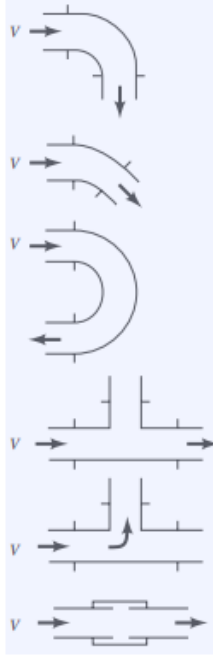
2

3

Appendix 1: Catalogue for HDPE pipe selection.



Appendix 2: Plate Heat exchanger drawing and details.

a. Elbows		
Regular 90°, flanged	0.3	
Regular 90°, threaded	1.5	
Long radius 90°, flanged	0.2	
Long radius 90°, threaded	0.7	
Long radius 45°, flanged	0.2	
Regular 45°, threaded	0.4	
b. 180° return bends		
180° return bend, flanged	0.2	
180° return bend, threaded	1.5	
c. Tees		
Line flow, flanged	0.2	
Line flow, threaded	0.9	
Branch flow, flanged	1.0	
Branch flow, threaded	2.0	
d. Union, threaded		0.08
*e. Valves		
Globe, fully open	10	
Angle, fully open	2	
Gate, fully open	0.15	
Gate, $\frac{1}{4}$ closed	0.26	
Gate, $\frac{1}{2}$ closed	2.1	
Gate, $\frac{3}{4}$ closed	17	
Swing check, forward flow	2	
Swing check, backward flow	∞	
Ball valve, fully open	0.05	
Ball valve, $\frac{1}{4}$ closed	5.5	
Ball valve, $\frac{3}{4}$ closed	210	

*See Fig. 8.36 for typical valve geometry

Appendix 5: K_r coefficient for different joints and intersections [11].

Gradual enlargement

Gradual contraction

Some loss coefficients for gradual enlargement:

θ°	5	7	10	20	30	40
K_r	0.17	0.22	0.28	0.45	0.59	0.73

Some loss coefficients for gradual contraction:

θ°	30	45	60
K_r	0.02	0.04	0.07

Appendix 6: K_r coefficient for enlargement and contraction.



SHANDONG GRAD GROUP CO.,LTD

NO.6,GRAD ROAD,TIANQU INDUSTRIAL PARK,DEZHOU,SHANDONG,CHINA

TEL:0086-534-2730845

FAX:0086-534-2730191

CEL:0086-15505347810

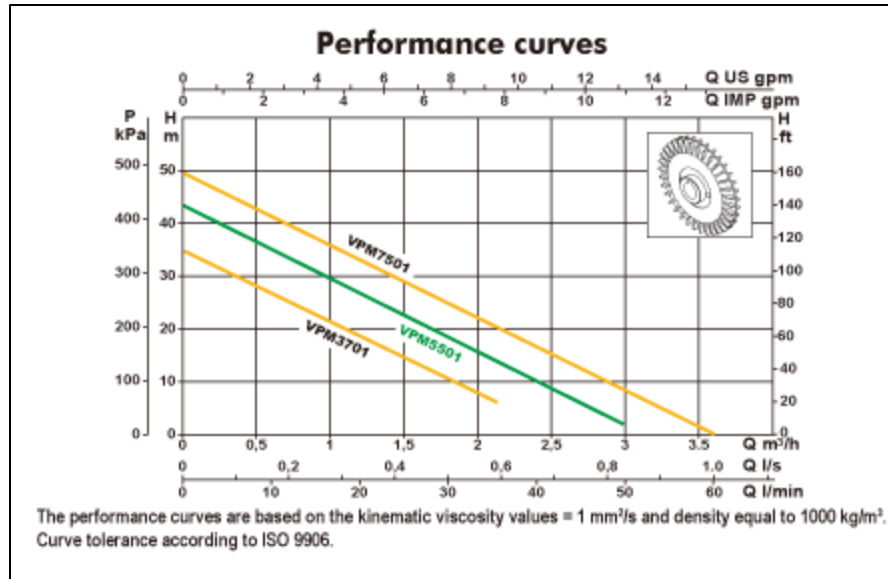
GRAD Ceiling Air Handling Unit Quotation

