Department of Mechanical and Aerospace Engineering

Decarbonising the Citadel: River Source Heat Pump Feasibility Study

Author: Nicholas Plunkett

Supervisor: Dr Nick Kelly

A thesis submitted in partial fulfilment for the requirement of degree in
Master of Science in Sustainable Engineering: Renewable Energy Systems & the Environment

2024

Copyright Declaration

Signed: *Nicholas Plunkett* Date: 07/08/2024

Abstract

The aim of this project is to assess the feasibility of using a river source heat pump to provide heating for the Citadel leisure centre in Ayr.

Currently, the heating of buidlings represents a large portion of overall greenhouse gas emissions, with the combustion of natural gas remaining the primary means of energy consumption. Recent developments in national policy have placed a duty on Scottish local authorities to deliver heat and energy efficiency plans relating to domestic and public buildings. South Ayrhsire Council have an ambition to decarbonise public buildings within their estate and have identified a river source heat pump as a potential solution for the Citadel.

The methodology used to assess the feasibility of implementing this technology takes into account the existing mechanical systems at the Citadel, the characteristics of the waterbody source, the River Ayr, and the complex heating loads unique to a facility which includes an indoor swimming pool. A detailed investigation of the energy demands of the building is carried out by conducting a site visit to examine the existing pool water and air heating systems. Additionally, technical documentation, monitored demand data and historical weather data is used to create a numerical model of the heat losses exhibited by the swimming pool area. The demand is characterised and an overall heat pump thermal capacity is determined and the amount of water required to be abstracted from the waterbody is calculated. Three design options for the heat pump system are presented and compared in terms of decarbonisation potential, cost of energy and reduction in overall energy usage. Potential sites are identified for the necessary heat pump plant equipment.

The swimming pool area was found to make up a large proportion of the overall site demand, with pool occupancy presenting a large effect on evaporative heat loss. The large demand of the pool area was found to be well served by a lower flow temperature than the rest of the building, with a mix of low and high temperature heat pumps providing the greatest efficiency. Retaining the existing gas boilers for conventional space heating was found to provide significant reductions in CO_2 emissions and a reasonable reduction in annual energy costs.

Acknowledgements

Firstly, I'd like to thank Dr Nick Kelly for his support and advice throughout the project. I'd also like to thank Alan Roseweir for his facilitation of contact with parties involved with the Citadel and his assistance and enthusiasm for the project. I'd also like to thank Pauline Bradley, David Thornton and Bob from South Ayrshire Council for sharing valuable information relevant to the site, and for allowing me to come and ask questions and explore the plant rooms of the Citadel. I'd also like to thank William Balloch for sharing his expertise in heat recovery ventilation systems.

I'd also like to thank all of the staff and fellow students who helped to make the past year such an enjoyable and challenging experience.

Finally, I'd like to thank my girlfriend, Zoë. This past year would have been more difficult without her continued emotional support and inspiration.

Table of Contents

1: Introduction....................11
1.1: Aim....................12
1.2: Objectives....................12
1.3: Overview....................13
2: The Citadel....................14
2.1: Heating System....................14
2.2: Pool Water Heating....................14
2.3: Pool Air Ventilation System....................16
3: Swimming Pool Operations....................17
3.1: Energy Use....................17
3.2: Pool Water Heating....................17
3.3: Pool Air Heating & Ventilation....................18
4: Heat Pump....................19
4.1: Thermodynamics....................19
4.2: Performance....................21
4.3: Heat Sources & Applications....................22
4.3.1: Air Source Heat Pump (ASHP)....................22
4.3.2: Ground Source Heat Pump (GSHP)....................24
4.3.3: Water Source Heat Pump (WSHP)....................25
4.3.4: Closed Loop System....................25
4.3.5: Open Loop System....................26
4.4: Environmental Considerations....................27
4.5: Case Studies....................28
4.5.1: Queens Quay, Clydebank....................28
4.5.2: Crystal Leisure Centre, Dudley....................29
5: Methodology....................30
5.1: Demand Split....................30
5.1.1: Demand Profile....................30
5.1.2: Daily Profile Analysis....................30
5.1.3: Citadel Zones....................30
5.1.4: Pool Zone Details....................31
5.1.5: Pool Zone Losses....................32
5.2: Pool Air Heat Losses....................33
5.2.1: Ventilation Losses....................34
5.2.2: Heat Recovery: LMTD Method....................34
5.2.3: Heat Recovery: Effectiveness Method....................36
5.2.4: Exhaust Air Recirculation....................37
5.2.5: Fabric Losses....................40
5.3: Pool Water Heat Losses....................40
5.3.1: Evaporative Losses....................41
5.3.2: Convective Losses....................43
5.3.3: Conductive Losses....................43
5.3.4: Radiative Losses....................43

