A TECHNICAL AND ECONOMIC INVESTIGATION OF THE USE OF DEEP COLD SEAWATER FOR AIR CONDITIONING IN COASTAL COMMUNITIES IN GHANA

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CERTIFICATION

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

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This is to certify that I have read this thesis and that in my opinion it is fully adequate, in scope and quality, as a thesis for the degree of Mater of Science.

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DECLARATION

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

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ABSTRACT

A TECHNICAL AND ECONOMIC INVESTIGATION OF THE USE OF DEEP COLD SEAWATER FOR AIR CONDITIONING IN COASTAL COMMUNITIES IN GHANA

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Air conditioning has become a very important feature of buildings in Ghana today for maintaining comfortable indoor conditions for occupants. On the other hand, besides its high capital cost, air conditioners are expensive to operate because of their high power demand especially in the warm and humid tropical climate of Ghana. Investigation carried out in the country has shown that they contribute over 60% of the power consumed by air conditioned buildings. This has made it necessary as a nation to look for ways to reduce the power demand of air conditioners and already energy labels on air conditioners has been introduce to help the consumer to choose efficient models.

Another direction in the search to reduce power consumed by air conditioners and other appliances in buildings is in the use of renewable energy. Already advances have been made in the introduction of solar water heaters and solar lighting in the country but use of renewable energy in conventional air conditioning systems is still far from reaching large commercial application. One area of renewal energy air conditioning technology that has remained untapped in the country is the use of the naturally-occurring cold seawater from deep down the ocean for air conditioning application ashore. This technology is referred to as seawater air conditioning (SWAC).

At a depth of 1km and below, the seawater temperature is 5^{o}C which is suitable for air conditioning application. There are several places in the world today where this technology has been successfully applied to air condition coastal communities or for air conditioning of coastal communities. By replacing the use of conventional electrically-driven chillers in air conditioning systems, this technology has a very high potential for reducing the high energy demand used today for air conditioning in communities close to the shore.

This thesis carried out investigation into the technical and economic viability of applying SWAC technology for coastal communities in Ghana. The investigation was conducted on four locations along the coastline, namely, Keta, Accra, Cape Coast and Takoradi for cooling loads ranging from 1,000 TR (tons cooling) to 13,000 TR.

The first stage of the investigation was technical, which was aimed to determine whether or not the cold seawater pumped from the ocean deep met the technical requirement for air conditioning application when it arrived ashore. Cape Coast and Takoradi sites could not meet this requirement for the range of load under investigation. Accra met the technical requirement for loads starting from 5,500 TR whilst Keta met the requirement from 5,000 TR. The second stage of the thesis is the economic investigation for the two sites that passed the technical investigation, namely, Accra and Keta. Using the method of Cost of Ownership calculation, SWAC technology at Keta site turned out to be more economical than the conventional chiller system at loads from 8,500 TR. At Accra site, SWAC technology was more expensive than the conventional chiller system but showed a good potential to catch up at higher cooling loads above the 13,000 TR limit investigated in this thesis.

The conclusion of this work is that the potential for SWAC application in Ghana looks very promising and further research is recommended, especially for higher cooling loads beyond the 13,000 TR limit used in this work. It is also recommended for the future research to investigate the use of the SWAC technology in hybrid combination with conventional absorption cooling system powered by solar heating.

DEDICATION

I dedicate

this thesis to my Parents and Uncle,

Mr. John Kissi Bimpong, Miss. Roberta Dadeboe and Mr. Richard William Quaynor

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I thank the Almighty God for guiding me and leading me through this research.

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TABLE OF CONTENT

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

TR	Tons Refrigeration
A/C	Air Conditioning
SWAC	Sea Water Air Conditioning
T	Temperature, ^{o}C [K]
$\dot{m}$	Mass Flow Rate of Seawater , kg/s
$\dot{q}$	Heat Flux, W/m^2 [kW/m^2]
$\dot{Q}$	Air Conditioning Load, W [kW]
Q	Volume Flow Rate of Seawater, m^3/s
ΔT_S	Change in Temperature of Seawater Across Heat Exchanger, ^{o}C [K]
c_p	Specific Heat Capacity of Seawater at Constant Pressure, J/kg K [kJ/kg K]
ρ	Density, kg/m^3
Pr	Prandtl Number
ν	Kinematic Viscosity, m^2/s
μ	Dynamic Viscosity, kg/ms
k_p	Thermal Conductivity of HDPE Pipe, W/mK
k_s	Thermal Conductivity of Seawater, W/mK
V	Average Velocity of Seawater in Supply Pipeline , m/s
V_f	Average Velocity of Seawater outside Supply Pipeline, m/s
T_s	Average External Surface Temperature of Seawater Supply Pipeline, ^{o}C [K]
R	Thermal Resistance as a Result of Heat Transfer Across Seawater Supply Pipeline, m^{2}K/W
T_{ch}	Average Temperature of Seawater in the Supply Pipeline, ^{o}C
$T_{ch,h}$	Final temperature of Seawater at the exit of the supply pipeline as a result of wall transmission load, ^{o}C [K]
$\Delta T_{ch,h}$	Temperature rise due to wall transmission load, ^{o}C [K]
$\Delta T_{ch,f}$	Temperature rise due to frictional effects, ^{o}C [K]
$T_{ch,t}$	Total temperature rise due to both frictional effects and wall tranmsission, ^{o}C [K]
U	Overall Heat Transfer Coefficient of Seawater Supply Pipeline, W/m^2 K
h	Convective Heat Transfer Coefficient, W/m^{2}K
h_{LF}	Head loss due to frictional effects, m

h_{Lx}	Head loss at heat exchanger, m
h_{LFS}	Head loss due to both frictional effects and specific losses, m
D	Pipe Diameter, m
LMTD	Log Mean Temperature Difference, ^{o}C [K]
r	Radius of Pipe, m
P	Cold Sea Water Pumping Power, W [kW]
h_w	Sea Water Pumping Head, m
A	Area, m^2
η	Sea Water Pump Efficiency, %
h_L	Head Losses due to Major and Minor Losses, m
f	Frictional Factor
K	Shape Factor
L	Total Length of Seawater Supply Pipeline, m
L_e	Equivalent length, m
Re	Reynolds number
Nu	Nusselt Number
ε	Roughness of pipe, mm
PN	Nominal Pressure, bar
SDR	Standard Dimension Ratio
C	Cost of Item, $
Cr	Cost of Reference Item, $
S	Capacity of Item, kW or m^3/s etc.
Sr	Capacity of Reference Item, kW or m^3/s etc
m	Exponent of Cost Size Relationship
P	Present Value, $
A	Amount, $
N	Number of years, yrs
i	Interest Rate, i

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Air conditioning has become a very important feature of buildings in Ghana today. As a result of the warm and humid tropical climate, occupied buildings such as hotels, shopping malls, restaurants and private homes alike all rely on air conditioning to keep their occupants comfortable. On the other hand, air conditioning is expensive to operate mainly as a result of the high power it consumes especially in the warm and humid tropical climate. With demand for electric power outstripping supply, outages have become an everyday feature of power supply in Ghana today, thereby worsening the economic woes of major users of air conditioning when they have to switch to expensive generators to run their air conditioning systems during periods of power outage.

To cut down on the energy use of air conditioners requires energy-efficient practices on the part of the building designer, the air conditioning service provider, and finally the user (occupant). The actions or inactions of any of the three parties affect the energy use (or misuse) of air conditioners. Several strategies have been proposed to deal with the high power consumption of comfort air conditioning in buildings such as adoption of renewable energy solutions. Solar-driven air conditioning technology is one renewable energy solution that is receiving intensive research, especially in the area of absorption plants, but this is yet to achieve large scale commercial application. One other renewable energy option for air conditioning application in the country is the large and stable mass of cold seawater that exists deep down in the ocean.

The sun's energy warms the surface of the earth's oceans while the depths of the ocean remain at relatively low temperatures. These thermal gradients in the ocean, maintained by the sun, can be utilized by heat engines of various sorts, to generate electricity. This is known as Ocean Thermal Energy Conversion (OTEC). The use of cold seawater for air conditioning (SWAC) is a concept derived from OTEC research, and can provide benefits over more conventional forms of air conditioning. This technology has been successfully applied in some tropical island areas, where air conditioning demands are high and the physical distance to cold seawater is at a minimum. Such a system (SWAC) does not require large, electrically-powered conventional chillers, and would also not use refrigerants such as CFCs that are considered harmful to the environment.

1.2 JUSTIFICATION

From surveys carried out by the Energy Foundation in Ghana, among others, air conditioners represent over 60% of the power usage in air conditioned buildings (Uba et al, 2011). Therefore economizing the energy use by air conditioners will have major impact on building's total energy requirement.

The two cooling systems that are essentially used in conventional air conditioning systems today are the vapour compression system which operates on electric power and the absorption system which is heat driven. Considering the fact that electric power is generated partly by thermal plants, it implies that both cooling systems contribute to global warming as a result of combustion of fossil fuels associated with their operation. Global warming is a problem that has engaged the concern of the world's scientific community. The Exponential growth in the build-up of combustion products trapped within Earth's atmosphere is implicated as the primary cause of the greenhouse effect. (Marland, Boden and Andres et al, 2001). Thus with the global concern on the greenhouse effect there is the need to look into renewable sources of energy, especially for air conditioning in Ghana.

Research carried out by Makai Ocean Engineering Company, one of the world's leading engineering company on seawater air conditioning projects, has recommended the coastlines of Africa as potential areas where cold seawater air conditioning is viable. This has provided the motivation for this thesis.

1.3 OBJECTIVES

1.3.1 Main Objective

The main objective of this research is to carry out a technical and economic assessment of the use of cold seawater for air conditioning of coastal communities in Ghana.

1.3.2 Specific Objective

The specific objectives of the thesis include:

1. Review the state of development of SWAC technology.
2. Carry out a technical assessment of the application of SWAC systems for air conditioning in coastal communities in Ghana.
3. Assess the economic viability of the use of SWAC system in Ghana compared to the conventional air conditioning systems.

1.4 METHODOLOGY

Literature study was carried out on the fundamental principles of SWAC technology and review made of sample SWAC installations in the world. Consultations were made with the

Oceanography and Fisheries Department of the University of Ghana (Legon) to obtain data on the topography of the seabed along the coastline in Ghana. Technical analysis was carried out to assess the potential of applying the SWAC technology over a range of air conditioning loads for a selected number of towns along the coastline of Ghana. The highlights of the technical assessment is the determination of the terminal temperature of the cold seawater after it has been pumped through the long pipeline spanning from the cold deep sea through rising seawater temperature to the shore. The size of cold seawater pipes, pump size, pumping power and required heat exchanger surface area was also assessed.

The KNUST library, the internet, etc. were accessed for information. Appropriate methods and software (Matlab and Microsoft Excel 2007) were used, during the technical stage, to determine;

- Total mass and volume flow rate flow rate of seawater
- Heat transfer along the seawater pipeline
- Total head loss and seawater pumping power
- Heat transfer surface area of heat exchanger

For comparison of the SWAC system with the conventional air conditioning system, the comparison was limited to the production of chilled fresh water for air conditioning, on one side using the SWAC technology and the other using a conventional chiller. The reason is that the remaining part of the air conditioning system, namely, chilled water distribution pipe work, circulating pump(s), terminal devices such as air handling units and fan coil units, ductwork, controls, etc. will be the same irrespective of whether the chilled fresh water is produced using SWAC system or a conventional chiller.

Cost of Ownership was the method used to perform economic comparison of the SWAC system and conventional chiller system.

1.5 SCOPE OF WORK AND THESIS ORGANIZATION

The scope and organization of the thesis is as follows:

Chapter 1 deals with the introduction to the topic and the objective of embarking on such a research. Chapter 2 presents a review of conventional air conditioning systems and introduces the SWAC technology and examples of real-life application of the technology.

Chapter 3 presents a technical analysis of the SWAC piping system for a range of air conditioning loads from 1,000 TR (Tons Refrigeration) to 13,000 TR (1 TR = 3.516 kW) to determine the final temperature of the cold seawater when it arrives onshore which will be the determining factor on whether or not it met the technical requirement for air conditioning application. Chapter 4 takes a look at the economic analysis. Chapter 5 presents recommendations and conclusions.

CHAPTER 2

LITERATURE REVIEW

2.1. REVIEW OF CONVENTIONAL AIR CONDITIONING SYSTEM

Air conditioning systems can be classified into a number of broad categories. Among this classification are the unitary systems and the central systems.

Unitary systems are the simplest air conditioning equipment. They are basically factory assembled units with all the components contained in one or two enclosures. They are further classified into two sub-groups as Window Units and Split Unit. The latter is also referred to as split-packaged which may be Single Split or Multi-Split such as the Variable Refrigerant Volume (VRV) system.

The central system comes in three main versions, namely, the All-Air Systems, Air-and-Water Systems and All-Water Systems. In the All-Air System, the air is centrally treated (cooled and dehumidified) externally in equipment room located outside the occupied space (for example in a central mechanical equipment room) and supplied to the occupied space via ducts to provide total cooling of the space. No additional cooling is necessary within the occupied space. It may be designed in two versions as Constant Air Volume (CAV) system or Variable Air Volume (VAV) system.

In the Air-and-Water System, both treated air and chilled water are distributed via ducts and pipes from an external equipment room to terminal units installed in the occupied spaces. Note that cooling occurs partly in the occupied space.

In the All-Water Systems, chilled water is circulated from a central refrigeration system through cooling coils in terminal units (such as fan coil units) located in the occupied space. Total cooling occurs in the space. The central system is more easily adaptable for providing air conditioning for multiple spaces or buildings and gives better air conditioning due to several factors such as better air filtration and external location of all or part of the equipment. It is however more expensive.

For the purpose of comparison with SWAC system, a central Chiller which can be used to supply chilled water for operating any of the three central systems is considered. The chiller may be a Mechanical Compression Chiller or Absorption chiller. The mechanical compression chiller may be fitted with a reciprocating, screw, centrifugal or scroll compressor and may also operate with diverse types of refrigerants. The Absorption chiller is usually of the water-lithium bromide type for air conditioning where the water is the refrigerant. The basic equipment layouts of both systems are shown in Figure 2.1 and Figure 2.2. Both systems may operate with a water-cooled condenser coupled to a cooling tower or an air-cooled condenser.

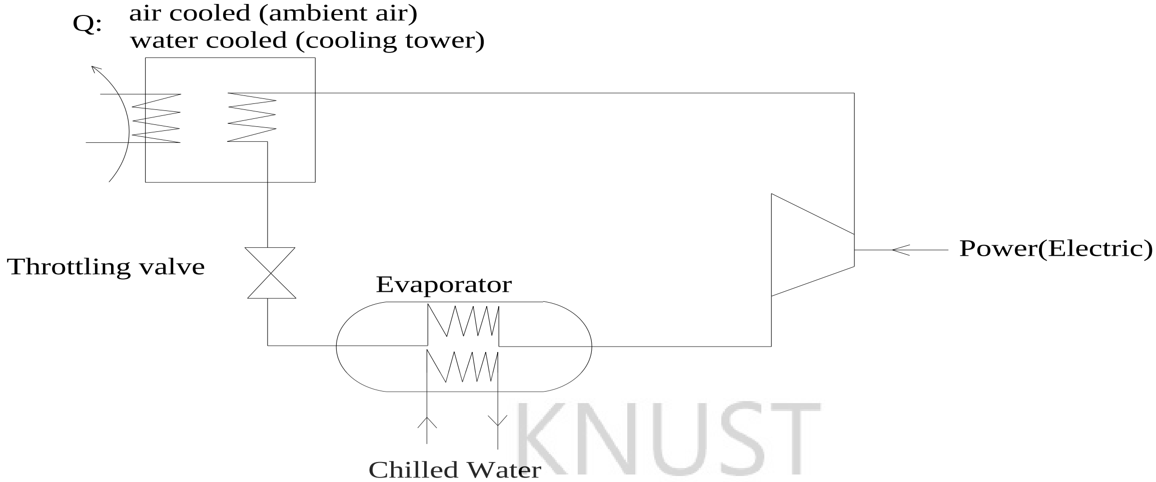

Figure 2.1: Schematic of a mechanical compression chiller

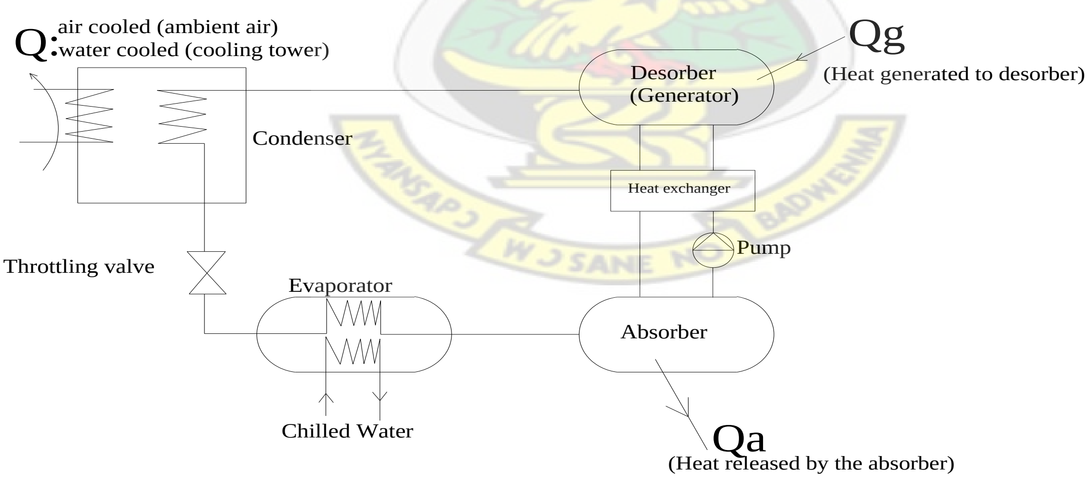

Figure 2.2: Schematic of an absorption chiller

2.2. CHILLED WATER TEMPERATURE

With a dry bulb (db) and wet bulb (wb) temperature of around 33^{o}C and 27^{o}C respectively, the weather condition in Ghana is highly humid. This is typical of all tropical regions. This means that the latent component of air conditioning cooling load is considerably high and the system must be capable of removing humidity in the air without necessarily over-cooling it. One way to deal with high humidity is to design for deep cooling of the air and use reheat to control the supply air temperature from being too cold, though reheating is unnecessary waste of energy. Design indoor conditions are assumed to be 25^{o}C db and 50% relative humidity which have a dew point temperature of 14^{o}C.

Air conditioning chillers usually operate with chilled water supply/return temperature of 6/12^{o}C. This is the recommended value taking into account the economics of the refrigeration process and the freezing temperature of water. If the water temperature is sank lower, there will be the risk of the temperature on the coil surface reaching zero which will cause freezing on the surface. Industrial systems requiring lower temperature use brine or antifreeze solutions. The room dew point temperature also sets constraint on the chilled water temperature inside the cooling coil of the terminal device. If the coil is to be able to remove moisture from the air, then its surface temperature should not be higher than the dew point temperature of the air, which in the above stated indoor condition is 14^{o}C. This sets the recommended limit of chilled water temperature in the coil at 12^{o}C if it is to be able to do latent cooling everywhere on the coil surface. At higher chilled water temperature, the coil can only remove sensible heat from the air and no latent cooling will occur. The room condition will then be difficult to maintain.

