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Potential energy savings when using saline water for cooling chillers in Malta

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Potential Energy Savings When Using Saline Water for Cooling Chillers in Malta

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Abbreviations

CO_2	Carbon dioxide
SO_2	Sulphur dioxide
NO_x	Nitrogen oxides
CFC	Chlorofluorocarbon
HCFC	Hydrochlorofluorocarbon
DSWC	Direct surface water cooling
SWHP	Surface water heat pump
HSWHP	Hybrid surface water heat pump
GSHP	Ground source heat pump
DHC	District heating and cooling
HDPE	High density polyethylene
SDI	Silt density index
kgoe	Kilograms of oil equivalent
BMS	Building management system
EPB	Energy performance of a building
EPC	Energy performance certificate
EU	European Union
US	United States of America
%	Per cent
€	Euro currency

Notation

T_i	[K]	Temperature
$Q_{i\text{-}j}$	[kJ/kg]	Heat flow between two states
$W_{i\text{-}j}$	[kJ/kg]	Work input/output between two states
COP		Coefficient of performance
h_i	[kJ/k]	Enthalpy
s_i	[kJ/kg K]	Entropy
P_i	[kPa]	Pressure
E_{RES}	[W]	Amount of energy from renewable sources captured by heat pumps
Q_{usable}	[W]	Estimated total usable heat delivered by heat pumps
SPF		Estimated seasonal performance factor
η		Ratio between total gross production of

		electricity and the primary energy consumption for electricity
$\dot{m}$	[kg/s]	Mass flow rate of refrigerant
Refrigerating effect	[kJ/kg]	Area under the line joining states *4* and *1* in a temperature-entropy diagram
Refrigerating capacity	[W]	The refrigerating effect multiplied by the mass flow rate of refrigerant
Power	[W]	Power required by several components which make up systems
Potential savings	[W]	Energy savings resulting from the use of water cooled systems as opposed to air cooled systems

Subscripts

i, j	States
hp	Heat pump
ref	Refrigerator

Units

J	Joule
kJ	Kilojoule
W	Watt
kW	Kilowatt
MW	Megawatt
kWh	Kilowatt hour
MWh	Megawatt hour
GWh	Gigawatt hour
K	Kelvin
°C	Degree Celsius
°F	Degree Fahrenheit
µg	Microgram
kg	Kilogram
l	Litre
µS	MicroSiemens
kPa	KiloPascal
s	Second
min	Minute
h	Hour
mm	Millimetre
cm	Centimetre
m	Metre

Abstract

Several cooling systems found in facilities on the coast of St. Julian's were found to be making use of chillers whose condensers are cooled with saline water as opposed to air. The former practice is advantageous since it results in better chiller performance which can be explained through thermodynamic principles. The principle of operation of water cooled chillers is to reject heat from the condenser to saline groundwater obtained from boreholes through a heat exchanger. The higher temperature saline ground water is then rejected to the sea.

Chillers cooled with saline groundwater are subject to the Borehole Drilling and Excavation Works within the Saturated Zone Regulations and to the Protection of Groundwater against Pollution and Deterioration Regulations. Under both the Energy End-Use Efficiency and Energy Services Regulations and the Promotion of Energy from Renewable Sources Regulations, Ground Source Heat Pump systems can contribute to the set targets. Chillers configured to have their condensers cooled by saline groundwater qualify as Ground Source Heat Pump systems. Heat pumps and district heating and cooling are both regarded as adequate means for improving the energy performance of buildings under the Energy Performance of Buildings Regulations.

By using the principles of the standard vapour-compression cycle, the system Coefficient of Performance of an air cooled chiller system was found to be 4.0 while that of a water cooled chiller system was found to be 5.9. This translates into an energy saving potential of 63.3 GWh of electricity per year and a total of 55,733 tonnes of carbon dioxide emissions avoided per year. 63.3 GWh represents approximately 2.85% of the generated electricity in Malta per year.

Chapter 1

Introduction

1 Introduction

In recent years, the Maltese Islands have seen an increase in both the number and size of industrial, commercial and residential edifices. Such large buildings necessitate systems that provide heating and cooling in order to obtain a comfortable temperature within. Malta's hot climate means that more importance is given to cooling rather than heating in buildings. Cooling is usually achieved by circulating chilled water through the structures (Davidson, 2003). The chilled water is in turn provided by chillers. Chillers, which provide cooling in buildings, require an energy input in order to provide the necessary commodity of temperature control.

Chillers are essentially heat pumps operating on a refrigerating cycle. They harvest heat from the indoor air thus using it as their heat source. Work must then be done to transfer the harvested heat to a heat sink. A lower temperature sink will enable better performance of the chiller in cooling the source since the chiller will require less work to transfer the heat energy. Heat pumps have reasonably high performance factors since they transfer energy rather than convert it (NREL, 2001) (Omer, 2008).

The conventional heat sink for chillers is ambient air. However, the temperature of air is usually high in the summer season and therefore it limits the performance of the chiller thus resulting in more energy consumption for a given refrigerating effect.

With rises in energy costs and increased concern over the by-products of combustion, the use of saline water as the heat sink in chillers is becoming a more attractive solution.

Water, with its high heat capacity, shows a delay in temperature change with respect to seasons. During summer, water bodies are cooler than the surrounding air and absorb solar radiation which slowly increases their temperature. When winter comes, water bodies would have warmed up and slowly start cooling down again until the following summer. The temperature of water bodies is thus in ante-phase with the air temperature. This phenomenon can be exploited to our advantage in cooling chillers in summer.

Saline water in general exhibits lower temperatures than air in the summer season when cooling is necessary. It therefore improves the performance of the chiller and reduces its energy needs making it more efficient.

Additionally, such systems make use of renewable energy since both the ground from which the saline water is extracted and water bodies themselves store renewable solar energy naturally (Cao, Han, Gu, Zhang, & Hu, 2009) (Omer, 2008). The resource is virtually unlimited and can be used in a non-polluting way (Davidson, 2003).

Moreover, Malta, being a small island state and having high-density development along much of its coast where saline water is abundant, might be an ideal site for the application of this technology (Chua, Chou, & Yang, 2010).

1.1 The Research Problem

The questions that this dissertation will seek to answer are:

- Are there any systems currently making use of saline water for cooling in Malta? How do they work?
- Do saline water cooled chillers qualify as contributors to any targets set by Maltese law?
- By which Maltese laws are such systems regulated?

- How much better will the performance of water cooled systems be when compared to air cooled systems?
- How much energy and greenhouse gas emissions can be avoided if more systems making use of saline water are implemented?

This research will establish whether systems making use of saline water cooling are viable in the Maltese Islands. If such systems prove to be viable, environmental benefits can result from the reduced emissions as well as cost reductions which arise from the decrease in the use of energy. Also, the implementation of similar systems could contribute towards reaching targets established by regulations such as the Energy Performance of Buildings Regulations (Subsidiary Legislation 513.01, 2012) and the Promotion of Energy from Renewable Sources Regulations (Subsidiary Legislation 423.19, 2010).

Answering the questions that make up the research problem will enable the acceptance or rejection of the following hypothesis:

"The use of saline water as the heat sink for chillers results in significant energy savings when compared to the use of air as the heat sink for chillers."

The ultimate objective is to obtain sensible values for the coefficient of performance of the chillers when operating with both sinks. This will enable the quantification of the amount of energy that can be saved by using this alternative technology. Additionally, the carbon dioxide emissions avoided will also be quantified.

The research will unravel in seven chapters:

- Chapter 1 is an introduction to the research problem and gives a background to the study.
- The relevant aspects of seawater and ground source heat pumps from literature are highlighted in Chapter 2 and aspects of thermodynamic principles, types of systems, requirements, system components and sustainability, environmental and social considerations are explained.
- Chapter 3 looks into local systems and summarises the visits made together with any information obtained.
- A review of pertinent regulations, their meaning and relevance to chiller systems cooled by saline water is presented in Chapter 4.
- Chapter 5 describes the methodology used to reach the objective of the research and the results obtained.
- The results obtained are discussed in Chapter 6 along with some of the difficulties encountered in obtaining the results.
- Conclusions are drawn in Chapter 7 and some recommendations are given.

Chapter 2

Literature Review

2 Literature Review

2.1 The Heat Pump

Heat flows naturally from regions of higher temperature to regions of lower temperature. In a heat engine the natural flow of heat from a higher temperature reservoir to a lower temperature reservoir is exploited and thus work is extracted. This principle is used in power generation.

The heat engine can be operated in reverse and in such cases it is called a heat pump. In a heat pump, energy is transferred from a lower temperature reservoir to a higher temperature reservoir – in order to do this, a work input is required to overcome the natural temperature gradient and force heat to flow in the opposite direction. This situation is analogous to water being pumped from a reservoir at a lower elevation to a reservoir at a higher elevation – work needs to be done against the natural gravitational field (Eastop & McConkey, 1993, p. 485).

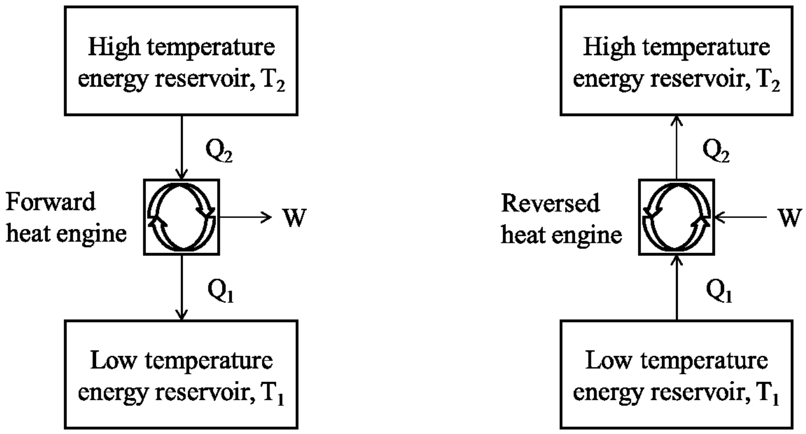

Figure 2-1 Forward and reversed heat engines – *T* is the temperature, *Q* is the heat flow and *W* is the work done (Eastop & McConkey, 1993, p. 89).

The reversed heat engine can be used in one of two ways; to supply heat, in which case it is referred to as a heat pump or to remove heat, in which case it is referred to as a refrigerator. When the reversed heat engine is being used as a heat pump, the reservoir of interest is the high temperature reservoir since the purpose of the heat pump is to heat the high temperature reservoir further. As a result of this the parameter of interest is the heat flow Q_2 (refer to Figure 2-1). Conversely, when it is being used as a refrigerator, the reservoir of interest is the low temperature reservoir since the purpose of the refrigerator is to further cool the low temperature reservoir. As a result of this the parameter of interest is the heat flow Q_1 (refer to Figure 2-1). The work required to cause the heat to flow determines the power required by the system and constitutes the majority of the running cost (Eastop & McConkey, 1993, p. 486).

Due to the two ways in which a heat pump can be operated, two coefficients of performance (COP) are defined depending on the mode of operation:

For a heat pump (Eastop & McConkey, 1993, p. 487):

$$COP_{hp} = \frac{Q_2}{\sum W}$$

2-1

For a refrigerator (Eastop & McConkey, 1993, p. 486):

$$COP_{ref} = \frac{Q_1}{\sum W}$$

2-2

The main components of a rudimentary heat pump system are shown in Figure 2-2. It can be shown that the highest possible efficiency is obtained when heat is supplied at a constant temperature and rejected at a lower constant temperature (Eastop & McConkey, 1993, p. 125). This is known as the Carnot cycle. All the processes carried out as part of the Carnot cycle are thermodynamically reversible (Stoecker & Jones, 1982, p. 188) and thus, the Carnot cycle gives the ideal, theoretical, highest possible efficiency; however, in practice, the efficiency achievable is much lower than the Carnot efficiency due to irreversibilities and deviations from the ideal cycle made for practical reasons (Eastop & McConkey, 1993, p. 125). Despite this, the Carnot cycle is still of interest since it can serve as a standard for comparison and since it provides a guide to the temperatures between which the cycle should operate for maximum effectiveness (Stoecker & Jones, 1982, p. 188). The Carnot cycle is independent of the working substance used as long as the refrigerant has a suitable wet vapour state (Eastop & McConkey, 1993, pp. 125, 487).

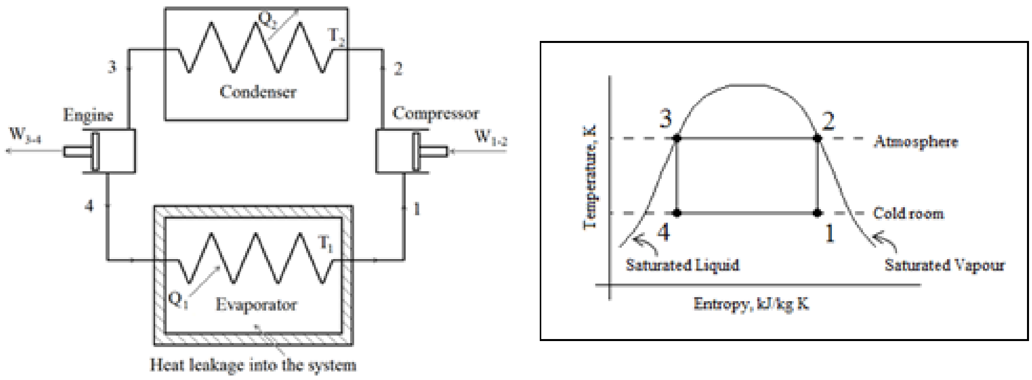

Figure 2-2 Left: Reversed heat engine system operating on the Carnot cycle (Eastop & McConkey, 1993, p. 487); Right: Corresponding temperature-entropy diagram with refrigerant as the condensing and evaporating fluid (Stoecker & Jones, 1982, p. 194).

Referring to Figure 2-2, the refrigerant in the wet vapour state at *1* enters the compressor and is compressed reversibly and adiabatically (without heat transfer to or from the fluid), that is, isentropically, to state *2*. At this point the vapour enters the condenser where it condenses and thus releases heat (Q_2). The condensation occurs at constant temperature and pressure until the refrigerant is completely liquid. The liquid refrigerant at state *3* then expands behind the engine piston and does work in the process. The expansion is also isentropic. The low temperature and pressure liquid at state *4* enters the evaporator where it is evaporated thanks to the heat supplied (Q_l) by the cold source. (Eastop & McConkey, 1993, p. 487)

A secondary working fluid is used to exchange heat with the condenser or evaporator (depending on the heat pump mode of operation) and distribute it through pipes encircling the entire building; the secondary fluid can be water or any other safe substance. This is done in order to avoid hazards from leakage of refrigerants in the building (Eastop & McConkey, 1993, p. 499). The cost of the system increases and the COP decreases when the secondary fluid is used – furthermore, there is an additional cost involved in pumping the secondary fluid (Eastop & McConkey, 1993, p. 499).

From Equations 2-1 and 2-2 and incorporating thermodynamics principles it can be shown that (Eastop & McConkey, 1993, pp. 488-489):

$$COP_{hp} = \frac{T_2}{T_2 - T_1}$$

2-3

$$COP_{ref} = \frac{T_1}{T_2 - T_1}$$

2-4

$$COP_{hp} = COP_{ref} + 1$$

2-5

Heat pump performance depends on the temperature of the reservoirs as shown by Equations 2-3 and 2-4 – the smaller the temperature difference between the two reservoirs, the higher the COP. When the atmosphere is used as the heat source or sink in a heat pump, the system is less effective due to the fluctuations in the atmospheric temperature (Eastop & McConkey, 1993, p. 490).

Seawater or saline groundwater, when used as the heat source or sink, can provide a lower, more constant temperature (Eastop & McConkey, 1993, p. 490). This is especially true as the depth from which the water is extracted increases. Seawater and saline groundwater are colder than the atmosphere in summer and thus make for a suitable high temperature reservoir (T_2) to be used in a refrigerating cycle as they can reduce the temperature difference between the high and low temperature (building at T_1) reservoirs (refer to Figure 2-1). In summary, seawater and saline groundwater can provide a colder high temperature reservoir in summer and thus show great potential in improving the COP.

Heat pumps are becoming increasingly popular in heating and cooling applications since: they are capable of recovering heat from certain sources such as seawater (Cao, Han, Gu, Zhang, & Hu, 2009) (Chua, Chou, & Yang, 2010) (Mitchell & Spitler, 2013); they can have high COPs (Chua, Chou, & Yang, 2010) (Cao, Han, Gu, Zhang, & Hu, 2009); and they have evolved to become more energy efficient (Chua, Chou, & Yang, 2010). Also, when used for space heating and heat generation, heat pumps have been shown to produce less greenhouse gases, particularly carbon dioxide (CO_2) emissions, as well as other environmentally harmful gases such as sulphur dioxide (SO_2) and nitrogen oxides (NO_x) (Chua, Chou, & Yang, 2010) (Omer, 2008).

2.1.1 The Vapour-Compression Cycle

Most refrigerators make use of the vapour-compression cycle (Stoecker & Jones, 1982, p. 187) (Eastop & McConkey, 1993, p. 491). The name of the cycle derives from the fact that a liquefiable vapour is used as the refrigerant in the system (Eastop & McConkey, 1993, p. 491).

The Carnot cycle described in the previous section is an idealised cycle and cannot be duplicated due to the irreversibilities that occur in real cycles (Stoecker & Jones, 1982, p. 193). The vapour-compression cycle is a modification of the Carnot cycle that arises from practical considerations (Stoecker & Jones, 1982, pp. 187, 193) (Eastop & McConkey, 1993, p. 491).

2.1.1.1 Replacement of the Expansion Engine by a Throttle Valve

In the Carnot cycle, expansion takes place isentropically and the work resulting from the expansion process is used to aid the compression process; however, the process of extracting work from the expansion of a fluid in a mixed liquid and vapour state poses difficulties in lubricating the expansion engine and this, together with the cost of the expansion engine, make it unfeasible to have an expansion engine given the small amount of work that can be produced by the expansion process (Stoecker & Jones, 1982, p. 195).

In order to simplify the system, the expansion engine can be replaced by a throttle valve (Eastop & McConkey, 1993, p. 491). During a throttling process, the refrigerant has its pressure reduced without experiencing changes in its potential and kinetic energy (Stoecker & Jones, 1982, p. 195). Additionally there is no transfer of heat and thus the enthalpy before and after the throttling device remains the same (Stoecker & Jones, 1982, p. 195). The entropy increases due to the fact that the process is highly irreversible (Stoecker & Jones, 1982, p. 195).

The use of a throttle valve results in a decrease in the refrigerating effect (Eastop & McConkey, 1993, p. 491). The resulting cycle appears on a temperature-entropy diagram as shown in Figure 2-3.

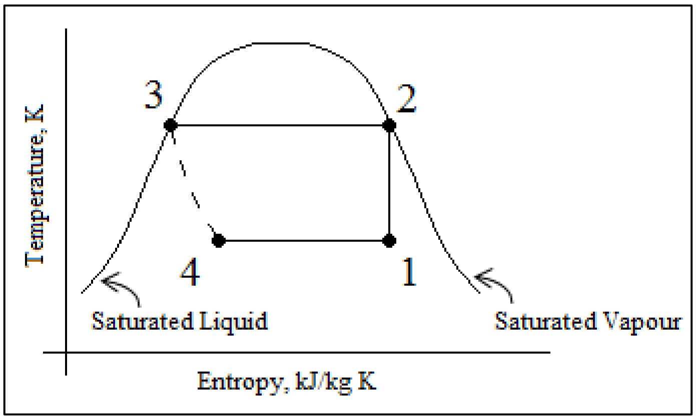

Figure 2-3 Cycle using a throttle valve (Eastop & McConkey, 1993, p. 491).

2.1.1.2 Condition at the Compressor Inlet

In the Carnot cycle, the compression process is referred to as being wet since the refrigerant is found in both the liquid and vapour phases (Stoecker & Jones, 1982, p. 193). In practice, it is highly undesirable to have a fluid in a mixed state at the compressor inlet due to several reasons (Stoecker & Jones, 1982, p. 194):

- The valves or cylinder head may become damaged if liquid droplets remain trapped in the head of the cylinder when the piston rises.
- Lubrication of the walls of the cylinder may prove to be difficult or ineffective since the liquid refrigerant washes away the lubricant thus increasing wear rates in the cylinder.

It is therefore desirable for the refrigerant to be completely in the vapour state before entering the compressor (Stoecker & Jones, 1982, p. 194). This increases the work required for compression but protects the compressor from damage and wear

(Stoecker & Jones, 1982, p. 195) (Eastop & McConkey, 1993, p. 492). The resulting temperature after compression is higher than the condensing temperature meaning that the refrigerant is superheated (Stoecker & Jones, 1982, p. 195). The amount of superheat should be as small as possible (Eastop & McConkey, 1993, p. 492).

The superheated condition of the refrigerant upon entry to the condenser means that the heat rejection from the condenser cannot occur at a constant temperature (Eastop & McConkey, 1993, p. 492).

The resulting cycle appears on a temperature-entropy diagram as shown in Figure 2-4.

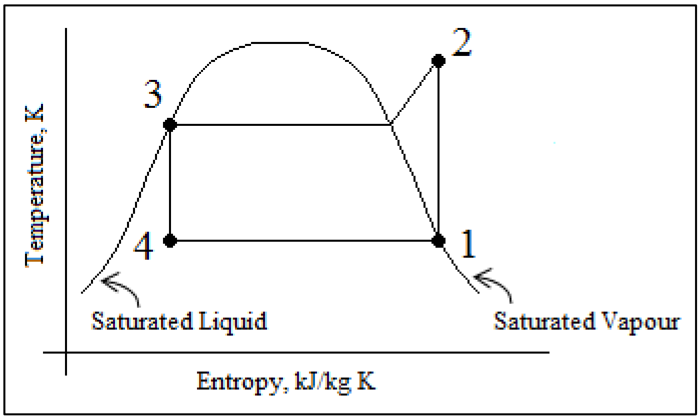

Figure 2-4 Cycle using dry compression (Stoecker & Jones, 1982, p. 194).

2.1.1.3 The Resulting Vapour Compression Cycle

The resulting vapour-compression cycle appears as in Figure 2-5. The processes making up the ideal vapour-compression cycle are (Stoecker & Jones, 1982, pp. 195-196):

1-2 Reversible adiabatic compression from saturated vapour to the condenser pressure.

2-3 Reversible heat rejection at constant pressure resulting in desuperheating and condensation of the refrigerant.

3-4 Irreversible expansion at constant enthalpy from saturated liquid to the evaporator pressure.

4-1 Reversible addition of heat at constant pressure causing evaporation to saturated vapour.

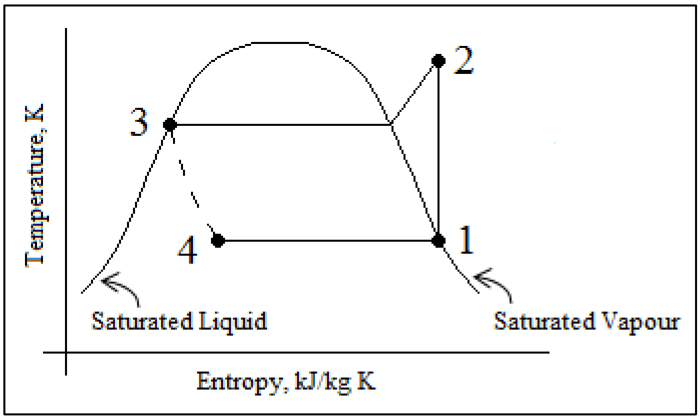

Figure 2-5 Standard vapour-compression cycle (Stoecker & Jones, 1982, p. 195).

From the steady-flow energy equation and neglecting changes in kinetic and potential energy (Stoecker & Jones, 1982, p. 197):

$$h_1 + Q = h_2 + W$$

2-6

Where *h* represents the enthalpy, *Q* represents the heat rejected or absorbed and *W* represents the work done. The subscripts *1* and *2* refer to the part of the cycle being analysed.

Since no heat is transferred in the compression process due to its adiabatic nature the term *Q* is equal to zero and therefore the work of compression in a standard vapour-compression cycle is given by (Stoecker & Jones, 1982, p. 197):

$$W_{1-2} = h_2 - h_1$$

2-7

The refrigerating effect is the heat transferred in process *4-1* and is given by (Stoecker & Jones, 1982, p. 198):

$$Q_{4-1} = h_1 - h_4$$

2-8

The resulting coefficient of performance of a standard vapour-compression cycle is therefore equal to (Stoecker & Jones, 1982, p. 198):

$$COP_{chiller} = \frac{h_1 - h_4}{h_2 - h_1}$$

2-9

The standard vapour-compression cycle, despite being closer to real cycles than the Carnot cycle still consists of some idealisations. Additionally the actual vapour-compression cycle departs from the standard vapour-compression cycle both intentionally and as a result of phenomena which are unavoidable (Stoecker & Jones, 1982, p. 202).

When the refrigerant leaves the condenser it may be subcooled due to the cooling medium cooling the refrigerant to a temperature below the saturation temperature (Stoecker & Jones, 1982, p. 203). Subcooling is desirable in order to ensure that the refrigerant is completely liquid before entering the expansion device (Stoecker & Jones, 1982, p. 203).

In real cycles, there are pressure drops across both the evaporator and condenser due to friction losses (Stoecker & Jones, 1982, p. 203). This means that in the actual vapour-compression cycle more work will be required at the compressor than in the standard vapour-compression cycle (Stoecker & Jones, 1982, p. 203). The compression process is usually also not isentropic (Stoecker & Jones, 1982, p. 203).

Some superheating usually also occurs in the evaporator to ensure that the refrigerant entering the compressor is completely in the vapour state (Stoecker & Jones, 1982, p. 203).

2.2 Heat Pump Components

2.2.1 The Compressor

The compressor is the core of the vapour-compression system (Stoecker & Jones, 1982, p. 205). The two factors determining the performance of a compressor are its refrigeration capacity and its power requirements (Stoecker & Jones, 1982, p. 207). The suction and discharge pressures from the compressor control these two parameters of the compressor (Stoecker & Jones, 1982, p. 207).

There are four main types of compressors; reciprocating, screw, centrifugal and vane (Stoecker & Jones, 1982, p. 205). Each type has its virtues and flaws and in time has adapted for specific situations and system sizes.

The reciprocating compressor makes use of a piston or multiple pistons which move back and forth in a cylinder (Stoecker & Jones, 1982, p. 205). Suction and discharge valves are specifically arranged to produce a pumping action (Stoecker & Jones, 1982, p. 205). Reciprocating compressors may be hermetic or semihermetic (Stoecker & Jones, 1982, p. 206). This type of compressor performs better at part-load operation (Stoecker & Jones, 1982, p. 230).

The other three types of compressors all make use of rotating elements (Stoecker & Jones, 1982, p. 205). The screw and vane compressors are positive displacement machines while the centrifugal compressor, as the name implies, utilises centrifugal forces to produce a pumping action (Stoecker & Jones, 1982, p. 205).