5.3.5: Refilling Losses....43
5.4: Final Demand Breakdown....44
5.4.1: Comparison of Pool Water Heating and Pool Air Heating....44
5.5: River Characteristics....44
5.5.1: Location....44
5.5.2: River Temperature....44
5.5.3: Flow Rate....45
5.5.4: Water Abstraction....47
5.6: Site Selection....47
5.6.1: Site Survey....47
5.6.2: Geographical Analysis....47
5.7: Heat Pump....49
5.7.1: Flow Temperature....49
5.7.2: Thermal Capacity....50
5.7.3: Design Option....50
5.7.4: Heat Pump Performance....51
5.8: Carbon Emissions....52
5.9: Energy Cost....52
6: Results....53
6.1: Citadel Demand Profile....53
6.2: Winter Scenario....53
6.3: Pool Air Losses....55
6.3.1: Heat Exchange....56
6.3.2: Air Mixing....58
6.3.3: Ventilation Losses....61
6.3.4: Fabric Losses....62
6.3.5: Total Pool Air Losses....63
6.4: Pool Water Losses....63
6.4.1: Evaporation Losses....63
6.4.2: Radiation....64
6.4.3: Refill Losses....65
6.4.4: Total Pool Water Losses....65
6.5: Demand Split....66
6.6: Heat Pump Sizing....66
6.7: River Water Abstraction Level....67
6.8: Heat Pump Configuration....67
6.8.1: Design Option 1 (DO1)....68
6.8.2: Design Option 2 (DO2)....68
6.8.3: Design Option 3 (DO3)....69
6.9: Heat Pump Performance....70
6.10: Energy Cost....72
6.11: Carbon Emissions....72
6.12: System Comparison....73
7: Conclusions....75
7.1: Energy in River....75

7.2: Demand Split & Design Options........75
7.3: Suitability of Site........75
7.4: Heat Pump Performance........76
7.5: Future Work........76
    7.5.1: Localised River Monitoring........76
    7.5.2: Fabric-First Approach........76
    7.5.3: Heat Pump Detailed Design........77
    7.5.4: District Heating Applications........77

Table of Figures

Figure 1: Main pool water pumps (Photograph taken on site visit in July 2024)....................15
Figure 2: Plate heat exchanger for the diving pool (Photograph taken on site visit in July 2024)...15
Figure 3: Rendering of Dantherm DanX XKS air handling unit (Dantherm)..........................16
Figure 4: Schematic of AHU with heat recovery showing air flows and main components (Dantherm)..16
Figure 5: Schematic of gas powered pool water heating system (Li, et al, 2020)...................17
Figure 6: Ideal vapour compression cycle with T-s diagram (Tuohy, 2014)............................20
Figure 7: Vapour compression cycle with intermediate heat exchange and p-h Diagram (Tuohy, 2014)...20
Figure 8: Schematic of an air source heat pump system in a domestic setting (energy.nl)......22
Figure 9: Typical ASHP performance with varying ambient conditions (fluidmechanics.co.uk) 23
Figure 10: Schematic of a ground source heat pump system in a domestic setting (greenmatch.co.uk)...24
Figure 11: Schematic of a water source heat pump with intermediate closed loop (Kensa Heat Pumps)..25
Figure 12: Abstraction equipment for an open loop water source heat pump (ACR Journal). 26
Figure 13: Heat pump plant room at Queen's Quay, Clydebank (Star Refrigeration).............28
Figure 14: Heat pump plant room at Crystal Leisure Centre, Dudley (Cotswold Energy)......29
Figure 15: Site plan drawing of Citadel at pool hall level with pool areas highlighted (South Ayrshire Council)...31
Figure 16: Simplified isometric view of the pool hall area with boundaries...........................32
Figure 17: Illustration of heat losses within an indoor pool area..33
Figure 18: Diagram illustrating the exchange of heat between two fluids in counterflow (Tuohy, 2008)...35
Figure 19: Effect of heating on the moisture content of air, plotted on a psychrometric chart 37
Figure 20: Properties of mixed air, plotted on a psycrhometric chart (CIBSE).......................39
Figure 21: Average surface sea temperature for Clyde Estuary between 1997 and 2012 (Marine Scotland)..45
Figure 22: Location of Mainholm monitoring station, relative to proposed water intake location (OpenStreetMap)... 46
Figure 23: Measured flow data for River Ayr at Mainholm (UK National River Flow Archive) 46
Figure 24: Citadel location showing adjacent land uses..48
Figure 25: Unoccupied space for pipework routing..49
Figure 26: Annual gas demand profile for Citadel 2023/2024...53
Figure 27: Demand profile for 17/10/23...54
Figure 28: Citadel AHU nameplate...55
Figure 29: Existing heat exchanger data sheet...56
Figure 30: Side elevation section view of Citadel AHU...56
Figure 31: Plot of air streams on a psychrometric chart...59
Figure 32: Plot of mixed air stream on a psychrometric chart..60
Figure 33: Evaporative heat loss and occupancy rate...64