Though terminal devices such as air handling and fan coil units can operate with a design supply water temperature of 4.4 to12.8^{o}C, the capacity drop off drastically at higher entry water temperature in addition to the problem of little or no latent cooling at the higher chilled water temperature discussed above. Illustrated in Table 2.1 is the change in the cooling capacity of a Fan coil unit (source: York International) which shows that the total cooling capacity of the unit for a room air wet bulb temperature of 20^{o}C drops from 9.2 kW for chilled water in/out temperature of 6/12^{o}C to 4.2 kW when the chilled water in/out temperature rises to 12/18^{o}C corresponding to over 50% drop in cooling capacity. Thus operating chilled water air conditioning systems at higher chilled water temperature than recommended is both ineffective and expensive in capital and operating cost.

Table 2.1 Cooling Capacities of Fan Coil Unit (York International)

FCU type YSD/W8		**TOTAL COOLING CAPACITY, kW**				
CHILLED WATER TEMP., ^{o}C		**WET BULB TEMPERATURE, ^{o}C**				
OUT	IN	14	16	18	20	21
11	5	4.1	5.8	7.5	10	-
12	6	3.5	5.1	7.0	9.2	10.5
12	7	3.3	4.8	6.8	8.0	10.2
16	10	-	2.3	3.9	5.8	6.7
18	12	-	-	2.6	4.2	5.0

Source: York International

2.3 REVIEW OF SWAC

2.3.1 SWAC System

The SWAC system is an alternate energy system that uses the cold water from the deep ocean (and in some cases a deep lake) to cool buildings. In some areas it is possible to reduce dramatically the power consumed by air conditioning (A/C) systems; SWAC can be a cost-effective and attractive investment (Makai et al, 2004).

2.3.2 Benefits of a SWAC System

According to Makai (2004) the seawater air conditioning system taps into a significant and highly valuable natural energy resource that is available at some coastal locations. The benefits of a seawater air conditioning system include:

- Large energy savings approaching 90%
- Proven technology
- Short economic payback period
- Environmentally friendly
- Costs are nearly independent of future energy price increases
- No evaporative water consumption
- Cold seawater availability for secondary applications

2.3.3 Seawater Air Conditioning Concepts

According to Makai (2004) along many ocean coastlines and lake shorelines, there is reasonable access to naturally cold water that is as cold as or colder than the water used in conventional air conditioning systems. If this water can be tapped, then the power required to operate mechanical chillers can be eliminated. The basic concept of SWAC is to take advantage of available deep cold seawater to cool the chilled water in one or more buildings as opposed to using more energy intensive refrigeration systems.

A schematic of a SWAC system is shown in Figure 2.3. The buildings to the far right are identical internally to buildings cooled with conventional chillers. Chilled fresh water moves through these buildings with the same temperatures and flows of conventional systems. The seawater and chilled water pumps and heat exchangers would typically be located at the shoreline in a cooling station.

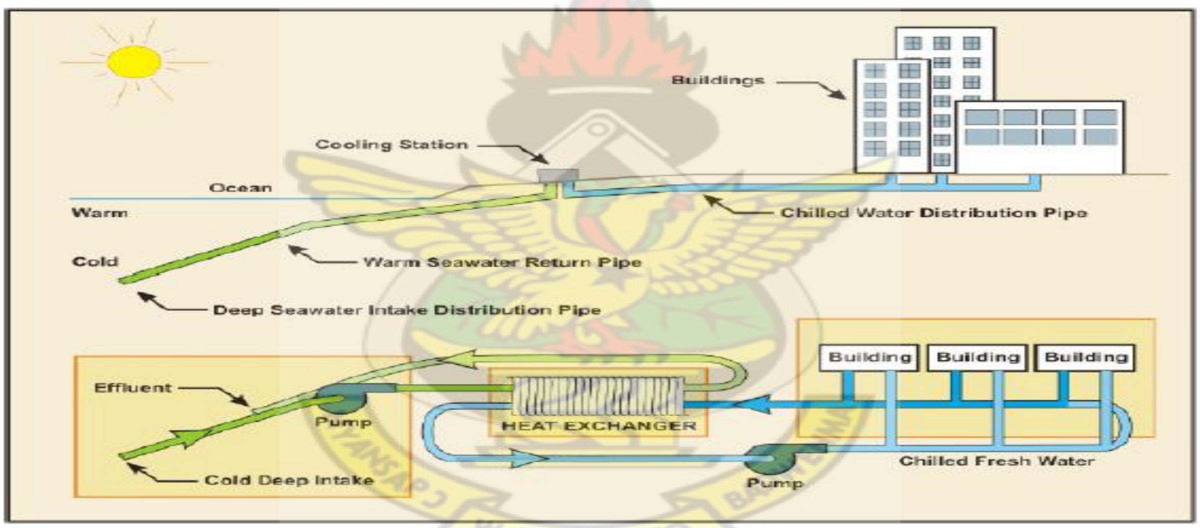

Figure 2.3: SWAC Schematic diagram

2.3.4 SWAC Components

According to Elsafty and Saeid (2009) the main components of a SWAC are the seawater supply system, the heat exchanger or cooling station and the fresh water distribution system.

- Seawater supply system: this consist of pipes for cold water intake and for returning warm water. These pipes are made out of seawater-resistant material, such as polyethylene. Adequate filters are needed to prevent accumulation of solid particles in the system.

- Heat exchanger (“cooling station”): this allows the cold seawater to cool the recirculating closed loop fresh water used for air conditioning application. Titanium heat exchangers are used for this since titanium combines resistance to salty water with high thermal conductivity.
- Chilled water distribution net: this system carries the cold chilled water into the buildings for air conditioning

2.3.5 History of SWAC

The feasibility of using cold seawater to directly cool buildings has been studied and analyzed for many years. At certain locations, successful installation and operation has occurred. The following is a brief history of seawater air conditioning according to Makai (2004):

In 1975, the US Department of Energy funded a program entitled "Feasibility of a District Cooling System Utilizing Cold Seawater" (Hirshman et al, 1975). Several locations were studied and the two most favorable sites were Miami/Ft. Lauderdale and Honolulu. The study, however, noted that one of the limiting technical factors was the inability to deploy large diameter pipelines to depths of 1500ft. and more. This technical challenge has since been addressed and demonstrated with Makai-designed deep-water pipelines at the Natural Energy Laboratory of Hawaii, Keahole Point, Hawaii.

In 1980, the Naval Material Command at Port Hueneme, California, conducted a study entitled "Sea/Lake Water Air Conditioning at Naval Facilities" (Ciani et al, 1980). Computer models were developed which provided reasonable estimates of the capital cost and energy use of seawater air conditioning systems at Point Mugu, California and Pearl Harbor, Hawaii. The study concluded that at a hypothetical typical Navy facility, a SWAC system will use 80% less energy than conventional A/C, but the capital costs of SWAC systems are 60% greater. The Life Cycle Cost of SWAC at a typical Naval facility would be 25% lower than the life cycle cost of conventional A/C.

In 1986, a joint project between the Canadian government and Purdy's Wharf Development, Ltd. demonstrated the use of ocean water as a source for building cooling to a 350,000 square ft. office complex along the waterfront in Halifax, Nova Scotia. Due to the geographic conditions and annual low water temperatures, a small diameter pipeline was deployed to a depth of less than 100ft. This was a major factor in limiting the overall expense of installing the cooling system. Total investment for this project was $200,000. The project was very successful and savings were identified in the following areas: a saving of $50-60,000 per year in avoided electrical cost, fewer maintenance staff, reduction in fresh water, savings in water treatment, and savings in cooling tower maintenance and replacement. The financial result in terms of a simple payback period was two years. (Building Cooling) Today, Purdy's Wharf continues to utilize successfully an expanded seawater air conditioning system for their waterfront properties.

In 1986, the Natural Energy Laboratory of Hawaii Authority, Keahole Point, Hawaii began the successful utilization of SWAC in their main laboratory building. Deep-water pipelines were already installed to provide cold, nutrient rich, seawater for research purposes in alternate energy and aquaculture. Since a cold water supply was already incorporated into the infrastructure, it was decided to utilize the cold water for cooling. This proved to be a very sound economic decision that resulted in monthly electric savings of $400. Today, the use of SWAC has been expanded to a new administration building and a second laboratory. Estimated monthly saving in electricity is $2000.

In 1990, the US Department of Energy funded a study entitled: "Waikiki District Cooling Utility." The purpose of this brief study was to evaluate whether it was economically and technically feasible to utilize seawater air conditioning as a means to provide cooling to the hotels in Waikiki and to create a Waikiki Cooling Utility. Waikiki was targeted because of the high density of hotels, high electrical consumption and a large demand for air conditioning. It was estimated by Hawaiian Electric Company that of the 107 MW consumed in Waikiki, 51.4 MW were used for air conditioning. This study concluded that economically and technically, Waikiki could be cooled by utilizing seawater air conditioning.

In 1995, Stockholm Energy started supplying properties in central Stockholm with cooling from its new district cooling system. Most of the cooling is produced by using cold water from the Baltic Sea. The temperature of the cooling water leaving the plant is 6°C or lower and the return temperature from the distribution grid is 16°C at high load and a few degrees lower at low load. The district cooling system is designed for a maximum load of 60 MW.

In 1999, the Cornell Lake Source Cooling Project installed a 63in. diameter pipeline into nearby Lake Cayuga. This Makai-designed pipeline was 10,000ft in length and installed to a depth of 250ft. Cold water from this pipeline, at approximately 4°C, provides air conditioning for the Cornell University Campus. This system is capable of providing in excess of 20,000 tons of cooling; the system started operation in mid-2000.

In 2004, the Deep Water Cooling Project for Toronto, Canada, commenced cooling of downtown Toronto with a peak capacity of 58,000 tons. Three 63in. pipelines reach far into Lake

Ontario for both potable and cooling water for the city. Makai assisted in the engineering of this project.

In 2004, Makai completed the design for a 450 ton SWAC system in French Polynesia for a major hotel.

In 2004, a SWAC system design and planning was underway in the Caribbean for 2500 tons AC for a complex of hotels, meeting centers and university.

In 2004, a SWAC system design commenced for very large SWAC systems to service Honolulu, Hawaii.

2.3.6 Makai Ocean Engineering Inc. Offshore Pipelines – Experience

According to Makai (2004) the key cost and risk component of any SWAC system is the offshore pipeline. The lack of a low-cost methodology for the installation of these pipelines prevented SWAC development in the 1970's and 80's. Today, the technology for the successful installation of pipelines to depths of 3000ft. and greater is available. Numerous deep water intake pipelines have been installed – nearly all of the world's successful pipes have been Makai designs. A typical seawater pipeline installed by Makai is shown in Figure 2.4.

Figure 2.4: Typical HDPE pipe been deployed by Makai

All of the deep seawater intake pipelines designed by Makai have used polyethylene as the pipeline material. Polyethylene has significant advantages for these pipelines in that it is inert and will neither corrode nor contaminate the water. Polyethylene lengths are heat fused together to form a long, continuous pipeline with joints that are as strong as the pipeline itself. Polyethylene has excellent strength and flexibility and is buoyant in water. These characteristics allow a great deal of design flexibility and deployment ease. The wall thickness can be varied depending on strength requirements for deployment and operating suction over the lifetime of the pipe.

Makai's approach for a deep water pipe deployment is to minimize the time at-sea. The pipeline is basically designed for the deployment process since this represents the major cost and risk of the installation. The pipeline is assembled complete on shore, launched floating using the pipeline to support all anchors and fastenings, towed to the site, and controllably submerged while carefully monitoring pipe tensions and pressures. The procedure is stable and reversible.
The major risk to the pipeline is during deployment. These are standard marine construction risks and would be covered by the installer's insurance. Once deployed, the likelihood of deep-water failure is very remote.

Makai has been designing and working with deep water pipelines since 1979 and has designed a number of down-the-slope polyethylene intake pipelines and suspended pipelines. In addition, Makai has been involved with a variety of field research programs studying the installation and loading on large diameter pipelines - both deep and shallow. A brief summary of Makai's experience in HDPE (High Density Polyethylene) deep water pipeline design, analysis and deployment is looked into from Makai (2004).

(a) Lake Source Cooling

Makai was selected by Gryphon International Engineering Services and Cornell University to design a 63in. diameter HDPE intake and a 48in. diameter outfall pipeline in Cayuga Lake, NY to provide 20,000 tons of centralized cooling for the university. The intake pipeline is two miles long with an intake at 250ft. depth. The pipeline provides 32,000 gpm of cold water and has a 75-year lifetime. Construction was completed in 1999.

Figure 2.5: 63in. diameter HDPE intake and a 48in. diameter outfall pipeline

(b) Toronto Pipelines

The city of Toronto is now operating a district cooling system with a maximum capacity of 58,000 tons. The cooling is provided with deep lake water from Lake Ontario. Makai assisted in the design of three, nearly 4 miles long, and 63in. diameter intake pipelines for this project.

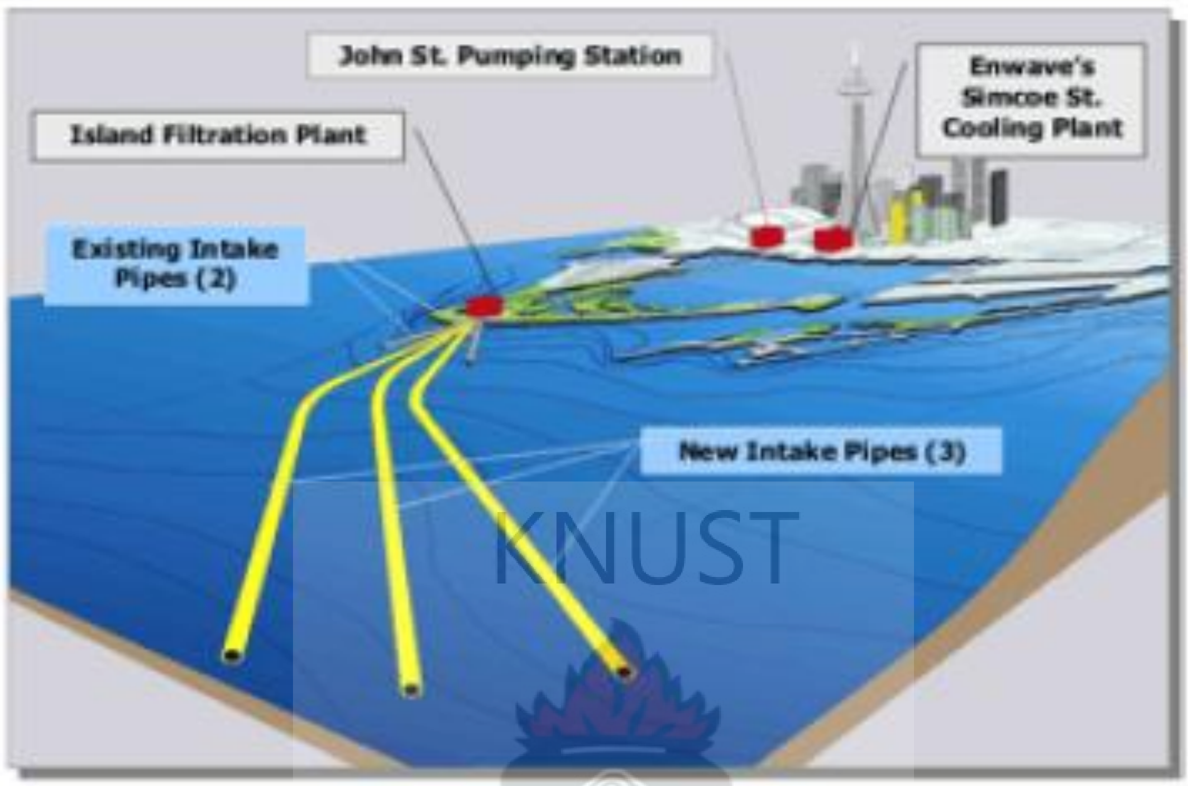

Figure 2.6: Schematic of 4 miles long, and 63in. diameter intake pipelines from Lake Ontario

(c) Indian OTEC Pipeline

Makai provided conceptual designs and design guidance to the National Institute of Ocean Technology (NIOT) in Madras, India, for OTEC intake pipeline and mooring system for a floating OTEC research barge in the Indian Ocean. This pipeline will be 1m in diameter and will provide water from a 1km depth.

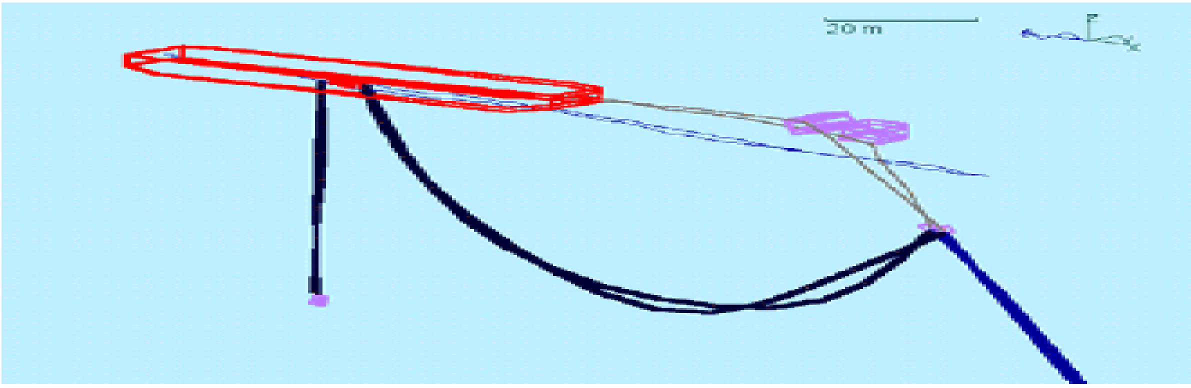

Figure 2.7: Schematic of a 1 meter diameter OTEC pipeline

(d) 55in., 3000ft. deep Pipeline

Makai engineered the main seawater supply source for the Hawaii Ocean Science Technology Park (HOST Park) at Keahole Point, Hawaii. The supply system consisted of a cold-water pipeline (55in. diameter, 3000ft. deep, and two miles long), a 55in. diameter warm water intake pipe, a tunneled shoreline crossing and a shore-based pumping station. The system has the capacity to deliver 27,000 gpm of 4 ^{o}C water and over 40,000 gpm of warm water to the technology park. Makai received a national award from the American Society of Civil Engineers for this project as one of the six most outstanding CE projects in 2003.

Figure 2.8: 55in., 3000ft. deep Pipeline

(e) 40in. Intake Pipeline Design and Installation

In a project with R.M. Towill Corporation, funded by the State of Hawaii and the U.S. Department of Energy, Makai was tasked with the design of a 40in. polyethylene cold water pipe to be used jointly by the Natural Energy Laboratory and the Hawaii Ocean Science and Technology (HOST) Park sites on the Big Island. It is the largest deep-water intake pipeline in the world. This pipe is a larger and more rugged version of the previous MOE 12in. pipe design at NELH and includes a 3000ft. long buoyant section. Makai assisted in the deployment of this pipe to a depth of 2200ft in August 1987. It is currently a main source of water for the Natural Energy Laboratory.

Figure 2.9: 40in. Intake Pipeline Design and Installation

(f) 18in. Cold Water Pipeline

Makai designed and provided construction management for an 18in. down-the-slope cold-water intake at the Natural Energy Laboratory of Hawaii. The goal is to install a reliable, minimal cost, deep-water intake system to 2000ft. This polyethylene design differs from previous NELH pipelines in that the deep water pipe is buoyed approximately 40ft off the bottom on a series of pendants; the deployment was accomplished without major offshore equipment. This pipeline was successfully deployed in October, 1987, and is still operational.

Figure 2.10: 18in. Cold Water Pipeline

(g) Long Operating OTEC Cold Water Pipeline

Makai conceived, designed and managed the construction of an experimental, down-the-slope polyethylene OTEC pipeline, 12in. in diameter, for the State of Hawaii. This one-mile long pipeline has an intake at 2000ft. and utilizes a unique 3000ft. long free-floating catenary section to avoid contact with the steep, rocky bottom. The pipeline was installed in 1981 off Keahole Point, Hawaii. In spite of its "temporary" design life of 2 years, it has survived many major storms including a hurricane and was operational for over twelve years.