Screw compressors are more efficient when operating near full load, have fewer moving parts than a reciprocating compressor and have long operating lives (Stoecker & Jones, 1982, p. 230).

Table 2-1 Most commonly used type of compressor for a given refrigeration capacity (Stoecker & Jones, 1982, p. 230).

Refrigeration Capacity	Type of Compressor
Up to 300 kW	Reciprocating or vane for domestic applications
300 kW – 500 kW	Screw
500 kW and above	Centrifugal

2.2.2 Condensers and Evaporators

Condensers and evaporators are essentially heat exchangers (Stoecker & Jones, 1982, p. 233). They are very often of the shell-and-tube type or of the finned-coil type and can have the refrigerant flowing inside the tubes or outside (Stoecker & Jones, 1982, p. 233). The fluid used for cooling the condenser can be either in the liquid phase or in the gas phase (Stoecker & Jones, 1982, p. 233). Also, the fluid being cooled by the evaporator can be either in the liquid phase or in the gas phase (Stoecker & Jones, 1982, p. 233). These possible configurations of condensers and evaporators are used in order to make it possible to classify them.

Condensers and evaporators have certain features in common, such as the physical principles that dictate the movement of fluid through the shell and over the tubes (Stoecker & Jones, 1982, p. 233). However, conditions are very different when refrigerant boils and when it condenses (Stoecker & Jones, 1982, p. 233).

Two very important parameters of a heat exchanger are its heat transfer coefficient and the heat transfer area (Stoecker & Jones, 1982, p. 233). These two parameters, when multiplied with the mean temperature difference between the fluids give the rate of heat transfer (Stoecker & Jones, 1982, pp. 233, 235).

Different heat transfer mechanisms operate between one fluid and the outside surface of a tube, between the outside and the inside surfaces of a tube and between the inside surface of a tube and the second fluid (Stoecker & Jones, 1982, p. 235). Under steady-state conditions, the rates of heat transfer between these interfaces are all equal (Stoecker & Jones, 1982, p. 235).

2.2.2.1 Shell-and-Tube

Pressure drops are also important in heat exchangers and occur as the fluid moves inside the straight tubes, as it flows through bends and as it enters and exists the heat exchanger (Stoecker & Jones, 1982, p. 237). The pressure drop is proportional to the square of the flow rate (Stoecker & Jones, 1982, p. 237) and the pressure drop for a given flow rate is predictable (Stoecker & Jones, 1982, p. 237).

Pressure drops also affect the liquid flowing through the shell over tubes (Stoecker & Jones, 1982, p. 239). The pressure drops can be accurately predicted at different flow rates if the value of the pressure drop is known at one specific flow rate (Stoecker & Jones, 1982, p. 239). The heat transfer coefficient of liquid that flows over tubes is complicated and can be estimated from empirical correlations (Stoecker & Jones, 1982, p. 238).

2.2.2.2 Finned-Coils

The outside resistance of a tube is said to be the controlling resistance, which means that it is the resistance that contributes most to the overall heat transfer coefficient of the heat exchanger (Stoecker & Jones, 1982, p. 239). In order to decrease this resistance, the heat transfer area is increased by using fins (Stoecker & Jones, 1982, p. 239).

2.2.2.3 Condensers

The refrigerant changes phase from vapour to liquid as it passes through the condenser. In doing so, the refrigerant rejects heat energy to its surroundings. In an ideal Carnot cycle, the condensation process occurs at constant pressure and temperature.

The fluid to which the condenser heat is rejected is usually either air or water (Stoecker & Jones, 1982, p. 244). The rate of heat transfer in the condenser is related to the refrigerating capacity and the evaporation and condensation temperatures (Stoecker & Jones, 1982, pp. 244, 245). "The condenser rejects both the energy absorbed by the evaporator and the heat of compression added by the compressor" (Stoecker & Jones, 1982, p. 245). After some time in operation, the heat transfer coefficient of a water cooled condenser will decrease due to fouling caused by the water (Stoecker & Jones, 1982, p. 247).

For condensers, the heat transfer mechanisms are more important than the pressure drop that occurs as the refrigerant flows in the tubes.

2.2.2.4 Evaporators

The refrigerant enters the evaporator in a mixed state of vapour and liquid. The relative amount of vapour to liquid increases as the refrigerant flows through the evaporator and evaporates. In the process of evaporation the refrigerant absorbs heat from its surroundings. In an ideal Carnot cycle, the evaporation process occurs at constant pressure and temperature.

The flow to the evaporator is regulated by an expansion device and is usually such that the refrigerant vapour is superheated upon leaving the evaporator (Stoecker & Jones, 1982, p. 253). This is done in order to ensure that only vapour enters the compressor. Any liquid entering the compressor can cause damage to it. Some systems make use of devices which separate any liquid which remains in the refrigerant stream after the evaporator and send it back to the evaporator (Stoecker & Jones, 1982, p. 247).

If the pressure upon exit from the evaporator is too low, more work will have to be done by the compressor in order to increase its pressure (Stoecker & Jones, 1982, pp. 256, 257). If pressure drop is allowed however, the velocity of the refrigerant can be kept high thus resulting in an improvement in the heat transfer coefficient (Stoecker & Jones, 1982, pp. 256, 257).

2.2.3 Expansion Devices

The purpose of the expansion device is to “reduce the pressure of the liquid refrigerant” and “regulate the flow of refrigerant to the evaporator” (Stoecker & Jones, 1982, p. 260). Several types of expansion devices exist including the capillary tube, the constant-pressure expansion valve, the float valve, and the superheat-controlled expansion valve (Stoecker & Jones, 1982, p. 260). The most frequently used expansion devices are the capillary tube and the superheat controlled expansion valve (Stoecker & Jones, 1982, p. 260).

2.2.3.1 Capillary Tubes

Capillary tubes are used in systems having a refrigerating capacity up to 10 kW (Stoecker & Jones, 1982, p. 260). The capillary tube is usually between 1 m and 6 m long and the inside diameter is usually between 0.5 mm and 2 mm (Stoecker & Jones, 1982, p. 260). The inside diameter of capillary tubes is too large for capillary action to occur therefore the name is a misnomer (Stoecker & Jones, 1982, p. 260).

Pressure drops occur across capillary tubes because of friction and acceleration of the refrigerant (Stoecker & Jones, 1982, p. 260). Flashing of liquid into vapour may also occur in a capillary tube (Stoecker & Jones, 1982, p. 260).

Several combinations of length and inside diameter can be used for a given pressure drop and flow rate, however, once the capillary tube is installed, it cannot adjust to variations in discharge pressure, suction pressure, or load (Stoecker & Jones, 1982, p. 260).

The compressor and expansion device must be able to achieve suitable suction and discharge conditions (Stoecker & Jones, 1982, p. 261). The conditions have to be such that the amount of refrigerant pumped by the compressor from the evaporator is equal to the amount of refrigerant fed to the evaporator by the expansion device (Stoecker & Jones, 1982, p. 261). However, the suction pressure fixed by the capillary tube and the compressor has to satisfy the heat transfer relationships of the evaporator (Stoecker & Jones, 1982, p. 261). If the heat transfer in the evaporator is not suitable, the result is said to be an unbalanced condition and the evaporator may end up starved or overfed (Stoecker & Jones, 1982, p. 261).

Capillary tubes have the advantage of being simple, having no moving parts and being inexpensive, however they do not adjust when load conditions change, are prone to clogging and require the mass of refrigerant charge to be held within close limits (Stoecker & Jones, 1982, p. 263). Holding the mass of refrigerant charge within close limits means that capillary tubes can only be used in hermetically sealed systems. (Stoecker & Jones, 1982, p. 263).

2.2.3.2 Constant-Pressure Expansion Valves

The pressure of the refrigerant as it comes out of the constant-pressure expansion valve remains constant and therefore the supply of refrigerant to the evaporator is always at a set, constant pressure (Stoecker & Jones, 1982, p. 271). The valve opens wider when the evaporator pressure falls below the control point and closes when the evaporator pressure rises above the control point (Stoecker & Jones, 1982, pp. 271, 272).

Constant-pressure expansion valves are used in units having refrigerating capacity not higher than 30 kW (Stoecker & Jones, 1982, p. 272).

2.2.3.3 Float Valves

Another type of expansion device is the float valve. The float valve operates by keeping the liquid in a vessel or an evaporator at constant level (Stoecker & Jones, 1982, p. 272). Float valves are controlled by float switches which open when the level drops below the control point and close when the level reaches the control point (Stoecker & Jones, 1982, p. 272). Balanced conditions are always present between the compressor and the float valve (Stoecker & Jones, 1982, p. 272).

This type of expansion device is used in large air conditioning installations (Stoecker & Jones, 1982, p. 273).

2.2.3.4 Superheat-Controlled Expansion Valves/Thermostatic Expansion Valves

The amount of superheat the refrigerant has upon leaving the evaporator controls this type of valve (Stoecker & Jones, 1982, p. 273). The rate of flow of liquid refrigerant is regulated according to the rate of evaporation inside the evaporator (Stoecker & Jones, 1982, p. 273).

Superheat-controlled expansion valves are most commonly used in moderately sized systems (Stoecker & Jones, 1982, p. 273).

2.2.4 The Refrigerant

The physical and chemical properties of a refrigerant are important for the refrigeration cycle, however, one also needs to take into account the risks associated with fire, explosions or poisoning in case leaks occur as well as the environmental impact such as ozone depleting potential (Eastop & McConkey, 1993, pp. 522, 523).

It is desirable for a refrigerant to have the following characteristics (Eastop & McConkey, 1993, p. 523):

- Saturation pressure at the desired low temperature above atmospheric but not much higher as leakage would then be harder to prevent and the components would need to withstand the higher pressures.
- A high specific enthalpy of vaporisation at the low temperature so as to be able to have lower mass flow rates of refrigerant for a given refrigeration capacity.
- Low specific volume at the suction to the compressor to keep the size of the compressor small.
- Inertness and miscibility with the lubricating oil.
- Low fire and explosion risks.
- Non-toxic or at least low toxicity in the amount of time it takes for evacuation of personnel.

2.2.4.1 Chlorofluorocarbons (CFCs) and Hydrochlorofluorocarbons (HCFCs)

CFCs and HCFCs both contain atoms of chlorine and therefore when released into the environment can cause ozone depletion. CFCs have been phased out and HCFCs are in the process of being phased out.

A very popular CFC was R12. When R12 was invented it was thought to be an ideal refrigerant since it has ideal thermodynamic properties, is non-toxic and has zero fire and explosion risk (Eastop & McConkey, 1993, p. 524), however, it is very harmful to the ozone layer.

R22 is a very commonly used HCFC. Systems making use of R22 still exist but they are being retrofitted with new refrigerants. HCFCs will be phased out completely by the year 2030 in developed countries according to the Montreal Protocol (US EPA, 2012).

2.2.4.2 Alternative Refrigerants

R134a and R717 (ammonia) are considered as suitable long-term solutions and can be used as alternative refrigerants since they have no chlorine atoms and therefore zero ozone depleting potential with respect to R11 (Eastop & McConkey, 1993, p. 527). R134a has a greenhouse potential of 0.37 with respect to R11 while ammonia has zero greenhouse potential (Eastop & McConkey, 1993, p. 527).

Ammonia has the disadvantage of being toxic (Eastop & McConkey, 1993, p. 528), however, its pungent odour makes leakages easily identifiable. Additionally, risks can be reduced by confinement of systems making use of ammonia in well-ventilated plant rooms ideally situated at the top of buildings and equipped with detectors and warning systems (Eastop & McConkey, 1993, p. 527). Air conditioning systems which make use of ammonia would need to make use of secondary or even tertiary coolant systems to confine the risk.

2.3 Heat Pump Modifications

Some common variations and improvements over the heat pump described in Section 2.1 will be discussed in this section.

2.3.1 Multistage Cycles

The use of multistage cycles is one of the improvements that can be carried out on a heat pump in order to improve the energy efficiency (Chua, Chou, & Yang, 2010). The concept of using multiple stages can be applied to both the compression part of the cycle and the evaporation part of the cycle (Chua, Chou, & Yang, 2010). For multistage compression one may have either compound or cascade systems (Bertsch & Groll, 2008). In a compound system, the two (or more) compression stages are in series (Chua, Chou, & Yang, 2010) while in a cascade system, there would be two single-stage refrigeration systems operated independently and connected by a cascade condenser (Bertsch & Groll, 2008).

A compound two-stage or multi-stage compression system (Friotherm, 2012) (Chua, Chou, & Yang, 2010) allows for a smaller compression ratio but higher compression efficiency for each stage of compression, greater refrigerating effect, lower discharge temperature at the high-stage compressor and greater flexibility (Chen, Li, Sun, & Wu, 2008) (Agrawal & Bhattacharyya, 2007). Compression ratios are usually chosen to be almost equal in the two stages so as to obtain high COPs (Agrawal & Bhattacharyya, 2007). Figure 2-6 (a) and Figure 2-6 (b) show two variations of the compound two-stage compression system.

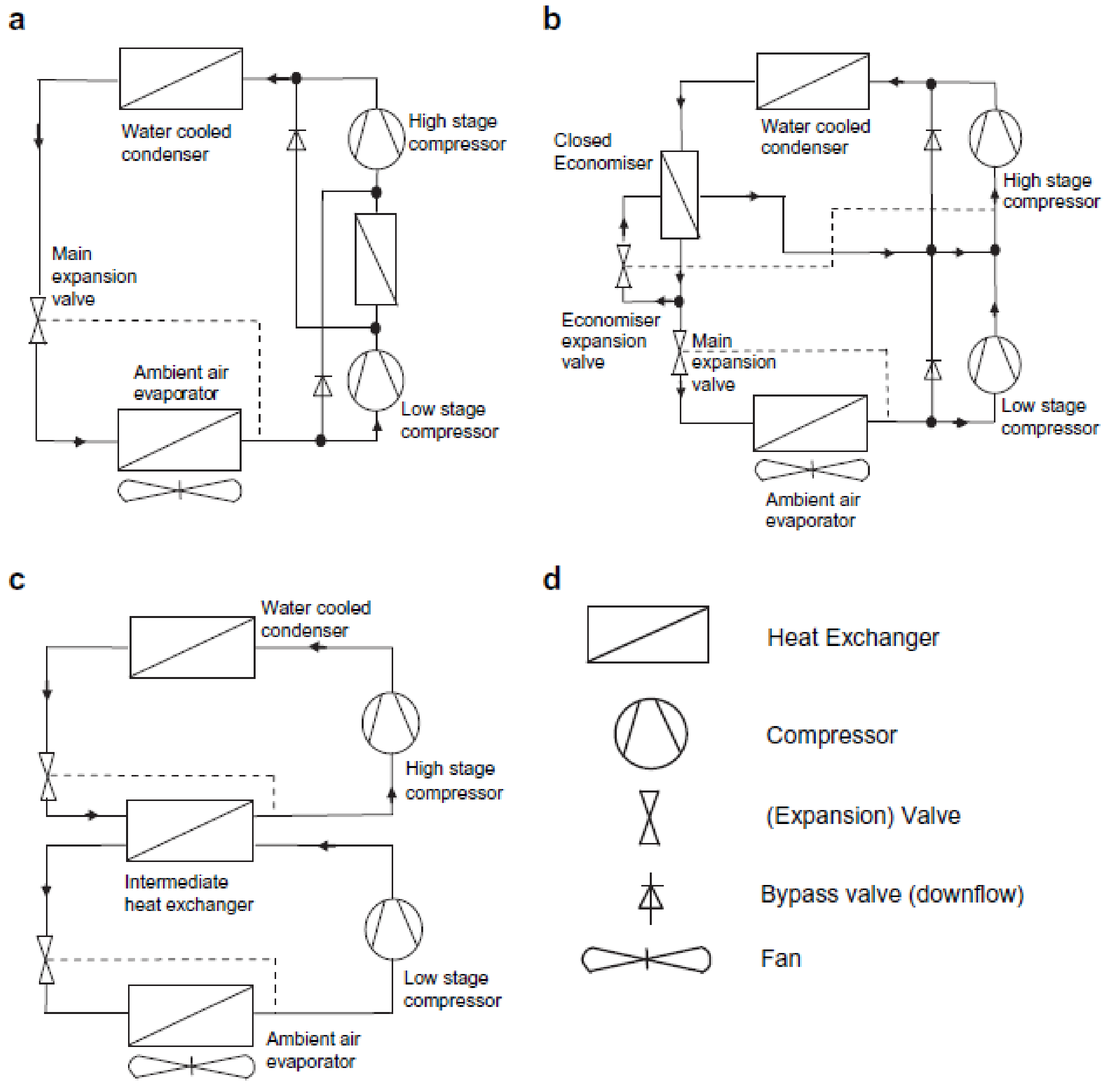

Figure 2-6 Schematics of (a) two-stage cycle with intercooler, (b) two-stage cycle with closed economizer, (c) cascade cycle, and (d) legend (Bertsch & Groll, 2008).

In a cascade system, such as the one shown in Figure 2-6 (c), the lower system produces a refrigerating effect while operating at a lower evaporating temperature (Bertsch & Groll, 2008). Through the cascade condenser, the heat rejected by the condenser of the lower system is extracted by the evaporator in the higher system which therefore operates at a higher evaporating temperature (Chua, Chou, & Yang, 2010).

In a two-stage evaporator heat pump system the physical area available for heat transfer is increased over that of a single-stage evaporator heat pump system (Chua, Chou, & Yang, 2010). This leads to drastic increases in the performance of the heat pump (Chua & Chou, 2005).

2.3.2 Improving Compressor Performance

As discussed in Section 2.1, much energy is consumed by the compression stage in a heat pump (Eastop & McConkey, 1993, p. 486). Therefore one way of improving the energy efficiency of a heat pump is by improving the performance of the compressor (Chua, Chou, & Yang, 2010).

This can be achieved by using a scroll compressor as opposed to a reciprocating compressor – the former is approximately 10% more efficient with respect to the latter (Chua, Chou, & Yang, 2010) due to three main reasons (Winandy & Lebrun, 2002): (i) suction and discharge are separate and therefore the suction gas does not receive any heat as it goes into the compressor, (ii) compression occurs slowly and over 540 degrees rather than 180 degrees of rotation, and (iii) the suction and discharge valves are made redundant when scroll compression is introduced and therefore can be removed thus reducing pressure losses. Additionally, fewer moving parts are found in a scroll compressor making it more reliable and operable in sluggish conditions (Chua, Chou, & Yang, 2010).

Another option is the revolving vane compressor which uses a rotating cylinder that moves with the compressing mechanism to cut down on loss of energy from friction and leaks (Teh & Ooi, 2009).

Improvements in compressor performance can also be achieved by keeping the compressor temperature low (Chua, Chou, & Yang, 2010). Two ways of doing this are by providing external cooling for the compressor motor and removing heat from the compressor so as to make the compression process isothermal (Wang, Hwang, & Radermacher, 2008).

2.3.3 Ejector-Compressor System

An ejector is used to compress the refrigerant up to a certain state by using low-grade thermal energy (Chua, Chou, & Yang, 2010). A gas-liquid separator is then used to separate the gas from the liquid (Chua, Chou, & Yang, 2010). Subsequently, the higher pressure gas is passed to the compressor and the liquid is passed to the evaporator (Chua, Chou, & Yang, 2010). Therefore upon entry to the compressor, the gas would already be compressed to a certain degree, meaning that the power requirement of the compressor is reduced (Chua, Chou, & Yang, 2010). In certain configurations where waste heat is available, the compressor, and therefore its power requirement, can be eliminated completely with the compression load undertaken entirely by the ejector (Chua, Chou, & Yang, 2010).

2.4 Some Options for Cooling with Water

Water for cooling can be obtained from a variety of sources including surface water[1] bodies, ground water bodies, inland water bodies and coastal water bodies.

Malta does not have any significant inland surface water bodies (MEPA, 2005) and the freshwater aquifer is under pressure from over abstraction. This only leaves the options of using the saline groundwater found under the fresh groundwater lens or seawater from coastal water bodies.

[1] Surface water consists of any body of water that is exposed to the atmosphere including ponds, lakes, oceans, and rivers (Mitchell & Spitler, 2013)

Systems making use of saline groundwater can classify as ground source systems. On the other hand, systems making use of seawater directly obtained from the sea are referred to as surface water heat pumps.

2.4.1 Systems Making Use of Saline Groundwater

One way of obtaining water for cooling is through boreholes which extract seawater from below the fresh groundwater lens (also known as seawells). Such systems classify as ground source systems since they exploit the relatively constant low temperature of the ground and groundwater as their cooling medium and are thus capable of reaching high efficiencies (US Department of Energy, 2012).

Ground source systems can be further subdivided into more classifications. One way in which they are classified is by the type of loop used; whether closed or open (US Department of Energy, 2012).

Closed loop systems circulate an antifreeze solution through pipes which are buried deep in the ground or submerged in a pond or lake (US Department of Energy, 2012). The solution rejects heat to the ground and then passes through a heat exchanger found above ground where it absorbs more heat from the refrigerant which would have previously absorbed heat from indoors. Closed loop systems can be of the vertical, horizontal, or pond/lake type referring to the configuration of the pipes buried under ground (US Department of Energy, 2012).

Horizontal closed loop systems have the advantage of lower trenching costs than well drilling costs but may need longer pipe lengths than their vertical counterparts (Omer, 2008). Due to the longer pipe lengths, horizontal systems also require higher pumping power (Omer, 2008). Also, they require large ground areas and can be affected by seasonal variance if the loop is found at a shallow depth (Omer, 2008). In general, horizontal systems exhibit lower efficiencies than vertical systems (Omer, 2008).

Vertical closed loop systems require less land area, less length of pipe and less pumping energy than horizontal systems; however, they require high drilling costs (Omer, 2008). Due to the depths at which the loop is installed, vertical systems are not affected by seasonal variation (Omer, 2008).

The pond/lake type of closed loop systems require pipes to be anchored some distance above the bottom of the pond or lake (Omer, 2008). This alternative requires the least pipe length and involves the least costs due to no trenching or drilling (Omer, 2008). However, its viability largely depends on the presence of a large water body which is suitable for this application (Omer, 2008).

Open loop systems extract surface or groundwater and circulate it through a heat exchanger where it absorbs heat from the refrigerant used in the chiller (US Department of Energy, 2012). The higher temperature water is then either rejected back to the aquifer through a separate well, or allowed to run off (US Department of Energy, 2012) or rejected to a surface water body such as a stream, pond, ditch, drainage tile, river or lake (Natural Resources Canada, 2009). Some factors that need to be studied when considering an open loop system are water quality, fouling, corrosion, blockage, adequacy of available water to provide the necessary flow rate, regulations and effluent discharge (Omer, 2008).

Open loop systems have simple designs, lower drilling requirements and lowest costs (Omer, 2008). However, they have the highest pumping requirements and may be subject to permitting and restrictions for water extraction and disposal (Omer, 2008).

Open loop systems are more adequate for cooling capacities exceeding 100 kW (Abesser, 2013) (Omer, 2008) and can obtain better efficiencies than closed loop systems under the same operating conditions (Cao, Han, Gu, Zhang, & Hu, 2009) (Omer, 2008).

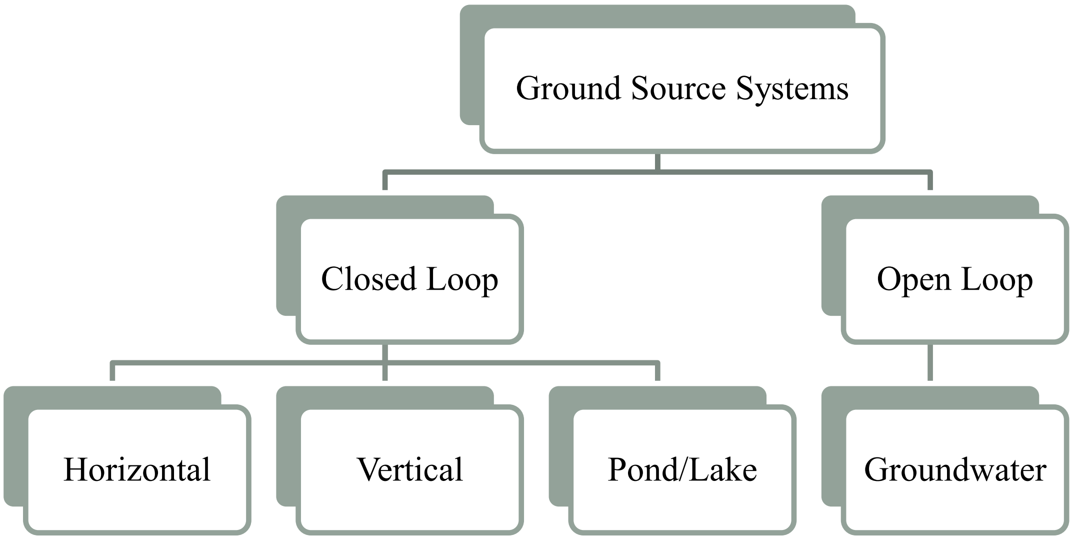

Figure 2-7 Classification of ground source systems (created from information obtained from: US Department of Energy, 2012).

In both closed and open loop systems and where temperatures permit, the heat pump system can be omitted and the antifreeze solution or ground water circulated directly around the area to be cooled (Natural Resources Canada, 2009). This type of system is referred to as a passive cooling system (Natural Resources Canada, 2009) (Omer, 2008).

2.4.2 Systems Making Use of Surface Water

The seawater temperatures and the temperatures desired indoors will play a very important role in choosing the type of system. Three options are available when using surface water for heating and/or cooling (Mitchell & Spitler, 2013):

1. Direct surface water cooling (DSWC) – water from the sea or a lake is used directly to cool buildings without a heat pump or chiller. Intermediate heat exchangers may be used to isolate surface water from the actual cooling system due to the fouling problems and corrosion that it may create.
2. Surface water heat pump (SWHP) – surface water acts as the source or sink for heat pumps or chillers used to provide heating and/or cooling. In spring and autumn two heat pumps can be used to provide heating and cooling simultaneously in different parts of a District Heating and Cooling (DHC) network (Hani & Koiv, 2012).