Figure 34: Demand split of the building....66
Figure 35: Design Option 1 schematic....68
Figure 36: Design Option 2 schematic....69
Figure 37: Design Option 3 schematic....70
Figure 38: Seasonal COP for design options....71
Figure 39: System comparison....74

Index of Tables

Table 1: Citadel opening hours....32
Table 2: Latent heat of vapourisation in recommended pool temperature range....41
Table 3: Occupancy rate and number of swimmers for the Citadel pool area....42
Table 4: Humidity ratios at saturation for recommended water and air temperatures....42
Table 5: Air properties at target time (from Weather Underground)....54
Table 6: Air properties at winter scenario conditions....57
Table 7: Air properties of fresh and exhaust air streams....58
Table 8: U-values and surface area of construction types in the pool zone....63
Table 9: Summary of heat loss from the pool air....63
Table 10: Summary of heat loss from the pool water....65
Table 11: Energy use for each design option....72
Table 12: Annual energy cost (based on 2023 prices)....72
Table 13: Annual carbon emissions (based on 2024 conversion factors)....73
Table 14: Design option advantages and disadvantages....75

1: Introduction

In May 2022, the Scottish Government passed the Local Heat and Energy Efficiency Strategies (Scotland) Order, a policy framework to drive the development of decarbonised heat in Scotland. This is to be achieved by placing a statutory duty on Scottish local authorities to develop and maintain a Local Heat and Energy Efficiency Strategy (LHEES), with the goal of decarbonising domestic and public properties (Scottish Government, 2022).

South Ayrshire Council (SAC) is in the process of developing its own LHEES, and are working in collaboration with East and North Ayrshire Councils to develop a regional-scale Ayrshire Energy Masterplan. SAC has an ambition to transition energy systems to net-zero and recognises the socio-economic impacts possible from such a transition. SAC have over 100 buildings in their estate that use gas as fuel for heating, consuming over 36,000 MWh of energy per year (Ricardo, 2023)

The Citadel leisure centre in Ayr is currently undergoing a £10 million redevelopment. This includes new leisure facilities and repairs to the building, with some fabric improvements to make the building wind and watertight. In line with the council's ambition to decarbonise heating supply within its own estate, several options are being assessed to achieve this at the Citadel.

1.1: Aim

Large scale river source heat pump systems have been successfully employed in Scotland and overseas, harnessing the energy in flowing water bodies to deliver clean heat to homes and businesses. The Citadel's proximity to the River Ayr makes this an attractive technology. This report will aim to assess the feasibility of installing such a system, considering the existing mechanical systems within the building, the local environmental conditions, and the operational reality of running a busy leisure centre.

1.2: Objectives

The overall aim of this project will be achieved by completing the following objectives:

- Characterise the demand of the building by using a numerical model to calculate the multiple heat losses from the pool area at peak demand.
- Select three heat pump design options to meet the demand of the building, taking into account the character of the demand.
- Create a seasonal COP model to compare the performance of each heat pump design option.
- Determine the energy available in the River Ayr and calculate the amount of water which is required to be abstracted to meet the energy demand.
- Calculate the cost of energy for each design option based on wholesale energy prices.
- Calculate the reduction in CO_2 emissions for each design option based on up-to-date conversion factors.