Figure 2.11: 12in. diameter OTEC pipeline for the State of Hawaii

Also Makai engineered several portions of the Mini-OTEC project under contract to Dillingham Corp. This project was a full demonstration of Ocean Thermal Energy Conversion (OTEC) and jointly funded by the State of Hawaii, Lockheed, Dillingham and Alfa Laval. Makai designed a 2ft. diameter polyethylene pipe that served not only as an intake pipe from a 2000' depth, but also as the "mooring line" for the 120ft. x 35ft. barge. The initial design for the barge layout, seawater intakes (cold and warm), effluent lines, and pumps was also done by Makai. Makai developed and planned the deployment scheme and participated in the at-sea deployment. On August 2, l979, Mini-OTEC developed 50 kW of power and consumed 40 kW, for a net positive output of l0 kW. This was the first time that a positive output had been achieved from any OTEC facility.

Figure 2.12: 2ft diameter polyethylene pipe OTEC pipeline

2.3.7 Review of Examples of SWAC Applications

Although new in the air conditioning industry, SWAC systems have been successfully installed and operating at a number of locations around the world. The following section provides an overview of current applications of SWAC.

(a) Natural Energy Laboratory, Keahole Point Hawaii

In 1986, the Natural Energy Laboratory of Hawaii Authority, Keahole Point, Hawaii began the successful utilization of SWAC in their main laboratory building.

Several deep-water pipelines have been installed by the state of Hawaii and are operating at Keahole Point on the island of Hawaii, bringing in deep, cold seawater for OTEC and aquaculture research and development. These pipelines range in diameter from 300mm to 1m and have intake depths ranging from 650m to 700m. The 1-meter pipeline has been in continuous operation for nearly five years. A smaller, 300mm pipeline of the same design has been servicing Keahole for more than ten years. From these pipelines, the two main buildings at the National Energy Laboratory in Hawaii are air conditioned with deep cold seawater (Ryzin and Leraand et al, 1992).

(b) Cornell University

The Cornell University lake cold water air condition system started operation in mid-2000.

Although this application does not use seawater, it does follow the same principle idea as deep seawater air conditioning. This principle idea is namely, if a large reservoir of renewably available cold liquid exists (whether seawater or other) near a large air conditioning demand, than the use of that cold liquid for direct air conditioning application warrants investigation. In many cases, it may be more beneficial in economic terms than typical vapour compression air conditioning. In fact, Cornell University does believe the use of deep lake water to meet their air conditioning demands to be economically beneficial.

"The university [Cornell] has embarked on a $55-million project called Lake Source Cooling, which by mid-2000 will be using cold water from the lake to reduce Cornell's energy consumption for air conditioning by 80 percent.

To make this happen, the university is laying underground a huge pipeline in two distinct segments. One segment, 1.6m (5.25 feet) in diameter, will draw water from Cayuga (the longest of the Finger Lakes) from a depth of 76 meters, raise it to a heat-exchange facility on shore and then discharge it into a shallower part of the lake. A separate line of somewhat smaller underground pipes will carry water downhill from Cornell, about three kilometers (two miles) away, to the heat exchanger and back up again. The Cornell water and the lake water will never mix, but heat will be transferred through stainless-steel plates. It's a big undertaking, but something has to be done, the university says. Its present chilled-water system relies heavily on ozone-depleting chlorofluorocarbons, whose manufacture was banned in 1996. Less harmful refrigerants are now available, but deep-water cooling is expected to be more economical in the long run, although the break-even point may be 30 years away." (Ryzin and Leraand et al, 1992). A diagram of the Cornell University, deep lake water cooling project is provided as Figure 2.13.

Figure 2.13: Cornell University Deep Lake Water Cooling Project

(c) Intercontinental Resort and Thalasso Spa Bora Bora, French Polynesia

According to Makai (2004) on May 1st, 2006, the world's first commercial deep seawater air conditioning system opened for business at the Intercontinental Resort and Thalasso Spa Bora Bora, French Polynesia.

The new hotel’s air conditioning pipeline supplies frigid 5°C (41°F) pure seawater from 900m (2,950 feet) deep to eliminate typical air conditioner machinery driven by large electric motors. The seawater passes through a titanium heat exchanger to cool the hotel-wide freshwater cooling network. By using the naturally cold water, the hotel’s 15 kilowatt seawater pump provides cooling that would otherwise consume 300 kilowatts of electricity. The annual electrical savings will be ninety percent.

The Bora Bora pipeline is two kilometers long and 400mm in diameter. The system uses high-density polyethylene pipeline technologies similar to other Makai-designed deep cooling pipelines used by the city of Toronto, Cornell University in New York State, and a government facility at the Natural Energy Laboratory of Hawaii.

CHAPTER 3

TECHNICAL INVESTIGATION

The technical investigation involved the detailed engineering calculation to find the energy demand for both air conditioning with cold seawater and air conditioning with vapour compression system.

3.1 SITES SELECTED FOR INVESTIGATION

Four strategic sites were chosen for this study. The sites were chosen based on their closeness to the source of cold seawater at a depth of 1000m.The sites were also chosen based on economic activities that takes place within them. Keta which has the shortest distance to reach the cold seawater at a depth of 1000m was chosen. Accra, Takoradi and Cape-Coast due to their economic location are also chosen. Thus these four sites were investigated to see their viability for a SWAC project.

(i) Accra

Accra is the capital city of Ghana. It lies within Coordinates 5.55°N 0.25°W. The beaches of the Atlantic coast are popular with visitors and Ghanaians alike. La Pleasure and Kokrobite Beach, just 25km west Accra, are particularly popular at weekends. Among the highlights of Accra are the National Museum, with its splendid display of exhibits that reflect the heritage of Ghana from prehistoric times to modern times; the National Theatre with its distinctive modern architecture, the Centre for National Cultural Centre, Independence Square, the Kwame Nkrumah Mausoleum; the fishing port at James Town and Makola Market.

(ii) Keta

Keta is the capital of the Keta Municipality, one of the 18 administrative districts of the Volta Region. It was carved out of the former Anlo District, which also comprised Akatsi and Ketu

Districts. The district lies within Longitudes 0.30E and 1.05E and Latitudes 5.45N and 6.005N. It is located east of the Volta estuary, about 160km to the east of Accra, off the Accra-Aflao main road. It shares common borders with Akatsi district to the north, Ketu district to the east, South Tongu district to the west and the Gulf of Guinea to the south.

(iii) Cape Coast

Cape Coast is the capital of the Central Region of Ghana and is also the capital city of the Fante (Fanti) people, or (Mfantsefo). It is situated 165 km west of Accra on the Gulf of Guinea. The district lies within Coordinates 05°06′00″N 01°15′00″W.

(iv) Takoradi

Takoradi is one of the twin cities of Sekondi and Takoradi, the capital of the Western Region of Ghana. It is Ghana's third largest city and an industrial and commercial center. The district lies within Coordinates 04°55′00″N 01°46′00″W.

3.2 RANGE OF INVESTIGATION OF AIR CONDITIONING LOAD

According to Ryzin and Leraand (1992) SWAC technology is uneconomical for cooling capacities below 1,000 tons (3,516 kW). This recommendation was adopted to set the lower limit of air conditioning cooling load that was investigated in this project at 1,000 tons. To set the upper limit of cooling load for the investigation, it was noted that the cooling load of many hotels fall below the recommended 1,000 tons lower limit for SWAC. For example, at the usual guest room cooling load of 3 kW in the tropics, a 200 guest room hotel will have total cooling load of 600 kW for the guest rooms. Add twice this amount for the public areas in the hotel (reception, restaurants, bars, shops, gymnasiums, etc.) gives a projected air conditioning load of 1,800 kW. For example, Kumasi Golden Tulip City Hotel which has about 200 guest rooms operates with eight mini chillers, each of cooling capacity 200 kW (total chiller capacity 1,600 kW) in addition to twelve multi-split VRV air conditioning units, each of cooling capacity 28 kW (total 336 kW), which add up to total installed cooling capacity of 1,936 kW (550.6 tons). This is far below the recommended SWAC minimum of 1000 tons.

The conclusion here is that the SWAC technology will not be economical for cases of isolated beach hotels or resorts in Ghana due to their low cooling load. This realization directed the investigation towards coastal communities that could have a mix of hotel(s), dwelling apartments, public areas, etc. Thus, the range of cooling load selected for the SWAC investigation is 1,000 - 13,000 tons (3,516 - 45,719 kW) at intervals of 500 tons. This will represent the air conditioning cooling load of diverse mix of coastal communities.

3.3 SWAC SYTEM ENERGY DEMAND

3.3.1 Factors that Determine SWAC Energy Demand

The SWAC energy demand is the electrical energy consumed by the seawater pump. This energy demand depends on the following factors:

(a) Depth of cold layer of seawater

(b) exit temperature of seawater from the supply pipe line

(c) bathymetry of the sea bed

(d) length of the seawater pipe supply and effluent pipe

(e) characteristics for the seawater pipe such as the material and diameter

(f) air conditioning load and

(g) cold seawater requirements

3.3.2 Depth of Cold Layer of Seawater

The inlet temperature can be determined from the temperature profile for the Gulf of Guinea. The temperature profile for the Gulf of Guinea is shown in Figure 3.1. From the temperature profile the best point to tap the seawater will be 1000m since after this depth the temperature of the seawater almost remains a constant. The temperature at a depth of 1000m is 5.5^{O}C and this will be the inlet temperature of the seawater to the seawater supply pipe.

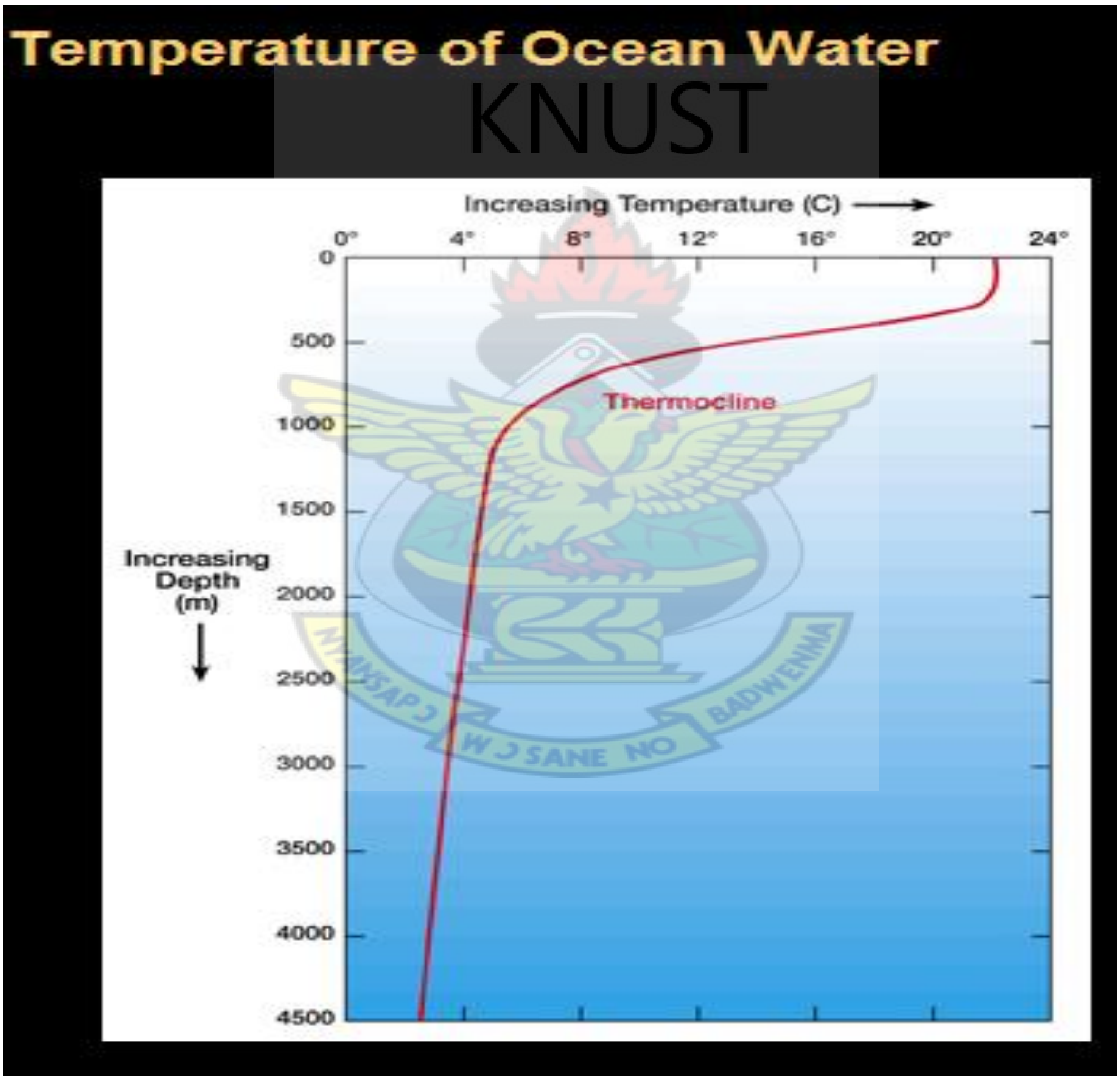

Figure 3.1: Temperature Profile for the Gulf of Guinea

Source: Ghana National Petroleum Company, GNPC

3.3.3 Length of Seawater Supply Pipeline

The distance to cold seawater from the seawater is very complex to determine since it is dependent on the topography of the sea bed.

The distance will however be determined approximately by using the Pythagoras Theorem and applying a factor of safety of 10% to account for modulation of the pipe on the sea bed.

From figure 3.2 the approximate length of the pipeline shall be determined as L. In figure 3.2, D represents the depth of the seawater and H represents the horizontal length from the sea shore to the depth of 1000. H was determined using Figure 3.3.

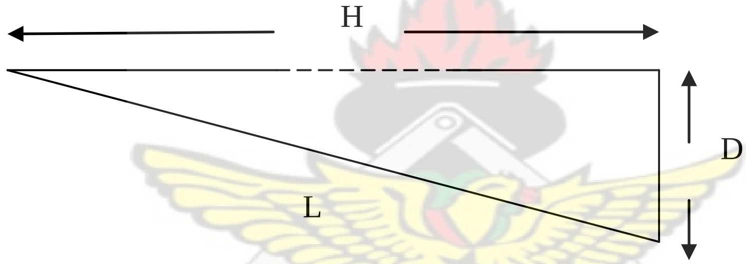

Figure 3.2: Diagram showing how the length of pipeline is approximated

The approximate length, L, projected from Figure 3.2 for all the four sites is shown in Table 3.1.

.Table 3.1: Length of Seawater Pipe for Selected Sites

Item	Location(Site)	Scaled Horizontal length to 1000m depth(km)	Seawater Supply Pipe length(km)	Pipe length corrected by 10%(km)
1	Keta	10.54	10.59	11.65
2	Accra	13.18	13.21	14.53
3	Cape Coast	31.62	31.64	34.80
4	Takoradi	36.01	36.03	39.63

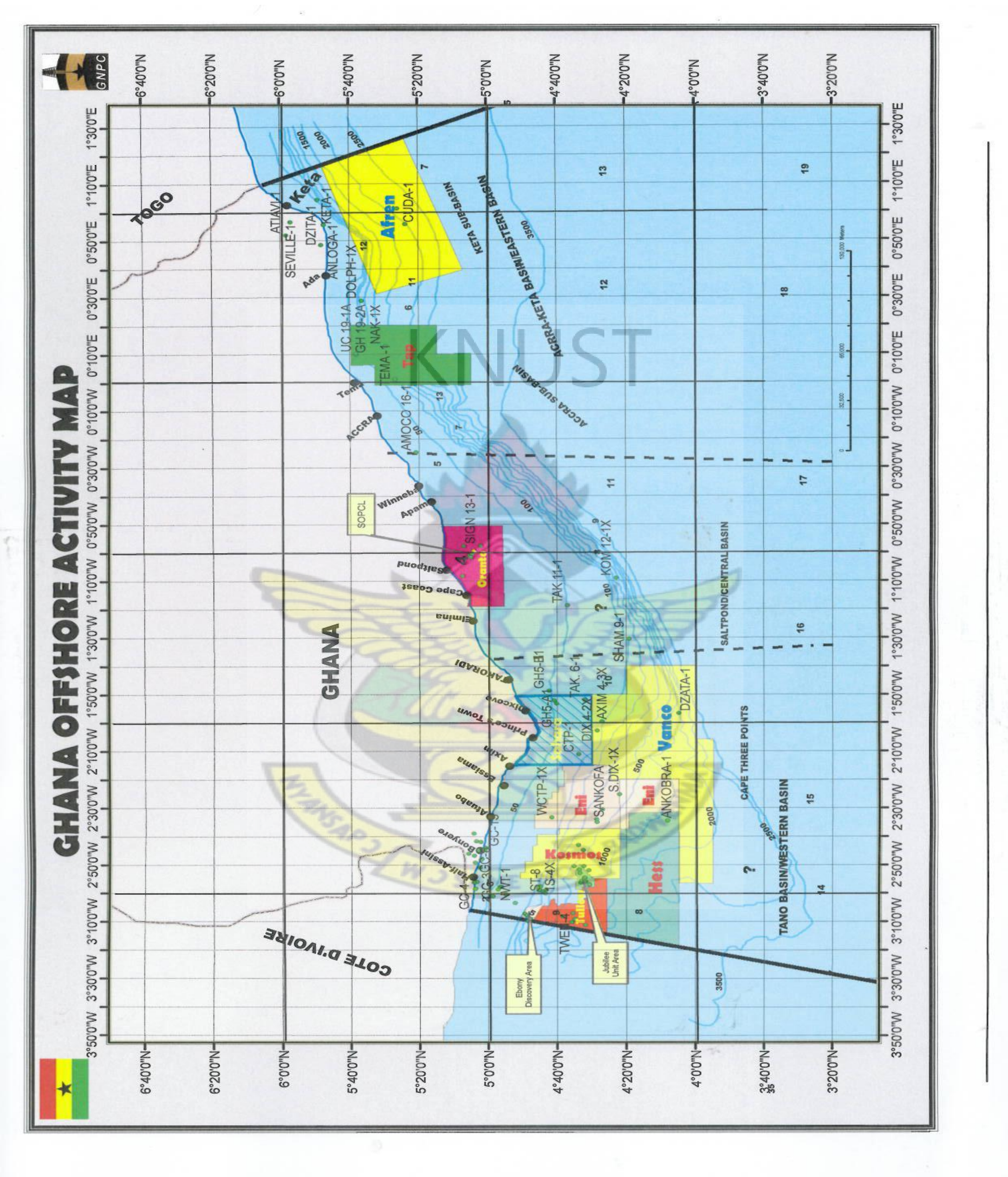

Figure 3.3: Ghana Offshore Activity Map

Source: Ghana National Petroleum Company

3.3.4 Cold seawater requirements

The cold seawater requirements for this study were based on a range of air conditioning load. The air conditioning loads is between 1000 to 13000 TR. With this range of air conditioning load, enough cold seawater must be supplied to the heat exchangers to remove the thermal energy from the fresh water which circulates in the air conditioning system. By definition, 1 TR equals 3.5 kW of thermal energy. From the experience of similar projects, a 5°C increase of the seawater temperature in the heat exchangers can be assumed (Williams et al, 1994).