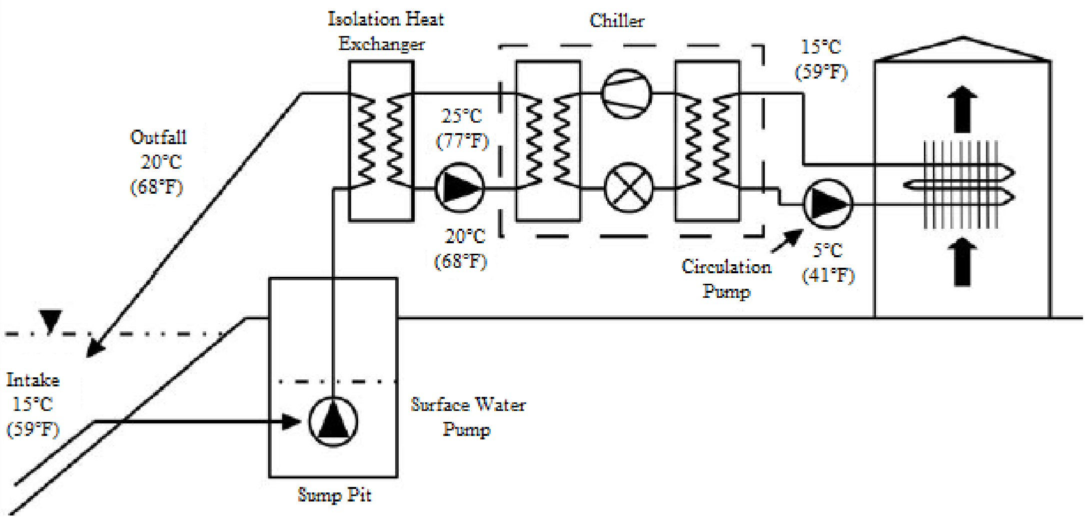

Figure 2-8 SWHP system configured for cooling purposes with an isolation heat exchanger and wet sump pump configuration (Mitchell & Spitler, 2013).

3. Hybrid surface water heat pump (HSWHP) – water is used directly for cooling when the temperature is low enough. When the temperature does not permit, heat pumps or chillers provide heating and/or cooling.

2.4.3 The Maltese Scene

As will be shown in Chapter 3, systems that are present in Malta make use of saline groundwater which is obtained from seawells. It is then common practice to discharge the higher temperature saline groundwater to sea. The practice of extracting seawater from seawells is carried out because of some advantages it has over direct extraction from the sea. The said advantages will be discussed in Chapter 3.

The Maltese systems therefore closely resemble the open loop ground source systems described previously. However, since the systems make use of what is essentially seawater, they are more closely related to surface water systems due to the water temperatures they experience and the harsh saline conditions to which they are exposed. In this regard, the review of literature focuses more on surface water systems.

The type of surface water system that most approximates the systems used in Malta is the SWHP wherein seawater is pumped to a heat exchanger where it absorbs heat from the chiller and is then rejected back to the sea.

2.5 Existing Systems

The properties of some existing systems are summarised in Table 2-2, Table 2-3 and Table 2-4.

For DSWC, water at a temperature as high as 18°C can be used to provide cooling (Ciani, 1978). This means that in some instances, surface and near-surface water is cold enough to be used in DSWC systems (Davidson, 2003). As can be seen from Table 2-2, distances from the cold water source to the installation tend to be large due to the low temperatures required for DSWC. Due to the large distances and associated infrastructure costs, only large systems are viable (Mitchell & Spitler, 2013). The economic feasibility of large scale DSWC systems is advantageous because the per litre cost of obtaining seawater decreases as the size of the pipeline increases (Davidson, 2003). The system COP for large DSWC systems can therefore be very high.

Table 2-2 Summary of existing DSWC systems (Mitchell & Spitler, 2013).

City	Helsinki	Ithaca	Mollis	Bora Bora
State/Province	Southern Finland	New York	Glarus Nord	French Polynesia
Country	Finland	United States	Switzerland	France
Latitude	60° 10' N	42° 26' N	47° 6' N	16° 30' S
Longitude	24° 56' E	76° 30' W	9° 5' E	151° 44' W
Water source	Gulf of Finland	Lake Cayuga	Lake Walen	Pacific Ocean
Cooling capacity, (MW)	-	70	-	1.6
Intake depth, (m)	7.5	76	60	900
Intake length, (m)	-	3200	-	2350
Intake temp, (°C)	8	4	6	4
Return depth, (m)	-	4.3	30	-
Flow rate, (m^3/h)	-	7300	2530	-
Operation date	2012	2000	-	2006
System COP	-	25.8	-	-
References	(Miller R. , 2011)	(Peer & Joyce, 2002) (Looney & Oney, 2007) (Zogg, Roth, & Broderick, 2008)	(Hamilton, 2010)	(Makai Ocean Engineering, 2011) (War, 2011)

As can be seen in Table 2-3, SWHP temperatures can be much higher than those required for DSWC. This means that the water inlet does not need to be deep down into the sea. However, COPs are lower in general than those of DSWC systems.

Table 2-3 Summary of existing SWHP systems (Mitchell & Spitler, 2013).

City	Nagasaki	Xiangtan	Drammen	Stockholm
State/Province	Kyushu	Hunan	Buskerud	Sodermanland
Country	Japan	China	Norway	Sweden
Latitude	32° 47’ N	27° 51’ N	59° 44’ N	59° 20’ N
Longitude	129° 52’ E	112° 54’ E	10° 12’ E	18° 4’ E
Water source	Nagasaki Harbour	Mengze Lake	Oslo Fjord	Baltic Sea
Heating capacity, (MW)	-	8.1	15	180
Cooling capacity, (MW)	4	15	-	-
Intake depth, (m)	-	2	-	15
Flow rate, (m^3/h)	762	-	-	-
Operation date	2000	2003	2010	1984
Inlet temp (cooling), (°C)	-	25-35	-	-
System COP (cooling)	2.9	3.8-4.2	-	-
Heat Pump COP (cooling)	-	4.3-4.8	-	-
Inlet temp (heating), (°C)	-	7-15	-	3
System COP (heating)	-	3.4-4	-	-
Heat Pump COP (heating)	-	3.8-4.3	3	3.75
References	(Song, Akashi, & Yee, 2007)	(Chen, Zhang, Peng, Lin, & Liu, 2006)	(Lind, 2010)	(Friotherm, 2012)

Table 2-4 summarizes the properties of some of the existing HSWHP.

Table 2-4 Summary of existing HSWHP systems (Mitchell & Spitler, 2013).

City	Oslo	Stockholm	Geneva	Toronto	Halifax
State/Province	Vestfold	Sodermanland	Geneva	Ontario	Nova Scotia
Country	Norway	Sweden	Switzerland	Canada	Canada
Latitude	59° 57' N	59° 20' N	46° 12' N	43° 43' N	44° 51' N
Longitude	10° 45' E	18° 4' E	6° 9' E	79° 20' W	63° 12' W
Water source	Oslo Fjord	Baltic Sea	Lake Geneva	Lake Ontario	Atlantic Ocean
Heating capacity, (MW)	2	27	-	-	-
Cooling capacity, (MW)	4.2	43	23.5	176	3.5
Intake depth, (m)	32	0-20	35	83	18
Operation date	-	1995	2009	2006	1989
References	(Smebye, Midttømme, & Stene, 2011)	(Fermbäck, 1995) (Friotherm, 2012)	(Viquerat, Lachal, Weber, Mermoud, & Pampaloni, 2008) (Viquerat, Lachal, & Hollmuller, 2012) (Viquerat, 2012)	(Fotinos, 2003) (Zogg, Roth, & Broderick, 2008) (Newman & Herbert, 2009)	(Eliadis, 2003) (Newman & Herbert, 2009) (War, 2011)

Although the focus of this research is on SWHP, it is worthwhile to notice that in Malta there might be potential for DSWC; the temperature of the mean sea-level aquifer is approximately 20°C (MRA, 2009) and seawater temperatures reported by managers of existing systems in Malta are between 16°C and 25°C (refer to Table 3-1). This might indicate that saline groundwater at bigger depths could potentially be colder than the required 18°C for DSWC (Ciani, 1978). Additionally, in its Consultation Paper regarding the 'Application of Groundwater Heating/Cooling

Schemes in Malta and Gozo', the Malta Resources Authority includes direct DSWC systems (MRA, 2009). If this should not prove to be the case, HSWHP could prove to be a solution since they can be used as DSWC systems when water temperatures are cold enough and SWHP systems when water temperatures are higher.

2.6 Requirements, Constraints and Difficulties

Operation of a SWHP requires water temperatures higher than 2°C for heating and temperatures lower than 26°C for cooling (Cao, Han, Gu, Zhang, & Hu, 2009) (Zhen, Lin, Shu, Jiang, & Zhu, 2007) and can be installed in a wide variety of climates (Mitchell & Spitler, 2013). The outdoor climatic conditions together with the building size will determine the heating and cooling loads (Hani & Koiv, 2012).

Depths at which the necessary water temperatures occur are another determining factor in establishing the cost of the system – costs and technical difficulties increase with increasing depth (Cao, Han, Gu, Zhang, & Hu, 2009) (Zhen, Lin, Shu, Jiang, & Zhu, 2007). A water temperature profile will in general be needed to assess the feasibility of the system (Hani & Koiv, 2012). In Malta water temperature profiles for seawells seem to be lacking or not made available because they are held by companies responsible for designing systems. Otherwise it is possible that system designers simply assume that the temperature of the water extracted from the seawells will satisfy their requirements.

For SWHP systems, the distance from the shore at which suitable sea temperatures occur is another important constraint since despite the piping being insulated (Hani & Koiv, 2012), transporting cold sea water through warmer areas (and vice versa) may result in an increase (or decrease) in its temperature, thus causing the system to not remain as efficient. This is not so much of a problem for the systems used in Malta since seawater is only transported vertically and the temperature of the ground and groundwater is more stable than that of seawater obtained from open seas (Miller D. W., 1956) (Rodríguez-Estrella & Pulido-Bosch, 2009).

For a ground source heat pump system (GSHP) one would need to consider the geology of the coastal area where it is proposed (Hani & Koiv, 2012). This factor will determine whether it is possible to feasibly construct a seawell.

2.6.1 Corrosion and Fouling

Seawater tends to be extremely corrosive and can also be rich in living organisms; corrosion and fouling impact negatively on system performance since they increase frictional losses, cause components to fail and alter the heat transfer mechanisms which are very important to the cooling process. It is therefore beneficial to limit the number of components that come into direct contact with seawater.

The use of a heat exchanger enables heat to be extracted or rejected from or to the seawater without circulating the seawater around the entire system. Therefore only a few components will be in direct contact with the seawater. The heat in the seawater is transferred to freshwater (or vice versa depending on the mode of operation) which is then allowed to circulate in the system. The freshwater has a much lower potential for corrosion and fouling and simple measures can be taken to prevent these.

The components that come into contact with the seawater are essentially the piping to and from the heat exchanger, the heat exchanger itself, any filters used and in some cases, the pump used to extract the seawater (Cao, Han, Gu, Zhang, & Hu, 2009).

The salty nature of seawater poses an important challenge for material selection of components that come into direct contact with it. Additionally, other seawater parameters such as suspended solids, water temperature and pH will also play a role in the selection of the material (Mitchell & Spitler, 2013). Salinity and water temperature are critical to electrochemical corrosion while suspended solids will cause erosion which leads to increased clearances in components such as the pump and thus result in reduced pump performance (Mitchell & Spitler, 2013). Materials such as stainless steel, carbon steel and copper alloys will be attacked by dissolved chlorides that are found in seawater and will thus suffer from pitting, crevice corrosion and inter-granular corrosion (Mitchell & Spitler, 2013). Stress corrosion cracking is also of concern in stainless steel components if warmer waters are present – this is especially true for the heat affected zones at welds (Mitchell & Spitler, 2013).

Fouling of the system, which occurs due to the larvae and eggs of marine creatures, can be prevented through the use of sodium hypochlorite which can be generated by a seawater electrolytic process (Cao, Han, Gu, Zhang, & Hu, 2009) although extracting seawater from a seawell should help in reducing this problem. The pump and heat exchanger can be cathode polarised to prevent galvanic corrosion (Cao, Han, Gu, Zhang, & Hu, 2009).

Further details relating to material selection of specific components are given in the following section.

2.7 Major System Components and Sub-Systems

2.7.1 Seawell

Seawells can be from 20 m to 200 m deep and their diameter can range from 100 mm to 200 mm (Yang, Cui, & Fang, 2010). Boreholes that are intended for use as seawells need to be as close as possible to the coastline (30 m to 50 m) (Rodríguez-Estrella &

Pulido-Bosch, 2009). This ensures that only seawater is extracted and that salinization of other boreholes further inland is prevented (Rodríguez-Estrella & Pulido-Bosch, 2009). The distance between boreholes should also be such that well interference is avoided (Maliva & Missimer, 2011).

The cost for drilling the seawell is a major contributor to the high initial costs of GSHP systems, for this reason, more cost-effective drilling technology should be sought so as to improve the life-cycle performance of GSHP systems (Yang, Cui, & Fang, 2010). Drilling should also be carried out with the potential problem of land subsidence in mind (Maliva & Missimer, 2011). Precautions need to be taken in this respect.

The borehole annulus is usually cemented or backfilled over the freshwater band, mixing zone and several metres below the mixing zone with special material called grout so as to avoid contamination of the fresh groundwater with saline water (Yang, Cui, & Fang, 2010) (Jorreto, Pulido-Bosch, Gisbert, Sánchez-Martos, & Francés, 2009) (Rodríguez-Estrella & Pulido-Bosch, 2009). Cementation below the mixing zone is required as a precaution against changes in the geometry of the contact zone once extraction begins (Rodríguez-Estrella & Pulido-Bosch, 2009). Backfilling of boreholes is an expensive, specialised process requiring heavy equipment and trained personnel (MRA, 2009). Despite this, failures are not uncommon, the cement quickly deteriorates and isolation of the aquifer is not guaranteed (MRA, 2009). Therefore monitoring of the salinity of the water being extracted is important in order to identify any changes that could lead to problematic scenarios (Maliva & Missimer, 2011).

2.7.2 Intake Piping and Screening

Seawater is obtained through intake piping which is equipped with screens that prevent debris from entering the system. The intake piping and screening make up the majority of the capital cost of a SWHP system (Mitchell & Spitler, 2013) due to the long lengths of piping required to transport the seawater. However, this might not apply to the type of systems making use of seawells since these are usually located close to the location where they are to be used and thus long stretches of piping are avoided. Drilling a borehole nevertheless might prove to be a significant part of the initial cost of the system (FWT Ltd) especially if very deep boreholes are needed in order to obtain the required flow rate for the system.

Different locations with different characteristics will require intake systems with different designs and sizes (Mitchell & Spitler, 2013). For a SWHP, the bathymetry and temperature of the seawater will play a very important role in establishing the design of the pipeline and the depth from which seawater is to be extracted (Mitchell & Spitler, 2013). At the design stage, consideration should also be given to the scouring effect, that is, “the increased transportation of sediments underneath and/or around structures involving a moving fluid” (Elahee & Jugoo, 2013). The scouring effect is a threat to stability (Elahee & Jugoo, 2013). In systems making use of seawells, these problems are somehow minimised and it is the type of rock present which plays a more important role rather than the bathymetry.

It is also important to note that for seawater extracted directly from the sea, as the seawater travels through warmer layers closer to the surface it will warm up (Mitchell & Spitler, 2013). The same applies for systems making use of seawells. However, since the temperature of seawater in seawells is more stable, the effects will be less pronounced.

Some materials used for the intake piping include steel lined with glass or concrete and covered with insulation or concrete (Hirshman, Whithaus, & Brooks, 1975), high density polyethylene (HDPE) (Hirshman, Whithaus, & Brooks, 1975) and fiberglass (Ciani, 1978) (Hirshman & Kirklin, 1979). Piping made from materials which are prone to corrosion can be protected by using epoxy resin paint or other such coatings where appropriate – these coatings serve to isolate any metal from the seawater (Cao, Han, Gu, Zhang, & Hu, 2009). The industry standard is currently HDPE since it provides certain benefits such as; chemical inertness, fusibility, flexibility, strength, lower density than seawater (therefore will float and make installation easier), low thermal conductivity (Yang Y. , 2007) (thus preventing warming up of the seawater as it passes through warmer layers of water) (Mitchell & Spitler, 2013) and smoothness (which decreases frictional pumping losses) (Hirshman, Whithaus, & Brooks, 1975).

In systems making use of seawells the choice of piping material usually depends on the type of pump used. When borehole pumps are used, piping is usually not made out of plastics because of the weight of the borehole pump which needs to be suspended inside the borehole itself. However, when the water level inside the seawell permits the use of a surface pump, the piping can be made of plastics since these do not need to be suspended inside the borehole itself.

A coarse, primary, radial screen is suggested at the intake of the SWHP system in order to obtain water from a single layer of the water column and also not to disrupt other layers or the sea floor (Mitchell & Spitler, 2013). This dynamic is of less concern in systems making use of seawells. The size of the screen should be designed in such a way to maintain an adequate face water velocity and thus avoid the entrainment of marine organisms (Mitchell & Spitler, 2013), especially those having low mobility and those whose size is smaller than the mesh (Avery & Wu, 1994). Marine organisms which are drawn in are then exposed to impingement and to a rapid change in environment (Elahee & Jugoo, 2013). This endangers their survival and may cause damage to the system (Elahee & Jugoo, 2013).

Seawater obtained from seawells usually requires less filtration due to the fact that it would already have been filtered through the rock and due to the fact that it would contain fewer organisms. However, a coarse filter is still generally used to prevent the entrainment of debris.

If the geology permits, an alternative to the radial screen is using the seabed itself as a filter by drilling horizontally under it (Mitchell & Spitler, 2013). This practice is similar in principle to the use of seawells.

2.7.3 Pumps and Pumping Configurations

The surface water pump is one of the major energy consumers in a SWHP system (Mitchell & Spitler, 2013). It is desirable to have the minimum altitude possible between the heat exchangers and the water body in order to reduce the pump power requirements (Hani & Koiv, 2012). Also, there has to be a compromise between the choice of pump and the diameter of the intake piping – a large pipe diameter will enable the use of a smaller pump, and therefore reduce the operating cost, but will increase the capital cost of the system (Mitchell & Spitler, 2013). Hence one has to consider the lifecycle cost in order to strike a balance between the intake pipe diameter and the pump to be used (Mitchell & Spitler, 2013). Intake pipe diameter and pump size are also determined by the flow rate needed to cool the condenser. It is usually desirable for the pumps to be accessible in order to carry out maintenance more easily.

SWHP systems can have their pumps configured in one of two predominant ways – the wet sump configuration or the dry sump configuration (Mitchell & Spitler, 2013).

The wet sump configuration requires the pump in the sump to be immersed in water. When the system is not being used, the water level in the sump is equal to the sea level (Mitchell & Spitler, 2013). When water is pumped up to the building, there is a difference in head between the sump and the sea level and thus more water flows into the sump (Mitchell & Spitler, 2013). It is also possible to operate the system in reverse, that is, if water is pumped into the sump, the head of the water in the sump will exceed that of the sea level and water will flow out from the sump to the sea to balance the head (Mitchell & Spitler, 2013). The reverse flow can be useful to clean any filters in the intake system (Mitchell & Spitler, 2013). Careful design will also make it possible to incorporate a mechanical pipe pig that removes certain marine organisms such as zebra mussels or contaminants that attach themselves to the inner walls of the piping (Cornell University, 2005).

The only constraint on the wet sump configuration is that it requires that the difference between the head at the inlet and the head at the outlet causes enough flow rate in the pipeline for proper operation of the system (Mitchell & Spitler, 2013). Other considerations may include any dredging needed to install the pipeline (Hirshman, Whithaus, & Brooks, 1975) and vortex formation at the pump inlet (Mitchell & Spitler, 2013).

In the dry sump configuration, water is pumped directly by the pump to the building (Mitchell & Spitler, 2013) and thus – as the name implies – the pump remains dry.

The pump still needs to be situated at or below the sea level with the only constraint being that of requiring the net positive suction head available to be greater than the net positive suction head required (Cengel & Cimbala, 2010). However, if possible, it is desirable to situate the pump below sea level since this has the added benefit of reducing the probability of pump cavitation (Mitchell & Spitler, 2013).

The wet sump configuration is somewhat similar to having a seawell the difference being that when seawater is obtained directly from the sea one needs to create a piping system that supplies water to the sump. In the case of a seawell the infiltration of water occurs naturally through the rocks thus avoiding the need for a piping system and reducing the initial cost of the system. However, seawells have to be deeper than sumps.

The type of pump mainly used in open loop ground source systems is the submersible pump (Rafferty, 2001). However, surface pumps are also an option and are sometimes preferred due to them having fewer parts which come in contact with seawater.

It is suggested that the pump be made of titanium and nickel alloys which are resistant to seawater corrosion and erosion (Keens, 1977). However, these materials are expensive and other materials can be used. Pump components that experience lower flow rates (up to 15 m/s) can be made out of high-nickel ductile iron and nickel-aluminium-bronze since these are resistant to erosion and galvanic corrosion (Mitchell & Spitler, 2013). Components that experience these conditions include the suction bell and column, or discharge head (Mitchell & Spitler, 2013). The impeller, shroud and casing experience higher flow rates and therefore stainless steel 316L makes a suitable candidate (Antunes, Cornman, & Hartkopf, 1981) if a sacrificial anode is used to protect it from pitting and other forms of galvanic corrosion to which it is susceptible (Mitchell & Spitler, 2013).

2.7.4 Isolation Heat Exchanger

The choice of the heat exchanger depends on the heat transfer rate required, maintenance requirements, expected fouling resistance, pressure drop, space requirements and cost (Mitchell & Spitler, 2013). The heat exchanger area depends on the temperature of the seawater supplied to it and on the temperature required for the chilled freshwater (Davidson, 2003). In general, the lower the temperature of the supplied seawater, the smaller the heat exchanger needs to be and hence the cost of the heat exchanger will be lower (Davidson, 2003). Optimization of the balance between the intake depth and the heat exchanger size should be sought (Davidson, 2003). The heat exchanger and associated pumps should be located as close to the shore as possible (Davidson, 2003).

The heat exchanger is susceptible to electrochemical corrosion since it comes into contact with the seawater therefore titanium is the most suitable material for the heat exchanger (Fermbäck, 1995) (Leraand & Van Ryzin, 1995) (Newman & Herbert, 2009) (War, 2011) (Smebye, Midttømme, & Stene, 2011) (Hani & Koiv, 2012) (Davidson, 2003). Titanium heat exchangers have the benefit of having high thermal efficiencies and being available 'off-the-shelf' (Davidson, 2003).

The heat exchanger is vulnerable to performance degradation due to fouling by marine organisms which includes biological film, algae, slime and/or molluscs (Mitchell & Spitler, 2013). Fouling degrades performance since it increases the thermal resistance and gives rise to the need for higher pumping power (Mitchell & Spitler, 2013). The use of a seawell can help reduce the effects of fouling.

Biocides can be used in order to overcome problems caused by biological organisms (Mitchell & Spitler, 2013) (Moreira, Brown, & Coleman, 2010). Biocides can be grouped into two main types; oxidising biocides such as chlorine dioxide, sodium hypochlorite, ozone and bromine and non-oxidising biocides such as amines, copper salts and quaternary ammonium salts (Moreira, Brown, & Coleman, 2010). Oxidising biocides are extensively utilized, however, these tend to corrode pipes, require high doses for effectiveness and produce by-products which are chlorinated and halogenated and may be toxic (Abarnou & Miossec, 1992). For chlorine-based oxidising biocides, it was noticed that corrosion increased with an increase in the amount of chlorine used (Fava & Thomas, 1978). However, the use of chlorine-based biocides used to be the most efficient and cost effective solution against fouling by marine organisms (Chien, Tse, & Yeung, 1986).

Film-forming biocides such as amines are now substituting oxidising biocides (Moreira, Brown, & Coleman, 2010). The former are less corrosive, require smaller doses and have less ecological impacts (Moreira, Brown, & Coleman, 2010).

Unfortunately, there is inadequate information regarding the environmental impacts of biocides and therefore decisions relating to which biocide should be used are usually more biased towards cost (Moreira, Brown, & Coleman, 2010).

In addition to biocides one can use mechanical fouling control which substantially reduces the dose of harmful chemicals used (Nebot, Casanueva, Casanueva, Fernández-Bastón, & Sales, 2006). One example of mechanical fouling control in shell-tube heat exchangers is the installation of a permanent brush system (Mitchell & Spitler, 2013). Other options include disassembly and cleaning, pre-treatment of used seawater to remove chemicals prior to disposal (Elahee & Jugoo, 2013), adding an extra heat exchanger to compensate for the fouling and operating the heat exchangers intermittently in order to prevent them from fouling at the same time (Mitchell & Spitler, 2013).

2.7.5 Heat Pump or Chiller

This component would be specifically designed for the site, application, operating temperatures and conditions and in some cases will allow for both heating and cooling or either one (Mitchell & Spitler, 2013).

2.7.6 Return Piping

The water that is returned from the system to the water body is at a different temperature than when it entered the system therefore if the water is returned at the surface there is a risk that the surface warms up while the deeper layers cool further due to the extraction (Avery & Wu, 1994). This implies that care should be taken when rejecting the water back to the water body in order to avoid temperature disturbances in the marine ecosystem. Such disturbances may result in reduced hatching success of eggs, inhibition of larvae development (Elahee & Jugoo, 2013) and in more severe cases death among corals and fishes (Pelc & Fujita, 2002). The outfall location should be chosen such that the water is discharged back into the sea at the isothermal depth (Avery & Wu, 1994) and over a large area to avoid temperature gradients (Mitchell & Spitler, 2013). In the presence of currents of fluctuating intensity and direction and trade winds, these effects may be reduced or even neglected (Avery & Wu, 1994) (Elahee & Jugoo, 2013).

Water extracted from seawells will have different nutrient contents than seawater therefore when rejected back to the water body it is important to ensure that the nutrient balance of the sea is not upset at the outfall location (Mitchell & Spitler, 2013). Nutrient enriched zones near the coast may cause higher concentrations of jellyfish, attract predators such as sharks and cause toxic algal blooms (Elahee & Jugoo, 2013). These are all threats to humans which additionally may affect shore-based businesses and the fishing industry (Elahee & Jugoo, 2013).

Alternatively, instead of discharging the water back into the sea, one could pump it to a desalination plant. This approach saves on pumping costs at the desalination plant and prevents temperature and nutrient disturbances in the water body (Mitchell & Spitler, 2013). Cooling ponds or cooling towers may also be used in order to cool the seawater to a temperature comparable to that of the water body, prior to being discharged (Mitchell & Spitler, 2013) (Wallis & Aull, 2009).

Refer to Section 2.7.2 for return piping material requirements and selection.

2.8 Sustainability Considerations

2.8.1 Economic and Policy Considerations

SWHP as well as GSHP systems tend to involve large capital costs but have longer lifespans and lower operating costs than conventional systems (Helm, 2010) (Zhen, Lin, Shu, Jiang, & Zhu, 2007) (US Department of Energy, 2012). The initial cost depends on several factors including the temperature of the seawater (Van Ryzin & Leraand, 1991) (Yu, 2008), the distance from the shore and depth at which suitable sea temperatures occur for SWHP systems or the depth of the borehole required in order to satisfy the flow rate needed for the heat pump in GSHP systems, the distribution system required onshore, the size of the system (Samuel, Nagendra, & Maiya, 2013) and the support from government and local authorities (Van Ryzin & Leraand, 1991) (Yu, 2008). Additional costs are also involved due to the higher maintenance required because of corrosion and fouling (Yang, Cui, & Fang, 2010) and any effluent treatment required (Van Ryzin & Leraand, 1991) (Yu, 2008). The viability of the system will also depend on percentage utilisation, local cost of electricity and secondary uses of effluent (Van Ryzin & Leraand, 1991) (Yu, 2008).