1.3: Overview

This report will assess the feasibility of providing heat and hot water to the facility using a RSHP. Section 2 of this report will review the Citadel, describing its existing facilities and heating systems. Section 3 is a review of literature relating to the common practice of indoor swimming pool operation, discussing the heating requirements, ventilation requirements and temperatures required to maintain an acceptable level of thermal comfort and minimise heat loss. Section 4 introduces heat pump technology, describing the thermodynamic processes through which they deliver heat, methods used to determine performance, the various types of heat sources and applications of heat pumps. This is followed by a brief case study section describing heat pumps installed at Queen's Quay in Clydebank and the Crystal Leisure Centre in Dudley, West Midlands.

The methodology section describes the steps taken to assess the feasibility of installing a river source heat pump system to serve the Citadel. The methods used are employed to achieve the main objectives of this report as outlined in the previous section. The feasibility is explored through the analysis of the site demand, the characteristics of the River Ayr which will be the heat source, the suitability of the site for the construction of a RSHP system and the thermal, economic and environmental performance of three design option scenarios.

Section 6 presents the findings gained from application of the methodology using relevant data relating to the site, and section 7 presents the conclusions and opportunities for future research.

2: The Citadel

The Citadel Leisure Centre in Ayr was constructed between 1969 and 1972 and is one of several leisure centre facilities owned and operated by South Ayrshire Council (SAC). The pool itself dates from the original construction, while additional facilities were added in the 1990's. Today the extension includes a gym, a youth club and a salon.

Various proposals for a replacement facility have been mooted in recent years, however SAC are now committed to refurbishment of the existing venue, including an upgrade of gym facilities and improvements to the building fabric to minimise outdoor air and water leakage. In line with plans to decarbonise heating at facilities within their estate, SAC are currently in the process of exploring different pathways through which this could be achieved at the Citadel.

Ultimately, the Citadel is a vital community space and is well utilised seven days a week, year-round. Beyond regular opening hours, the pool is used for swimming clubs, diving clubs and scuba training. Many in the local community depend on its reliable operation, and so it is important that any future upgrade or replacement of mechanical and electrical equipment allow the facility to operate as normal.

2.1: Heating System

The building is served by six 400 kW capacity gas boilers, giving a combined capacity of 2.4 MW. Water is heated to 80°C and distributed through the building to provide space heating, hot water, pool water heating and pool air heating. The capacity far exceeds the demand of the building. This is intended for redundancy, allowing the facility to operate if a boiler is needed to be shut down, and provides the ability to quickly ramp up supply if required.

2.2: Pool Water Heating

The pool water is heated by a conventional plate heat exchanger system which dates from the original construction of the building. The water from the pool is continually circulated in a closed loop to satisfy cleaning requirements, and as this process occurs, the water is maintained at the appropriate temperature by passing through a plate heat exchanger where it interfaces with the hot water provided by the gas boiler heating system.

Much of the existing equipment in the pool plant room is legacy equipment, and operators have expressed difficulty in measuring exact flow rates for water recirculation. Much of the pipework is of unknown specification, and a number of the pumps used to circulate the pool are showing signs of wear.

Figure 1: Main pool water pumps (Photograph taken on site visit in July 2024)

Figure 2: Plate heat exchanger for the diving pool (Photograph taken on site visit in July 2024)

2.3: Pool Air Ventilation System

The existing pool air ventilation system is a relatively new addition to the building. The unit, manufactured by Dantherm, controls humidity and temperature within the pool hall, allowing for the mixing of outdoor air with recirculated indoor air to maintain thermal comfort for pool users. The unit also makes use of a crossflow heat exchanger to increase efficiency by taking advantage of the warm exhaust air from the pool hall. Operators have noted that the new system as resulted in a 20 to 30% decrease in energy demand for ventilation and heating. Pool air ventilation systems will be further explored in section 3.