The heat gain or lost by a fluid at constant pressure is given by

$$\dot{Q} = \dot{m}\, Cp\, \Delta T_s \tag{3.1}$$

Where

$\dot{m}$ is the mass flow rate of seawater, kg/s

$\dot{Q}$ is the air conditioning load, W

ΔT_S is the change in seawater temperature across heat exchanger, °C

Cp is the specific heat capacity of seawater at constant pressure, J/kg K

Thus the mass flow rate

$$\dot{m} = \frac{Air\ Conditioning\ Load, \dot{Q}}{c_p \times \Delta T} = \frac{\dot{Q}}{c_p \times \Delta T_s} \tag{3.2}$$

With the mass flow rate known the volumetric flow rate (Q) can be determined:

$$Q = \frac{mass\ flow\ rate\ of\ seawater, \dot{m}}{density\ of\ seawater, \rho} = \frac{\dot{m}}{\rho} \tag{3.3}$$

3.3.5 Cold Seawater Pipe Design

In undertaking the cold seawater pipe design for the project some assumptions were made. These include:

- ✓ the flow is incompressible so as to satisfy continuity
- ✓ the internal flow is completely bounded. This means that the flow velocity is the same over the entire length of pipe

- Seawater Properties

The properties of seawater were read at an average temperature of 14^{o}C and a salinity of 35g/kg from the Seawater Property Table (Appendix 1). These are the average temperature and salinity for the Gulf Of Guinea. The average temperature was based on the temperature profile for the Gulf of Guinea as shown in figure 1 .The properties are:

- ✓ Specific Heat Capacity at Constant Pressure, c_p = 3997.39 J/ kg K
- ✓ Density of Seawater, ρ = 1026.15 kg/m^3
- ✓ Prandtl Number, Pr = 8.574
- ✓ Kinematic Viscosity, ν = 12.362E-7 m^2/s
- ✓ Dynamic Viscosity,μ = 1.269E-3 kg/ms
- ✓ Thermal Conductivity, k_s = 0.5931 W/mK

All calculations were based on these seawater property values.

- Pipe material

In many older designs for ocean pipelines the basic pipeline material used in the petroleum industry, for example, is steel. External and internal coatings were specified for corrosion protection and hydraulic flow enhancement. Newer technologies make things easier.

Plastic pipe, now available in large diameters, has several important advantages over steel pipe in this application. It will not corrode in seawater and it has good thermal insulation properties. Plastic pipe, now available in large diameters, has several important advantages over steel pipe in this application. It will not corrode in seawater and it has good thermal insulation properties. Two candidate materials are, high density polyethylene pipe (HDPE) and fiberglass reinforced resin pipe (FRP) (Williams et al, 1994).

The best candidate plastic pipe material is high density polyethylene pipe (HDPE) due to its unique characteristics and low cost compared to FRP. HDPE has successfully been deployed and used for almost all seawater air-conditioning projects. HDPE lengths are heat fused together to form a long, continuous pipeline with joints that are as strong as the pipeline itself. HDPE has excellent strength and flexibility and is buoyant in water. These characteristics allow a great deal of design flexibility and deployment ease. Pipes as large as 63" in diameter are commercially available.

Due to its proven stable and economic use, for almost all SWAC projects worldwide, HDPE shall be used for this study.

- Pipe diameter

Makai Ocean Engineering Company has carried out a research which clearly shows that the 1m diameter HDPE pipeline, if used exclusively for air conditioning purposes, could supply 5,000 tons of air conditioning load (Ryzin and Leraand et al, 1992). Based on this research this pipe diameter shall be used as the reference diameter of pipe for this study. This pipe diameter shall be accepted for this study since a large pipe diameter shall ensure minimum temperature rise; this is a major requirement for a SWAC project.

Interplast Ghana manufactures HDPE pipes. The data on their HDPE pipes is shown in Appendix (4).

- Average Velocity of seawater in pipe

Since the volumetric flow rate and diameter of the seawater pipe is known then the average velocity, V, of the seawater may be written as

$$V = \frac{volumtric\ \ flow\ rate\ of\ \ seawater\ \left(\dot{Q}\right)}{cross\,\mathrm{sec}tional\ area\ of\ seawater\ pipe, A_s} = \frac{\dot{Q}}{A_s} = \frac{4\dot{Q}}{\pi D_i^{\ 2}} \qquad (3.4)$$

where D_i is the inside diameter of seawater pipe, m

The average velocity of seawater in the pipe was calculated using the Matlab Program for the determination of the seawater exit temperature from the seawater supply line. The program is shown in Appendix (2).

According to Jones (1997) the recommended maximum velocity of cold liquid flow in pipes is 2.4m/s. Thus pipe sizes were determined ensuring that velocities do not exceed the maximum of 2.4m/s and the need to reduce temperature rise in the pipe due to frictional effects. Based on these criteria an HDPE pipe with an external diameter of 800mm was used for air conditioning loads from 1000 to 5000 TR, and an external diameter of 1200mm was used for air conditioning loads from 5500 TR to 13000.

3.3.6 Temperature Rise of Cold Seawater

The heat gains of the cold seawater as it travels in the delivery pipe which runs from the cold deep sea through rising seawater temperature to the shore will determine the entry temperature of the cold seawater in the SWAC heat exchanger. The heat gain will be due to frictional effects encountered as it flows through the pipe and the heat transfer from the relatively warmer outside seawater to the cold seawater in the pipe.

The temperature rise as a result of frictional effects can be determined after calculating the pumping power.

- **Temperature Rise as a Result of Heat Transfer across the Seawater Pipe**

The temperature of the seawater outside the pipe changes as a function of depth as shown in figure 3.1, and the depth also changes as a function of the length with respect to the pipe. Therefore the temperature is an unknown function of both depth and length of pipe. To calculate the heat loss over the length of the pipe, an average temperature outside the pipe was assumed. A better solution was arrived at by assuming an average temperature over various sections of the pipe and calculating the heat loss and temperature change for each section. The heat loss and temperature change in each section was then added. The pipe was divided into ten sections to permit for a better segmenting of the seawater pipeline. This is the method employed.

The heat transferred across the seawater supply line is absorbed by the seawater inside the pipe. Thus

$$\dot{m}Cp(\Delta T) = U\left[T_S - \left(T_{ch} + \frac{\Delta T}{2}\right)\right] \tag{3.5}$$

Where

$\dot{m}$ is the mass flow rate of the seawater in the pipeline, kg/s

Cp is the specific heat capacity of seawater at constant pressure, J/kgK

U is the overall heat transfer coefficient of seawater supply line, W/K

ΔT_h is the change in temperature of the seawater in the supply pipeline as a result of temperature rise of seawater due to supply pipe wall transmission, ^oC

T_{ch} is the average temperature of seawater in the supply pipeline, ^oC

T_s is the average temperature on the external surface of the supply pipeline, ^oC

Making the change in temperature, ΔT, of the seawater in the seawater supply line as the subject then

$$\Delta T_{ch,h} = \left[\left(\frac{2U(Ts - T_{ch})}{2\dot{m}C_P + U} \right) \right] \tag{3.6}$$

With the change in temperature determined then the final exit temperature of the seawater at the exit of the seawater supply pipeline as a result of wall transmission load can then be determined as

$$T_{chh} = 5.5 + \Delta T_{ch,h} \tag{3.7}$$

From equation (3.6) for the change in temperature of the seawater in the supply line to be determined, then the overall heat transfer coefficient, U, must be determined.

The overall heat transfer coefficient is given as

$$U = \frac{1}{R} \tag{3.8}$$

Where

R is the thermal resistance for heat transfer through the seawater supply pipeline

The overall thermal resistance,

$$R = \frac{1}{h_i A_i} + \frac{In(\frac{r_o}{r_i})}{2\pi L k_p} + \frac{1}{h_o A_o} \qquad (3.9)$$

Where

h_i is the convective heat transfer coefficient of the seawater flowing inside the supply line, kg/m^{3}s

h_o is the convective heat transfer coefficient of the seawater flowing outside the supply line, kg/m^{3}s

r_i is the inside radius of the seawater supply line, m

r_o is the outside radius of the seawater supply line, m

k_p is the thermal conductivity of the seawater supply pipeline, thus HDPE pipe, W/mK

The convective heat transfer coefficient of the seawater flowing outside the supply line can be determined from the equation

$$h_o = \frac{k_s \times Nu_o}{D_o} \qquad (3.10)$$

Where

k_s is the thermal conductivity of seawater

Nu_o is the Nusselt Number at outside pipe conditions

D_o is the outside pipe diameter, m

From equation (3.10) the convective heat transfer coefficient can only be determined after determining the Nusselt number at outside pipe conditions.

The Nusselt number at outside pipe conditions is determined by the Dittus-Boulter equation as

$$Nu_o = 0.023 \times (\mathrm{Re}^{0.8} \times \mathrm{Pr}^{0.4}) \qquad (3.11)$$

Where

Re_o is the Reynold's number at outside pipe conditions

Pr is the Prandtl number at outside pipe conditions

The convective heat transfer coefficient of the seawater, flowing inside the supply pipeline, can be determined just like that of the outside conditions but now at inside pipe conditions.

With the convective heat transfer coefficients determined, then the overall thermal resistance, R, is determined. Thus the overall heat transfer coefficient can be determined.

The detailed calculation was done using a Matlab program shown in Appendix 2.

The result for all the four sites is shown in Tables 3.2 to 3.5.

Table 3.2 : Temperature Rise of Seawater due to Supply Pipe Wall Transmission, Keta

Item	**Load(TOR)**	**m(kg/s)**	**V(m/s)**	**$\Delta T_{ch,h}$(^{O}C)**
External Diameter of Pipe : 800mm				
1	1000	175.1	0.4	6.0
2	1500	262.7	0.6	4.5
3	2000	350.2	0.8	3.6
4	2500	437.8	1.0	3.0
5	3000	525.3	1.2	2.6
6	3500	612.9	1.4	2.2
7	4000	700.5	1.6	2.0
8	4500	788.0	1.8	1.8
9	5000	875.6	2.0	1.6
External Diameter of Pipe : 1200mm				
10	5500	966.1	1.0	1.5
11	6000	1054.0	1.1	1.4
12	6500	1142.0	1.2	1.3
13	7000	1230.0	1.2	1.2
14	7500	1317.0	1.3	1.1
15	8000	1405.0	1.4	1.1
16	8500	1493.0	1.5	1.0
17	9000	1581.0	1.6	0.9
18	9500	1669.0	1.7	0.9
19	10000	1757.0	1.8	0.9
20	10500	1839.0	1.9	0.8
21	11000	1926.0	2.0	0.8
22	11500	2014.0	2.0	0.7
23	12000	2101.0	2.1	0.7
24	12500	2189.0	2.2	0.7
25	13000	2277.0	2.3	0.7

Table 3.3 : Temperature Rise of Seawater due to Supply Pipe Wall Transmission, Accra

Item	Load(TOR)	m(kg/s)	V(m/s)	$\Delta T_{ch,h}(^{O}C)$
External Diameter of Pipe : 800mm				
1	1000	175.1	0.4	7.0
2	1500	262.7	0.6	5.3
3	2000	350.2	0.8	4.3
4	2500	437.8	1.0	3.6
5	3000	525.3	1.2	3.1
6	3500	612.9	1.4	2.7
7	4000	700.5	1.6	2.4
8	4500	788.0	1.8	2.2
9	5000	875.6	2.0	2.0
External Diameter of Pipe : 1200mm				
10	5500	966.1	1.0	1.8
11	6000	1054.0	1.1	1.7
12	6500	1142.0	1.2	1.6
13	7000	1230.0	1.2	1.5
14	7500	1317.0	1.3	1.4
15	8000	1405.0	1.4	1.3
16	8500	1493.0	1.5	1.2
17	9000	1581.0	1.6	1.1
18	9500	1669.0	1.7	1.1
19	10000	1757.0	1.8	1.0
20	10500	1839.0	1.9	1.0
21	11000	1926.0	2.0	1.0
22	11500	2014.0	2.0	0.9
23	12000	2101.0	2.1	0.9
24	12500	2189.0	2.2	0.8
25	13000	2277.0	2.3	0.8

Table 3.4 : Temperature Rise of Seawater due to Supply Pipe Wall Transmission, Cape Coast

Item	**Load(TOR)**	**m(kg/s)**	**V(m/s)**	**$\Delta T_{ch,h}(^{O}C)$**
External Diameter of Pipe : 800mm				
1	1000	175.1	0.4	11.3
2	1500	262.7	0.6	9.5
3	2000	350.2	0.8	8.1
4	2500	437.8	1.0	7.1
5	3000	525.3	1.2	6.2
6	3500	612.9	1.4	5.6
7	4000	700.5	1.6	5.1
8	4500	788.0	1.8	4.6
9	5000	875.6	2.0	4.3
External Diameter of Pipe : 1200mm				
10	5500	966.1	1.0	3.9
11	6000	1054.0	1.1	3.6
12	6500	1142.0	1.2	3.4
13	7000	1230.0	1.2	3.2
14	7500	1317.0	1.3	3.2
15	8000	1405.0	1.4	2.8
16	8500	1493.0	1.5	2.7
17	9000	1581.0	1.6	2.6
18	9500	1669.0	1.7	2.4
19	10000	1757.0	1.8	2.3
20	10500	1839.0	1.9	2.3
21	11000	1926.0	2.0	2.2
22	11500	2014.0	2.0	2.1
23	12000	2101.0	2.1	2.0
24	12500	2189.0	2.2	1.9
25	13000	2277.0	2.3	1.9

Table 3.5 : Temperature Rise of Seawater due to Supply Pipe Wall Transmission, Takoradi

Item	Load(TOR)	m(kg/s)	V(m/s)	$\Delta T_{ch,h}$(^{O}C)
External Diameter of Pipe : 800mm				
1	1000	175.1	0.4	11.9
2	1500	262.7	0.6	10.1
3	2000	350.2	0.8	8.7
4	2500	437.8	1.0	7.7
5	3000	525.3	1.2	6.8
6	3500	612.9	1.4	6.2
7	4000	700.5	1.6	5.6
8	4500	788.0	1.8	5.1
9	5000	875.6	2.0	4.7
External Diameter of Pipe : 1200mm				
10	5500	966.1	1.0	4.4
11	6000	1054.0	1.1	4.1
12	6500	1142.0	1.2	3.8
13	7000	1230.0	1.2	3.6
14	7500	1317.0	1.3	3.4
15	8000	1405.0	1.4	3.2
16	8500	1493.0	1.5	3.1
17	9000	1581.0	1.6	2.9
18	9500	1669.0	1.7	2.8
19	10000	1757.0	1.8	2.7
20	10500	1839.0	1.9	2.6
21	11000	1926.0	2.0	2.4
22	11500	2014.0	2.0	2.4
23	12000	2101.0	2.1	2.3
24	12500	2189.0	2.2	2.2
25	13000	2277.0	2.3	2.1

3.3.7 Calculation of Cold Seawater Pumping Power Requirement

The power requirement to pump the cold seawater can be calculated as follows

$$P = \frac{\rho\ g\ hw V\ A}{\eta} \tag{3.12}$$

Where

P is pumping power required, kW

ρ is the density of seawater, kg/m^3

h_w is the pumping head, m

V is the average flow velocity, m/s

A is the cross sectional area of HDPE pipe, m^2

g is acceleration due to gravity, m/s^2

η = pump efficiency: a pump efficiency of 70% was assumed

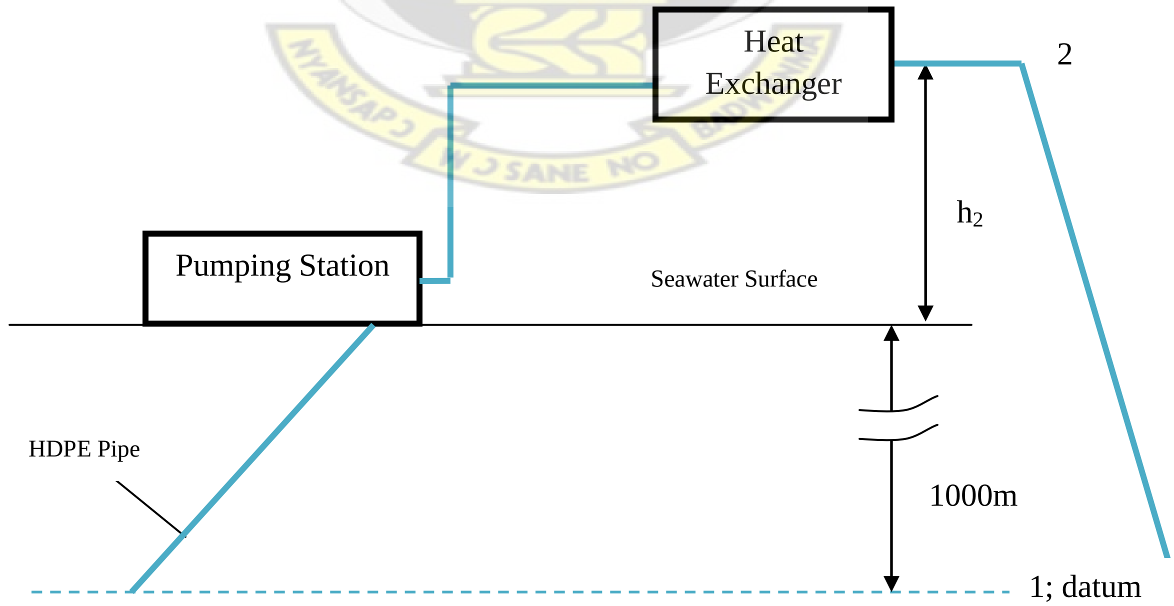

Figure 3.4: Schematic of SWAC

From figure 3.4 the energy equation taken between the inlet of the seawater supply pipeline (taken as point 1) and the exit of the heat exchanger (taken as point 2) can be written as

$$\left(\frac{P_1}{\rho g}+\frac{U_1^{\;2}}{\rho g}+Z_1\right)-\left(\frac{P_2}{\rho g}+\frac{U_2^{\;2}}{\rho g}+Z_2\right)=h_L-h_W \qquad (3.13)$$

Where

P_1 is the absolute pressure of seawater at the inlet of the seawater supply pipeline, Pa.

From figure 3.4

$$\left(\frac{P_1}{\rho g}\right)=\left(\frac{P_{atm}}{\rho g}\right)+1000 \qquad (3.14)$$

P_2 is the absolute pressure of seawater at the outlet of the seawater supply pipeline, Pa

Since beyond the point 2 the seawater can flow gradually under gravity and with some kinetic energy it will be assumed that the pressure at point 2 will be equal to atmospheric pressure.

U_1 is the initial velocity of seawater at the inlet of the seawater supply pipeline, m/s. Thus U_1 equals zero (0).

U_2 is the final velocity of seawater at the outlet of the heat exchanger, m/s

Z_1 is the depth at the datum line, m; Z_1= 0

Z_2 is the sum of depth of seawater from the datum line and the heat exchanger installation height, m

The installation height of the heat exchanger, h_2, is assumed to be 3m.

Thus $Z_2 = 1000 + 3 = 1003$m

h_L is the head losses due to major and minor losses, m

hw is the pumping head, m

Substituting variables and working out the pumping head, then the pumping head,

$$h_W = h_L + \left(\frac{U_2^{\ 2}}{\rho g} + 3 \right) \tag{3.15}$$

The head losses are as a result of frictional effects (h_{LF}), other specific losses due to bends, valves etc. and losses at the heat exchanger (h_{LX}).