The cost of producing or removing thermal energy when using seawater as the source or sink will be lower than that for producing thermal energy using air source heat pumps since a more efficient heat source and sink is used. Additionally, cooling with water is more effective than cooling with air due to the higher specific heat capacity of water (Omer, 2008).

During the delivery of the seawater to the system, it is of utmost importance to minimise heat losses or heat gains.

The costs can be reduced if the system is incorporated in the planning phase of a construction rather than retrofitted (Samuel, Nagendra, & Maiya, 2013). Another advantage of systems making use of water is that they can be constructed gradually (Pavković, Delač, & Mrakovčić, 2012). The water pipelines would be placed at the construction phase of a facility and heat pumps could then be brought online according to demand until the required capacity is obtained (Pavković, Delač, & Mrakovčić, 2012). This advantage is especially evident for DHC systems.

The time-value of money must also be taken into consideration when assessing the feasibility of a system. At the time of construction, money will have a different value than at the time of operation when savings from the system can be made thus the discount rate has to be chosen carefully (Hawkey, Webb, & Winskel, 2013).

The payback time of systems making use of seawater is of a few years since the capital cost can be offset by the energy savings obtained from the system due to its higher efficiency (US Department of Energy, 2012). The rate of return will depend on many factors including cost of electricity, cost of monitoring and pollution control (Elahee & Jugoo, 2013).

For return on investment, legislation should aim to monetize the externalities associated with fossil fuel use and the benefits associated with sustainable systems (Euroheat & Power, 2013). Such benefits include avoided energy imports, price stability, energy security and environmental savings (Euroheat & Power, 2013) (IEA DHC/CHP Executive Committee, 2002). One way of monetizing the externalities associated with fossil fuel use is to impose taxes on fossil fuel users (Reihav & Werner, 2008) (IEA DHC/CHP Executive Committee, 2002).

Sustainable urban development policies should encourage the use of sustainable cooling systems (IEA DHC/CHP Executive Committee, 2002). Policies aiming to improve the penetration of renewable energy sources of heating are still being developed and refined (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012). Support for renewable sources of energy for heating and cooling can be both financial and non-financial (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012). Financial instruments include (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012):

- Grants or investment subsidies that defray the high capital costs.
- Public procurement, that is, implementing technologies in public buildings.
- Tariff or bonus mechanisms such as feed-in schemes and bonuses on top of the revenue made

- Tendering mechanisms involving competitive bidding for contracts among system developers.
- Levies and tax related instruments which exempt the desired technology and thus induce a behaviour change. Some examples of tax related instruments are tax credits, value added tax and enhanced depreciation.
- Soft loans to provide capital below the market rate.
- Support for research, development and demonstration

Non-financial instruments include (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012):

- Use obligations requiring a minimum amount of energy to come from renewable sources.
- Skills, education and training to support the growing industry.
- Information, awareness and promotion strategies.
- Standardisation of systems and setting minimum performance standards to increase consumer confidence.

All the instruments mentioned have advantages and disadvantages and can be modified according to the different aims and scenarios in different countries. In some instances, one instrument will be more suitable for targeting large scale developments while another instrument will be more appropriate for supporting smaller scale installations. Due to the very different scenarios present it may be useful to combine different policy instruments and perhaps even combine some obligations for renewable sources of heat with building regulations (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012). However, it is important to keep in mind the interaction of policy instruments aimed at supporting renewable sources of heat with other policies as in some cases there may be conflicts (Connor, Bürger, Beurskens, Ericsson, & Egger, 2012).

2.8.2 Environmental Considerations

Cooling using seawater as the heat sink is conducive to better environmental conditions through increased efficiency, its renewable nature and the use of energy which would otherwise go unused. This implies that carbon emissions can be reduced as well as air pollution, stratospheric ozone depletion and acid rain (Samuel, Nagendra, & Maiya, 2013). Heat pumps having a COP higher than 2.5 are considered to be less environmentally harmful than gas fired boilers (Hani & Koiv, 2012).

However, such systems are not completely free from environmental concerns. Power is still required for operating the compressor; therefore the degree to which environmental impacts can be averted depends on the source of the energy used to produce that power (Omer, 2008). There might also be concerns over carbon dioxide outgas which occurs when seawater is extracted from great depths (Elahee & Jugoo, 2013). The world's oceans are an important carbon sink – carbon dioxide is stored in its dissolved state in seawater and the concentration of dissolved carbon dioxide increases with depth (Elahee & Jugoo, 2013). Upon bringing seawater to the surface, its pressure is reduced and its temperature is increased thus the solubility of the dissolved gas is decreased and carbon dioxide gas is released (Elahee & Jugoo, 2013). Impingement and entrainment of marine organisms, the scouring effect, the use of biocides, upsetting of the nutrient balance in the marine environment and the thermal pollution caused in the sea are additional concerns that need to be addressed. These have been discussed in Sections 2.7.2, 2.7.4 and 2.7.6 respectively.

2.8.3 Social Considerations

The energy-efficient nature of SWHPs and GSHPs can aid in reducing electricity peak loads in summer when cooling demands are higher (Euroheat & Power, 2013) (IEA DHC/CHP Executive Committee, 2002) and could potentially lead to the avoidance of power cuts and therefore a more reliable supply of electricity to all consumers. Additionally, owners of SWHP and GSHP systems can benefit from reduced cooling and operation costs (MacRae, 1992).

The use of cooling systems which reject heat to seawater can reduce emissions due to higher efficiency and lower fuel consumption (MacRae, 1992). This results in an improvement in environmental quality and ultimately human health (MacRae, 1992).

The implementation of systems is constrained by the socio-technical aspect of the respective location which results in systems that are place-specific (Hawkey, Webb, & Winskel, 2013). Socio-technical considerations include physical characteristics and other factors such as ownership of land and buildings, existing systems and energy contracts, consumer habits, expectations and willingness to use the systems and interfaces with other systems (Hawkey, Webb, & Winskel, 2013) (Summerton, 1992).

Chapter 3

Visits

3 Visits

Several visits were carried out to facilities making use of saline water for cooling chillers. This chapter gives some information about the systems being used in the visited facilities. All of the facilities visited are hotels located in the St. Julian's area. Other facilities making use of saline water for cooling chillers are present in Malta.

3.1 InterContinental (Borg, 2013)

The InterContinental Hotel in St. Julian's uses seawater (as opposed to air) to cool the chillers which provide cooling of the facilities. This is possible since the hotel is located close to the sea.

Twenty-one metre deep boreholes are used to obtain seawater. Three boreholes are used at the InterContinental hotel for cooling purposes, one each for each of the three 284 kW chillers present. Access to the boreholes is through the car park which is the lowest level in the facility. The car park is approximately 10 m below street level and 1 m above sea level.

At the moment (July), the temperature of the seawater obtained from the boreholes is approximately 19°C. In high summer, the temperature of the seawater obtained from the boreholes rises to approximately 22°C (also due to fouling) while in winter the seawater temperature goes down to 16°C. No filters are used at the borehole since the seawater would have already been filtered through the rocks.

Following the pumping of seawater from the borehole by means of a 30 kW submersible multistage pump coupled to an inverter, the seawater is passed through a coarse strainer and a plate heat exchanger. It is then discharged back into the sea, thus making this an open loop.

No precautions are taken when water is thrown back to sea. At peak time, temperatures of the seawater flowing out from the heat exchanger are close to 32°C.

The pipes making up the open loop seawater circuit are made of stainless steel and plastic. The pressure in the open loop seawater circuit only goes up to a maximum of 2 bar thus making plastic suitable for the application.

The condenser pump, heat exchanger and condenser make a closed loop. The 13 kW or 15 kW condenser pump, pumps freshwater in the closed loop circuit so that the freshwater can exchange heat with the seawater by means of the heat exchanger and cool down the condenser. The freshwater in this closed loop circuit is used for cooling the condenser instead of having fan blowers which blow air to cool the condenser.

Of the three chillers present, one of the chillers is always being used while the other two are mostly left unused. Of the latter two chillers, one acts as an aid to the chiller that is always in operation during peak times while the other is dedicated to providing cooling for the part of the hotel used for conferences.

The chillers pass water to headers through the use of 7.5 kW pumps which are equipped with speed drive so as to control the speed and consequently reduce energy consumption. Two headers are present; one for public areas and one for guest rooms. One can decide where to send cool water to decide which place to cool.

Every operating parameter of the chillers can be controlled through the Building Management System (BMS) of the hotel.

The chiller system using seawater for condenser cooling was incorporated at the planning stage of the hotel. The implementation of the system was possible since, at the hotel's location, the cost incurred to dig to the depth at which the seawater can be found is not prohibitive.

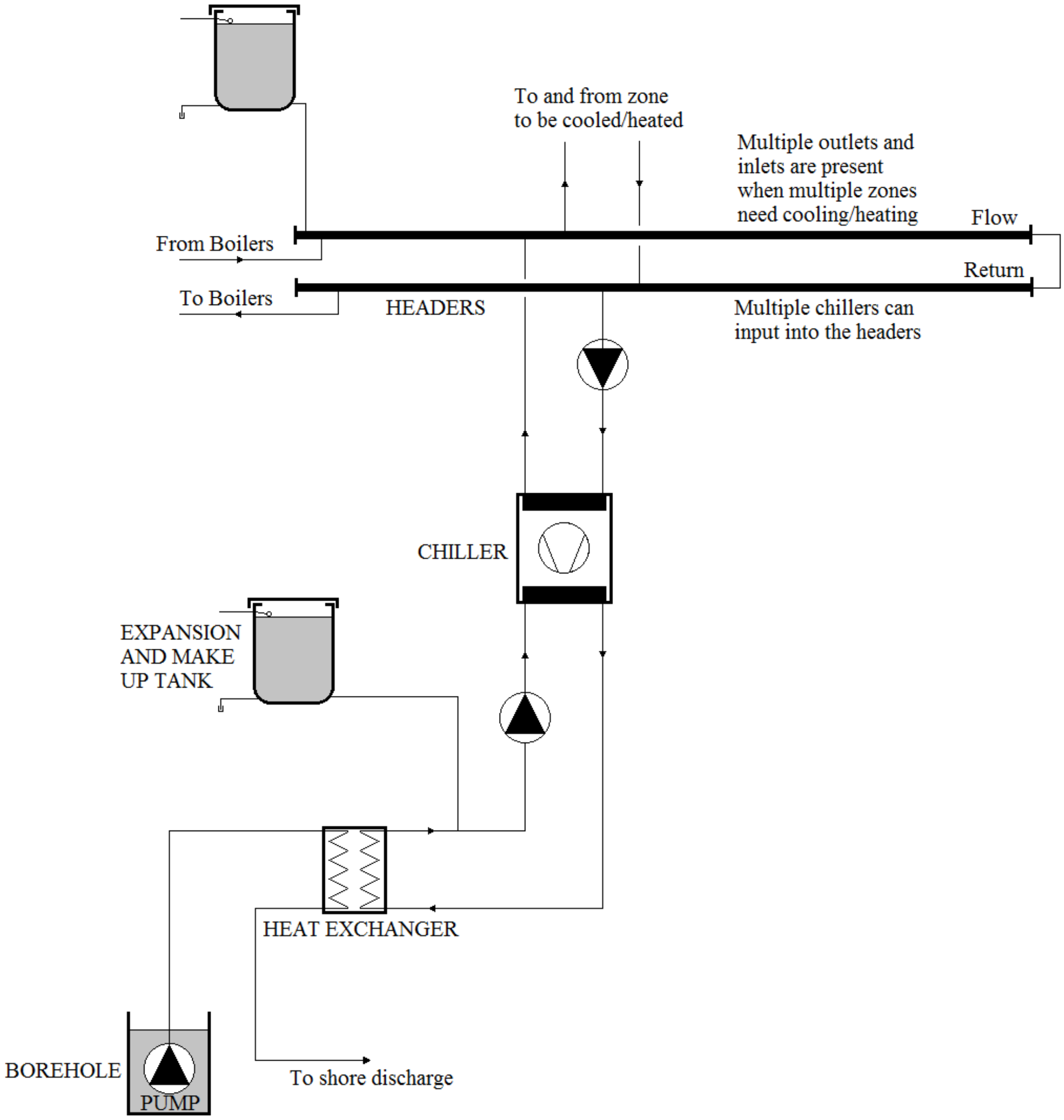

Figure 3-1 Schematic diagram of the InterContinental Hotel's chiller system (reproduced with permission).

3.1.1 Other Energy Efficiency and Energy Saving Measures

Despite the energy saving nature of this system, precautions are still taken to try to reduce energy consumption and losses overall in the hotel. One such example is the installation of transparent sheet filters on glass panes. Another example is that since guest rooms themselves are not controlled by the BMS, a system is going to be implemented, by the end of this year, wherein sensors detect the presence of people in a room and their activity, for example, whether they are sleeping. Depending on the information gathered by the sensors, the temperature, and therefore the air conditioning, is controlled. This is beneficial in avoiding having air conditioning in use when no one is present in the room.

In winter, when occupancy of the hotel is low, sections of the hotel can be switched off and isolated so that heating in certain areas is not done unnecessarily.

When the chiller is switched on, a large amount of power is consumed since the temperature difference would still be too large. It is preferable to leave the chiller running for longer time periods with smaller temperature differences than using the chiller for short periods of time to overcome a large temperature difference since the latter consumes larger amounts of power. Leaving the chiller running continuously is also advantageous since it prevents water in the chiller from warming up. This would require more power to get the water back to the lower temperature desired and is thus strenuous on the machinery. Due to these two reasons, the hotel management tries to avoid over-loading the chillers and does not have specific operating parameters for the chiller for specific months of the year, so the system is always controlled and managed according to the outdoor temperature. Also, during non-peak times, chillers are limited so that they do not ramp and peak and therefore power is saved. Some room for variations is however always allowed.

Where public areas are concerned, the aim of the hotel management is to provide comfort and not to make people feel cold indoors. Limiting the chillers means that the cooling effect is still provided; however, it is reduced – the limited value to which the chillers can provide cooling is still enough for indoor comfort. Excess cooling that would make people feel cold indoors is not needed and is considered a waste. As little as 0.5°C can make a large difference in costs.

Cooling in the guest rooms is left to the guests' discretion; however, more cooling is usually provided in guest rooms since one would already have passed from the public areas to get to the guest rooms and would thus have got used to the temperature indoors. If no conferences are being held at the hotel, the focus of cooling is more aimed towards the public areas and the guest rooms. In the evenings and early mornings more power is dedicated to cooling rooms since most people in the hotel are in the rooms.

Everything in the hotel is metered so that management can know where power is being consumed.

3.1.2 Costs

The hotel spends € 1.2 million per year on utilities, that is, electricity, water, fuel and gas. This equates to € 100,000 per month spent on utilities. On average, € 90,000 per month is spent on electricity but this depends on the outdoor climatic conditions. It is estimated that € 80 to € 100 per hour are spent on making the chillers work. This equates to approximately € 57,600 to € 72,000 per month spent on electricity to run the chillers. This serves to emphasize the great impact that air conditioning has on utility costs.

Maintenance depends on the location of a system and maintenance practices are partly developed through experience.

Maintenance on the boreholes is carried out twice a year – the borehole pumps are washed with freshwater. In winter, when the boreholes are not used, they are still sometimes switched on to ensure that they are still functioning. Additionally, since the InterContinental hotel is in a valley, when it rains, large amounts of debris are carried through the valley with the rain water runoff and can end up in the borehole. In order to avoid problems, the boreholes are left running at low speed during storms so that debris is continuously removed from the borehole and is not allowed to accumulate and clog the system. Chillers are cleaned once a year while heat exchangers are cleaned twice a year.

3.2 Hilton (Restall, 2013)

The main way in which seawater is used at the Hilton hotel is to cool the chillers; however, some heat recovery is also made.

3.2.1 The Original System

The original system, installed at the same time as the hotel was being built, consists of chillers on a closed freshwater circuit which exchange heat, by means of titanium heat exchangers, with seawater that goes through an open loop circuit.

The seawater is obtained from three 80 m deep boreholes (with respect to sea level). Its temperature does not vary much since the boreholes are very deep and since water filters into the boreholes. The difference in temperature between summer and winter is of 2°C at most which means that water is always between approximately 19°C and 21°C. In the original system, 120 kW submersible borehole pumps were used at a fixed speed to obtain seawater from a depth of around 20 m to 30 m. The boreholes are 80 m deep in order to satisfy the original flow rate requirement of 70 l/s. When the Water Services Corporation bores a borehole, boring is carried out until the specified flow rate is obtained. The fissures in the rock will supply a certain amount of water and one has to keep boring until the required flow rate is achieved. 80 m was the depth at which the 70 l/s flow rate requirement was met.

Seawater at the higher temperature is pumped back into the sea at a few metres depth.

The Hilton system is made up of three chillers – two of 2.3 MW and one of 450 kW. Separate boreholes supply each chiller individually. The 2.3 MW chillers cannot be in operation simultaneously; however, the 450 kW chiller can operate in conjunction with either of the 2.3 MW chillers or by itself. In winter only the smaller chiller is used. The 2.3 MW chiller is usually operated between 60% and 90% of the load depending on whether the conference centre is being used.

30 kW condenser pumps are used to circulate freshwater in the closed loop that includes the heat exchanger and the condenser of the 2.3 MW chillers. The 450 kW chiller has a 4 kW condenser pump. The circulating pumps responsible for delivering cooled water to the zones that require cooling consume a total of 101 kW.

When the hotel was designed around 15 years ago, the latest most energy-efficient system commercially available at the time, that is, the centrifugal seawater cooling chiller was implemented because theoretically it could reach on its own COPs of 5 to 6. The chief engineer believes that the system before the improvements could actually reach these COPs.

3.2.2 The Modified System

About two years ago, two major changes were carried out on the original system. It was thought that the continuous operation of the submersible borehole pump was wasteful and that something needed to be done in order to avoid this waste.

Additionally, the chillers were rejecting approximately 1MWh of low-grade heat into the seawater. It was therefore decided that an attempt at recovering this heat should be made.

3.2.2.1 From Borehole Pump to Surface Pump and Inverter

The system was changed from having a borehole pump to having a surface pump. A borehole pump consumes 120 kW and has yearly maintenance costs of € 2,000 to € 3,000. The surface pump consumes 33.6 kW and will cost € 2,000 for maintenance over its lifetime. The initial cost for a borehole pump is approximately € 16,000 while the initial cost for a surface pump is € 3,000 to € 4,000.

In the original system, seawater was always being pumped at the same rate even if a lower rate would have sufficed to meet the cooling demand. Whatever the load on the chiller, whatever heat exchange was needed to cool it down, the borehole pump was always continuously pumping a fixed volume of water (70 l/s) whether the chiller needed it or not.

In order to avoid the continuous operation of the pump, the ideal water temperature entering the chiller was calculated (approximately 23°C) and an inverter was installed such that seawater can be supplied to the chiller at the necessary flow rate to obtain the ideal temperature. The seawater exchange is therefore only used when needed. The return temperature into the chiller is always kept at the ideal temperature by means of the varying flow rate which is achieved thanks to the inverter.

With the current configuration, the borehole can still supply up to a maximum of 70 l/s but in reality this high flow rate is never used unless the chillers are operating at full load. At the moment the surface pump is extracting water from a depth of 7.5 m.

The system now consumes 18% of the energy that the original system was consuming.

3.2.2.2 Heat Recovery

The circuit used for circulating seawater was changed. After the seawater cools the condenser of the chiller through the heat exchanger, it passes from a 671 kW heat pump. The heat pump heats fresh water which is then used for the indoor (heated) pool, the calorifiers (which are part of the boiler system) and space heating. This has led to the boilers not being used at all in the last year which has saved the hotel € 140,000 worth of gas. Typically, in the previous configuration, the temperature of the seawater rejected back to sea used to be around 38°C. With the new configuration, which includes heat recovery due to the heat pump installed, the temperature of the rejected seawater is around 26°C.

The COP of the heat pump is about 8.5 even though the heat pump is operated outside of its operational efficiency range. This is quite remarkable since the heat pump was not designed to operate with water which is already at a higher temperature but was designed to operate with cold water and raise its temperature. The heat pump is now receiving hot water and raising its temperature by fewer degrees Celsius than before.

Additionally, in winter when the chiller is not used, heat is being recovered from the seawater through the heat pump; this was an unplanned, but welcome, consequence. Typically the freshwater going into the boiler would be at a temperature of around 18°C. The seawater never goes down to such temperatures and always remains at around 20°C (even in winter). These 2°C temperature difference mean that heat is being recovered in an inverse way from the sea. Even in winter, when the chiller is not used as much, the heat pump also operates very efficiently because heat is being recovered from the sea.

3.2.2.3 Materials

The specifications for the original system were of having titanium heat exchangers and stainless steel 316 borehole pumps. However, the borehole pumps only lasted around six months due to corrosion. They were then changed to pumps made of stainless steel 904. These still suffer from corrosion but corrode much more slowly.

When the borehole pump was being used, the piping had to be made out of stainless steel to support the weight of the borehole pump. Now, with the surface pump, plastic is being used.

3.2.2.4 Operating Costs

The electricity costs of the assembly including the heat pump vary between € 3,000 and € 4,500 per month but around € 10,000 to € 15,000 are being saved on gas every month. Yearly € 143,000 are being saved (net savings) from the avoided use of gas and electricity.

3.2.2.5 Maintenance

The system is expected to last the lifetime of the hotel.

The heat exchangers are opened every 6 months for cleaning and after 13 years of operation they are not showing any signs of corrosion. The pumps are serviced once a year. The freshwater in the closed loop circuit is analysed and treated to eliminate corrosion in the chiller itself.

3.2.3 The Drive for Implementing the System

The drive for upgrading the system was dual – the hotel wanted to achieve energy saving for environmental purposes and to get the voluntary EU Ecolabel.

The EU Ecolabel was obtained by Hilton Malta in 2006 for a number of environmental initiatives. Hilton Malta is the only hotel in Malta to have obtained the EU Ecolabel.

Additionally in 2005, Hilton itself, as an international brand, instituted an award system for environmental friendliness and Hilton Malta won this award. Hilton aims to be the leading hotel in energy saving.

3.3 Westin (Cassar, 2013)

Seawater is used to provide cooling of the chiller although it never comes into direct contact with it. Seawater is obtained from a borehole and passed through a plate heat exchanger so as to exchange heat with the freshwater that circulates in a closed loop and cools the condenser. The seawater is then rejected back to sea thus making the seawater circuit an open one.

The system that is currently in use at the Westin was installed in April 2013. The previous system also made use of seawater to cool the condensers.

The boreholes used to extract seawater are approximately 45 m deep. The temperature of the water obtained from the boreholes goes up to a maximum of 25°C in summer and a minimum of 18°C in winter. The previous system used submersible borehole pumps which are very maintenance-intensive. The new pumps are surface pumps and consume 65 kW.

A pit, dug into the rock, is used to discharge the seawater back to the sea. The warm seawater is placed into this pit and is then allowed to filter back into the sea through fissures in the rock.

Piping that comes into contact with seawater is made out of plastic while the heat exchanger is made out of titanium. Pumps have a cast iron bronze impeller.

The system consists of two new 1.2 MW chillers and a 0.97 MW chiller which was retained from the previous system as backup. The smaller chiller will be decommissioned by 2015. Since the chillers were installed in April 2013, not much historical data has been gathered yet and thus the actual power consumption profile is not yet known. However, in the summer months, the demand on the chillers usually doubles. In May and June of 2013, the hotel management noticed a 9% energy saving compared to May and June of 2012. However, this might also be due to the fact that the outdoor temperatures in May and June of 2013 were not as hot as those in May and June of 2012. Nevertheless, the chief engineer at Westin is hopeful that as the summer progresses and outdoor temperatures and consequently demand increase, more savings will be obtained.

The hotel can generally operate with just one chiller; however, in peak summer, the hotel management prefers to use two chillers running at lower loads than leaving one chiller running at full load. The chillers are rarely used in winter, except when there are large functions going on in the hotel. When the demand is such that one chiller can satisfy it, the two 1.2 MW chillers operate alternately for a week each so as not to cause excessive wear.

The circulation pumps used to deliver cold freshwater to the areas requiring cooling consume a maximum of 22 kW but they operate according to the need by means of an inverter.

The new system also includes a control panel which incorporates a BMS.

Since the system is new, maintenance procedures and frequency have not yet been completely established since these are also dependent on experience. However, from experience with the previous system, sacrificial anodes are checked twice a year, the heat exchanger is cleaned every six months and strainers are cleaned once a month. Frequent cleaning of the strainers is especially required in the beginning of operation of a new system since the system would be getting rid of any debris left over from installation work carried out.

3.3.1 The Need for a New System

The new system cost around € 300,000 to the hotel. This figure includes the cost of the chillers, the new pumps, heat exchangers, boring, etc. The system was mainly changed since it made use of R22 as its refrigerant. R22 will become illegal to use in 2015 and is no longer being produced. The only R22 available for purchase is recycled R22 and it is very expensive.

Since a new system was being installed, new, bigger boreholes were bored closer to the hotel to reduce the friction losses which in the previous system accounted for 400 l/min.

Moreover, the heat exchangers of the previous system were thought to be undersized and it is the chief engineer's belief that they were limiting the efficiency of the chiller. The original system was designed such that the seawater would cool the condenser directly without the use of a heat exchanger but the condenser corroded after a short time and a heat exchanger was needed urgently. This led to the hasty purchase of a heat exchanger which was undersized.

When the new system was installed, outlets from the heat exchanger were created but left unused. The scope of these outlets is to implement a system of heat recovery in the future. The heat recovered from the seawater could be used to raise the temperature of the water going into the gas boilers by a few degrees so that savings can be made from the decreased consumption of gas. If the water going into the boilers is even a few degrees warmer, significant savings can be achieved.

3.4 Bay Street (Cortis, 2013)

At the Bay Street Hotel seawater is used to cool the condensers of the chillers.

One borehole with a 5 kW submersible pump is used to supply seawater to a makeup tank. Biocide, to prevent fouling, is added in the makeup tank. The borehole is 27 m deep and the temperature of the seawater is around 19°C throughout the year but can fluctuate from 18°C to 22°C. The borehole pump is coupled to an inverter and is used to maintain an adequate level of seawater in the makeup tank. Therefore the seawater pump does not operate continuously but switches off automatically when there is enough water in the makeup tank. Water flows from the makeup tank to a secondary tank which feeds the chillers.

The temperature at which seawater is returned to sea is of 47°C and seawater is filtered through a mesh to prevent any debris from going to sea.

Three chillers are installed; each chiller has two circuits and each circuit can provide 65 kW of cooling. Therefore the total cooling capacity of the Bay Street system is 390 kW. Three titanium, plate heat exchangers are used; one each for every chiller.