To	Mr Ali	Trade term:	FOB Qingdao	
From:	Alexandra Zhu	E-mail:	trade5@gradgroup.com	
Date:	GJZ -2017-10-28	Validity:	30days	
Subject:	GRAD Ceiling Air Handling Unit Quotation			
Model	Description	Unit Price (USD)	Qty.	Sub-total Price (USD)
FP70*2D (4 rows)	1.Cooling capacity:79.9KW; 2.Heating capacity:153KW; 3.Air Volume:14000m³/h; 4.External Static:350pa; 5.Power of the fan:2.2*2kw; 6.Water flow:13.9m³/h; 7.Weight:352kg; 8.Size:2580*1200*1300mm; 9.Noise:≤67dB(A); 10.Voltage: 380V/50HZ 11.Fan quantity:2 set	1,414	1	1,414
Transportation and port fee(USD)				833
FOB Qingdao total (USD)				2,247

Appendix 7: Quotation for AHU.

B-NM	NM	P ₂		Q m³/h l/min	15	16,8	18,9	21	24	27	30	33	37,8	39	42	45	48	54
		kW	HP		250	280	315	350	400	450	500	550	630	650	700	850	800	900
B-NM 40/12F	NM 40/12F/A	1,1	1,5	H m	14	13,5	13	12	11	9,5	8	6						
B-NM 40/12C	NM 40/12C/A	1,5	2		17,5	17	16,5	16	15	13,5	12	10,5	7,5	6,5				
B-NM 40/12A	NM 40/12A/A	2,2	3		22	22	21,5	21	20	19	18	16,5	14	13	11,5			
B-NM 40/16C	NM 40/16C/A	2,2	3		23	22,5	22	21,5	20	18,5	16,5	14,5	11	10				
B-NM 40/16B	NM 40/16B/A	3	4		29	28,8	28	27,5	26,5	25	23,5	21,5	18	17	14			
B-NM 40/16A	NM 40/16A/A	4	5,5		37	36,5	36,5	36	35	33,5	32	30,5	27	26	23,5	20	17	
B-NM 40/20D	NM 40/20DE	4	5,5		39	38	37	35,5	33,5	30,5	27	22,5	14					
B-NM 40/20C	NM 40/20CE	4	5,5		41,5	40,5	39,5	38	36	33,5								
B-NM 40/200BE	NM 40/20BE	5,5	7,5		50	49,5	48,5	47,5	45,5	43,5	41,5	37,5	30,5					
B-NM 40/200AE	NM 40/20ARE	5,5	7,5		55	54,5	54	53	51	49								
B-NM 40/200AE	NM 40/20AE	7,5	10		57,5	57	56,5	55,5	54,5	52,5	50,5	48	42,5	40,5	35			
B-NM 4025/C/A	NM 40/25C/A	9,2	12,5		61	61	60,5	59,5	58,5	56,5	53,5	49,5	41,5	40	33,5			
B-NM 4025/B/A	NM 40/25B/A	11	15		69,5	69,5	69	68,5	67	65,5	63,5	60,5	53,5	51	45			
B-NM 4025/A/A	NM 40/25A/A	15	20		90	90	89,5	89	88,5	87	85	83	77,5	76	70,5			

Appendix 8: Volume flow rate and head for chilled water pump selection.



Appendix 9: Performance curve for model pump.



HongKong Naser Machinery Co.,Ltd

DongGuan Naser Machinery Co.,Ltd (Factory)

Tel:+86-731-88683203

Fax:+86-731-88683203

Website: <http://www.naserland.hk>

E-mail: naser06@naserland.com

Address: Comprehensive Zone, Hengli Town, Dongguan City, Guangdong Province, China

QUOTATION

NO.NSQ20171027002

TO:	Muzammil Ali	E-MAIL:	s11109547@student.usp.ac.fj		
FM:	Shirley Gui/+86-13974913392	Date:	Oct.27th.2017		
Spe	Specification	Quantity	EXW price	Total	Note
"NASER" Air Cooled Industrial Chiller NWS-30AC (HS code:8418699090)		1	US\$7,250	US\$7,900	CIF Suva, Fiji
Ship by sea, from China port to Suva, Fiji		1	US\$650		

1	Model:NWS-30AC
2	Voltage: 380V/3Phase/50HZ (Schneider Electric)
3	Cooling Capacity: 91KW=78260Kcal/h=25.8USRT
4	Compressor: 7.5kw*3units (Japan Sanyo scroll compressor)
5	Refrigerant: R22 Emerson expansion valve
6	Condenser: Japan Kobe High Effective Red Copper Pipes Covered Into Aluminum Fins+Mute fan
7	Axial Fans – air flow: 31000 m³/h
8	Evaporator: stainless steel water tank with high effective copper coil type
9	Water tank capacity: 365L
10	Diameter of standard pipe: 2"
11	Standard freezing water flow: 15.6M³/H
12	Built in pump: 2.2Kw, Lift: 16m, flow rate: 39m³/h (Taiwan MINAMOTO)
13	Machine weight: 1400KG
14	Machine Size: L2200*W1100*H1880mm
15	Microcomputer temperature control/1set