The Head loss as a result of frictional effects, $h_{LF,}$ and the other specific losses due to bends, valves etc. shall be defined as h_{LFS}:

$$h_{LFS} = \frac{U_1^{\ 2}}{2g} \left(\frac{fL}{D} + \sum K \right) \tag{3.16}$$

Where

f is the frictional factor

K is the shape factor which accounts for the specific losses

L is the total length of the pipeline

D is the diameter of pipe, m

The frictional factor shall be determined using the Reynold's number and the relative roughness of the pipe.

The specific losses were written in terms of an equivalent length, say L_{ek}, so that the specific losses are equated to frictional loss over an additional length. Thus an equivalent length may normally be used where

$L_e = L + L_{ek}$, and hence the head loss

$$h_{LFS} = \frac{U_1^{\ 2}}{2g}\left(\frac{fL_e}{D}\right) \qquad (3.17)$$

The specific loss which includes the losses in the valves, bends etc. was assumed to be 50% of the total loss due to frictional effects.

The frictional factor, f, is calculated differently depending on whether or not flow is laminar or turbulent. Pipe flow may be either laminar or turbulent depending on the Reynolds number,

$$\mathrm{Re} = \frac{\rho VD}{\mu} \qquad (3.18)$$

Where

ρ is the density of seawater at average temperature flow, kg/m^3

V is the average velocity of flow, m/s

D is the diameter of pipe (characteristic length of pipe),

μ is the kinematic viscosity of fluid at average temperature,

If Re < 2300 then the flow is assumed to be laminar. Re > 2300 is assumed to be turbulent. The frictional loss factor, f, for laminar flow is computed analytically with the following equation

$$f = \frac{64}{\mathrm{Re}} \qquad (3.19)$$

With the frictional factor determined then the frictional head loss is determined.

From equation (3.18) for laminar flow, the frictional loss in the pipe is dependent on Reynolds number only, not on the roughness of the pipe surface.

For fully developed turbulent flow, the friction factor is determined experimentally. The frictional head loss in this case is known to depend on the Reynolds number and the pipe roughness.

The relative roughness, roughness of the HDPE divided by the diameter of pipe, must be determined from the pipe characteristics. The friction factor can then be determined from the Moody chart (Appendix 6).

The maximum allowable pressure drop assumed across the heat exchanger was 100kPa (approximately 10m head) based on recommendation by Alfa Laval, one of the world's leading manufacturers of heat exchangers.

An Excel sheet was used to quicken the calculation of the pumping power and is shown in Appendix (3).

- **Temperature Rise due to Frictional Effects**

With the pumping head determined the temperature rise due to frictional effects ($T_{ch,f}$) can then be determined. This temperature rise was added to that caused by heat transfer to obtain the total temperature rise for the cold seawater ashore on arrival to the heat exchanger ($T_{ch,t}$). This temperature rise due to frictional effects was however not determined for sites under loads which were not feasible for SWAC projects due to high temperature rise of seawater only by heat transfer across the pipe. This included Takoradi, from 1000 TR to 13000 TR, and Cape Coast from 1000 to 11500 TR.

An Excel sheet was used to quicken the calculation and is shown in Appendix (3).

The pumping power for Keta, Accra and Cape Coast are shown in Table 3.6 for all four sites.

The effect of wall transmission and pumping work on final cold seawater temperature is shown in Tables 3.7 to 3.9 for Keta, Accra and Cape Coast respectively. The initial temperature of cold seawater at a depth of 1km, $5.5^{O}C$, was added to arrive at this final temperature.

The only sites that qualified for SWAC projects were Keta from air conditioning load of 5000 TR and Accra from air conditioning load of 5500 TR.

Table 3.6: Pumping Power for Keta, Accra and Cape Coast

Item	Location	Keta	Accra	Cape Coast
	Load(TOR)	**Pumping Power (kW)**		
External Diameter of Pipe : 800mm				
1	4000	926.04	N/A	N/A
2	4500	1275.1	N/A	N/A
3	5000	1706.37	2053.5	N/A
External Diameter of Pipe : 1200mm				
4	5500	451.16	508.93	N/A
5	6000	550.25	626.44	N/A
6	6500	665.03	763.19	N/A
7	7000	716.28	822	N/A
8	7500	853.21	985.97	N/A
9	8000	1009.6	1173.74	N/A
10	8500	1186.2	1386.32	N/A
11	9000	1384.39	1625.42	N/A
12	9500	1605.6	1892.72	N/A
13	10000	1851.16	2189.91	N/A
14	10500	2115.6	2510.49	N/A
15	11000	2215.7	2870.29	N/A
16	11500	2522.34	3001.44	N/A
17	12000	2856.6	3407.46	7284.62
18	12500	3222.4	3852.14	8284.5
19	13000	3619.9	4335.67	9373.76

Table 3.7 : Effect of Wall Transmission and Pumping Work on Final Seawater Tempaerture, Keta

Item	Load(TOR)	$\Delta T_{ch,h}$(^{O}C)	$\Delta T_{ch,f}$(^{O}C)	$T_{ch,t}$(^{O}C)
External Diameter of Pipe: 800mm				
1	4000	2.0	0.23	7.72
2	4500	1.8	0.28	7.57
3	5000	1.6	0.34	7.47
External Diameter of Pipe: 1200mm				
4	5500	1.5	0.08	7.07
5	6000	1.4	0.09	6.97
6	6500	1.3	0.10	6.88
7	7000	1.2	0.10	6.79
8	7500	1.1	0.11	6.73
9	8000	1.1	0.13	6.68
10	8500	1.0	0.14	6.64
11	9000	0.9	0.15	6.59
12	9500	0.9	0.17	6.56
13	10000	0.9	0.18	6.53
14	10500	0.8	0.20	6.51
15	11000	0.8	0.21	6.49
16	11500	0.7	0.22	6.46
17	12000	0.7	0.24	6.45
18	12500	0.7	0.26	6.45
19	13000	0.7	0.28	6.44

Table 3.8 : Effect of Wall Transmission and Pumping Work on Final Seawater Tempaerture, Accra

Item	Load(TOR)	$\Delta T_{ch,h}(^{O}C)$	$\Delta T_{ch,f}(^{O}C)$	$T_{ch,t}(^{O}C)$
External Diameter of Pipe: 800mm				
1	5000	2.10	0.41	7.87
External Diameter of Pipe: 1200mm				
2	5500	1.8	0.09	7.39
3	6000	1.7	0.10	7.26
4	6500	1.6	0.12	7.17
5	7000	1.5	0.12	7.06
6	7500	1.4	0.13	7.0
7	8000	1.3	0.15	6.93
8	8500	1.2	0.16	6.87
9	9000	1.1	0.18	6.82
10	9500	1.1	0.21	6.80
11	10000	1.0	0.22	6.75
12	10500	1.0	0.24	6.73
13	11000	1.0	0.26	6.71
14	11500	0.9	0.26	6.67
15	12000	0.9	0.28	6.65
16	12500	0.8	0.31	6.65
17	13000	0.8	0.33	6.64

Table 3.9 : Effect of Wall Transmission and Pumping Work on Final Seawater Tempaerture, Cape Coast

Item	Load(TOR)	$\Delta T_{ch,h}(^{O}C)$	$\Delta T_{ch,f}(^{O}C)$	$T_{ch,t}(^{O}C)$
External Diameter of Pipe: 1200mm				
24	12000	2.0	0.61	8.05
25	12500	1.9	0.66	8.03
26	13000	1.9	0.72	8.03

3.3.8 Technical Specification of Heat Exchanger

Heat exchangers are available in a variety of types, sizes and materials. There are standard models available by many manufactures.

The type of heat exchanger considered in this work is a plate and frame heat exchanger. These are made of an assembly of pressed metal plates, aligned on or secured to metal frames or bars, and a containment or cover. Gaskets are set in the outside groups to contain the fluids and to direct the fluid flow distribution. The hot and cold fluids can flow in the grooves in the plate in opposite sides of the plates.

There are many advantages to plate and frame heat exchangers. The principle advantage is the accessibility of the heat exchange surfaces for cleaning. Other advantages of plate and frame heat exchangers over tube and shell exchangers are as follows:

1. Less surface area required for heat transfer, resulting in cost savings. The close spacing of the grooves between adjacent plates results in a geometry similar to that of very small tubes.
2. Less space required for the unit(s) as a result of the reduced surface area.
3. Better heat exchanger effectiveness.
4. The number of plates can be easily increased or decreased depending on future load changes

The only disadvantage over other types is the large number of surfaces sealed by gaskets. Due to the use of elastomeric materials in the gaskets, the internal temperature and pressures of the heat exchanger are generally limited to 300 ^{o}F and 400 psi (William et al, 1994).

- **Material**

For the SWAC system, the best choice of heat exchanger is a modular titanium plate heat exchanger with gasket joints in a counter-flow configuration (Elsafty and Saeid et al, 2009).

Titanium plate heat exchangers are very durable and well proven for heat transfer involving seawater; life expectancies are in excess of 20 years. Titanium flat plate and also shell and tube heat exchangers are in operation at the Natural Energy Laboratory of Hawaii. These titanium heat exchangers have been the subject of long term corrosion and biofouling experiments for OTEC research. With deep seawater, there are neither biofouling nor corrosion problems associated with titanium (Leraand and Ryzin et al, 1995).

- **Technical Specification**

The type of heat exchanger, the material used, the heat transfer required, and the temperature differences of the fluids on both the “hot” and “cold” sides determine the size and cost of the unit or units.

The size is determined by the surface area required over which the fluids must flow. The design and selection of heat exchangers is facilitated by employing the overall heat transfer coefficient, U, in the fundamental heat transfer relation:

$$\dot{Q} = U_h \ A_h \ (LMTD) \tag{3.20}$$

where, $\dot{Q}$ is the heat transfer rate (air conditioning load), U_h is the overall heat transfer coefficient, A_h is the effective heat transfer surface area, and LMTD is the log mean temperature difference. For a single pass heat exchanger, using figure 3.4 the log mean temperature difference, LMTD, is defined as:

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{In(\frac{\Delta T_2}{\Delta T_1})} \quad (3.21)$$

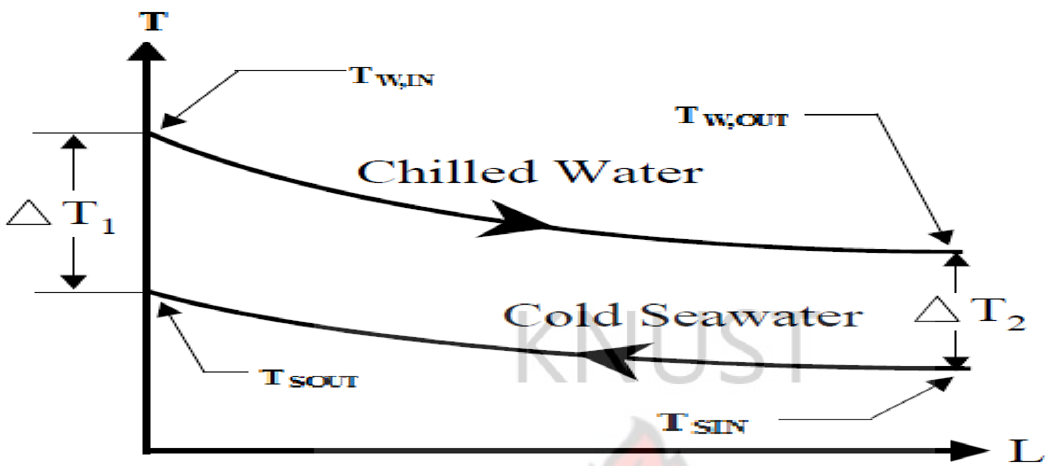

Figure 3.5: Temperature Profile in a Counter Flow Heat Exchanger

Where

$T_{W,IN}$ is the temperature of the chilled water at the inlet of the heat exchanger, ^oC

$T_{W,OUT}$ is the temperature of the chilled water at the outlet of the heat exchanger, ^oC

T_{SOUT} is the temperature of the seawater at the inlet of the heat exchanger, ^oC

T_{SIN} is the temperature of the seawater at the inlet of the heat exchanger, ^oC

- **Selection of Suitable SWAC Sites**

In Section (2.2) it was shown that, for the cooling coil in the conditioned space to remove humidity from the space, the surface temperature of the cooling coil should not be higher than the dew point temperature which is 14^{o}C at the indoor design condition of 25^{o}C db and 50% relative humidity.

From the example of Table (2.1) fan coil units the recommended minimum temperature change across the indoor units for economical heat exchanger surfaces is 5°C. Therefore at 14°C outlet temperature from the indoor unit the highest fresh chilled water entry temperature is 9°C. Allowing for a recommended temperature difference of 1.5°C between the entering cold seawater and the leaving chilled water; this sets the highest cold seawater temperature at heat exchanger entry at 7.5°C. Therefore Takoradi and Cape coast do not qualify for SWAC even up to 13,000 TR. Accra qualifies for SWAC application from 5,500 TR to 13000 TR. Keta also qualifies for SWAC application from 5,000 TR to 13,000 TR.

- **Surface Area of Heat Exchanger**

As can be seen, there are many variables involved with the heat exchanger, and any one can be varied to achieve changes in other variables. For example, a larger LMTD can be offset by a smaller heat transfer area, to achieve the same heat transfer rate. And, to achieve a larger LMTD, the temperatures of the fluids entering and exiting the heat exchanger must be modified. The variations are endless, in order to achieve the same heat transfer rate.

The heat exchanger surface area from equation (3.19) is

$$A_h = \frac{\dot{Q}}{U_h(LMTD)} \tag{3.22}$$

The coefficient of heat transfer or operating U value for titanium heat exchanger can be expected to be about 7166.4 W/m^2°C or 1262 BTU/ft^{2}hr^{O}F (William et al, 1994).

The heat transfer area for feasible site locations, under specific air conditioning loads, is shown in Tables 3.10 and 3.11.

Table 3.10: Required Surface Area of Heat Exchanger for Keta

Location	**KETA**		
Item	**Load(TOR)**	**LMTD(^{O}C)**	**A(m^2)**
	External Diameter of Pipe : 800mm		
1	5000	1.5	1596.0
	External Diameter of Pipe : 1200mm		
2	5500	1.9	1391.8
3	6000	2.0	1443.5
4	6500	2.1	1497.4
5	7000	2.2	1546.9
6	7500	2.3	1613.6
7	8000	2.3	1684.1
8	8500	2.4	1759.0
9	9000	2.4	1823.9
10	9500	2.4	1901.5
11	10000	2.5	1977.3
12	10500	2.5	2059.5
13	11000	2.5	2140.4
14	11500	2.5	2211.2
15	12000	2.6	2298.3
16	12500	2.6	2394.1
17	13000	2.6	2480.1

Table 3.11: Required Surface Area of Heat Exchanger for Accra

Location	ACCRA		
Item	**Load(TOR)**	**LMTD(^{O}C)**	**A(m^2)**
	External Diameter of Pipe : 1200mm		
1	5500	1.6	1668.4
2	6000	1.7	1684.1
3	6500	1.8	1734.7
4	7000	1.9	1762.2
5	7500	2.0	1831.5
6	8000	2.1	1887.5
7	8500	2.1	1949.0
8	9000	2.2	2016.3
9	9500	2.2	2109.0
10	10000	2.3	2170.6
11	10500	2.3	2259.1
12	11000	2.3	2346.0
13	11500	2.3	2410.5
14	12000	2.4	2493.9
15	12500	2.4	2597.8
16	13000	2.4	2690.3

3.4 ENERGY DEMAND OF CONVENTIONAL WATER CHILLER

Shown in Table 3.12 is sample cooling capacity against input power for a water-cooled chiller produced by York International of USA which is one of the world's leading manufacturers of air conditioning equipment. The performance ratings given are actual measured values from test run conducted in the factory. The result shows that the energy efficiency of the chiller increase with the size (capacity) which is often the case with most engineering equipment. Rated performance of chillers from other leading manufacturers such as Trane and Carrier, also US companies, shows that chillers of corresponding capacities have the same energy efficiencies practically speaking. This is not surprising since the energy rating is one of the yardsticks for the company's competitiveness. For this study, a conservative energy demand of 0.8 kW/TR is used which includes the energy demand of the condenser cooling water pump. The energy consumption for all the load ranges is shown in Table 3.13.

Table 3.12: Cooling Capacity against Input Power for York Chiller

MODEL	LEAVING CHILLED WATER TEMP., ^{O}C	CONDENSER LEAVING WATER TEMP., $30^{O}C$		kW/TR (1TR=3.516kW)
		COOLING kW	INPUT kW	
YCWM B 60	6	58	14.6	0.89
	7	60	14.8	0.87
	8	63	14.9	0.83
	10	67	15.2	0.80
222 717 SC	6	1769	305	0.61
YK N3 P5 -P3	6	7016	1495	0.75
	6	8152	1529	0.66

Source: York International

Table 3.13: Energy Consumption of Conventional Chillers

Item	**Load**	**kW/TR**	**Total Electrical Energy Demand (kW)**
1	5000	0.8	4000
2	5500	0.8	4400
3	6000	0.8	4800
4	6500	0.8	5200
5	7000	0.8	5600
6	7500	0.8	6000
7	8000	0.8	6400
8	8500	0.8	6800
9	9000	0.8	7200
9	9500	0.8	7600
10	10000	0.8	8000

CHAPTER 4

ECONOMIC ANALYSIS

4.1 INTRODUCTION

The technical analysis carried out in Chapter 3 has shown that SWAC technology may be applied for Keta and Accra area for air conditioning loads from 4,000 TR and 5,000 TR respectively. In this chapter, the economics of using the SWAC technology for these areas is compared with the use of conventional chiller.

In this analysis, it is assumed that the weather pattern in the country does not vary significantly throughout the year which can be said to be a typical trend for tropical regions. It is also assumed here that the cooling demand is 100% for about 8hrs during the day whilst for the remaining 16hrs, it is 50%. This is equivalent to saying that the air conditioning system works at 100% for 2/3 of any period under consideration such as day, month or year. This variation of air conditioning demand which has been assumed will affect the air conditioning systems being compared in the same way.

With the exception of the HDPE pipe which is locally manufactured in Ghana by Interplast, all the other components of the two systems and the related services can only be obtained offshore. Even obtaining the cost of HDPE pipes from Interplast does not lessen the difficulty in estimating installed cost of the sea water delivery pipe work as the task of deploying the pipe at sea including moorings, intake and other accessories constitute a complex and expensive component of the total cost. For this reason, the data and methodology used according to DeGarmo, Sullivan and Bontadelli (1989) has been used for the economic comparison of the two systems.

4.2 CAPITAL COST OF SYSTEMS

There are three key components of capital cost of the SWAC system. These are the cost of seawater pipe (delivery and effluent), heat exchanger and the seawater pump. For the conventional chiller system, the key capital cost are the chiller, cooling tower and the condenser cooling water pump and piping connection.

According to DeGarmo, Sullivan and Bontadelli (1989),

"In many instances, the first cost of an equipment varies in a power-law manner with its capacity. The physical basis for this power law and a rough estimate of the exponent can be obtained from the simplistic understanding that the cost is proportional to the amount of material in the item and that the capacity is proportional to the volume of the item.

Thus the cost C, is expected to be related to the capacity S as,

$$C = Cr\left(\frac{S}{S_r}\right)^m \tag{4.1}$$

C: Cost of item

C_r: Cost of reference item

S: Capacity of Item

S_r: Capacity of reference item

m: Exponent of cost size relationship"

4.2.1 CAPITAL COST OF CONVENTIONAL CHILLER SYSTEM

(i) Chiller

Conventional chillers of which the mechanical compression type is the most popular are manufactured in a wide range of capacities from as low as 1 TR to as high as 2500 TR or higher.