18 kW circulation pumps are then responsible to circulate the fresh, chilled water around the hotel. The freshwater that circulates around the hotel is treated and tested regularly for legionella.

Depending on the cooling demand, the BMS decides which chillers to use and how much each chiller should be loaded. One chiller would be the ‘master’ and then the BMS decides which other chiller to bring online if needed.

The system was incorporated in the planning phase of the hotel and it supplies cooling to the entire complex, that is, the hotel and the shops.

An overhaul of the system is carried out once a year. Every six months, approximately before summer starts and after summer ends, the system is shut down and checked. Every quarter, general maintenance, such as tightening of bolts and cleaning, is carried out. The heat exchangers are washed with water every quarter but during the overhaul, gaskets are changed, scale is removed and a more thorough cleaning in general is carried out.

The chief engineer at Bay Street believes that the drive for installing the system was to save space. The space dedicated to the chillers is rather restricted.

R22 used to be the refrigerant present in the chillers; however, since this is being phased out, some changes were made to the chillers so that they are now operating with 477D refrigerant.

No more details were given about the Bay Street Hotel’s chiller system.

3.5 Summary and Remarks

All of the hotels visited make use of saline groundwater for cooling of chillers in their facilities. The general operating principle is always the same – fresh water is circulated by means of a pump in a closed loop which includes the condenser and the plate heat exchanger. Heat from the condenser is rejected to the freshwater and in turn transferred to the saline groundwater through the heat exchanger. Cold saline groundwater is pumped to the heat exchanger and then discharged at a higher temperature at sea.

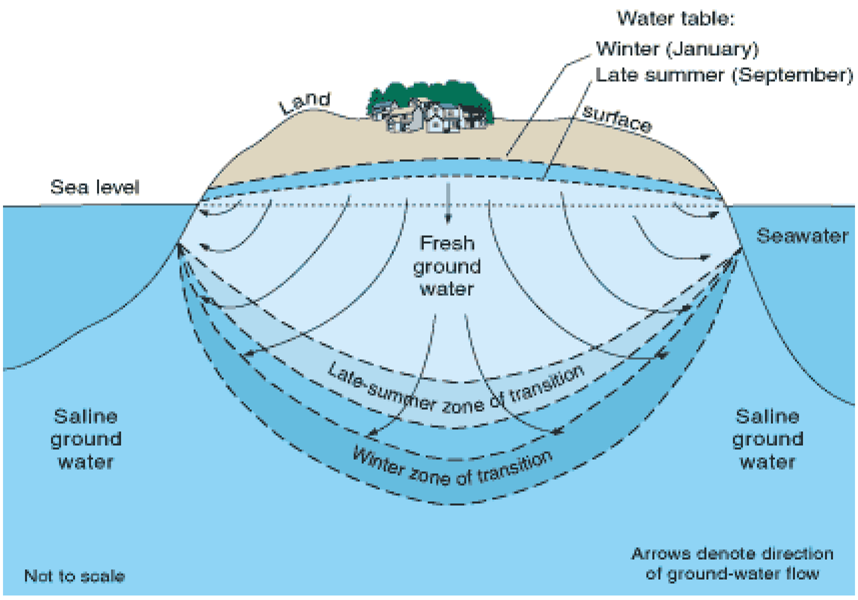

Figure 3-2 Typical island aquifer (Orr, 2013).

Seawater is obtained through boreholes as this provides advantages over seawater obtained directly from the sea through piping. Since the hotels are located along the coastline, and the Ghyben-Herzberg fresh groundwater lens is thinnest at the coast (as shown in Figure 3-2), seawater can be obtained through rocks at shallow depths by means of a borehole. The costs and difficulties associated with obtaining seawater through a borehole are lower when compared to the costs and difficulties associated with obtaining seawater from the sea. The latter would entail creating a pipe network from the sea, through roads, to the hotels. Moreover, the natural filtration provided by rock strata causes saline groundwater to have a lower silt density index (SDI[2]) and organic content (Rodríguez-Estrella & Pulido-Bosch, 2009) and to be more stable in terms of chemical quality characteristics, turbidity and suspended matter (Miller D. W., 1956) (Rodríguez-Estrella & Pulido-Bosch, 2009) (Maliva & Missimer, 2011). Therefore the use of saline groundwater causes less fouling and requires less filtering than if surface water was used. Additionally, the temperature of groundwater does not experience the wide range of variation experienced by surface water (Miller D. W., 1956) (Yang, Cui, & Fang, 2010) which is beneficial for engineering applications.

Borehole depths vary considerably depending on the flow rate needed for operating the chillers efficiently. The borehole depths reported vary from 21 m to 80 m and the cooling capacities vary from 390 kW to 5.05 MW.

Seawater temperature observations fluctuate from a low of 16°C in winter to a high of 25°C in summer. The saline groundwater, after having passed through the heat exchanger, is rejected back to the sea at higher temperatures which vary from 32°C to 47°C for systems which do not make use of heat recovery.

The seawater temperatures at inlet show considerable variation due to the differences in borehole depths. Another factor which might be affecting the inlet seawater temperatures is the discharge of warm seawater to the sea with only minimal precautions taken. Such precautions include discharging seawater at a few metres depth, discharging seawater by pumping it into a pit so that it can filter back through

[2] SDI is a measure of the fouling capacity of water (Rodríguez-Estrella & Pulido-Bosch, 2009).

the rocks to the sea and making use of a mesh to prevent discharging debris along with the seawater.

In order to obtain the best possible COP from the chillers it is desirable to have the coolest possible seawater. The COP will suffer if the seawater abstracted is of higher temperature because it is being influenced by the outflow of warm seawater. Thermal feedback can reduce the lifetime of a GSHP system by rendering it inefficient (Freedman, Waichler, Mackley, & Horner, 2012). In general, when speaking to hotel management personnel from different hotels it was noticed that there seems to be little concern over thermal pollution and it is in fact not recognised as a form of pollution.

The use of inverters and speed controllers to reduce the power consumption of pumps is becoming more common practice and the mention of such components was made in almost all cases. Also, there seems to be a transition from submersible to surface pumps going on. Submersible pumps are more expensive, more maintenance intensive and consume more electricity than surface pumps.

When asked about performance parameters, most hotel managers seemed to be unaware of actual figures (except for Hilton) and this could indicate a lack of knowledge or monitoring of performance parameters such as the coefficient of performance. Monitoring of performance parameters is of utmost importance since it might reveal potential for improvements and cost savings in the systems.

Awareness of operating costs was higher since two out of the four hotels visited were able to provide actual figures. The cost of providing cooling at Hilton is estimated to be between € 3,000 and € 4,500 per month while the cost of providing cooling at the InterContinental is estimated to be between € 57,600 and € 72,000 per month. This shows a large discrepancy between the two especially when considering the cooling capacities of the two hotels – 5.05 MW for Hilton and 852 kW for InterContinental. The discrepancy may be attributed to the heat recovery that is carried out at Hilton. The Hilton hotel is the only one out of the four hotels visited which is recovering heat from the condensers and making use of seawater for heating by means of a heat pump. Again, the encounters with management personnel from different hotels indicate that there seems to be a lack of knowledge about the potential for the use of seawater for space heating or the heat recovery from seawater after it has passed through the heat exchanger to provide domestic hot water and space heating.

The materials used in the cooling systems are mainly stainless steel and plastic. Titanium is generally used for the heat exchangers. The initial and maintenance costs of the system could theoretically be reduced if fresh groundwater was used, since saline groundwater provides very harsh conditions and thus requires: the use of corrosion resistant materials which are sometimes expensive; extensive maintenance; replacement of parts and cleaning; and isolation from main components (hence the use of the heat exchangers). However, given Malta's freshwater resource situation, it is not wise to place more pressure on what little freshwater resources are available especially when saline groundwater is so abundant and serves the purpose of cooling just as well.

The use of saline groundwater could actually be a solution to saltwater intrusion (Kacimov, Sherif, Perret, & Al-Mushikhi, 2009) (Sherif & Hamza, 2001) (Sherif & Kacimov, 2008) (Jorreto, Pulido-Bosch, Gisbert, Sánchez-Martos, & Francés, 2009) (Rodríguez-Estrella & Pulido-Bosch, 2009). When extracting saltwater from beneath the freshwater lens, the freshwater-saltwater interface shifts downwards and the mixing zone becomes thinner (Jorreto, Pulido-Bosch, Gisbert, Sánchez-Martos, & Francés, 2009) thus delaying or stopping saltwater intrusion. This occurs because the pressure difference between the freshwater and the saltwater is equalised (Miller D. W., 1956). This concept is illustrated in Figure 3-3.

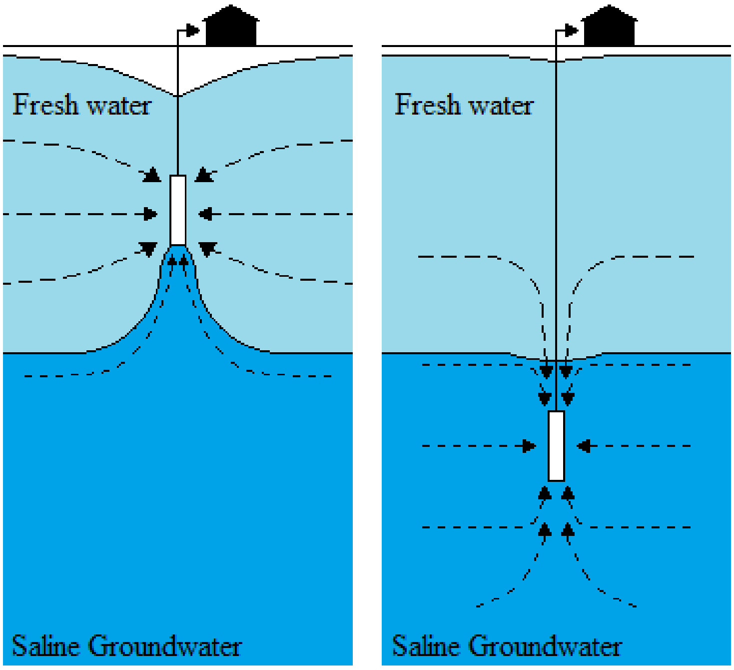

Figure 3-3 Left: upconing of saltwater when excessive amounts of freshwater are extracted from the aquifer (saltwater intrusion); Right: saline groundwater abstraction leading to downconing of freshwater due to equalisation of pressure (adapted from: Kooiman, Stuyfzand, Maas, & Kappelhof, 2004).

The use of Building Management Systems (BMS) has become very widespread although there seems to be some mistrust of such systems and system managers tend to prefer overriding the system in certain situations in which they deem their judgement superior to that of the BMS.

In most cases, cooling systems making use of seawater for cooling were implemented at the planning stage of the hotels. However, major alterations or renovations to the systems were made in the case of Hilton, Westin and Bay Street. In the latter two cases, the need for renovation was driven by the phasing out of the refrigerant R22.

Table 3-1 Summary of existing systems in Malta[3].

	Inter Continental	Hilton	Westin	Bay Street
Purpose	Cooling	Heating and cooling	Cooling	Cooling
Borehole depth with respect to sea level (m)	21	80	45	27
Number of boreholes	3	3	3	1
Seawater temperatures (°C)	16-22	19-21	18-25	18-22
Seawater pump type	Submersible pump with inverter	Surface pump with inverter	Surface pump	Submersible pump with inverter
Seawater pump size (kW)	30	33.6	65	5
Temperature of seawater rejected back to sea (°C)	32	26	-	47
Number of chillers	3	3	3	3
Chiller size (kW)	284, 284, 284	2300, 2300, 450	1200, 1200, 970	130, 130, 130
Total cooling capacity (kW)	852	5050	3370	390
Total heating capacity (kW)	N/A	671	N/A	N/A
Condenser pump size (kW)	13-15	30 (on large chillers) 4 (on small chiller)	-	-
Circulating pump size (kW)	7.5	101 (total)	22	18
COP cooling	-	5-6	-	-
COP heating	N/A	8.5	N/A	N/A
Cost of providing cooling (€/month)	57,600-72,000	3,000-4,500	-	-

[3] Fields marked with 'N/A' imply that the parameter is not applicable to the system.
Fields marked with '-' imply that no information was available regarding that parameter.

Chapter 4

Review of Relevant Laws

4 Review of Relevant Laws

Two acts from the Laws of Malta are of importance to this area of study; the Malta Resources Authority Act (Chapter 423) and the Building Regulation Act (Chapter 513). Specific subsidiary legislations are particularly relevant and will be discussed in the following sections.

4.1 Borehole Drilling and Excavation Works within the Saturated Zone Regulations (Subsidiary Legislation 423.32, 2008)

The Borehole Drilling and Excavation Works within the Saturated Zone Regulations aim to create a regulatory framework for such activities (Subsidiary Legislation 423.32, 2008). The concerned authority is the Malta Resources Authority.

For the purposes of the regulations, a borehole is “any shaft, adit, opening, hole, spring outlet or well artificially dug in the ground or artificially enlarged, which has access to groundwater or from which groundwater is extracted or can be extracted or which is vertically deeper than half the distance between the well head and the nearest water table at that point”. The regulations go on to establish that a seawell is a borehole “tapping the saline groundwater body lying underneath and in direct physical contact with sea level freshwater groundwater body and drilled at a distance of not more than 150 m from the nearest point along the coastline, and which is cased” or a borehole “tapping a confined saline groundwater body which is not in direct physical contact with any freshwater groundwater body”. The regulations specify that a confined saline groundwater body is one on top of which lies a groundwater body with notably lower conductivity and therefore lower salinity. In the context of these regulations, casing refers to any “conduit, pipe, fixture or constructional element which is inserted and attached to the circumference of the borehole to prevent any transfer of fluid between the borehole and its surrounding geological formation”.

Therefore the systems currently used and previously discussed are subject to the provisions of these regulations according to the definitions given.

Drilling in the saturated zone is forbidden unless a permit has been issued by the Malta Resources Authority. The regulations establish an application process for obtaining a permit from the Malta Resources Authority prior to the commencement of borehole drilling or excavation works carried out fully or partly in the saturated zone. A permit is granted under the criteria that any impacts on the water resource are insignificant, water is used in an efficient manner, the water environment is protected, discharges are minimal, and the public interest is protected. Additionally, the Authority specifies conditions, such as the duration of validity of the permit, in the permit. A permit may be suspended, revoked or cancelled when impacts on the groundwater body become substantial or when there is foul play by the permit holder. When a permit is revoked or cancelled, the Authority has the power of enforcement to order drilling to stop and the borehole to be backfilled and sealed. Provisions for drilling operations without a permit are made in the regulations.

According to the regulations, the Malta Resources Authority reserves the right to refuse any applications for the drilling of boreholes or excavations in the saturated zone for a time period established by the Authority itself. However, this does not apply to seawells.

Requirements are set out for casing of seawells in regulation three. Seawells found at different distances from the coastline and in contact with different bodies must be cased up to different depths:

- For seawells tapping confined saline groundwater bodies:

Table 4-1 Distance from coastline and corresponding depth of casing.

Distance from nearest point along the coastline (m):	Least depth of casing below sea level (m):
0-50	10
50-100	25
100-150	50

- For seawells tapping a saline groundwater body which is not in contact with a freshwater groundwater body, the casing must be such that it encases the seawell from the surface to at least 6 m below the blue clay-globigerina limestone interface.

These casing requirements are particularly relevant and important since they serve to prevent extraction and contamination of fresh groundwater. Fresh groundwater is an over-exploited and scarce resource in Malta.

Water testing by accredited laboratories must be carried out regularly on the water obtained from seawells and submitted to the Malta Resources Authority. This is the responsibility of the person who applied for the permit. The parameter of interest is the electrical conductivity of the water. Its value should not be lower than 50,000 µS/cm. In cases where the conductivity results lower than this stipulated value, the user must stop the extraction of water for a period established by the Authority.

Low conductivity in the water obtained from seawells could indicate that the fresh groundwater has moved downwards until, or even beyond, the borehole depth. This is disadvantageous since it would mean that fresh groundwater is being used for cooling – a purpose which saline groundwater can fulfil very easily.

Another situation where the Authority may order one to stop or reduce water extraction is when extraction can have a "negative impact on the quantitative and, or the qualitative status of a freshwater groundwater body". Therefore if the groundwater body is being depleted or contaminated and polluted in any way, the Authority may stop the extraction or works being carried out. The Authority may also terminate any borehole activities or excavation works and order the closure of a borehole.

4.2 Energy End-Use Efficiency and Energy Services Regulations (Subsidiary Legislation 423.34, 2008)

These regulations transpose into Maltese law Directive 32/2006/EC of the European Parliament and of the Council of 5 April 2006 on energy end-use efficiency and energy services and repealing Council Directive 93/76/EEC (Subsidiary Legislation 423.34, 2008).

The aim of these regulations is to "enhance the cost-effective improvement of energy end-use efficiency in Malta". This should be achieved through the removal of market barriers and imperfections and through the creation of "conditions for the development and promotion of a market for energy services and for the delivery of other energy efficiency improvement measures to final consumers". The responsible authority is the Malta Resources Authority.

The Energy End-Use Efficiency and Energy Services Regulations state that by 2017, 9% energy savings must be attained through "energy services and other energy efficiency improvement measures" as long as the measures implemented are cost-effective, practicable and reasonable. The regulations set the requirement of an Energy Efficiency Action Plan in which a strategy for reaching the 9% target is devised. The Authority is responsible for implementing the strategy for obtaining the target, monitoring and verifying energy savings, and reporting the results.

The third schedule is a non-exhaustive list of what the eligible energy efficiency improvement measures are under these regulations. Among the items on the list is heating and cooling by heat pumps and the installation or efficient upgrade of district heating or cooling systems. Also on the list are variable speed drives which are used in conjunction with the pumps used in heating and cooling systems and DHC systems.

In order for energy efficiency improvement measures to be eligible under these regulations, they must contribute towards energy savings that can be measured and substantiated or estimated. It is also important that such energy savings are not already being accounted for under some other regulation.

The fourth schedule provides a general framework for measurement and verification of energy savings. Energy savings are given in kilowatt hours (kWh), Joules (J) or kilograms of oil equivalent (kgoe). The regulations also make a distinction between measurements and estimates and establish methods which can be used for obtaining both.

Provisions for adjustment and normalisation of results due to external conditions which affect energy use are made in the fourth schedule.

4.3 Protection of Groundwater against Pollution and Deterioration Regulations (Subsidiary Legislation 423.36, 2009)

These regulations transpose into Maltese law Directive 2006/118/EC of the European Parliament and of the Council of 12 December 2006 on the protection of groundwater against pollution and deterioration (Subsidiary Legislation 423.36, 2009). The competent authority is the Malta Resources Authority.

Measures for the protection of groundwater include setting of criteria that define good groundwater chemical status, criteria which help to prove continued upward trends in pollution, and criteria for when and how to reverse upward trends.

In the context of these regulations, groundwater is any water "below the surface of the ground in the saturation zone and in direct contact with the ground or subsoil". A significant and sustained upward trend is an increase of statistical and environmental significance in the "concentration of a pollutant, group of pollutants, or indicator of pollution in groundwater for which trend reversal is identified as being necessary".

Biocidal products are defined as "active substances and preparations containing one or more active substances… intended to destroy, deter, render harmless, prevent the action of, or otherwise exert a controlling effect on any harmful organism by chemical or biological means" (Directive 98/8/EC of the European Parliament and of the Council, 1998). Such products are deemed to be pollutants in the context of the Protection of Groundwater against Pollution and Deterioration Regulations and the limit on the total sum of all the different biocides "detected and quantified in the monitoring procedure, including their relevant metabolites, degradation and reaction products" is 0.5 μg/l.

In view of these definitions, it is apparent that the use of saline groundwater for heating and cooling is subject to the provisions made in these regulations since biocidal products are sometimes added to saline groundwater to reduce fouling. While some systems provide for the addition of biocide when the seawater is still in the borehole, thus potentially contaminating groundwater by means of infiltration, other systems carry out the addition of biocidal products while the seawater is found in storage tanks prior to use. The latter practice is more favourable than the former since it provides for containment of biocides and prevents contamination of groundwater. However, when seawater is discharged back to the sea, the biocidal products used contaminate the marine environment and can potentially lead to the loss of marine life.

In order to prevent pollution and deterioration of groundwater bodies, the regulations stipulate that threshold values corresponding to good groundwater chemical status are to be established with due consideration for:

- The effect and inter-relationship of the groundwater body with any surface water body, terrestrial ecosystems and wetlands.
- Human toxicology and ecotoxicology.
- "Interference with actual or potential legitimate uses or functions of groundwater".

The established threshold values need to be published and updated when new information becomes available.

Measures implemented by the Authority should ensure the prevention of input of hazardous pollutants and the limitation on the input of non-hazardous pollutants.

The Authority is responsible for establishing and assessing the chemical status of groundwater bodies by means of properly situated monitoring sites the choice of which is made according to the Water Policy Framework Regulations.

For any groundwater bodies for which the Authority determines that there is a significant and sustained upward trend in pollution which may affect aquatic or terrestrial ecosystems, human health or the use of water resources, a starting point for the reversal of the trend must be defined by the Authority. In the case of plumes of pollution originating from point sources, the Authority must assess the propagation trends to ascertain that the plume does not expand. Diffuse sources of pollution should also be taken into account when possible.

The regulations place the responsibility of reversing significant and sustained upward trends on the Malta Resources Authority, however, there are no provisions made for situations in which point sources of pollution can be identified and traced back to responsible parties. This creates a situation in which enforcement of the regulations is weak and true culprits are not brought forward to justice hence there is no deterrent to prevent incidents from occurring.

4.4 Promotion of Energy from Renewable Sources Regulations (Subsidiary Legislation 423.19, 2010)

These regulations transpose into Maltese law Directive 2009/28/EC of the European Parliament and of the Council of 23 April 2009 on the promotion of the use of energy from renewable sources (Subsidiary Legislation 423.19, 2010). The competent authority is the Malta Resources Authority.

In the context of these regulations, renewable sources of energy are non-fossil sources such as wind, solar, aerothermal, geothermal, hydrothermal and ocean energy, hydropower, biomass, landfill gas, sewage treatment plant gas and biogases. More specifically, geothermal energy is heat stored under the earth's surface and hydrothermal energy is heat stored in surface water. Therefore the use of saline groundwater for heating and cooling is classified under geothermal energy.

The target for the share of energy from renewable sources from the gross final consumption of energy for Malta is at least 10% by 2020. The same target applies for the share of energy from renewable sources for transport.

The gross final consumption of energy from renewable sources is the sum of the energy coming from renewable sources consumed as electricity, consumed for heating and cooling and consumed in transport.

Under these conditions, geothermal and hydrothermal energy captured by heat pumps contributes towards the renewable energy target as long as the resulting effect in terms of energy is greater than the primary energy necessary to operate the heat pump.

The amount of energy from renewable sources captured by heat pumps (E_{RES}) is calculated as:

$$E_{RES} = Q_{usable} \times \left(1 - \frac{1}{SPF}\right)$$

4-1

Where Q_{usable} is the estimated total usable heat delivered by heat pumps and *SPF* is the estimated average seasonal performance factor.

Only energy from heat pumps whose *SPF* is greater than $1.15/\eta$ are to be considered, where η is the ratio between total gross production of electricity and the primary energy consumption for electricity production and shall be calculated as an EU average based on Eurostat data. This requirement ensures that the useful energy obtained from heat pumps is greater than the primary energy consumed for producing the electricity required for the heat pump to operate (Pavković, Delač, & Mrakovčić, 2012).

Under the Promotion of Energy from Renewable Sources Regulations, the Authority is to recommend the installation of electricity equipment that makes use of renewable energy sources and heating and cooling systems that make use of renewable energy sources and DHC. This should be done in all phases of construction be it planning, designing, building or renovating and should be done in both industrial and residential areas and in both local and regional administrative infrastructure as well as in the private sector. Additionally, the building regulations and codes should be updated to establish a minimum level of energy from renewable sources in the building sector and make it possible for these targets to be achieved "through district heating and cooling produced using a significant proportion of renewable energy sources".

Recommendations made by the Authority regarding heat pumps should promote heat pumps which fulfil the eco-labelling requirements.

The Authority is also responsible for enabling certification or qualification schemes for installers of small-scale biomass boilers and stoves, solar photovoltaic and solar thermal systems, shallow geothermal systems and heat pumps.

The Authority should provide guidance to planners, *periti* and engineers about the ideal balance between renewable energy sources, high efficiency technologies and district heating and cooling when planning, designing, building and renovating industrial or residential areas.

4.5 Energy Performance of Buildings Regulations (Subsidiary Legislation 513.01, 2012)

These regulations transpose into Maltese law Directive 2010/31/EU of the European Parliament and of the Council of 19 May 2010 on the energy performance of buildings (Subsidiary Legislation 513.01, 2012). The competent authority is the Malta Resources Authority.

The aim of these regulations is to encourage the enhancement of the energy performance of buildings with consideration for climatic and local conditions, indoor climate requirements and cost-effectiveness.

The regulations specify various definitions related to buildings. In its simplest form, a building is a "roofed construction having walls, for which energy is used to condition the indoor climate". The term "building" may refer to the entire construction or part of a construction which has been designed or modified to be used separately.

The energy performance of a building (EPB) is the "calculated or measured amount of energy, needed to meet the energy demand associated with a typical use of the building, which includes, … energy used for heating, cooling, ventilation, hot water and lighting".

In calculating the energy performance of a building one needs to account for several factors. Relevant factors include:

- The thermal characteristics of the building and its internal partitions (thermal capacity, insulation, passive heating, cooling elements and thermal bridges).
- Heating and hot water equipment and their insulation.
- Air conditioning equipment.

It is also necessary to classify buildings when establishing their energy performance characteristics. The identified building categories are; single-family houses of different types, apartment blocks, offices, educational buildings, hospitals, hotels and restaurants, sports facilities, wholesale and retail trade services buildings, and other types of energy-consuming buildings.

The regulations make provisions for establishing minimum energy performance requirements for buildings. In the case of new buildings, high-efficiency alternative systems need to be considered at the design stage. These high-efficiency alternative systems include the use of renewable sources of energy to supply energy in a decentralized manner, cogeneration, heat pumps, and district or block heating or cooling ideally powered by energy from renewable sources.

As of 2019, all new buildings must be nearly zero-energy buildings and targets need to be set for renovated buildings in order for them to strive for a nearly zero-energy status. This necessitates a National Energy Efficiency Action Plan.

A nearly-zero energy building is one whose energy performance is very high and whose energy needs are nearly entirely met by energy from renewable sources. Energy from renewable sources includes non-fossil fuel sources such as wind, solar, aerothermal, geothermal, hydrothermal and ocean energy, hydropower, biomass, landfill gas, sewage treatment plant gas and biogases. Thus the use of saline groundwater for heating and cooling could be an adequate tool for buildings to achieve a nearly zero-energy status.