Note: 1 Cooling capacity is based on: the chilled water inlet temperature 7 °C / 12 °C; cooling water inlet and outlet water temperature of 40 °C / 45 °C. 2. Scope of work: chilled water temperature range 6 °C ~ 30 °C; ambient temperature is not higher than 35 °C. The right to change due to product improvement the above parameters or dimensions without prior notice.

Remark:

1. Lead time : 15-20 working days after received the deposit money.
2. Payment way: By T/T, Credit Card (VISA) ,E-checking , 50% In advance , 50% Before Shipping
3. Remark: One year warranty period. The only promise is that the user must not violate the operation regulations.
4. Price valid: 30 days.
5. Package: Standard wooden case package.

Appendix 10: Quotation for chiller unit.

"M" Series



Model			7000BTU	9000BTU	12000BTU	18000BTU	24000BTU
			7K	9K	12K	18K	24K
			0.6Ton	0.75Ton	1Ton	1.5Ton	2Ton
Factory Model		Cooling	KF-20GW/M	KF-25GW/M	KF-35GW/M	KF-50GW/M	KF-70GW/M
		Cooling&Heating	KFR-20GW/M	KFR-25GW/M	KFR-35GW/M	KFR-50GW/M	KFR-70GW/M
Optional Power Supply		Ph-V-Hz	1Ph 220V-240V/50Hz Or 1Ph 208V-230V/60Hz				
Cooling	Capacity	w	2000	2500	3500	5000	7000
	power	w	760	920	1290	1820	2500
	Current	A	3.6	4.2	5.9	8.3	11.4
Heating	Capacity	w	2050	2550	3500	5050	7050
	power	w	740	910	1270	1800	2470
	Current	A	3.5	4.1	5.8	8.2	11.2
Operation temp		°C	16-32	16-32	16-32	16-32	16-32
Ambient temp		°C	-7-45	-7-45	-7-45	-7-45	-7-45
Noise- indoor/outdoor		dB(A)	36/50	36/50	38/52	39/55	42/62
Weight- indoor/outdoor		Kg	9/27	9/28	10/32	14/42	17/50
Indoor Dimension	Net	mm	680*188*245	680*188*245	790*187*270	1020*215*320	1020*215*320
	Package	mm	745*253*305	745*253*305	875*285*345	1100*290*390	1100*290*390
Outdoor Dimension	Net	mm	640*240*490	640*240*490	760*260*540	760*260*540	810*285*640
	Package	mm	720*310*550	720*310*550	860*340*600	860*340*600	910*390*710
Qty ' per 20' /40'HQ		set	135/350	135/350	120/265	100/215	70/170

Appendix 11: Specifications for split type unit.

Model	SX-A28	SX-A28	SX-A38	SX-A48	SX-A58	SX-A68
POWER	220V-50Hz/60Hz					
Rated cooling/heating(BTU)	9000/9400	12000/12200	18000/18500	24000/26300	30000/31000	36000/37000
Rated cooling/heating power(w)	860/880	1130/1100	1718/1700	2350/2490	3248/3490	3899/4170
Rated cooling/heating cyrrrent(A)	4.0/4.1	5.2/5.5	7.8/7.7	10.7/11.3	14.8/15.9	17.7/19.0
EER/COP	3.06/3.13	3.10/3.24	3.08/3.19	2.99/3.10	2.7/2.6	2.7/2.6
Packing Dimensions(L/W/H)mm(indoor unit)	860*240*345mm	860*240*345mm	1070*380*295mm	1070*380*295mm	1070*380*295mm	1070*380*295mm
Packing Dimensions(L/W/H)mm(outdoor unit)	750*318*530mm	9.15*330*530mm	985*400*775mm	985*400*775mm	985*400*775mm	1020*400*900mm
Applicable Room area(m²)	10-12	13-20	21-27	28-36	37-45	46-54
Noise Level dB(A)(indoor/outdoor)	39-51	40/52	42/53	44/54	44/56	45/57

Appendix 12: Specifications for split type unit.