For this study, it has been used as a combination of three large chillers of capacities of 2500 TR, 2000 TR and 1000TR. Using a base cost of $809,610 for a 5,000 kW (1,429) TR chiller in 2006 (Ref[18]) the cost of the three chillers above were estimated using Eq. 4.1 as follows:

1000TR: $$C = 809610 \times \left(\frac{1000}{1429}\right)^{0.66.} = \$639{,}668$$

2000TR: $$C = 809610\left(\frac{2000}{1429}\right)^{.66.} = \$1{,}010{,}728$$

2500TR: $$C = 809610\left(\frac{2500}{1429}\right)^{.66.} = \$1{,}171{,}103$$

The cost of the entire conventional chiller system for the various air conditioning loads was calculated using equation (4.1). The results are shown in Table 4.1.

Table 4.1: Total System Cost for Conventional Chiller

Load (TR)	Mechanical Vapour Compression Chiller Unit				Cooling Tower m=1; Cr=72000; Sr=3600kW			Cooling Water Pump and Distribution m=0.7; Cr=160000; Sr=1m^3/s			Total System Cost ($)
	Quantity of Chiller, 1000 TR	Quantity of Chiller, 2000 TR	Quantity of Chiller, 2500 TR	Total Cost of Chiller $	Required Capacity, kW	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	Required capacity, m^3/s	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	
5000	0	0	2	2,342,206	21500	256,088	352,696	0.9	43,725	60,219	2,755,121
5500	1	1	1	2,821,499	23650	274,017	377,389	0.9	43,725	60,219	3,259,107
6000	0	3	0	3,032,184	25800	291,479	401,439	1.0	43,863	60,410	3,494,033
6500	0	2	1	3,192,559	27950	308,524	424,913	1.1	43,989	60,583	3,678,055
7000	0	1	2	3,352,934	30100	325,192	447,869	1.2	44,104	60,741	3,861,545
7500	1	2	1	3,832,227	32250	341,518	470,354	1.3	44,210	60,887	4,363,469
8000	0	4	0	4,042,912	34400	357,531	492,409	1.4	44,308	61,023	4,596,343
8500	0	3	1	4,203,287	36550	373,257	514,066	1.5	44,400	61,149	4,778,503
9000	0	2	2	4,363,662	38700	388,716	535,357	1.5	44,400	61,149	4,960,169
9500	1	3	1	4,842,955	40850	403,928	556,308	1.6	44,486	61,268	5,460,531
10000	0	5	0	5,053,640	43000	418,910	576,941	1.7	44,567	61,379	5,691,961
10500	0	4	1	5,214,015	45150	433,675	597,277	1.8	44643.3	61484.7	5,872,777
11000	1	5	0	5,693,308	47300	448,238	617,334	1.89	44708.7	61574.8	6,372,217
11500	1	4	1	5,853,683	49450	462,611	637,129	1.97	44764.3	61651.4	6,552,463
12000	0	6	0	6,064,368	51600	476,803	656,675	2.06	44824.3	61734.1	6,782,777
12500	0	5	1	6,224,743	53750	490,825	675,986	2.14	44875.6	61804.7	6,962,534
13000	1	6	0	6,704,036	55900	504,685	695,075	2.23	44931.1	61881.1	7,460,992

4.2.2 CAPITAL COST OF SWAC SYSTEM

(i) Seawater pipeline

The all inclusive installed cost of seawater pipe is quoted in Leraand and Ryzin (1995) as \$15/1bm (pound mass) in 1999. Applying a consumer price Index (CPI) values of 167 in 1999 and 230 in 2012 (Source; US Department of Labour Bureau of Labour Statistics), this works out to \$45.45/kg of installed seawater pipe today (2012). The total length of seawater delivery and effluent pipe line is 12.65 km for Keta and 15.53 km for Accra. The mass of material in these pipe lengths is 1.95 x 10^6 kg for Keta and 2.4 x 10^6 kg for Accra.

Applying the cost rate of $45.45/kg, this gives the present value of the seawater pipe line as

Keta: 1.95 x 10^6 x 45.45 = $ 88,627,500 (outside diameter of pipe is 1200mm)

Keta: 1.23 x 10^6 x 45.45 = $ 55,903,500 (outside diameter of pipe is 800mm)

Accra: 2.4 x 10^6 x 45.45 = $ 109,080,000 (outside diameter of pipe is 1200mm)

Given the same pipe diameter, the above installed cost of seawater pipe will be constant for each location at the varying air conditioning loads.

(ii) Heat Exchanger and Seawater Pump

The cost of the heat exchanger and the seawater pump for the various SWAC loads are calculated using Eq.4.1 and tabulated in Tables 4.2 and 4.3 for Keta and Accra respectively.

Table 4.2 Capital Cost of SWAC System for Keta

Load (TR)	Seawater Pipeline $	Heat Exchanger m=0.71;Cr=42100;Sr=100m^2			Seawater Pump m=0.03;Cr=47000;Sr=10m^3/min			Total System Cost ($)
		Required Area, m^2	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	Required capacity, m^3/min	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	
5000	55,903,500	1596	3,009,092	4,144,258	51.2	49,360	67,981	60,115,739
5500	88,627,500	1392	2,730,351	3,760,364	56.5	49,506	68,182	92,456,046
6000	88,627,500	1444	2,802,024	3,859,075	61.6	49,635	68,359	92,554,934
6500	88,627,500	1497	2,875,919	3,960,847	66.8	49,756	68,526	92,656,872
7000	88,627,500	1547	2,943,119	4,053,397	72.0	49,868	68,680	92,749,577
7500	88,627,500	1614	3,032,647	4,176,699	77.0	49,968	68,818	92,873,017
8000	88,627,500	1684	3,126,109	4,305,419	82.2	50,066	68,953	93,001,873
8500	88,627,500	1759	3,224,234	4,440,562	87.3	50,157	69,078	93,137,140
9000	88,627,500	1824	3,308,162	4,556,151	92.4	50,242	69,196	93,252,847
9500	88,627,500	1902	3,407,561	4,693,048	97.6	50,325	69,309	93,389,858
10000	88,627,500	1977	3,503,417	4,825,065	102.7	50,402	69,415	93,521,981
10500	88,627,500	2059	3,606,198	4,966,620	107.5	50,471	69,511	93,663,631
11000	88,627,500	2140	3,706,185	5,104,327	112.6	50,541	69,608	93,801,435
11500	88,627,500	2211	3,792,890	5,223,740	117.8	50,609	69,701	93,920,941
12000	88,627,500	2298	3,898,359	5,368,997	122.8	50,673	69,790	94,066,287
12500	88,627,500	2394	4,013,001	5,526,887	128.0	50,736	69,875	94,224,263
13000	88,627,500	2480	4,114,870	5,667,186	133.1	50,796	69,958	94,364,644

Table 4.3: Capital Cost of SWAC System for Accra

Load (TR)	Seawater Pipeline $	Heat Exchanger m=0.71;Cr=42100;Sr=100m^2			Seawater Pump m=0.03;Cr=47000;Sr=10m^3/min			Total System Cost ($)
		Required Area, m^2	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	Required capacity, m^3/min	Cost,1999 CPI=167 $	Cost,2012 CPI=230 $	
5500	109,080,000	1668	3,105,402	4,276,901	56.5	49,506	68,182	113,425,083
6000	109,080,000	1684	3,126,109	4,305,419	61.6	49,635	68,359	113,453,778
6500	109,080,000	1735	3,192,529	4,396,896	66.8	49,756	68,526	113,545,422
7000	109,080,000	1762	3,228,398	4,446,296	72.0	49,868	68,680	113,594,976
7500	109,080,000	1831	3,317,943	4,569,622	77.0	49,968	68,818	113,718,440
8000	109,080,000	1887	3,389,703	4,668,453	82.2	50,066	68,953	113,817,406
8500	109,080,000	1949	3,467,724	4,775,908	87.3	50,157	69,078	113,924,986
9000	109,080,000	2016	3,552,342	4,892,447	92.4	50,242	69,196	114,041,642
9500	109,080,000	2109	3,667,502	5,051,050	97.6	50,325	69,309	114,200,360
10000	109,080,000	2171	3,743,321	5,155,472	102.7	50,402	69,415	114,304,888
10500	109,080,000	2259	3,850,993	5,303,763	107.5	50,471	69,511	114,453,274
11000	109,080,000	2346	3,955,599	5,447,831	112.6	50,541	69,608	114,597,439
11500	109,080,000	2411	4,032,547	5,553,807	117.8	50,609	69,701	114,703,508
12000	109,080,000	2494	4,131,114	5,689,558	122.8	50,673	69,790	114,839,348
12500	109,080,000	2598	4,252,601	5,856,875	128.0	50,736	69,875	115,006,751
13000	109,080,000	2690	4,359,522	6,004,133	133.1	50,796	69,958	115,154,091

4.3 ENERGY COST OF SYSTEMS

The current electricity tariff rate in Ghana is GH¢0.4120/kWh which at prevailing exchange rate corresponds to $0.2168/kWh. Using results from Chapter 3, the yearly energy cost of the two systems is set up in Table 4.4.

Table 4.4: Yearly Energy Cost of Conventional Chiller and SWAC System

Load, TR	Yearly Peak Hours, Hrs	Conventional Chiller		SWAC System, Keta		SWAC System, Accra	
		Peak Energy Demand, kW	Yearly Energy Cost, $	Peak Energy Demand, kW	Yearly Energy Cost, $	Peak Energy Demand, kW	Yearly Energy Cost, $
External Diameter of Pipe : 800mm							
5000	5840	4,000	**5,064,448**	1706	**2,159,987**	N/A	**N/A**
External Diameter of Pipe : 1200mm							
5500	5840	4,400	**5,570,895**	451.16	**571,017**	508.93	**644,451**
6000	5840	4,800	**6,077,338**	550.25	**696,361**	626.44	**792,586**
6500	5840	5,200	**6,583,782**	665.03	**841,964**	763.19	**966,043**
7000	5840	5,600	**7,090,227**	716.28	**906,536**	822	**1,040,744**
7500	5840	6,000	**7,596,672**	853.21	**1,079,994**	985.97	**1,248,386**
8000	5840	6,400	**8,103,117**	1009.6	**1,278,267**	1173.74	**1,486,415**
8500	5840	6,800	**8,609,562**	1186.2	**1,501,609**	1386.32	**1,754,831**
9000	5840	7,200	**9,116,006**	1384.39	**1,752,299**	1625.42	**2,057,432**
9500	5840	7,600	**9,622,451**	1605.6	**2,033,376**	1892.72	**2,396,750**
10000	5840	8,000	**10,128,896**	1851.16	**2,343,573**	2189.91	**2,772,785**
10500	5840	8400	**10,635,341**	2115.6	**2,679,093**	2510.49	**3,177,941**
11000	5840	8800	**11,141,786**	2215.7	**2,805,704**	2870.29	**3,633,741**
11500	5840	9200	**11,648,230**	2522.34	**3,193,134**	3001.44	**3,800,159**
12000	5840	9600	**12,154,675**	2856.6	**3,617,282**	3407.46	**4,314,226**
12500	5840	10000	**12,661,120**	3222.4	**4,079,413**	3852.14	**4,877,063**
13000	5840	10400	**13,167,565**	3619.9	**4,583,325**	4335.67	**5,489,862**

4.4 LIFE CYCLE COST ANALYSIS

The two systems are compared by the ownership cost of each cost. Calculation of the ownership cost is based on the following assumption;

i. The Total capital cost of each system takes place in the first year.

ii. Energy costs are considered to be a uniform series of payment over one year interval which are calculated to present values at interest rate of 7% prevailing on suppliers market

iii. Both cooling systems have a life cycle of 30 years.

iv. Maintenance and other operational costs are assumed to be equal for both systems and therefore it is not considered here.

The present value P of a uniform series of yearly payment of an amount A over N years at interest rate i is

$$P = A\left(\frac{[1+i]^N - 1}{i[1+i]^N}\right) \tag{4.2}$$

Shown in Table 4.5 and 4.6 is the result of cost of ownership which is further illustrated by the diagram in Figure 4.1.

Table 4.5: Total Ownership Cost for Keta

Load (TR)	Conventional Chiller System				SWAC System			
	Capital Cost, $	Energy Cost		Total Ownership Cost, $	Capital Cost, $	Energy Cost		Total Ownership Cost, $
		Yearly Energy Cost, $	Energy Cost, Present Value, $			Yearly Energy Cost, $	Energy Cost, Present Value, $	
5000	2,755,121	5,064,448	62,844,944	**65,600,065**	60,115,739	2,159,987	26,803,368	**86,919,107**
5500	3,259,107	5,570,895	69,129,465	**72,388,573**	92,456,046	571,017	7,085,773	**99,541,819**
6000	3,494,033	6,077,338	75,413,938	**78,907,970**	92,554,934	696,361	8,641,172	**101,196,107**
6500	3,678,055	6,583,782	81,698,422	**85,376,477**	92,656,872	841,964	10,447,966	**103,104,838**
7000	3,861,545	7,090,227	87,982,919	**91,844,464**	92,749,577	906,536	11,249,243	**103,998,819**
7500	4,363,469	7,596,672	94,267,416	**98,630,884**	92,873,017	1,079,994	13,401,690	**106,274,707**
8000	4,596,343	8,103,117	100,551,913	**105,148,256**	93,001,873	1,278,267	15,862,068	**108,863,940**
8500	4,778,503	8,609,562	106,836,409	**111,614,912**	93,137,140	1,501,609	18,633,528	**111,770,668**
9000	4,960,169	9,116,006	113,120,894	**118,081,063**	93,252,847	1,752,299	21,744,350	**114,997,197**
9500	5,460,531	9,622,451	119,405,391	**124,865,922**	93,389,858	2,033,376	25,232,247	**118,622,104**
10000	5,691,961	10,128,896	125,689,888	**131,381,848**	93,521,981	2,343,573	29,081,494	**122,603,475**
10500	5,872,777	10,635,341	131,974,384	**137,847,162**	93,663,631	2,679,093	33,244,975	**126,908,606**
11000	6,372,217	11,141,786	138,258,881	**144,631,099**	93,801,435	2,805,704	34,816,096	**128,617,531**
11500	6,552,463	11,648,230	144,543,366	**151,095,829**	93,920,941	3,193,134	39,623,731	**133,544,672**
12000	6,782,777	12,154,675	150,827,863	**157,610,640**	94,066,287	3,617,282	44,887,001	**138,953,288**
12500	6,962,534	12,661,120	157,112,360	**164,074,893**	94,224,263	4,079,413	50,621,604	**144,845,867**
13000	7,460,992	13,167,565	163,396,856	**170,857,848**	94,364,644	4,583,325	56,874,669	**151,239,312**

Table 4.6: Total Ownership Cost for Accra

Load (TR)	Conventional Chiller System				SWAC System			
	Capital Cost, $	Energy Cost		Total Ownership Cost, $	Capital Cost, $	Energy Cost		Total Ownership Cost, $
		Yearly Energy Cost, $	Energy Cost, Present Value, $			Yearly Energy Cost, $	Energy Cost, Present Value, $	
5500	3,259,107	5,570,895	69,129,465	**72,388,573**	113,425,083	644,451	7,997,019	**121,422,102**
6000	3,494,033	6,077,338	75,413,938	**78,907,970**	113,453,778	792,586	9,835,232	**123,289,011**
6500	3,678,055	6,583,782	81,698,422	**85,376,477**	113,545,422	966,043	11,987,667	**125,533,089**
7000	3,861,545	7,090,227	87,982,919	**91,844,464**	113,594,976	1,040,744	12,914,635	**126,509,611**
7500	4,363,469	7,596,672	94,267,416	**98,630,884**	113,718,440	1,248,386	15,491,273	**129,209,713**
8000	4,596,343	8,103,117	100,551,913	**105,148,256**	113,817,406	1,486,415	18,444,985	**132,262,391**
8500	4,778,503	8,609,562	106,836,409	**111,614,912**	113,924,986	1,754,831	21,775,770	**135,700,756**
9000	4,960,169	9,116,006	113,120,894	**118,081,063**	114,041,642	2,057,432	25,530,758	**139,572,401**
9500	5,460,531	9,622,451	119,405,391	**124,865,922**	114,200,360	2,396,750	29,741,369	**143,941,729**
10000	5,691,961	10,128,896	125,689,888	**131,381,848**	114,304,888	2,772,785	34,407,603	**148,712,491**
10500	5,872,777	10,635,341	131,974,384	**137,847,162**	114,453,274	3,177,941	39,435,201	**153,888,475**
11000	6,372,217	11,141,786	138,258,881	**144,631,099**	114,597,439	3,633,741	45,091,242	**159,688,681**
11500	6,552,463	11,648,230	144,543,366	**151,095,829**	114,703,508	3,800,159	47,156,330	**161,859,838**
12000	6,782,777	12,154,675	150,827,863	**157,610,640**	114,839,348	4,314,226	53,535,408	**168,374,756**
12500	6,962,534	12,661,120	157,112,360	**164,074,893**	115,006,751	4,877,063	60,519,676	**175,526,426**
13000	7,460,992	13,167,565	163,396,856	**170,857,848**	115,154,091	5,489,862	68,123,924	**183,278,015**

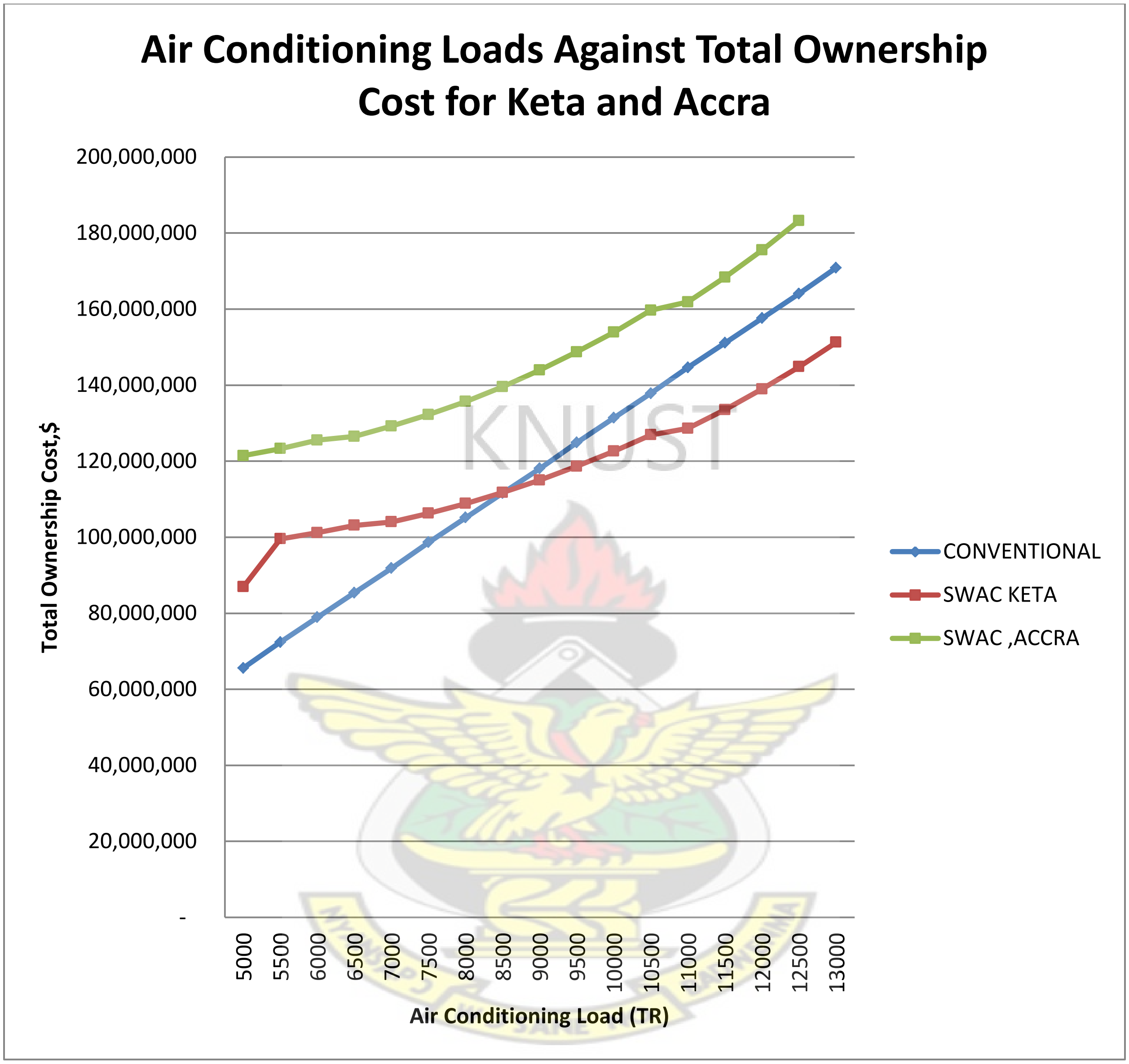

Figure 4.1: Air Conditioning Loads against Total Ownership Cost for Keta and Accra

CHAPTER 5

CONCLUSION AND RECOMMENDATION

The technical investigation carried out in this thesis has shown that SWAC technology is not feasible for Cape Coast and Takoradi areas for up to 13,000 tons cooling load due to the long distance from the shore to the 1km depth where the cold seawater available can be reached. Keta and Accra on the other hand involve relatively shorter distances and application of SWAC technology begins to look technically feasible after 5000 TR for Keta and 5500 TR for Accra. On the economic side, SWAC technology appears to be non-profitable for Accra area up to 13,000 TR. For Keta, on the other hand, SWAC overtakes the conventional chiller systems in terms of lower ownership cost at SWAC load above 8,500 TR.