Inspection of the accessible parts of building heating systems providing more than 20 kW of heating such as the heat generator, control system and circulation pump(s) with boilers should be carried out periodically.

The inspection should evaluate the boiler efficiency and boiler sizing. Different inspection frequencies should be established depending on the type and output of the heating system. For boilers greater than 100 kW, inspection should be carried out at least every two years. In the case of gas boilers, inspection may be carried out every four years.

Air conditioning systems providing more than 12 kW of cooling also need to be inspected and their efficiency and sizing needs to be evaluated. Different inspection frequencies should be established depending on the type and output of the air conditioning system.

The regulations establish a framework for the energy certification of buildings or building units which includes the issuing of energy performance certificates (EPCs).

Independent control systems for EPCs and heating/air conditioning system inspection reports are required.

4.6 Summary and Remarks

The extraction of saline groundwater is subject to the Borehole Drilling and Excavation Works within the Saturated Zone Regulations and to the Protection of Groundwater against Pollution and Deterioration Regulations.

The Borehole Drilling and Excavation Works within the Saturated Zone Regulations focus more on preventing the depletion of groundwater bodies and establish a permitting system in order to attempt to control the extent to which extraction from the aquifers occurs. Provisions are set for seawells and include the requirements for casing and testing to ensure that saline groundwater is being extracted.

The qualitative aspect of groundwater is given more importance in the Protection of Groundwater against Pollution and Deterioration Regulations. Provisions for establishing the limits on pollutants in order to sustain a good groundwater chemical status are made in the regulations. The Authority is in charge of identifying and reversing any continued upward trend in the concentration of pollutants in groundwater bodies. Additionally, the regulations place on the Authority, the responsibility of preventing the input of pollutants. This regulation is of importance due to the use of biocides in the operation of heat pumps cooled by saline groundwater.

Both the Energy End-Use Efficiency and Energy Services Regulations and the Promotion of Energy from Renewable Sources Regulations set percentage targets which must be reached by a specified year. In case of the Energy End-Use Efficiency and Energy Services Regulations the target is set at 9% energy savings by the year 2017. On the other hand, for the Promotion of Energy from Renewable Sources Regulations, the target is to obtain at least 10% of the gross final consumption of energy for Malta from renewable sources by 2020.

The Energy End-Use Efficiency and Energy Services Regulations and the Promotion of Energy from Renewable Sources Regulations both mention heat pumps and district heating and cooling as a means to achieve the said targets. However, in the former they are considered as energy-efficient systems while in the latter they are relevant only if the heat source or sink is a renewable source of energy. The Energy End-Use Efficiency and Energy Services Regulations emphasize the importance of avoiding double-counting. Hence if one system is already being accounted for under one regulation it may not be contributing to another target set by another regulation.

The energy performance of a building including any energy used for heating and cooling must conform to minimum requirements established in the Energy Performance of Buildings Regulations. The Regulations also make provisions for new buildings by stating that high-efficiency systems need to be considered for them. Heat pumps and DHC are both regarded as adequate means for improving the energy performance of buildings. Additionally, the regulations require that all new buildings achieve a nearly zero-energy status by 2019. Part of the regulations is exclusively dedicated to inspection of building heating systems and air conditioning systems thus showing the important role that such systems play in the energy performance of a building.

Chapter 5

Calculations and Results

5 Calculations and Results

5.1 Coefficient of Performance of a Water Cooled System

From data obtained from the Hilton, the evaporator temperature is equal to approximately 6°C and the condenser temperature is equal to approximately 30°C in August. The refrigerant used in the Hilton system is R134a and therefore the properties of this refrigerant shall be used for the purpose of these calculations. The properties of R134a can be found in Table 9-1 and Table 9-2. All subscripts used in the calculations refer to the state of the refrigerant.

Assuming a standard vapour-compression cycle (refer to Figure 2-5):

- The evaporator temperature corresponds to the saturated vapour temperature at state *1* and shall therefore be referred to as T_1.
  - By linear interpolation from Table 9-1, the saturation pressure at a temperature of 6°C is equal to 362.605 kPa. This pressure will be referred to as P_1.
  - By linear interpolation from Table 9-1, the enthalpy of saturated vapour at a saturation pressure of 362.605 kPa is equal to 250.665 kJ/kg. This enthalpy will be referred to as h_1.
  - By linear interpolation from Table 9-1, the entropy of saturated vapour at a saturation pressure of 362.605 kPa is equal to 0.91595 kJ/kg K. This entropy will be referred to as s_1.
- The condenser temperature corresponds to the saturated liquid temperature at state *3* and shall therefore be referred to as T_3.

- The saturation pressure at a temperature of 30°C is equal to 770.06 kPa. This pressure will be referred to as P_3.
- The enthalpy of saturated liquid at a saturation pressure of 770.06 kPa is equal to 91.94 kJ/kg. This enthalpy will be referred to as h_3.

- The standard vapour-compression cycle assumes that the entropy remains constant between state *1* and state *2* and therefore $s_1 = s_2 = 0.91595$ kJ/kg K.
  - Another assumption of the standard vapour-compression cycle is that the pressure between state *2* and state *3* remains constant and therefore $P_2 = P_3$ = 770.06 kPa.
  - Given the entropy and pressure of the refrigerant at state *2*, it can be deduced that the refrigerant is in a superheated state. By interpolating thrice from Table 9-2, the enthalpy at state *2* is found to be equal to 266.198 kJ/kg. This enthalpy will be referred to as h_2.
- In the standard vapour-compression cycle, the enthalpy at state *3* is equal to the enthalpy at state *4* due to the adiabatic nature of the throttling process. Therefore $h_3 = h_4 = 91.49$ kJ/kg.

From Equation 2-9:

$$COP_{chiller} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_{chiller} = \frac{250.665 - 91.49}{266.198 - 250.665}$$

$$COP_{chiller} = 10.25$$

This value of $COP_{chiller}$ however, does not include all the components of the system which require energy. Several pumps which consume energy are present in the system namely, the pump that extracts water from the borehole, the pump that circulates water between the heat exchanger and the condenser and the pump that circulates water between the evaporator and the rooms which require cooling.

The values for the evaporator and condenser temperatures quoted earlier are for a 2300 kW chiller which makes use of a 33.6 kW pump to extract seawater from its dedicated borehole, a 30 kW pump to circulate water between the heat exchanger and the condenser and a total of 101 kW in pumping power to distribute the cooling load to the respective locations throughout the facility.

The refrigerating effect from the above data can be calculated from Equation 2-8 as being:

$$Refrigerating\ Effect = h_1 - h_4$$

$$Refrigerating\ Effect = 250.665 - 91.49$$

$$Refrigerating\ Effect = 159.175\ kJ/kg$$

The mass flow rate of refrigerant required to obtain a refrigerating capacity of 2300 kW can be obtained by dividing the refrigerating capacity by the refrigerating effect:

$$\dot{m} = \frac{Refrigerating\ Capacity}{Refrigerating\ Effect}$$

5-1

$$\dot{m} = \frac{2300}{159.175}$$

$$\dot{m} = 14.45\ kg/s$$

Multiplying the work input to the cycle (Equation 2-7) by the mass flow rate of refrigerant will give the compressor power in Watts:

$$Compressor\ Power = \dot{m}(h_2 - h_1)$$

5-2

$$Compressor\ Power = 14.45(266.198 - 250.665)$$

$$Compressor\ Power = 224.45\ kW$$

This value represents the compressor power input.

Adding the power required for each of the pumps present in the system to the compressor power input gives the total work required for the system to operate:

$$Total\ Power = compressor\ power + seawater\ pump\ power + condenser\ pump\ power + circulation\ pump\ power$$

5-3

$$Total\ Power = 224.45 + 33.6 + 30 + 101$$

$$Total\ Power = 389.05\ kW$$

The system COP can now be calculated by dividing the refrigerating capacity by the total power:

$$COP_{system} = \frac{Refrigerating\ Capacity}{Total\ Power}$$

5-4

$$COP_{system} = \frac{2300}{389.05}$$

$$COP_{system} = 5.91$$

This value of the system COP is in very good agreement with the value quoted by the Hilton management of 5 to 6 (refer to Table 3-1).

5.2 Coefficient of Performance of an Air Cooled System

The Engineering Building of the University of Malta made use of an air cooled chiller. Apart from the chiller, the system also consisted of circulation pumps that circulate cold water around the building and fan units which provided the air cooling for the condenser. The characteristics of this system are summarized in Table 5-1.

Table 5-1 Characteristics of the Engineering Building chiller (Briffa, 2013).

Refrigerant	R22
Refrigerating capacity	259 kW
Circulation pump size	3.73 kW
Fan unit (2 fans for each condenser each consuming 1.12 kW)	2.24 kW
Evaporator temperature	2°C
Condenser Temperature[4]	50°C

Assuming a standard vapour-compression cycle (refer to Figure 2-5) and using Table 9-3 and Table 9-4:

- The evaporator temperature corresponds to the saturated vapour temperature at state *1* and shall therefore be referred to as T_1.
  - The saturation pressure at a temperature of 2°C is equal to 531.2 kPa. This pressure will be referred to as P_1.
  - The enthalpy of saturated vapour at a saturation pressure of 531.2 kPa is equal to 405.8 kJ/kg. This enthalpy will be referred to as h_1.
  - The entropy of saturated vapour at a saturation pressure of 531.2 kPa is equal to 1.748 kJ/kg K. This entropy will be referred to as s_1.

[4] The condenser temperature was quoted as being 15°C above ambient temperature. The mean maximum temperatures for June, July, August and September from 1947 to 2010 are 33.59°C, 36.38°C, 35.77°C and 32.39°C respectively (Galdies, 2011). The mean of these temperatures is 34.53°C which is approximately equal to 35°C. A temperature of 15°C above 35°C is equal to 50°C.

- The condenser temperature corresponds to the saturated liquid temperature at state *3* and shall therefore be referred to as T_3.
  - The saturation pressure at a temperature of 50°C is equal to 1943 kPa. This pressure will be referred to as P_3.
  - The enthalpy of saturated liquid at a saturation pressure of 1943 kPa is equal to 263.2 kJ/kg. This enthalpy will be referred to as h_3.
- The standard vapour-compression cycle assumes that the entropy remains constant between state *1* and state *2* and therefore $s_1 = s_2 = 1.748$ kJ/kg K.
  - Another assumption of the standard vapour-compression cycle is that the pressure between state *2* and state *3* remains constant and therefore $P_2 = P_3 = 1943$ kPa.
  - Given the entropy and pressure of the refrigerant at state *2*, it can be deduced that the refrigerant is in a superheated state. By interpolating thrice from Table 9-4, the enthalpy at state *2* is found to be equal to 438.3 kJ/kg. This enthalpy will be referred to as h_2.
- In the standard vapour-compression cycle, the enthalpy at state *3* is equal to the enthalpy at state *4* due to the adiabatic nature of the throttling process. Therefore $h_3 = h_4 = 263.2$ kJ/kg.

From Equation 2-9:

$$COP_{chiller} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$COP_{chiller} = \frac{405.8 - 263.2}{438.3 - 405.8}$$

$$COP_{chiller} = 4.4$$

This value of $COP_{chiller}$ however, does not include all the components of the system which require energy. Several components which consume energy are present in the system namely, the fan unit which provides air cooling and the pump that circulates water between the evaporator and the rooms which require cooling.

The values for the evaporator and condenser temperatures quoted earlier are for a 259 kW chiller which makes use of two 1.12 kW fans and a 3.73 kW pump to distribute the cooling load to the respective locations throughout the facility.

The refrigerating effect from the above data can be calculated from Equation 2-8 as being:

$$Refrigerating\ Effect = h_1 - h_4$$

$$Refrigerating\ Effect = 405.8 - 263.2$$

$$Refrigerating\ Effect = 142.6\ kJ/kg$$

The mass flow rate of refrigerant required to obtain a refrigerating capacity of 259 kW can be obtained by using Equation 5-1:

$$\dot{m} = \frac{Refrigerating\ Capacity}{Refrigerating\ Effect}$$

$$\dot{m} = \frac{259}{142.6}$$

$$\dot{m} = 1.8\ kg/s$$

Multiplying the work input to the cycle (Equation 2-7) by the mass flow rate of refrigerant will give the compressor power in Watts:

$$Compressor\ Power = \dot{m}(h_2 - h_1)$$

$$Compressor\ Power = 1.8(438.3 - 405.8)$$

$$Compressor\ Power = 58.5\ kW$$

This value represents the compressor power input.

Adding the power required for the fan unit and the circulating pump to the compressor power input gives the total power required for the system to operate (Equation 5-3):

$$Total\ Power = compressor\ power + fan\ unit\ power$$
$$+ circulation\ pump\ power$$

$$Total\ Power = 58.5 + 2.24 + 3.73$$

$$Total\ Power = 64.47\ kW$$

The system COP can now be calculated by dividing the refrigerating capacity by the total power (Equation 5-4):

$$COP_{system} = \frac{Refrigerating\ Capacity}{Total\ Power}$$

$$COP_{system} = \frac{259}{64.47}$$

$$COP_{system} = 4.0$$

5.3 Energy Savings

Two approaches will be adopted in order to estimate the energy savings resulting from the use of GSHP systems; one involves comparison of the electricity demand for each of the summer months[5] to the electricity demand of April while the other involves comparison of the electricity demand for each of the summer months to the electricity demand of November. The rationale for selecting these two particular months as a basis for comparison will be explained in each of the respective sections.

Table 5-2 Power generated in April, summer months and November from the years 2000 to 2011 (NSO, 2012).

Year	Power Generated (MWh)					
	April	June	July	August	September	November
2000	137,614	153,205	178,514	181,348	164,612	144,687
2001	142,302	162,927	192,528	193,355	173,815	150,932
2002	148,844	173,129	205,391	198,433	179,052	157,807
2003	156,528	193,973	232,954	227,086	188,770	162,410
2004	160,588	178,488	221,760	224,533	192,011	168,082
2005	163,433	188,104	226,791	217,974	200,292	166,497
2006	157,392	189,973	232,486	225,010	195,227	167,724
2007	164,451	200,405	232,255	237,344	204,716	175,633
2008	172,613	162,638	242,991	236,165	213,413	164,018
2009	156,629	183,332	222,045	231,631	198,169	158,076
2010	152,877	174,532	220,690	222,289	190,065	161,046
2011	157,549	181,076	222,627	224,596	201,814	165,061

For the years 2000 to 2002, the power generated in the summer months is noticeably lower than for 2003 to 2011 (refer to Table 5-2 and Figure 5-1). This might mean that the cooling demand in Malta increased between 2002 and 2003 thus resulting in the figures for the years 2000 to 2002 not being representative of the current cooling demand of the islands. In view of this, the mean values used for the purpose of the following calculations will be based on values from the years 2003 to 2011.

[5] June, July, August and September make up the summer months.

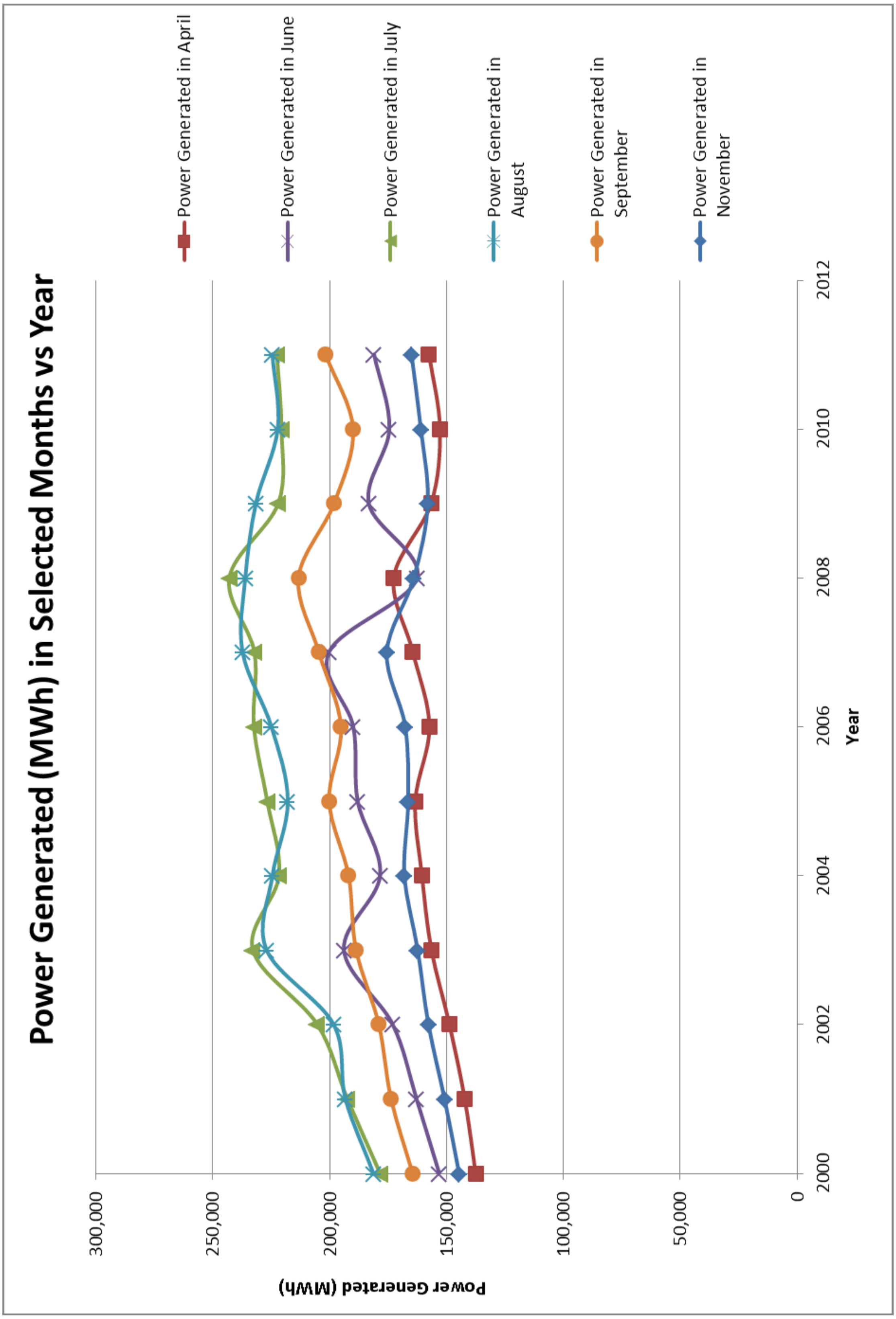

Figure 5-1 Graph showing power generated (MWh) in selected months for the years 2000 to 2011 (plotted using information from: NSO, 2012).

The penetration of cooling systems using saline water in the Maltese cooling market is still rather low and hence it is reasonable to assume that the electricity generated to meet the cooling demands is consumed by air cooled systems.

Malta's aquifer in 2005 was estimated to be approximately 110 m deep below sea level at its deepest point (refer to Figure 5-2) (MRA, 2005). In the absence of water abstraction from the aquifer, the estimated maximum depth would be approximately 160 m below sea level (refer to Figure 5-2) (MRA, 2005). Also, the highest point in Malta is 253 m above sea level (Savona Ventura, 2007). This means that it could be feasible to utilise cooling systems which make use of saline water all over Malta.

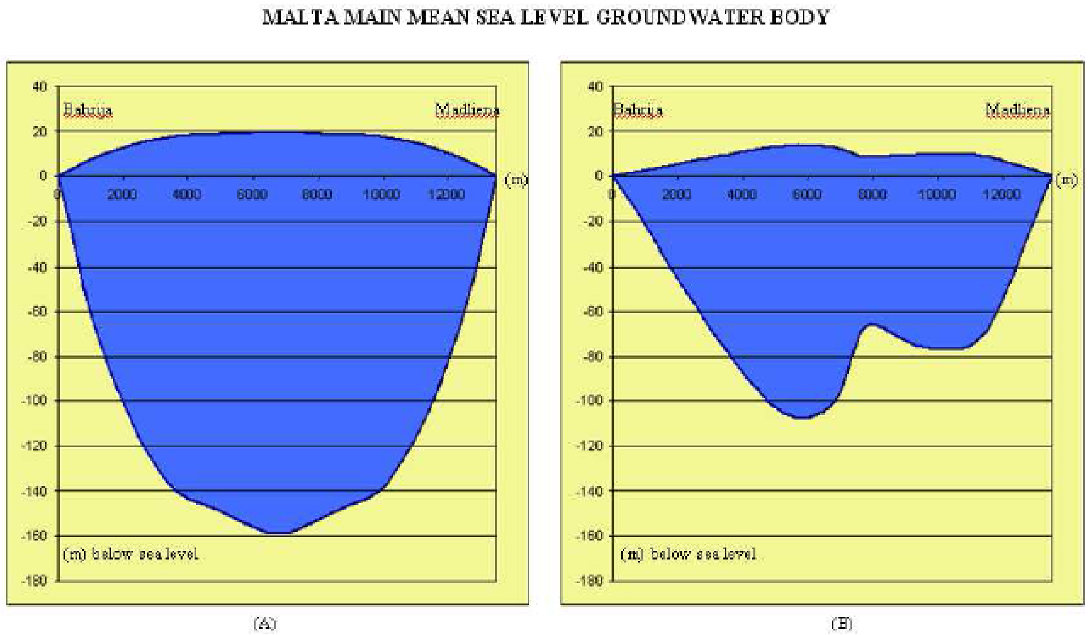

Figure 5-2 Vertical section of the Ghyben-Herzberg freshwater lens in the Lower Coralline Limestone aquifer; (A) at the levels it would stand if there were no abstraction of groundwater and (B) at the level it is today (MRA, 2005).

5.3.1 Comparison to April

The electricity required for meeting the cooling demand in Malta for each of the summer months can be estimated by comparing the power generated in April to that in each of the summer months. It is assumed that the difference in power generated can all be attributed to the cooling demand. April was chosen as a basis for comparison since it is regarded as a heating and cooling neutral month (Ecoheatcool, 2006). This methodology is used by 'Ecoheatcool' working groups to estimate the cooling demand.

Table 5-3 Difference in power generated between April and each of the summer months (table compiled from data obtained from: NSO, 2012).

Year	Difference in Power Generated between April and… (MWh)			
	June	July	August	September
2000	15,591	40,900	43,734	26,998
2001	20,625	50,226	51,053	31,513
2002	24,285	56,547	49,589	30,208
2003	37,445	76,426	70,558	32,242
2004	17,900	61,172	63,945	31,423
2005	24,671	63,358	54,541	36,859
2006	32,581	75,094	67,618	37,835
2007	35,954	67,804	72,893	40,265
2008	-9,975[6]	70,378	63,552	40,800
2009	26,703	65,416	75,002	41,540
2010	21,655	67,813	69,412	37,188
2011	23,527	65,078	67,047	44,265
Mean (2000-2011)	**25,540**	**63,351**	**62,412**	**35,928**
Mean (2003-2011)	**27,555**	**68,060**	**67,174**	**38,046**

From the year 2003 to 2011 considering April as the basis for comparison, the yearly total cooling demand has been 200,835 MWh. This value was calculated by summing up the means from the years 2003 to 2011 in Table 5-3.

[6] This value is an outlier and was therefore omitted from the calculation of the mean.

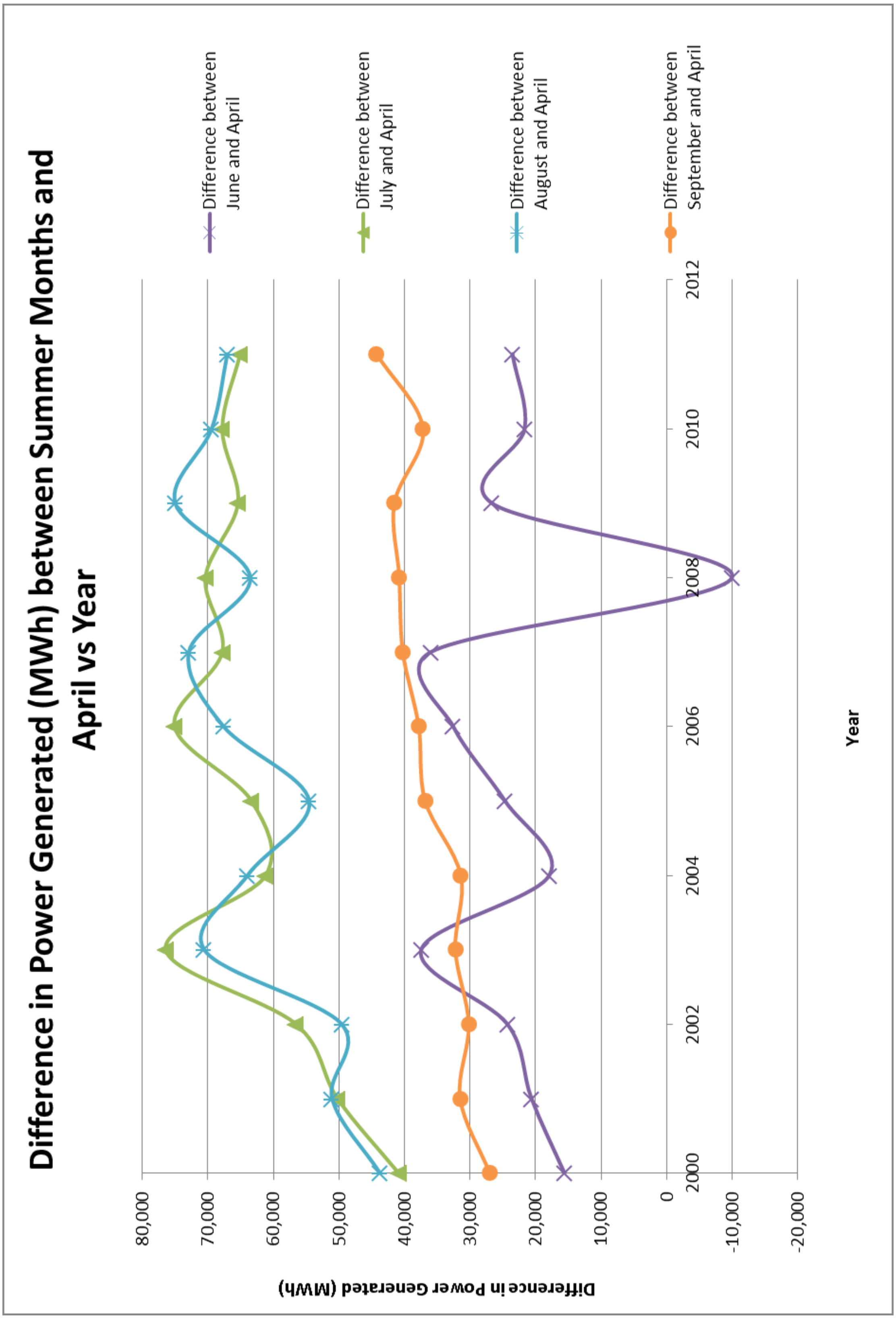

Figure 5-3 Graph showing the difference between power generated (MWh) in April and each of the summer months for the years 2000 to 2011 (plotted using information from: NSO, 2012).