The study shows that any region of Ghana coastline which has a shorter pipe travel than Keta to reach the 1km depth has a very good potential for application of SWAC technology.

The investigation also shows that the SWAC technology will not suit isolated beach resorts or hotels as these have air conditioning cooling load that in most cases fall short of 1,000 tons.

One area of SWAC application that was not part of the scope of this thesis but which has the potential to be feasible is the case of Hybrid cold seawater/Absorption system. When combined with solar collectors to drive the absorption plant, this could completely overtake conventional air conditioning systems in the tropical regions, especially in areas where the slope of the seabed is step to reduce the distance of pipe travel to the cold seawater depth.

The project encountered many challenges that need to be addressed in future work on this topic to make the result more reliable. The first challenge was the difficulty and by that the inaccuracy of extracting from a poorly scaled map of the Ghana coastline the distances from the shore to the point where the sea depth reached 1km; the contours were so jammed together on the map that reading errors was inevitable. At the early stage of the project, several attempts were made to obtain access to E-topo (an electronic map) at the Oceanography and Fisheries Department of the University of Ghana from which a more accurate reading of distances from the shore into the sea could be made but this was unsuccessful. Another important challenge is in obtaining reliable prices for the components of the SWAC system as a result of which this work had to rely on price rates of other researchers from many years back which could not be confirmed to be realistic today even though Consumer Price Indices was used to adjust them to today's value.

This research has shown that the potential for economical application of SWAC technology for coastal communities is good. The potential looks the more attractive at higher SWAC load beyond the 13,000 TR limit investigated in this work and can be rightly said that SWAC technology can completely revolutionalize air conditioning systems used in the coastal dwellings by overtaking the conventional air conditioning systems in the near future. In conclusion, it is recommended that further work be carried out that will address the challenges encountered in this work. Also, hybrid systems involving the use of cold seawater with absorption system is one area that must be included in future work.

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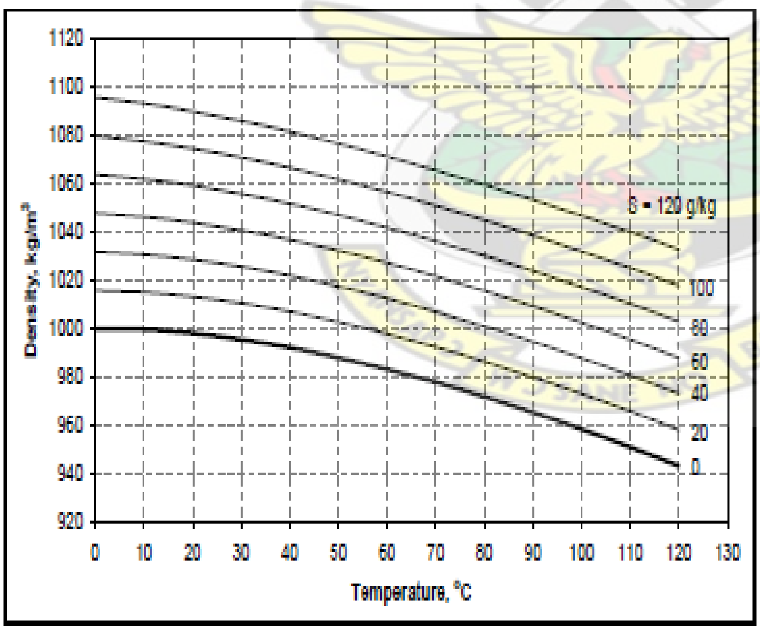
APPENDIX

APPENDIX 1: PROPERTIES OF SEA WATER

Table A.1: Density of Seawater at Varying Temperatures and Salinity

Density, kg/m^3

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	999.8	1007.9	1016.0	1024.0	1032.0	1040.0	1048.0	1056.1	1064.1	1072.1	1080.1	1088.1	1096.2
10	999.7	1007.4	1015.2	1023.0	1030.9	1038.7	1046.6	1054.4	1062.2	1070.1	1077.9	1085.7	1093.6
20	998.2	1005.7	1013.4	1021.1	1028.8	1036.5	1044.1	1051.8	1059.5	1067.2	1074.9	1082.6	1090.3
30	995.7	1003.1	1010.7	1018.2	1025.8	1033.4	1040.9	1048.5	1056.1	1063.6	1071.2	1078.7	1086.3
40	992.2	999.7	1007.1	1014.6	1022.1	1029.5	1037.0	1044.5	1052.0	1059.4	1066.9	1074.4	1081.8
50	988.0	995.5	1002.9	1010.3	1017.7	1025.1	1032.5	1039.9	1047.3	1054.7	1062.1	1069.5	1076.9
60	983.2	990.6	998.0	1005.3	1012.7	1020.0	1027.4	1034.7	1042.1	1049.5	1056.8	1064.2	1071.5
70	977.8	985.1	992.5	999.8	1007.1	1014.5	1021.8	1029.1	1036.5	1043.8	1051.2	1058.5	1065.8
80	971.8	979.1	986.5	993.8	1001.1	1008.5	1015.8	1023.1	1030.5	1037.8	1045.1	1052.5	1059.8
90	965.3	972.6	980.0	987.3	994.7	1002.0	1009.4	1016.8	1024.1	1031.5	1038.8	1046.2	1053.5
100	958.4	965.7	973.1	980.5	987.9	995.2	1002.6	1010.0	1017.4	1024.8	1032.2	1039.6	1047.0
110	950.9	958.3	965.8	973.2	980.6	988.1	995.5	1003.0	1010.4	1017.8	1025.3	1032.7	1040.2
120	943.1	950.6	958.1	965.6	973.1	980.6	988.1	995.6	1003.1	1010.6	1018.1	1025.6	1033.1



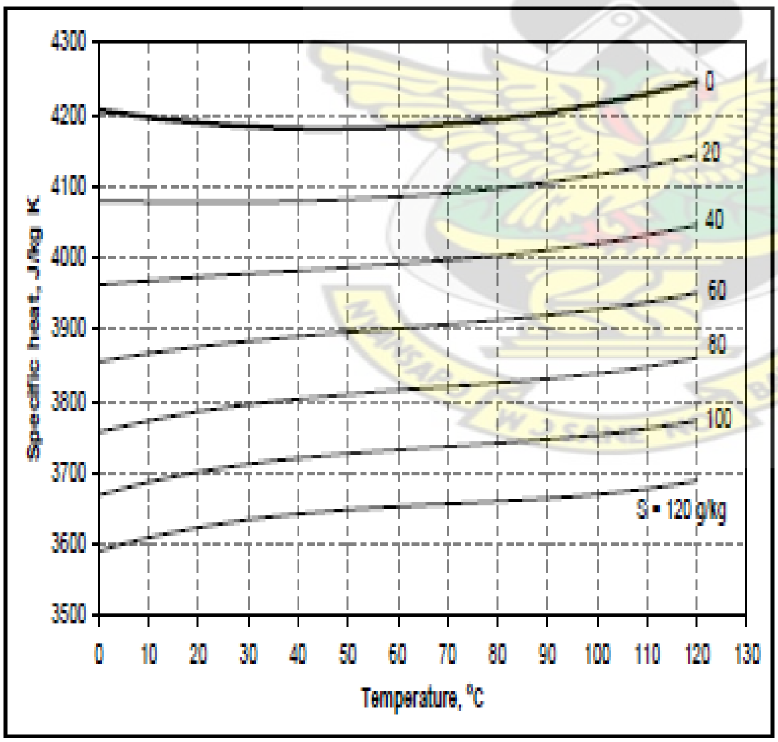
Accuracy ±0.1%

Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

Table A.2: Specific Heat at Constant Pressure of Seawater at Varying Temperatures and Salinity

Specific heat at constant pressure, J/kg K

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	4206.8	4142.1	4079.9	4020.1	3962.7	3907.8	3855.3	3805.2	3757.6	3712.4	3669.7	3629.3	3591.5
10	4196.7	4136.7	4078.8	4022.8	3968.9	3916.9	3867.1	3819.2	3773.3	3729.5	3687.7	3647.9	3610.1
20	4189.1	4132.8	4078.2	4025.3	3974.1	3924.5	3876.6	3830.4	3785.9	3743.0	3701.8	3662.3	3624.5
30	4183.9	4130.5	4078.5	4027.8	3978.6	3930.8	3884.4	3839.4	3795.8	3753.6	3712.7	3673.3	3635.3
40	4181.0	4129.7	4079.6	4030.7	3982.9	3936.4	3891.0	3846.7	3803.7	3761.8	3721.1	3681.6	3643.2
50	4180.6	4130.8	4081.9	4034.1	3987.3	3941.5	3896.6	3852.9	3810.1	3768.3	3727.5	3687.8	3649.0
60	4182.7	4133.7	4085.5	4038.3	3992.0	3946.5	3902.0	3858.3	3815.5	3773.7	3732.7	3692.6	3653.4
70	4187.1	4138.5	4090.6	4043.6	3997.3	3951.9	3907.4	3863.6	3820.6	3778.5	3737.2	3696.7	3657.0
80	4194.0	4145.3	4097.3	4050.1	4003.7	3958.1	3913.3	3869.2	3825.9	3783.5	3741.7	3700.8	3660.7
90	4203.4	4154.2	4105.9	4058.3	4011.5	3965.4	3920.2	3875.7	3832.0	3789.1	3746.9	3705.6	3665.0
100	4215.2	4165.4	4116.4	4068.2	4020.9	3974.3	3928.5	3883.6	3839.4	3796.0	3753.5	3711.7	3670.8
110	4229.4	4178.8	4129.1	4080.2	4032.2	3985.1	3938.7	3893.3	3848.6	3804.9	3761.9	3719.9	3678.6
120	4246.1	4194.7	4144.2	4094.6	4045.9	3998.2	3951.3	3905.4	3860.3	3816.2	3773.0	3730.7	3689.4



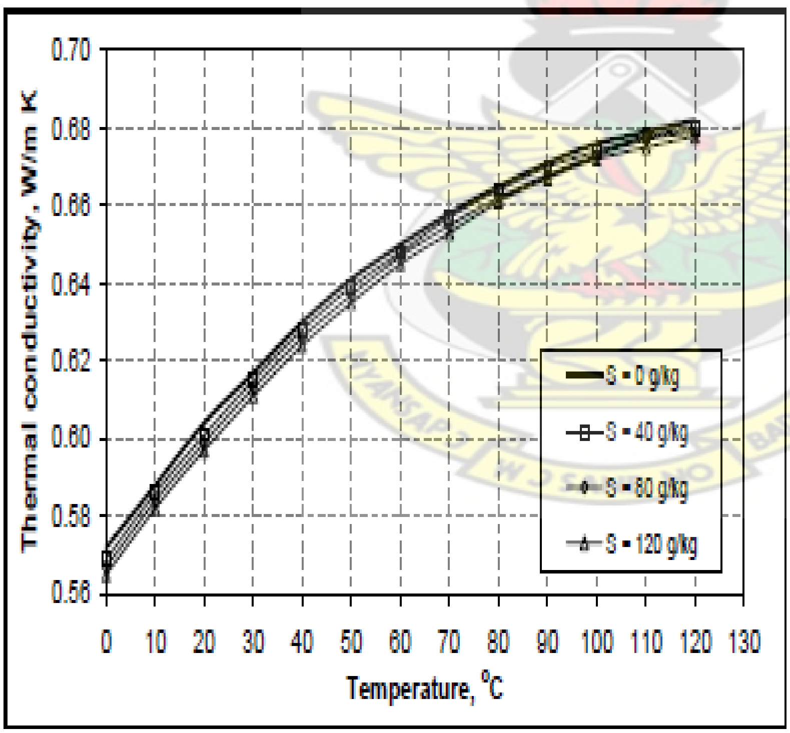
Accuracy ±0.28%

Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

Table A.3: Thermal Conductivity of Seawater at Varying Temperatures and Salinity

Thermal conductivity, W/m K

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	0.572	0.571	0.570	0.570	0.569	0.569	0.568	0.568	0.567	0.566	0.566	0.565	0.565
10	0.588	0.588	0.587	0.587	0.586	0.585	0.585	0.584	0.584	0.583	0.583	0.582	0.582
20	0.604	0.603	0.602	0.602	0.601	0.601	0.600	0.600	0.599	0.599	0.598	0.598	0.597
30	0.617	0.617	0.616	0.616	0.615	0.615	0.614	0.614	0.613	0.613	0.612	0.612	0.611
40	0.630	0.629	0.629	0.628	0.628	0.627	0.627	0.626	0.626	0.625	0.625	0.624	0.624
50	0.641	0.640	0.640	0.639	0.639	0.638	0.638	0.637	0.637	0.636	0.636	0.635	0.635
60	0.650	0.650	0.649	0.649	0.648	0.648	0.647	0.647	0.647	0.646	0.646	0.645	0.645
70	0.658	0.658	0.658	0.657	0.657	0.656	0.656	0.655	0.655	0.655	0.654	0.654	0.653
80	0.665	0.665	0.665	0.664	0.664	0.663	0.663	0.663	0.662	0.662	0.661	0.661	0.661
90	0.671	0.671	0.670	0.670	0.670	0.669	0.669	0.669	0.668	0.668	0.667	0.667	0.667
100	0.676	0.675	0.675	0.675	0.674	0.674	0.674	0.673	0.673	0.673	0.672	0.672	0.672
110	0.679	0.679	0.679	0.678	0.678	0.678	0.677	0.677	0.677	0.676	0.676	0.676	0.675
120	0.682	0.681	0.681	0.681	0.680	0.680	0.680	0.679	0.679	0.679	0.679	0.678	0.678



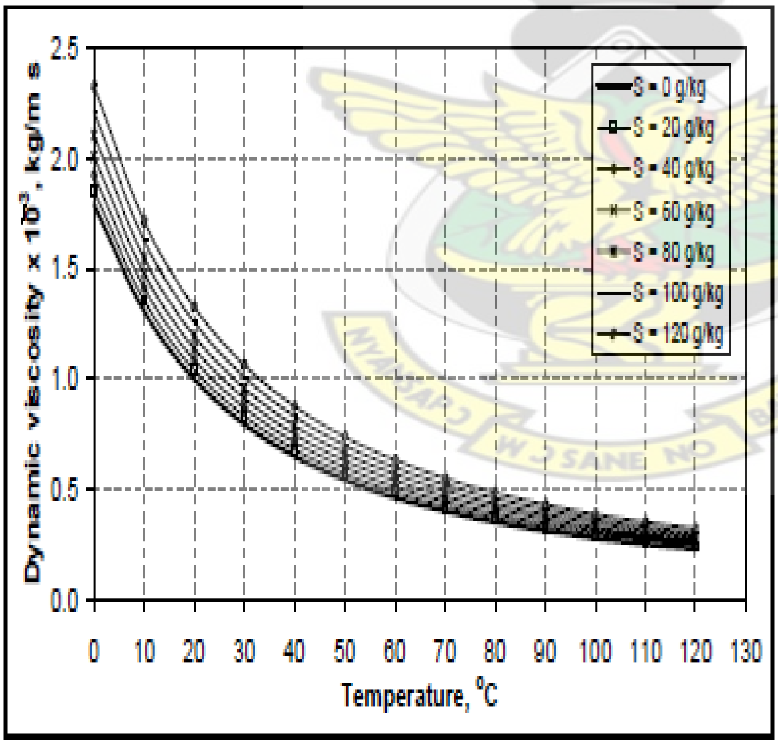
Accuracy ±3.0%

Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

Table A.4: Dynamic Viscosity of Seawater at Varying Temperatures and Salinity

Dynamic viscosity x 10^3, kg/m s

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	1.791	1.820	1.852	1.887	1.925	1.965	2.008	2.055	2.104	2.156	2.210	2.268	2.328
10	1.306	1.330	1.355	1.382	1.412	1.443	1.476	1.511	1.548	1.586	1.627	1.669	1.714
20	1.002	1.021	1.043	1.065	1.089	1.114	1.140	1.168	1.197	1.227	1.259	1.292	1.326
30	0.797	0.814	0.832	0.851	0.871	0.891	0.913	0.936	0.960	0.984	1.010	1.037	1.064
40	0.653	0.667	0.683	0.699	0.716	0.734	0.752	0.771	0.791	0.812	0.833	0.855	0.878
50	0.547	0.560	0.573	0.587	0.602	0.617	0.633	0.649	0.666	0.684	0.702	0.721	0.740
60	0.466	0.478	0.490	0.502	0.515	0.528	0.542	0.556	0.571	0.586	0.602	0.618	0.635
70	0.404	0.414	0.425	0.436	0.447	0.459	0.471	0.484	0.497	0.510	0.524	0.538	0.553
80	0.354	0.364	0.373	0.383	0.393	0.404	0.415	0.426	0.437	0.449	0.462	0.474	0.487
90	0.315	0.323	0.331	0.340	0.349	0.359	0.369	0.379	0.389	0.400	0.411	0.422	0.434
100	0.282	0.289	0.297	0.305	0.313	0.322	0.331	0.340	0.350	0.359	0.369	0.380	0.390
110	0.255	0.262	0.269	0.276	0.283	0.291	0.299	0.308	0.316	0.325	0.334	0.344	0.354
120	0.232	0.238	0.245	0.251	0.258	0.265	0.273	0.280	0.288	0.297	0.305	0.314	0.323