From the definition of the COP (Equation 5-4):

$$COP = \frac{Refrigerating\ Capacity}{Power\ Requirements}$$

$$\therefore Refrigerating\ Capacity = COP \times Power\ Requirements$$

Assuming that a typical air cooled system has a COP of 4.0 (as found in Section 5.2):

$$Refrigerating\ Capacity = 4.0 \times 200835$$

$$Refrigerating\ Capacity = 803340\ MWh$$

Also, from the definition of the COP (Equation 5-4):

$$Power\ Requirements = Refrigerating\ Capacity/COP$$

If all cooling requirements in Malta had to be met by water cooled systems and such systems on average had a COP of 5.9, the power requirements would be equal to:

$$Power\ Requirements = 803340/5.9$$

$$Power\ Requirements = 136159\ MWh$$

Therefore the potential savings would be:

$$\begin{aligned} &Potential\ Savings \\ &\qquad = Power\ Requirements\ of\ Air\ Cooled\ Systems \\ &\qquad - Power\ Requirements\ of\ Water\ Cooled\ Systems \end{aligned} \tag{5-5}$$

$$Potential\ Savings = 200835 - 136159$$

$$Potential\ Savings = 64676\ MWh$$

64.7 GWh could be saved annually if all cooling systems in Malta had a COP of 5.9 instead of 4.0 and the power generated in April is used as a basis for comparison.

5.3.2 Comparison to November

Figure 5-4 shows that in the year 2000, the least energy consumption occurred when the outdoor temperature was 18°C. Figure 5-5 on the other hand shows that the average monthly temperature closest to 18°C occurs in November in Malta. Therefore the electricity required for meeting the cooling demand in Malta for each of the summer months can be estimated by comparing the power generated in November to that in each of the summer months. It is assumed that the difference in power generated can all be attributed to the cooling demand.

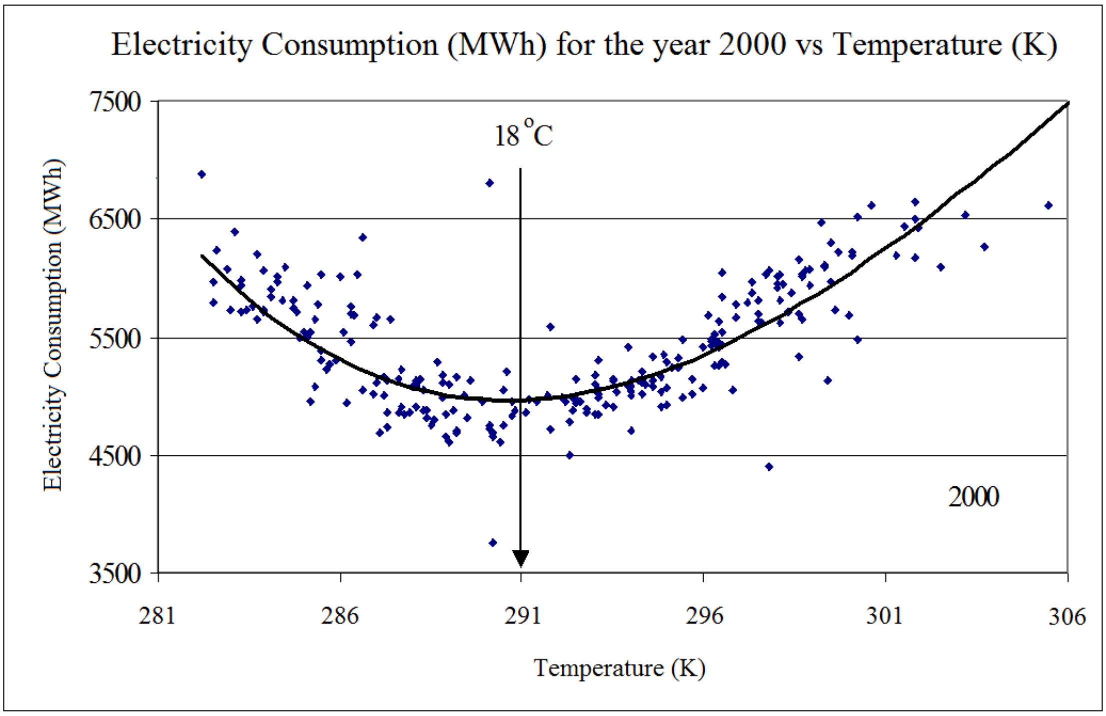

Figure 5-4 Electricity consumption against temperature (Fsadni, 2000).

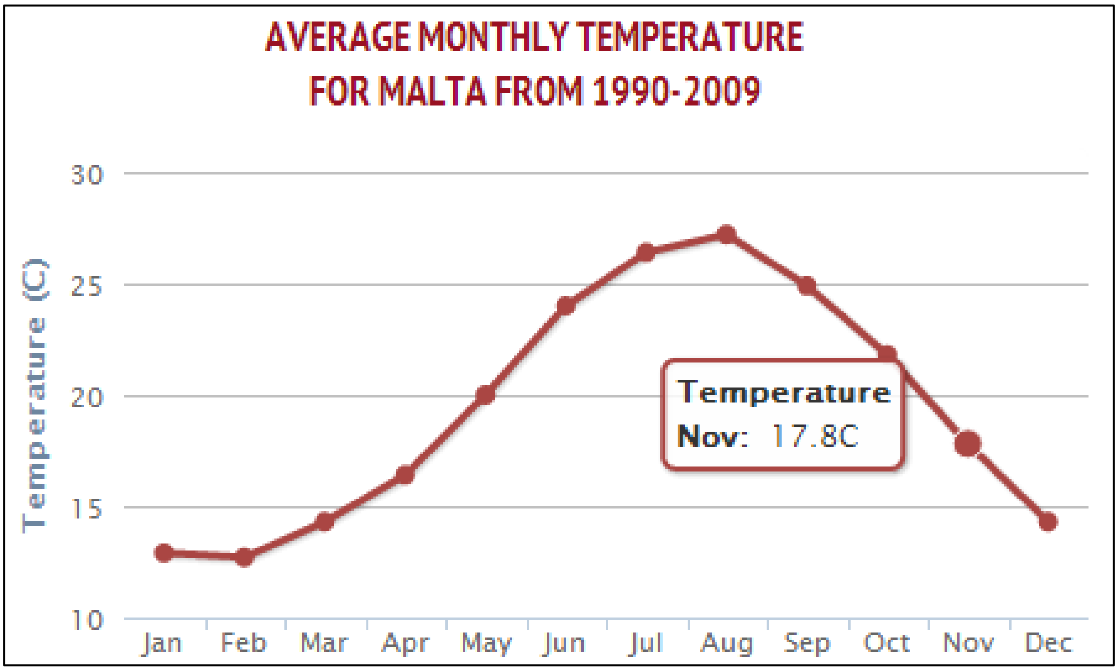

Figure 5-5 Average monthly temperature for Malta from 1990-2009 (The World Bank Group, 2013).

Table 5-4 Difference in power generated between November and each of the summer months (table compiled from data obtained from: NSO, 2012).

Year	Difference in Power Generated between November and… (MWh)			
	June	July	August	September
2000	8,518	33,827	36,661	19,925
2001	11,995	41,596	42,423	22,883
2002	15,322	47,584	40,626	21,245
2003	31,563	70,544	64,676	26,360
2004	10,406	53,678	56,451	23,929
2005	21,607	60,294	51,477	33,795
2006	22,249	64,762	57,286	27,503
2007	24,772	56,622	61,711	29,083
2008	-1,380[7]	78,973	72,147	49,395
2009	25,256	63,969	73,555	40,093
2010	13,486	59,644	61,243	29,019
2011	16,015	57,566	59,535	36,753
Mean (2000-2011)	**18,290**	**57,422**	**56,483**	**29,999**
Mean (2003-2011)	**20,669**	**62,895**	**62,009**	**32,881**

[7] This value is an outlier and was therefore omitted from the calculation of the mean.

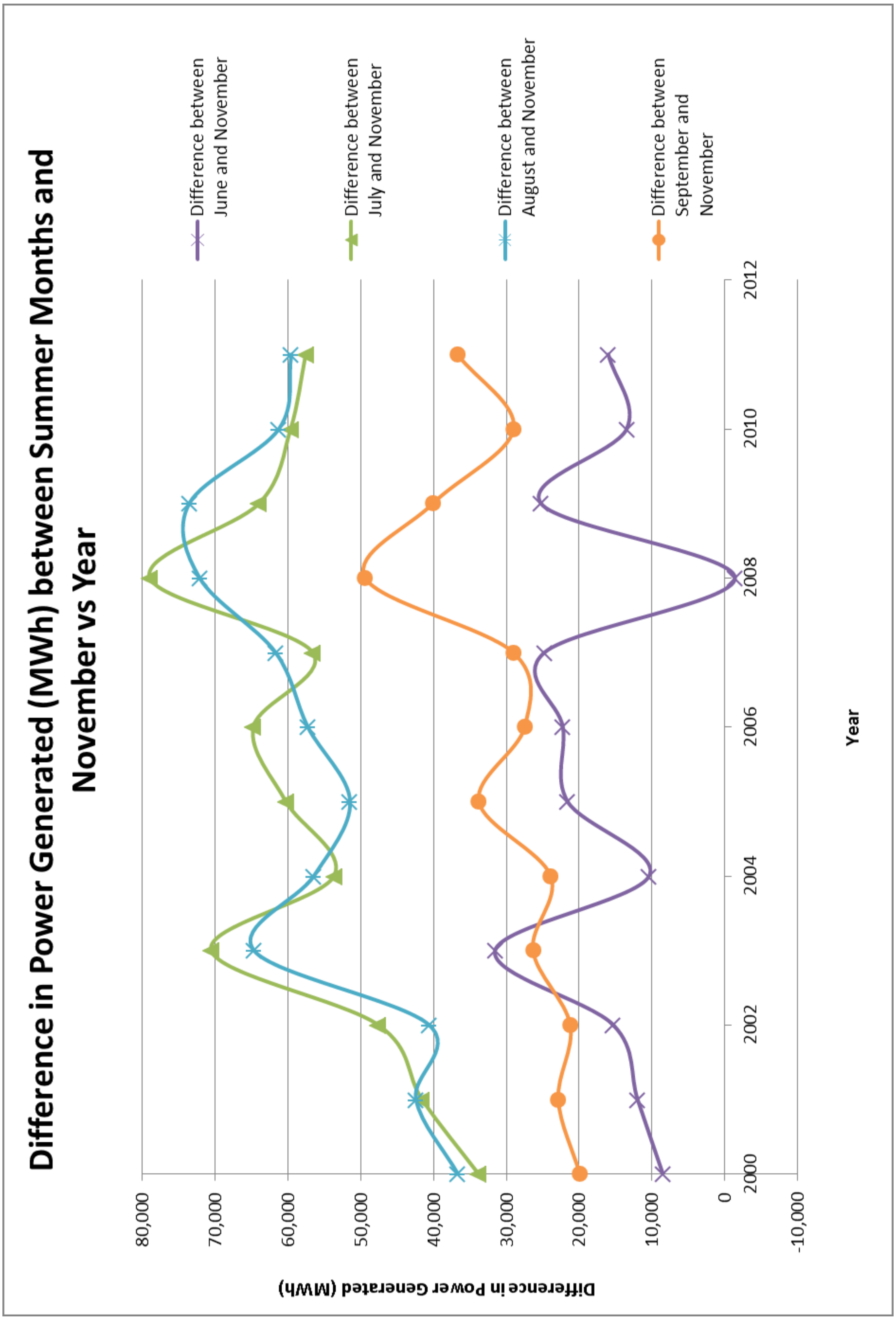

Figure 5-6 Graph showing the difference between power generated (MWh) in November and each of the summer months for the years 2000 to 2011 (plotted using information from: NSO, 2012).

From the year 2003 to 2011 considering November as the basis for comparison, the yearly total cooling demand has been 178,454 MWh. This value was calculated by summing up the means from the years 2003 to 2011 in Table 5-4.

As found in the previous section from Equation 5-4:

$$Refrigerating\ Capacity = COP \times Power\ Requirements$$

Assuming that a typical air cooled system has a COP of 4.0 (as found in Section 5.2):

$$Refrigerating\ Capacity = 4.0 \times 178454$$

$$Refrigerating\ Capacity = 713816\ MWh$$

Also from the previous section:

$$Power\ Requirements = Refrigerating\ Capacity/COP$$

If all cooling requirements in Malta had to be met by water cooled systems and such systems on average had a COP of 5.9, the power requirements would be equal to:

$$Power\ Requirements = 713816/5.9$$

$$Power\ Requirements = 120986\ MWh$$

Therefore the potential savings according to Equation 5-5 would be:

$$\begin{aligned} Potential\ Savings& \\ &= Power\ Requirements\ of\ Air\ Cooled\ Systems \\ &- Power\ Requirements\ of\ Water\ Cooled\ Systems \end{aligned}$$

$$Potential\ Savings = 178454 - 120986$$

$$Potential\ Savings = 57468MWh$$

57.5 GWh could be saved annually if all cooling systems in Malta had a COP of 5.9 instead of 4.0 and the power generated in November is used as a basis for comparison.

5.4 Avoided Carbon Dioxide Emissions

In Malta, every 1 kWh of electricity generated results in 0.88 kg of carbon dioxide[8] emitted to the atmosphere (Enemalta Corporation). This means that if 64.7 GWh of electricity is saved, a total of 56,936 tonnes of carbon dioxide emissions could be avoided every year. On the other hand if 57.5 GWh of electricity is saved, a total of 50,600 tonnes of carbon dioxide emissions could be avoided every year.

[8] This figure is conservative since it is the emission of carbon dioxide per unit of electricity generated. The figure per unit of electricity consumed would be higher since one would have to account for the transmission and distribution losses.

Chapter 6

Discussion of Results

6 Discussion of Results

6.1 Remarks about the Calculations

The results obtained were calculated using the principles of the standard vapour-compression cycle. The standard vapour-compression cycle departs from the actual vapour-compression cycle and actually assumes the cycle to be ideal. Therefore the values obtained for the COP are higher than it would be in real systems.

Also, the calculations carried out to obtain the values of the COP assume that all the components in the system are operating at full power. Pumps are increasingly being equipped with speed drives that control their speed according to the demand; therefore these types of pumps would not be consuming the same amount of power all the time. The chiller itself would also not always be supplying its rated cooling capacity. Due to these two factors, the COP of the system would vary throughout the operation of the system as the system would not always be operating under the same conditions.

A very common method of providing cooling in the zones requiring it is by the use of water-to-air systems. In different areas of a facility, fan units would be present to blow air over pipes through which chilled water flows. These units are present in both air cooled and water cooled chiller systems. The power requirements for these fans was not included in the calculations; however, it should not drastically alter the result as the power requirements for these fans would be small in comparison to the power requirements of the other system components.

Other components in the system might also require power such as the sensors which are present in multiple sub-systems and the control system to which the sensors feed information. The power consumption of sensors is very small and would not make a significant difference in the results. The power consumption of the control system was not included in the calculations since most often the control system would also be controlling other utility-providing systems (such as boilers for heating) and thus the power required would not be only for controlling the cooling system.

6.2 Remarks about the Different Systems

Different systems will consist of different components and therefore the COP will be different from system to system. Some systems might have a lower COP because some components would be unnecessarily oversized.

Adding the power requirements of the fan and/or pump components to the compressor power requirement causes the COP to go down by 0.4 in an air cooled system and by 4.4 in a water cooled system.

Recall Equation 5-4:

$$COP_{system} = \frac{Refrigerating\ Capacity}{Total\ Power}$$

This indicates that the air cooled system has fewer power-consuming components (or components that consume less power) aside from the compressor and therefore the denominator in Equation 5-4 is only slightly bigger than the denominator in:

$$COP_{chiller} = \frac{Refrigerating\ Capacity}{Compressor\ Power}$$

6-1

Resulting in a smaller variation between COP_{system} and $COP_{chiller}$.

Despite this, water cooled systems have a better COP because they operate between temperatures that are closer to each other. Having evaporator and condenser temperatures which are closer to each other reduces the compressor work. Therefore the evaporator and condenser temperatures together have a greater influence on the COP since the power required by the compressor is the major component of the power requirement of the entire system.

6.3 Remarks about the Results

Greater energy savings result from comparing the power generated in the summer months to April than when it is compared to November. This means that although the lowest electricity consumption in Malta occurs when the outdoor temperature is 18°C, it does not correspond to the month having the average daily temperature closest to 18°C, that is, November. From statistics published by the National Statistics Office, the minimum power generated never occurred in November in the years 2000 to 2011 (NSO, 2012). On the other hand, the minimum power generated occurred in April in nine out of the twelve years for which values of the power generated were published (NSO, 2012). In another two out of the twelve years for which values of the power generated were published, April had the second lowest power generated (NSO, 2012). These results can be seen in Figure 6-1.

In view of these findings and still assuming that the difference in power generated between the reference month and summer months can all be attributed to the cooling demand, the results obtained from comparison to April are going to be used and discussed further. The larger value for the power generated in November may be due to the heating demand.

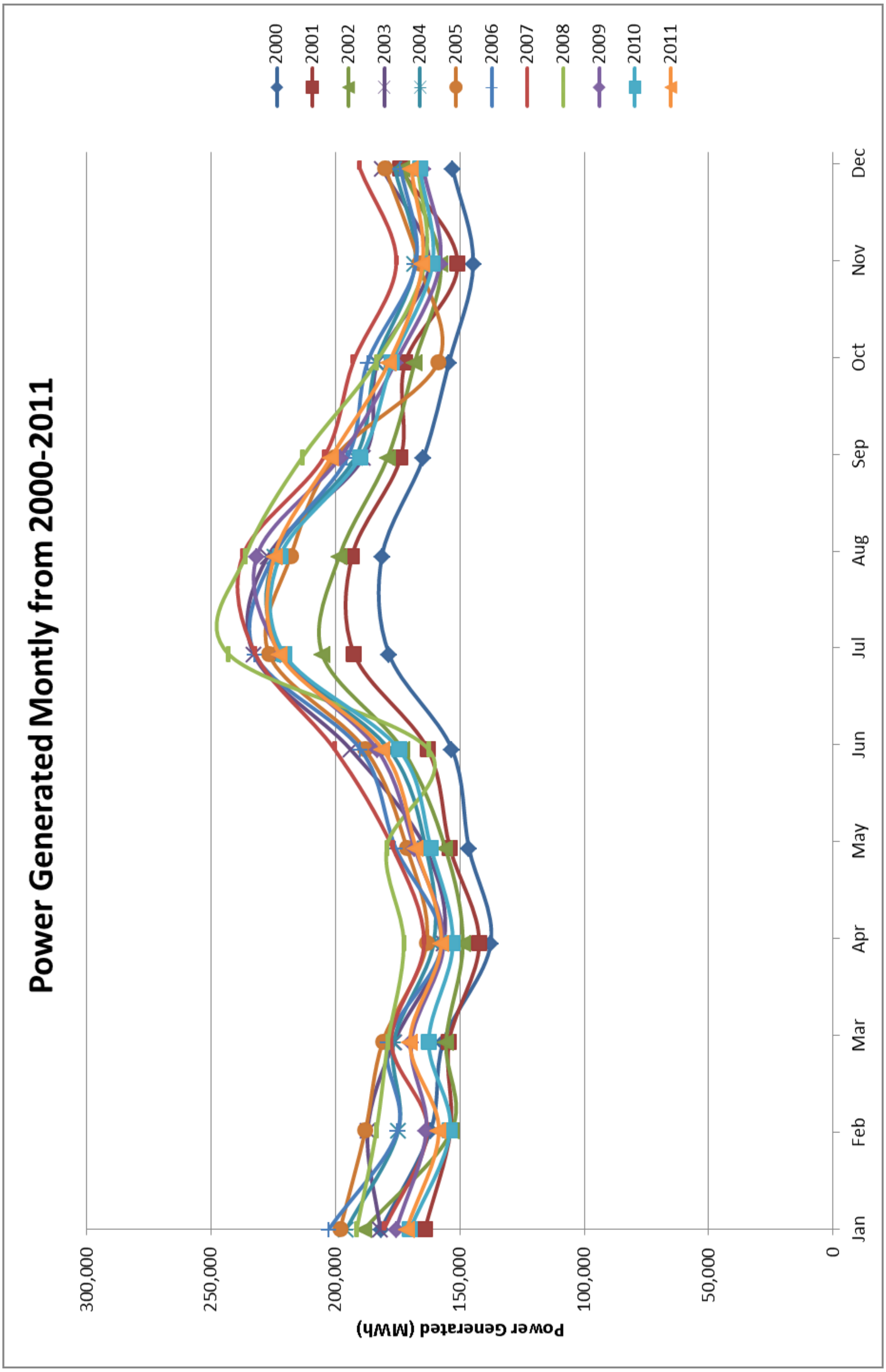

Figure 6-1 Graph showing power generated (MWh) monthly for the years 2000 to 2011 (plotted using information from: NSO, 2012).

The mean for the total power generated per year between 2003 and 2011 is 2,219,428 MWh and the potential energy savings that can be obtained from water cooled cooling systems is 64,676 MWh. Therefore approximately 2.91% of the generated electricity in Malta could be avoided if water cooled cooling systems were used instead of air cooled cooling systems.

Additionally, from the year 2001 to 2011 the trend has been to have the peak generation of power throughout the year occurring in July or August (refer to Figure 6-1). This peak probably occurs due to the demand for cooling in July or August since these also happen to be the hottest months in the year (refer to Figure 5-5). Therefore by using GSHP systems this peak could potentially be lowered due to the reduced consumption of electricity for the same amount of cooling required. Reductions of peak demands have advantages associated with them such as the reduced risk of power cuts and the avoided installation of extra generating capacity which is brought online only to supply the demand over the base load.

Despite the energy savings that can result from using water source chillers as opposed to air source chillers, it does not mean that the Maltese Islands should embark on some nation-wide project to convert cooling systems from one form to the other. A careful economic feasibility study would need to be carried out (among other types of studies) to establish whether it would be economically feasible to carry out a conversion from one type of system to the other. It is important to remember that GSHP systems suffer from the disadvantage of having high initial costs even though systems making use of groundwater have the lowest initial costs when compared to other GSHP systems (Natural Resources Canada, 2009) (ASHRAE, 1995).

6.4 Difficulties

Several difficulties were encountered in obtaining data to conduct calculations.

Obtaining data regarding the systems themselves was difficult both because of lack of monitoring and also because of lack of knowledge from system administrators.

Additionally, both at national and regional scale, data about and relating to the cooling requirements for Malta seem to be lacking or estimated. One document which reported missing and estimated data for Malta is the 'Ecoheatcool' report.

It would be interesting to have enough information to go into more depth in the systems themselves. For example, for the seawater heat exchanger it would be interesting to have seawater temperatures before and after the heat exchanger, values for the characteristics of the heat exchanger such as its heat transfer area and its overall heat transfer coefficient and the influence of the fouling factor. This would make it possible to calculate the heat exchanger effectiveness. For the heat exchanger in air cooled systems, the same information would be useful to have so that the effectiveness of the heat exchanger of an air cooled system could be compared to the effectiveness of the heat exchanger of a water cooled system. Additionally, this information would make it possible to have an idea of how much heat is rejected at the condenser for the two types of systems and what flow rates would be required for both air and water to obtain a definite amount of heat rejection. From these calculations one could get an idea of what size the heat exchangers and fans or piping for seawater (depending on the type of system) would need to be.

Metering of each component that consumes energy would help improve the accuracy of calculations drastically and can have the advantage of identifying potential wastes or faults in the system. However, the installation of meters might be a costly and hectic process.

It would also be useful to know the cooling requirements of different types and sizes of Maltese buildings and use them for conducting calculations. This information is available for many countries. Information that would help identify cooling requirements includes the R-value for a typical wall in different types of building and the influence of windows and doors on the cooling requirement when considering Malta's climate.

To be able to complete more accurate calculations regarding the potential energy savings that can be achieved by using water cooled chillers one would need more accurate data regarding the actual amount of power generated in order to meet the cooling demand. It would then be necessary to gather statistics relating to how many cooling systems are present, their type, size and performance characteristics. This would make it possible to quantify more accurately how much energy is being consumed by air cooled systems.

Chapter 7

Recommendations and Conclusions

7 Recommendations and Conclusions

7.1 Recommendations

7.1.1 District Heating and Cooling

DHC systems are systems that provide heating and cooling to multiple buildings through well-insulated, underground piping (Dincer & Rosen, 2007) (Euroheat & Power, 2013). A DHC system is usually composed of a thermal source plant, an outdoor pipeline network, hot/chilled water circulation pumps, and the heating/cooling systems inside buildings (Shu, Lin, Zhang, & Zhu, 2010). Residential, public, and commercial buildings together with low-temperature-heat-demanding industries can make use of DHC (Euroheat & Power, 2013). The benefits of having a DHC system are related to efficiency, economics and the environment (Nijjar, Fung, Hughes, & Taherian, 2009).

Primary energy sources are most often used to produce a heating or cooling effect (DHC Technology Platform, 2012). DHC can eliminate or reduce the need for using primary energy to produce a heating or cooling effect (DHC Technology Platform, 2012). Heat, cold or local fuel that would otherwise be lost or remain unused or heat from renewable sources is generally used in DHC systems (Euroheat & Power, 2013) (DHC Technology Platform, 2012). DHC systems are very versatile with regards to energy source (Rezaie & Rosen, 2012) (IEA, 2013) and can use heat from combined heat and power plants, waste-to-energy, biomass, geothermal, industrial excess heat (Persson & Werner, 2011), solar thermal, deep sea (Euroheat & Power, 2013), other renewables and indigenous or abundant fuels (IEA, 2013). Many systems all over the world take advantage of the cold temperatures found in deep waters to supply cooling through DHC systems (Zhen, Lin, Shu, Jiang, & Zhu, 2007) (Mitchell & Spitler, 2013) (Looney & Oney, 2007) (Zogg, Roth, & Broderick, 2008). The average market share of DHC in Europe is of 10% but this figure can go up to 50% or more for the regions of North, Central and Eastern Europe (Euroheat & Power, 2013). It is estimated that more than 80% of heat supplied by DHC networks is sourced from renewable energy or heat recovery (Euroheat & Power, 2013).

An extra 40 million tonnes of carbon dioxide could be avoided yearly if more DHC networks were to be created across Europe (Euroheat & Power, 2013). This figure corresponds to a 9.3% reduction in carbon dioxide emissions and would therefore cover the Kyoto target (Euroheat & Power, 2013).