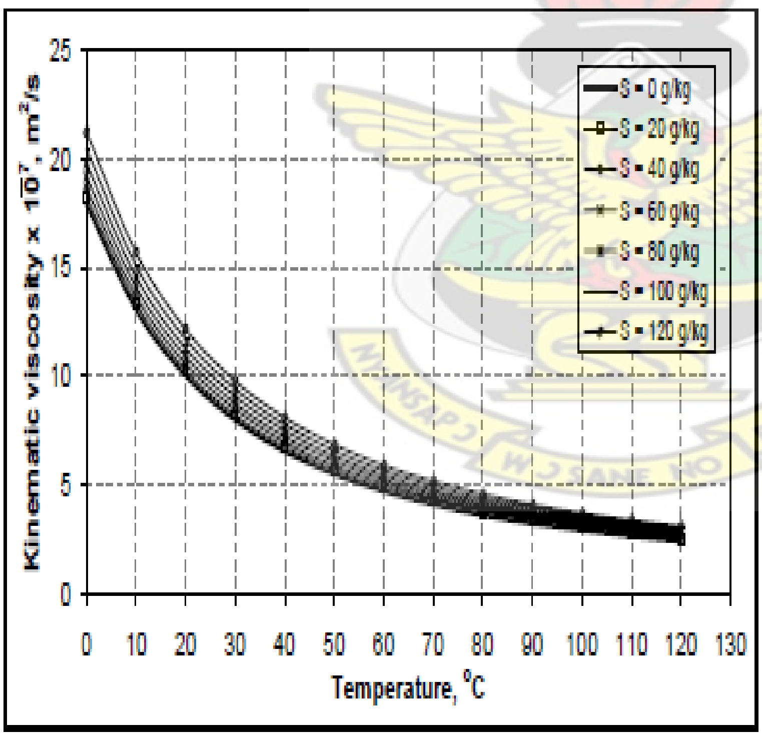
Accuracy ±1.5%

Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

Table A.5: Kinematic Viscosity of Seawater at Varying Temperatures and Salinity

Kinematic viscosity x 10^{7}, m^2/s

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	17.92	18.06	18.23	18.43	18.65	18.90	19.16	19.46	19.77	20.11	20.46	20.84	21.24
10	13.07	13.20	13.35	13.51	13.69	13.89	14.10	14.33	14.57	14.82	15.09	15.38	15.67
20	10.04	10.16	10.29	10.43	10.58	10.75	10.92	11.10	11.30	11.50	11.71	11.93	12.17
30	8.01	8.12	8.23	8.36	8.49	8.63	8.77	8.93	9.09	9.26	9.43	9.61	9.80
40	6.58	6.68	6.78	6.89	7.00	7.13	7.25	7.38	7.52	7.66	7.81	7.96	8.11
50	5.53	5.62	5.71	5.81	5.91	6.02	6.13	6.24	6.36	6.48	6.61	6.74	6.87
60	4.74	4.82	4.91	4.99	5.08	5.18	5.28	5.38	5.48	5.59	5.70	5.81	5.93
70	4.13	4.20	4.28	4.36	4.44	4.52	4.61	4.70	4.79	4.89	4.98	5.08	5.19
80	3.65	3.71	3.78	3.85	3.93	4.00	4.08	4.16	4.25	4.33	4.42	4.51	4.60
90	3.26	3.32	3.38	3.45	3.51	3.58	3.65	3.73	3.80	3.88	3.96	4.04	4.12
100	2.94	3.00	3.05	3.11	3.17	3.24	3.30	3.37	3.44	3.51	3.58	3.65	3.73
110	2.68	2.73	2.78	2.84	2.89	2.95	3.01	3.07	3.13	3.20	3.26	3.33	3.40
120	2.46	2.51	2.55	2.60	2.65	2.71	2.76	2.82	2.88	2.93	3.00	3.06	3.12



Accuracy ±1.5%

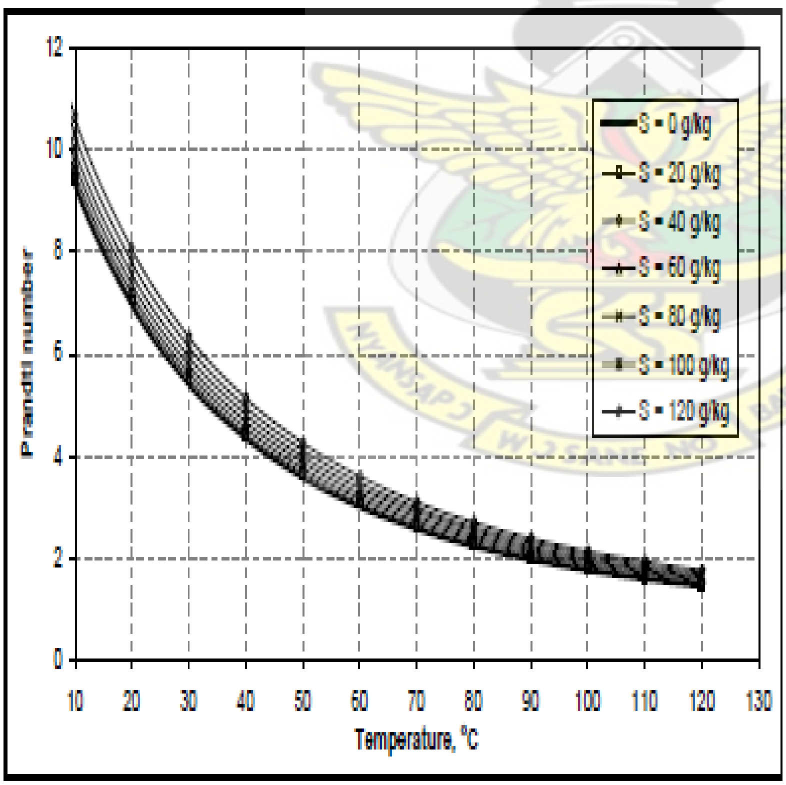
Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

Table A.6: Prandtl Number of Seawater at Varying Temperatures and Salinity

Prandtl number

	Salinity, g/kg												
Temp, °C	0	10	20	30	40	50	60	70	80	90	100	110	120
0	13.18	13.21	13.25	13.31	13.40	13.50	13.63	13.78	13.94	14.13	14.34	14.56	14.81
10	9.32	9.36	9.41	9.48	9.56	9.65	9.76	9.87	10.00	10.14	10.30	10.46	10.64
20	6.95	7.00	7.06	7.12	7.19	7.27	7.36	7.46	7.56	7.67	7.79	7.92	8.05
30	5.40	5.45	5.51	5.57	5.63	5.70	5.78	5.86	5.94	6.03	6.13	6.23	6.33
40	4.34	4.38	4.43	4.49	4.54	4.60	4.67	4.74	4.81	4.88	4.96	5.04	5.13
50	3.57	3.61	3.66	3.71	3.76	3.81	3.87	3.93	3.99	4.05	4.12	4.18	4.25
60	3.00	3.04	3.08	3.12	3.17	3.22	3.27	3.32	3.37	3.42	3.48	3.54	3.60
70	2.57	2.60	2.64	2.68	2.72	2.76	2.81	2.85	2.90	2.94	2.99	3.04	3.09
80	2.23	2.27	2.30	2.33	2.37	2.41	2.45	2.49	2.53	2.57	2.61	2.66	2.70
90	1.97	2.00	2.03	2.06	2.09	2.13	2.16	2.20	2.23	2.27	2.31	2.35	2.39
100	1.76	1.78	1.81	1.84	1.87	1.90	1.93	1.96	1.99	2.03	2.06	2.10	2.13
110	1.59	1.61	1.63	1.66	1.69	1.71	1.74	1.77	1.80	1.83	1.86	1.89	1.93
120	1.45	1.47	1.49	1.51	1.54	1.56	1.59	1.61	1.64	1.67	1.70	1.73	1.76

Accuracy ±3.3%



Source: Mostafa H. Sharqawy, John H. Lienhard V and Syed M. Zubair, Thermophysical Properties of Seawater, A Review of Existing Correlations and Data, Desalination and Water Treatment, 2010

APPENDIX 2: Matlab Program for Determining the Temperature Rise of Sea Water As A Result of Heat Transfer Across The Sea Water Supply Line

```
qt= ;%Tons(Air Conditioning Load)
q=qt*3500; %W (air conditioning load)
Cp=3997.39%J/kgK (specific heat capacity of sea water at constant pressure)
dTsw=5; %deg C(Change in temperature of sea water across heat exchanger)
m=q/(Cp*(dTsw));%kg/s(mass flow rate of sea water)
rho=1027.6; %kg/m3(density of sea water)
Q= m/rho; %m3/s (volume flow rate of sea water)
ID=1.1077; %m (inner diameter of pipe)
OD=1.2; %m (outer diameter of pipe)
AI= (pi*((ID) ^2))/4; %m2 (inner cross sectional area of pipe)
V=Q/AI; %m/s (average velocity of sea water flowing in pipe)
kp=1.5;%(thermal conductivity of HDPE pipe)
ks=0.5931;%(thermal conductivity of seawater)
Pr=8.574; % Prandtl number
v= 12.362e-7; %kinematic viscosity of sea water
ReI=(V*ID)/v;%Reynolds number for inside flow of sea water
NuI=0.023*(ReI^0.8)*(Pr^0.4);%Nusselt number (inside pipe conditions)
hi=(ks*NuI)/ID;%kg/m3s(convective heat transfer coefficient(@inside pipe
conditions))
ri=ID/2;%m(inner radius of pipe)
ro=OD/2;%m(outer radius of pipe)
Vf=1;%m/s flow velocity outside the pipe
ReO=(Vf*OD)/v;%Reynolds number for outside flow of sea water
NuO=0.023*(ReO^0.8)*(Pr^0.4);%Nusselt number (outside pipe conditions)
ho=(ks*NuO)/OD;%kg/m3s(convective heat transfer coefficient(@outside pipe
conditions))
```

```
L=[3093.3;3093.3;3093.3;3093.3;3093.3;3093.3;3093.3;3093.3;3093.3;6960];
%Segmental lengths for Cape Coast
%Segmental lengths for
Accra[1321;1321;1321;1321;1321;1321;1321;1321;1321;2642.7]
%Segmental lengths for
Keta[1035;1035;1035;1035;1035;1035;1035;1035;1035;2329];
%Segmental lengths for
Takoradi[3522;3522;3522;3522;3522;3522;3522;3522;3522;7926];
To=[5.8;7.6;9.4;11.2;13;14.8;16.6;18.4;20.2;22];%To is the average
temperature outside the pipe(deg C)
T1=[0;0;0;0;0;0;0;0;0;0];%T1 is the average temperature inside the pipe(deg
C)
Ao=[0;0;0;0;0;0;0;0;0;0];%outer surface area of seawater pipe(m2)
Ai=[0;0;0;0;0;0;0;0;0;0];%inner surface area of seawater pipe(m2)
R=[0;0;0;0;0;0;0;0;0;0];%thermal resistance(m2K/W)
Uo=[0;0;0;0;0;0;0;0;0;0];%overall heat transfer coefficient(W/m2K)
dT=[0;0;0;0;0;0;0;0;0;0];%temperature rise as a result of heat transfer(deg
C)
for k=1
    T1(1)=5.5;
end
for k=1:10
    Ao(k)=(pi)*(2*ro)*L(k);
    Ai(k)=(pi)*(2*ri)*L(k);
    R(k)= 1/(hi*Ai(k))+((log(ro/ri))/(2*pi*L(k)*kp))+(1/(ho*Ao(k)));
    Uo(k)=1/R(k);
    dT(k)=(2*Uo(k)*(To(k)-T1(k)))/((2*m*Cp)+(Uo(k)));
    T1(k+1)= T1(k)+ (dT(k)/2);%average temperature across the pipe
end
```

APPENDIX 3: EXCEL SHEET FOR FINDING THE PUMPING POWER FOR SWAC

Table A.7: Excel sheet for determining the pumping power for SWAC

DETERMINATION OF REYNOLD'S NUMBER	
Density Of Fluid(kg/m3)	
Velocity Of Flow(m/s)	
Mass Flow Rate of Sea Water (kg/s)	
Inside Diameter Of Pipe(m)	
Dynamic viscosity of sea water (kg/ms)	
Reynold's Number	
RELATIVE ROUGHNESS	
Roughness Of HDPE Pipe (K)(mm)	
Diameter Of Pipe(D)(mm)	
Relative Roughness, K/D	
FRICTION FACTOR READ FROM THE MOODY CHART	
((e/D)/3.7)^1.11 6.9/Re logarithsm equation f	
HEAD LOSS	
Total Length of supply pipe(m)	
Total Length of Effluent pipe(m)	
Acceleartion due to gravity (m/s^2)	
Frictional Head loss, h_L (m)	
Kinetic Head,m	
Installation height (m)	
Total Head,m	
Specific losses(50% of head Loss)	
Specific Losses in Heat Exchanger, m	
Overall Total Head Loss, m	
PUMPING POWER(P)	
Mass Flow Rate of Sea Water (kg/s)	
Volume flow rate of Sea Water (m3/s)/(L/S)	
Pumping Power (KW),Delivered to Fluid	
Efficiency(Mechanical and Electrical) of Pump	
Electrical Energy of Pump(kW)	
TEMPERATURE RISE AS A RESULT OF FRICTIONAL EFFECTS	
Heat Addition(KW)	
Mass flow rate of sea water(kg/s)	
Specific heat capacit at constant pressure,sea water(KJ/KgK)	
ΔT,f(^{O}C)	

APPENDIX 4: HDPE PIPE SCHEDULE FROM INTERPLAST, 2011

Table A.8: HDPE Pipe Schedule from Interplast, 2011

Standards	Pipe Size (mm)	PN (bar)	SDR	Wall Thickness (mm)	Weight (kg/m)
EN 12201 / ISO 4427 HDPE Water	50	6	26	2	0.31
EN 12201 / ISO 4427 HDPE Water	63	6	26	2.5	0.49
EN 12201 / ISO 4427 HDPE Water	75	6	26	2.9	0.67
EN 12201 / ISO 4427 HDPE Water	90	6	26	3.5	0.98
EN 12201 / ISO 4427 HDPE Water	110	6	26	4.2	1.43
EN 12201 / ISO 4427 HDPE Water	125	6	26	4.8	1.85
EN 12201 / ISO 4427 HDPE Water	140	6	26	5.4	2.33
EN 12201 / ISO 4427 HDPE Water	160	6	26	6.2	3.06
EN 12201 / ISO 4427 HDPE Water	180	6	26	6.9	3.81
EN 12201 / ISO 4427 HDPE Water	200	6	26	7.7	4.72
EN 12201 / ISO 4427 HDPE Water	225	6	26	8.6	5.93
EN 12201 / ISO 4427 HDPE Water	250	6	26	9.6	7.35
EN 12201 / ISO 4427 HDPE Water	280	6	26	10.7	9.17
EN 12201 / ISO 4427 HDPE Water	315	6	26	12.1	11.68

EN 12201 / ISO 4427 HDPE Water	355	6	26	13.6	14.76
EN 12201 / ISO 4427 HDPE Water	400	6	26	15.3	18.72
EN 12201 / ISO 4427 HDPE Water	450	6	26	17.2	23.67
EN 12201 / ISO 4427 HDPE Water	500	6	26	19.1	29.2
EN 12201 / ISO 4427 HDPE Water	560	6	26	21.4	36.6
EN 12201 / ISO 4427 HDPE Water	630	6	26	24.1	46.38
EN 12201 / ISO 4427 HDPE Water	710	6	26	27.2	58.95
EN 12201 / ISO 4427 HDPE Water	800	6	26	30.6	74.66
EN 12201 / ISO 4427 HDPE Water	900	6	26	34.4	94.81
EN 12201 / ISO 4427 HDPE Water	1000	6	26	38.2	116.52
EN 12201 / ISO 4427 HDPE Water	1200	6	26	46.15	-

Source: Interplast Ghana Limited

APPENDIX 5: US Consumer Price Index (CPI) DATA

Table A.9: US CPI from 1999 to 2012

Year	Average CPI
1999	166.6
2000	174.0
2001	177.1
2002	179.9
2003	184.0
2004	188.9
2005	195.3
2006	201.6
2007	207.3
2008	215.3
2009	214.5
2010	218.1
2011	224.9
2012	**230.0***

Note: Base year is chained; 1982-1984 = 100

*This figure was extrapolated based on past CPI records

Source: U.S. Department of Labor Bureau of Labor Statistic

APPENDIX 6: MOODY CHART

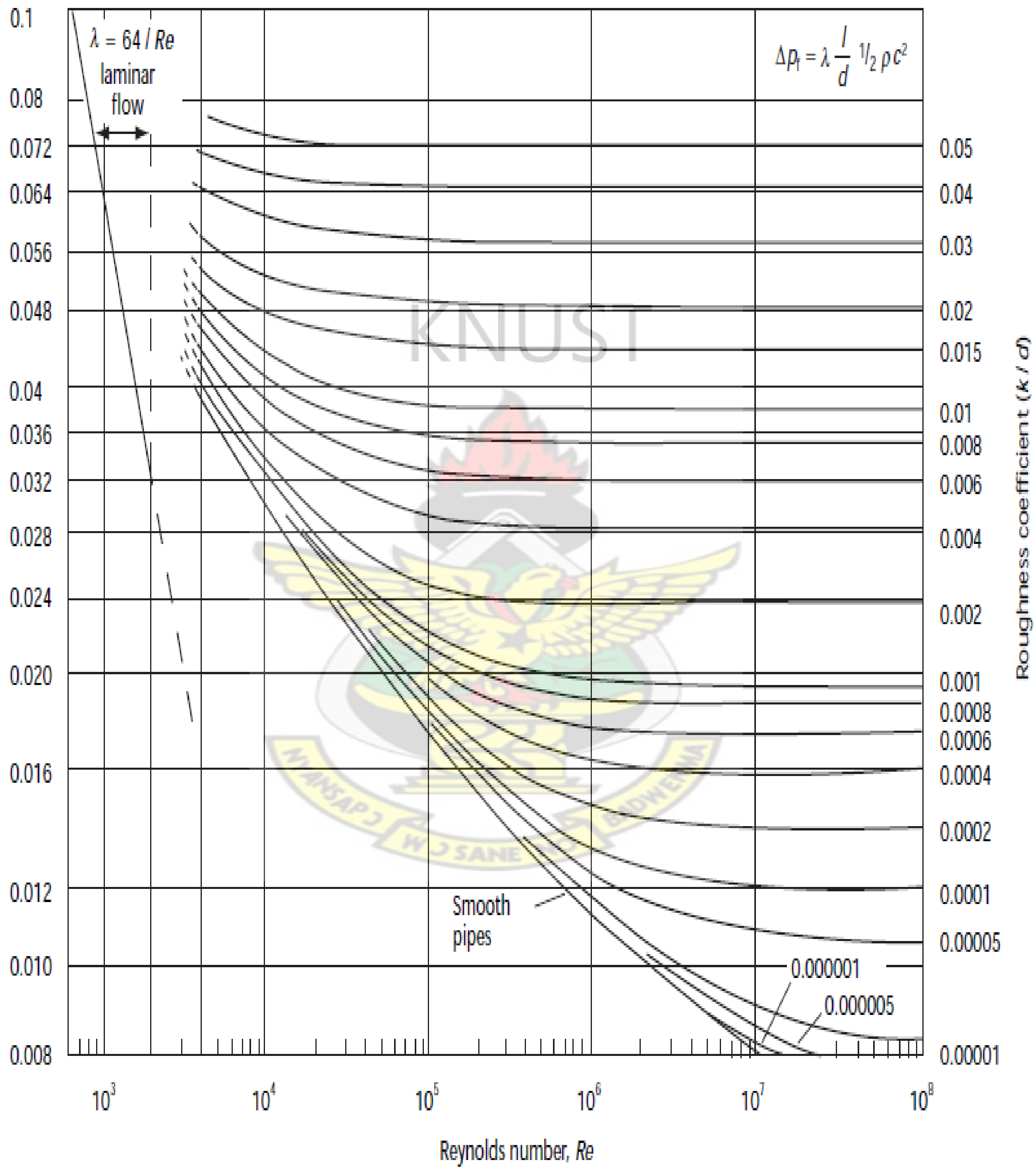


Figure A.1: Moody Chart