The use of DHC can save fuel and primary energy, reduce environmental concerns (air pollution, stratospheric ozone depletion and acid precipitation) and carbon dioxide emissions and increase energy security (IEA, 2013) (IEA DHC/CHP Executive Committee, 2002) (Samuel, Nagendra, & Maiya, 2013). Additionally, DHC technology is comfortable, reliable, mature and future-proof (Euroheat & Power, 2013). DHC systems can take advantage of economies of scale and the diversity in the cooling demand of different buildings (Shu, Lin, Zhang, & Zhu, 2010) (Yik, Burnett, & Prescott, 2001). Since more chiller units would be connected to a DHC network the efficiency of the DHC system can be optimized due to closer matching of the combination of chillers and pumps to be put into operation given a total cooling load on the system and also due to the better efficiencies of larger chillers and pumps (Yik, Burnett, & Prescott, 2001). In some types of DHC systems it is also possible to reduce noises, structural loads and required equipment area thus resulting in greater flexibility when designing a building (Shu, Lin, Zhang, & Zhu, 2010).

Despite having some major benefits, there are also some disadvantages associated with DHC systems, namely (MacRae, 1992):

- Limited knowledge and technical skills related to DHC systems.
- Large capital costs which necessitate hefty negotiations with investors.
- Difficulty in finding a suitable site where the heat source is close to the consumers.
- Economically feasible in densely populated areas, high density building clusters and industrial complexes but less advantageous in low-density residential areas.

Malta's size and the fact that it is an island mean that there is very easy access to seawater almost anywhere on the island. Therefore the potential for DHC should be explored especially in areas that are still being constructed which accommodate a high density of people. Creating a DHC network at the time of construction is less costly and less problematic than retrofitting.

7.1.2 A Tool Illustrating Potential for GSHP Systems

GSHP system performance and initial cost (including costs for boring and piping) rely on geological parameters and groundwater conditions (Nam & Ooka, 2011). Obtaining this information usually requires experiments and testing which greatly increase the cost of GSHP systems (Nam & Ooka, 2011). This aspect of GSHP systems emphasizes the need to accurately estimate geological parameters such as thermal properties, heat extraction rates and groundwater conditions (Nam & Ooka, 2011). One way of facilitating this process is through the creation of a tool that illustrates the potential for GSHP systems depending on the location.

Water temperature profiles for boreholes in different parts of Malta could be created to establish whether the temperatures of the saline groundwater are suitable for a GSHP and how deep one would have to dig to obtain saline groundwater at the required temperatures. The cost of drilling a borehole is proportionate to its depth and therefore the water temperature profiles could give an idea whether a GSHP system could be economically feasible.

In addition to water temperature profiles, there could be information about how deep one would have to drill in order to obtain a certain flow rate of water since this will also play an important role in establishing whether a GSHP system would be feasible.

These types of profiles could be done at key locations on the island and then depending on the location of the proposed system, one would use the data from the closest or most similar location.

The information from water temperature profiles, depths and regulatory requirements could then be combined into a tool that illustrates the locations having potential for GSHP systems in Malta. A similar tool exists for England and Wales.

The British Geological Survey in collaboration with the Environment Agency has created a web-based screening tool for England and Wales that can illustrate in which areas the right conditions for GSHP systems exist (Abesser, 2013). The conditions considered relate to the hydrogeological and economic aspects of the installation and include the presence of an aquifer, installation and pumping costs and regulatory requirements (Abesser, 2013). The tool is developed within a Geographic Information System and maps the potential for open loop GSHP installations with a heating or cooling output greater than 100 kW (Abesser, 2013).

Needless to say, more detailed analysis would then need to be carried out in order to obtain data for the specific location of the system. However, indicative, representative data could be a good start to establish where such systems can operate feasibly.

If a screening tool were to be developed it would have to be accompanied by informative and educational efforts aimed towards developers, stakeholders and the general public to explain why GSHP systems are beneficial and at the same time promote their use. Lack of knowledge about these novel systems results in reluctance from stakeholders to install these solutions even in favourable locations (Samuel, Nagendra, & Maiya, 2013).

7.2 Conclusions

GSHP systems are currently being used in Malta in a few establishments to provide cooling. Their principle of operation is to reject heat from the condenser to saline groundwater obtained from boreholes through a heat exchanger. The higher temperature saline ground water is then rejected to the sea.

Chillers cooled with saline groundwater are subject to the Borehole Drilling and Excavation Works within the Saturated Zone Regulations and to the Protection of Groundwater against Pollution and Deterioration Regulations.

Under both the Energy End-Use Efficiency and Energy Services Regulations and the Promotion of Energy from Renewable Sources Regulations, GSHP systems can contribute to the set targets however, if they are being accounted for in one target they may not be counted again as a contribution to the other target.

Heat pumps and DHC are both regarded as adequate means for improving the energy performance of buildings under the Energy Performance of Buildings Regulations.

The system COP of an air cooled cooling system was found to be 4.0 while that of a water cooled cooling system was found to be 5.9. This translates into an energy saving potential of 64.7 GWh of electricity per year and a total of 56,936 tonnes of carbon dioxide emissions avoided per year. 64.7 GWh represents approximately 2.91% of the generated electricity in Malta per year.

If the uptake of more GSHP systems were to be considered, the necessary regulatory framework would need to be put in place regarding the implementation of GSHP systems. Also, due consideration would need to be given to providing incentives or subsidies in order to encourage their uptake. Training of people in the operation and maintenance of such systems would also be required. Considerable investment by the government would need to be made to improve the dissemination of such technologies. Large investments by the private sector would also be required in order to cover the initial costs of GSHP systems.

The creation of DHC networks and the use of GSHP systems in Malta are hindered by common practice, mentality and culture. Maltese people tend to prefer owning property than renting it and the prices for buying property in Malta are rather high. For these reasons, contractors who develop blocks of apartments usually tend to opt for less costly options so as to keep the price of the property low and remain competitive on the market. Therefore contractors very often do not bother with installing insulation and do not go into the trouble of installing a centralised cooling system.

Different units in a building are usually then sold separately and in 'shell' form, meaning that individual owners of units in a building would then find contractors themselves to install utilities. Very often certain units remain vacant and therefore without utilities installed while other units start being used. When, and if, all the units in the building start being used, each unit would have its own systems and none of the owners would be willing to invest in a centralised system because they would each have invested in individual, very often less energy-efficient, systems. This does not leave much opportunity for the installation of systems such as GSHPs except in establishments such as hotels, large office buildings, industrial buildings and blocks of ‘high-end’ apartments.

Also, in Malta, very often air conditioning is regarded as a luxury and as a commodity which one can do without. Maltese homes very often only have a few air conditioned rooms and air conditioning units are not left running for long periods of time. For such conditions it is usually common practice to have a stand-alone, split-type air conditioning unit. At the household or individual level it is probably not feasible to invest in a SWHP system which has high capital costs. The cost of the system would probably not be recovered given the small amounts of area that require cooling and the short operating times.

Chapter 8

Appendix A – Baxter Visit

8 Appendix A – Baxter Visit (Scerri, 2013)

Baxter (Malta) Limited produces medical devices in its facilities in Marsa. The production of medical devices requires very strict environmental conditions and as such the facilities require constant and strict temperature control. This is achieved by means of chillers, the condensers of which are cooled by means of freshwater.

The freshwater used for cooling the condenser passes through a vertical ground loop and a cooling tower such that it rejects heat to the ground and to the atmosphere (refer to Figure 8-1). The system as such makes use of a closed loop and neither fresh groundwater nor saline groundwater are extracted or affected except for their temperature. Groundwater is only used as a heat sink and only comes into contact with the outer surface of the piping which makes up the vertical ground loop.

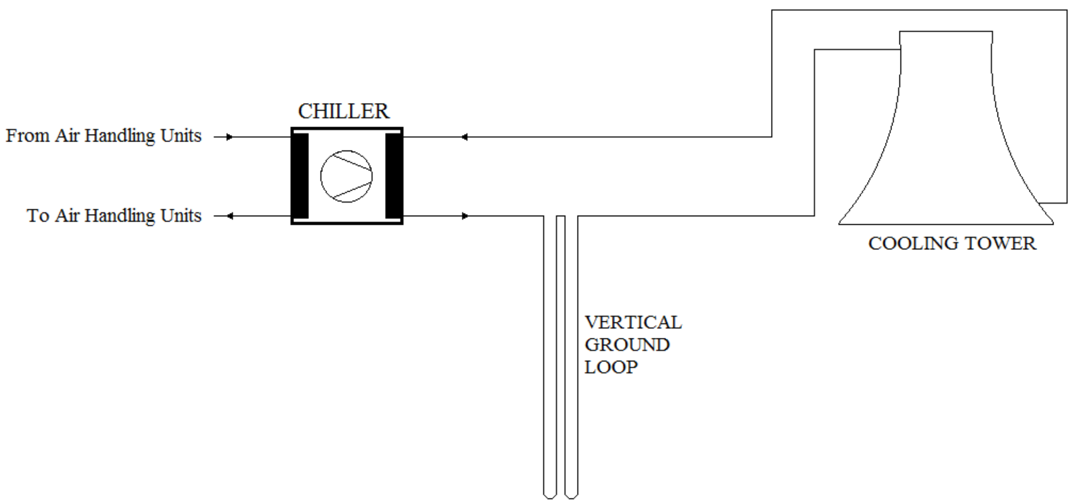

Figure 8-1 Schematic diagram of the Baxter system.

Two cooling towers are used to cool the condensers of two 600 kW turbo-core chillers. This type of chiller makes use of very efficient compressors in which the principle of magnetism is utilised rather than having bearings. Turbo-core chillers are not very common in Malta anymore. Another three chillers with approximately 300 kW cooling capacity also form part of the system. All chillers make use of refrigerant R134a. Typical chiller COP values are around 4.

Two 11 kW and one 8 kW circulation pumps are used in the system to circulate the chilled water around the facilities.

The 22 boreholes in which the vertical ground loops are found are 150 m deep. No water is extracted from these boreholes as they are only used as a means to reach deep into the ground and make use of its geothermal properties. Upon passing through the vertical ground loops, the temperature of the fresh water drops by around 1°C to 2°C. Another 5°C temperature drop is obtained by passing the freshwater through the cooling tower.

The geothermal loop is only used in the summer months (June to September), however; the cooling towers are used throughout the year. The use of the geothermal loop in winter is not feasible as the energy required to circulate the water through the loop becomes higher than the temperature drop achieved. This causes a reduction in system efficiency.

The freshwater that circulates on the condenser side of the system needs to be replenished due to evaporative losses that occur in the cooling tower. The freshwater is replenished from a freshwater reservoir which is in turn supplied by freshwater from a bowser. Rainwater is also collected into the reservoir from some of the roof areas of the facility, however, not all roofs drain into the reservoir and since rain falls mostly in winter in Malta, the supply from the bowser is required in summer. The water reservoir is relatively small when compared with the size of the facilities – in winter it overflows with the rainwater collected from only part of the roof area. The plant manager had plans for increasing the size of the water reservoir however they have been put aside due to the high cost of the project. The cost of increasing the size of the reservoir would be much greater than what the company is currently spending to get water bowsers.

The major problem with obtaining water from bowsers is the hardness of the water. Random tests are carried out to monitor this aspect of water quality. Water hardness should ideally be low for use in cooling towers since it can cause scale which greatly reduces the efficiency of cooling towers. The cooling tower is set to reject water whose hardness is higher than a set limit. The hardness of the freshwater will inevitably increase as the water goes through several cycles in which evaporation occurs and hence the concentration of elements which contribute to hardness (such as calcium and magnesium) in the water increases. With this aspect in mind it is therefore desirable to replenish the freshwater in the loop with freshwater having as low a hardness as possible as otherwise the amount of cycles during which the water remains in circulation decreases and the requirement to replenish water increases. Freshwater is treated at the cooling towers for legionella.

8.1 The Drive for Implementation of Water Cooling

Cooling towers were installed approximately seven years ago. Prior to that all chillers were air cooled. However, air cooling was not effective enough in summer and used to be supplemented by cooling with water jets on the outside surface of the condensers.

The main drive to implement water cooled systems was for energy savings. Water cooling has its drawbacks; in the case of Baxter, sub-contracting water supplies itself is somewhat of a hassle since it is not always easy to get a contractor to supply water and it may not be easy to find satisfactorily good quality water. Other drawbacks of using water include the corrosion and fouling that it causes. This gives rise to the need for more frequent maintenance and cleaning. Cleaning of the condensers is carried out every two months and it involves a cycle of cleaning with certain acids and flushing with water

8.2 Heat Recovery and System Upgrade

Heating is never required for the production zones. The chillers are used in summer to provide chilled air to the production zones. In winter only one chiller is operated and the facilities also make use of winter free cooling systems which circulate the cold, outside air to cool the facilities.

The cooling system at Baxter is currently being upgraded to include heat recovery. The heat rejected by the chillers will start being recovered and used for heating offices. When the heating load is greater than what the heat recovery process can supply, a heat pump coupled to the ground loop will come into operation.

Energy savings apart, some of the air conditioning units had to be changed due to the fact that they made use of the refrigerant R22 which has to be decommissioned by 2015. The management saw an opportunity in this and upgraded the system to include heat recovery.

Baxter Malta was awarded the Ambassador’s Award for Environmental Excellence in 2008. The award, instituted by the US Embassy in Malta, rewards US businesses operating in Malta for their environmental efforts.

Chapter 9

Appendix B – Refrigerant Property Tables

9 Appendix B – Refrigerant Property Tables

Table 9-1 Saturation temperature table for R134a (ASHRAE, 1988).

Saturated R134a -- Temperature Table							
		Spec. Volume		Enthalpy		Entropy	
°C	MPa	m3/kg		kJ/kg		kJ/kg.K	
Temp.	Sat. press.	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor
T °C	p_sat@T	v_f	v_g	h_f	h_g	s_f	s_g
-24	0.11160	0.0007296	0.1728	19.29	232.85	0.0798	0.937
-22	0.12192	0.0007328	0.1590	21.77	234.08	0.0897	0.9351
-20	0.13299	0.0007361	0.1464	24.26	235.31	0.0996	0.9332
-18	0.14483	0.0007395	0.1350	26.77	236.53	0.1094	0.9315
-16	0.15748	0.0007428	0.1247	29.30	237.74	0.1192	0.9298
-12	0.18540	0.0007498	0.1068	34.39	240.15	0.1388	0.9267
-8	0.21704	0.0007569	0.0919	39.54	242.54	0.1583	0.9239
-4	0.25274	0.0007644	0.0794	44.75	244.9	0.1777	0.9213
0	0.29282	0.0007721	0.0689	50.02	247.23	0.197	0.919
4	0.33765	0.0007801	0.0600	55.35	249.53	0.2162	0.9169
8	0.38756	0.0007884	0.0525	60.73	251.8	0.2354	0.915
12	0.44294	0.0007971	0.0460	66.18	254.03	0.2545	0.9132
16	0.50416	0.0008062	0.0405	71.69	256.22	0.2735	0.9116
20	0.57160	0.0008157	0.0358	77.26	258.35	0.2924	0.9102
24	0.64566	0.0008257	0.0317	82.90	260.45	0.3113	0.9089
26	0.68530	0.0008309	0.0298	85.75	261.48	0.3208	0.9082
28	0.72675	0.0008362	0.0281	88.61	262.5	0.3302	0.9076
30	0.77006	0.0008417	0.0265	91.49	263.5	0.3396	0.907
32	0.81528	0.0008473	0.0250	94.39	264.48	0.349	0.9064
34	0.86247	0.0008530	0.0236	97.31	265.45	0.3584	0.9058
36	0.91168	0.0008590	0.0223	100.25	266.4	0.3678	0.9053
38	0.96298	0.0008651	0.0210	103.21	267.33	0.3772	0.9047
40	1.01640	0.0008714	0.0199	106.19	268.24	0.3866	0.9041
42	1.07200	0.0008780	0.0188	109.19	269.14	0.396	0.9035
44	1.12990	0.0008847	0.0177	112.22	270.01	0.4054	0.903
48	1.25260	0.0008989	0.0159	118.35	271.68	0.4243	0.9017
52	1.38510	0.0009142	0.0142	124.58	273.24	0.4432	0.9004
56	1.52780	0.0009308	0.0127	130.93	274.68	0.4622	0.899
60	1.68130	0.0009488	0.0114	137.42	275.99	0.4814	0.8973

Table 9-2 Superheated vapour table for R134a (Bhattacharjee, 2006).

Superheated R134a						
	Spec. Volume	Enthalpy	Entropy	Spec. Volume	Enthalpy	Entropy
°C	m^3/kg	kJ/kg	kJ/kg.K	m^3/kg	kJ/kg	kJ/kg.K
	p = 0.70 MPa (Tsat = 26.72 °C)			p = 0.80 MPa (T_{sat} = 31.33 °C)		
T	*v*	*h*	*s*	*v*	*h*	*s*
Sat.	0.02918	261.85	0.9080	0.02547	264.15	0.9066
30	0.02979	265.37	0.9197	-	-	-
40	0.03157	275.93	0.9539	0.02691	273.66	0.9374
50	0.03324	286.35	0.9867	0.02846	284.39	0.9711
60	0.03482	296.69	1.0182	0.02992	294.98	1.0034
70	0.03634	307.01	1.0487	0.03131	305.50	1.0345
80	0.03781	317.35	1.0784	0.03264	316.00	1.0647
90	0.03924	327.74	1.1074	0.03393	326.52	1.0940
100	0.04064	338.19	1.1358	0.03519	337.08	1.1227
110	0.04201	348.71	1.1637	0.03642	347.71	1.1508
120	0.04335	359.33	1.1910	0.03762	358.40	1.1784
130	0.04468	370.04	1.2179	0.03881	369.19	1.2055
140	0.04599	380.86	1.2444	0.03997	380.07	1.2321
150	0.04729	391.79	1.2706	0.04113	391.05	1.2584
160	0.04857	402.82	1.2963	0.04227	402.14	1.2843

Table 9-3 Saturation temperature table for R22 (DuPont, 2005).

Saturated R22 -- Temperature Table							
		Spec. Volume		Enthalpy		Entropy	
°C	kPa	m3/kg		kJ/kg		kJ/kg.K	
Temp.	Sat. press.	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor
T °C	p_sat@T	v_f	v_g	h_f	h_g	s_f	s_g
-10	354.8	0.0008	0.0653	188.4	401.2	0.957	1.766
-9	367.5	0.0008	0.0631	189.6	401.6	0.962	1.764
-8	380.5	0.0008	0.0610	190.7	402.0	0.966	1.763
-7	393.9	0.0008	0.0590	191.9	402.4	0.970	1.761
-6	407.7	0.0008	0.0571	193.0	402.8	0.974	1.760
-5	421.8	0.0008	0.0553	1942.0	403.2	0.979	1.758
-4	436.3	0.0008	0.0535	195.3	403.5	0.983	1.757
-3	451.1	0.0008	0.0518	196.5	403.9	0.987	1.755
-2	466.4	0.0008	0.0502	197.7	404.3	0.992	1.754
-1	482.0	0.0008	0.0486	198.8	404.7	0.996	1.752
0	498.0	0.0008	0.0471	200.0	405.0	1.000	1.751
1	514.4	0.0008	0.0457	201.2	405.4	1.004	1.749
2	531.2	0.0008	0.0442	202.4	405.8	1.008	1.748
3	548.4	0.0008	0.0429	203.5	406.1	1.013	1.746
4	566.1	0.0008	0.0416	204.7	406.5	1.017	1.745
5	584.1	0.0008	0.0403	205.9	406.8	1.021	1.744
6	602.6	0.0008	0.0391	207.1	407.2	1.025	1.742
7	621.5	0.0008	0.0380	208.3	407.5	1.030	1.741
8	640.9	0.0008	0.0368	209.5	407.9	1.034	1.739
9	660.7	0.0008	0.0358	210.7	408.2	1.038	1.738
10	680.9	0.0008	0.0347	211.9	408.6	1.042	1.737
11	701.7	0.0008	0.0337	213.1	408.9	1.046	1.735
12	722.9	0.0008	0.0327	214.3	409.2	1.051	1.734
13	744.5	0.0008	0.0318	215.5	409.5	1.055	1.733
14	766.7	0.0008	0.0309	216.7	409.9	1.059	1.732
15	789.3	0.0008	0.0300	217.9	410.2	1.063	1.730
16	812.4	0.0008	0.0291	219.1	410.5	1.067	1.729
17	836.1	0.0008	0.0283	220.4	410.8	1.071	1.728
18	860.2	0.0008	0.0275	221.6	411.1	1.076	1.726
19	884.8	0.0008	0.0267	222.8	411.4	1.080	1.725

Table 9-3 Saturation Temperature Table for R22 (DuPont, 2005) *continued…*

Saturated R22 -- Temperature Table							
		Spec. Volume		Enthalpy		Entropy	
°C	kPa	m3/kg		kJ/kg		kJ/kg.K	
Temp.	Sat. press.	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor
T °C	p_sat@T	v_f	v_g	h_f	h_g	s_f	s_g
20	910.0	0.0008	0.0260	224.1	411.7	1.084	1.724
21	935.7	0.0008	0.0253	225.3	411.9	1.088	1.722
22	961.9	0.0008	0.0246	226.5	412.2	1.092	1.721
23	988.7	0.0008	0.0239	227.8	412.5	1.096	1.720
24	1016.0	0.0008	0.0232	229.0	412.8	1.100	1.719
25	1044.0	0.0008	0.0226	230.3	413.0	1.105	1.717
26	1072.0	0.0008	0.0220	231.5	413.3	1.109	1.716
27	1101.0	0.0009	0.0214	232.8	413.5	1.113	1.715
28	1131.0	0.0009	0.0208	234.1	413.8	1.117	1.714
29	1161.0	0.0009	0.0203	235.3	414.0	1.121	1.712
30	1192.0	0.0009	0.0197	236.6	414.3	1.125	1.711
31	1223.0	0.0009	0.0192	237.9	414.5	1.129	1.710
32	1255.0	0.0009	0.0187	239.2	414.7	1.133	1.709
33	1288.0	0.0009	0.0182	240.5	414.9	1.138	1.707
34	1321.0	0.0009	0.0177	241.8	415.1	1.142	1.706
35	1355.0	0.0009	0.0172	243.1	415.3	1.146	1.705
36	1389.0	0.0009	0.0168	244.4	415.5	1.150	1.704
37	1424.0	0.0009	0.0164	245.7	415.7	1.154	1.702
36	1460.0	0.0009	0.0159	247.0	415.9	1.158	1.701
39	1497.0	0.0009	0.0155	248.3	416.1	1.162	1.700
40	1534.0	0.0009	0.0151	249.6	416.2	1.166	1.698
41	1571.0	0.0009	0.0147	251.0	416.4	1.171	1.697
42	1610.0	0.0009	0.0143	252.3	416.6	1.175	1.696
43	1649.0	0.0009	0.0140	253.7	416.7	1.179	1.695
44	1689.0	0.0009	0.0136	255.0	416.8	1.183	1.693
45	1729.0	0.0009	0.0133	256.4	417.0	1.187	1.692
46	1770.0	0.0009	0.0129	257.7	417.1	1.191	1.691
47	1812.0	0.0009	0.0126	259.1	417.2	1.196	1.689
48	1855.0	0.0009	0.0123	260.5	417.3	1.200	1.688
49	1899.0	0.0009	0.0119	261.9	417.4	1.204	1.687

Table 9-3 Saturation Temperature Table for R22 (DuPont, 2005) *continued…*

Saturated R22 -- Temperature Table							
		Spec. Volume		Enthalpy		Entropy	
°C	kPa	m3/kg		kJ/kg		kJ/kg.K	
Temp.	Sat. press.	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor	Sat. liquid	Sat. vapor
T °C	p_sat@T	v_f	v_g	h_f	h_g	s_f	s_g
50	1943.0	0.0009	0.0116	263.2	417.4	1.208	1.685
51	1988.0	0.0009	0.0113	264.6	417.5	1.212	1.684
52	2033.0	0.0009	0.0110	266.0	417.6	1.216	1.682
53	2080.0	0.0009	0.0108	267.5	417.6	1.221	1.681
54	2127.0	0.0009	0.0105	268.9	417.6	1.225	1.680
55	2175.0	0.0010	0.0102	270.3	417.7	1.229	1.678
56	2224.0	0.0010	0.0100	271.8	417.7	1.233	1.677
57	2274.0	0.0010	0.0097	273.2	417.7	1.238	1.675
56	2324.0	0.0010	0.0094	274.7	417.6	1.242	1.674
59	2375.0	0.0010	0.0092	276.1	417.6	1.246	1.672

Table 9-4 Superheated vapour table for R22 (DuPont, 2005).

Superheated R22						
	Spec. Volume	Enthalpy	Entropy	Spec. Volume	Enthalpy	Entropy
°C	m³/kg	kJ/kg	kJ/kg.K	m³/kg	kJ/kg	kJ/kg.K
	p = 1.90 MPa (Tsat = 49.03 °C)			p = 2.00 MPa (T_{sat} = 51.27 °C)		
T	*v*	*h*	*s*	*v*	*h*	*s*
Sat.	0.0119	417.4	1.687	0.0113	417.5	1.683
50	0.0120	418.4	1.690	-	-	-
55	0.0126	423.7	1.706	0.0116	421.6	1.696
60	0.0130	428.7	1.721	0.0121	426.9	1.712
65	0.0135	433.6	1.736	0.0126	431.9	1.727
70	0.0139	438.3	1.750	0.0130	436.8	1.741
75	0.0143	442.9	1.763	0.0134	441.5	1.755
80	0.0147	447.5	1.776	0.0138	446.1	1.768
85	0.0151	451.9	1.788	0.0142	450.7	1.781
90	0.0155	456.4	1.801	0.0146	455.2	1.793
95	0.0159	460.8	1.813	0.0149	459.6	1.806
100	0.0162	465.1	1.825	0.0153	464.1	1.817
105	0.0166	469.5	1.836	0.0156	468.5	1.829
110	0.0170	473.8	1.847	0.0160	472.8	1.841
115	0.0173	478.1	1.859	0.0163	477.2	1.852
120	0.0176	482.4	1.870	0.0166	481.5	1.863
125	0.0180	486.7	1.880	0.0170	485.8	1.874
130	0.0183	490.9	1.891	0.0173	490.1	1.885
135	0.0186	495.2	1.902	0.0176	494.4	1.895
140	0.0190	499.5	1.912	0.0179	498.7	1.906
145	0.0193	503.8	1.922	0.0182	503.0	1.916
150	0.0196	508.1	1.932	0.0185	507.3	1.926
155	0.0199	512.3	1.943	0.0188	511.7	1.936
160	0.0202	516.6	1.953	0.0191	516.0	1.946
165	0.0205	520.9	1.962	0.0194	520.3	1.956
170	0.0208	525.2	1.972	0.0197	524.6	1.966
175	0.0211	529.6	1.982	0.0200	529.0	1.976
180	0.0214	533.9	1.991	0.0203	533.3	1.986
185	0.0217	538.2	2.001	0.0206	537.7	1.995
190	0.0220	542.6	2.010	0.0209	542.0	2.005
195	0.0223	546.9	2.020	0.0211	546.6	2.014
200	0.0226	551.3	2.029	0.0214	550.8	2.023

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