Figure 3: Rendering of Dantherm DanX XKS air handling unit (Dantherm)

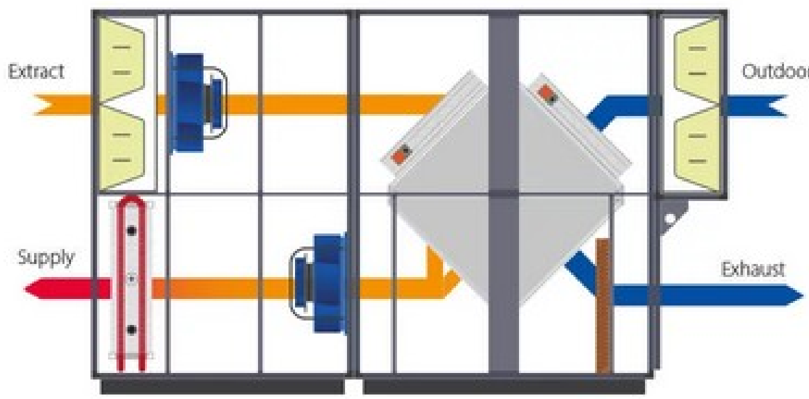


Figure 4: Schematic of AHU with heat recovery showing air flows and main components (Dantherm)

3: Swimming Pool Operations

3.1: Energy Use

Indoor swimming pools use a large amount of energy when compared to other types of building. According to the Carbon Trust Swimming Pool Guide, this can be as much as five times greater per m^2 of floor area. High temperatures must be maintained in the pool water and air to provide sufficient thermal comfort, a large volume of fresh air must be continuously supplied to the zone, and operational hours are typically greater than that of a conventional public or commercial building. The two separate processes of pool water and pool air heating contribute to high energy demand.

3.2: Pool Water Heating

Pool water must be continually heated during operational times to maintain thermal comfort for any occupants and replace heat lost by water evaporation and other mechanisms. The PWTAG Code of Practice, 2019, gives a recommended temperature range of 27-29°C for leisure pools.

A conventional gas heating system makes use of a heat exchanger to transfer heat from the heating system's distribution network to the circulating pool water. Figure 5 shows a simplified illustration of such a system. This means that, much like a conventional space heating system, hot water supplied by a heat pump may be employed to replace the gas heated water. Depending on the flow temperature from the heat pump, heat exchanger design must be carefully considered.

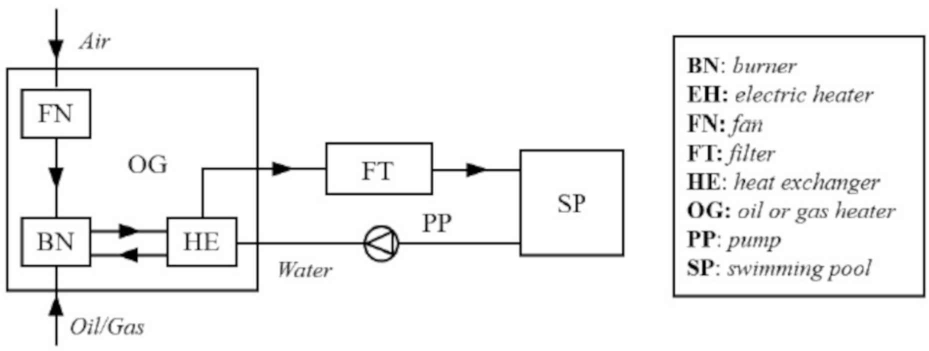


Figure 5: Schematic of gas powered pool water heating system (Li, et al, 2020)

To reduce heat exchanger size, it is common practice to heat a portion of the recirculated water to a higher temperature than the target, which is then mixed back into the recirculated water to achieve a mixed stream at the required temperature (Tim Smith Heat Exchangers). In other cases, as is the case with the Citadel, the entire volume of the pool is recirculated through the heating system post-cleaning, maintaining the desired temperature.

A simple tube heat exchanger may be used if the flow temperature is high, however a plate type heat exchanger achieves a higher efficiency by increasing the contact area between the hot and cold fluids and is therefore more suited to lower flow temperatures from the distribution system.

3.3: Pool Air Heating & Ventilation

The PWTAG Code of Practice, 2019, recommends that pool hall air temperatures must be maintained at 1°C above pool water temperature. This is to maintain a reasonable level of occupant thermal comfort, as pool users will experience cooling from evaporation when at the pool side. Maintaining hall temperature close to water temperature also helps to minimise pool water evaporation (Carbon Trust Swimming Pool Guide, 2008).

In addition to space heating, humidity control is another important factor in the operation of an indoor pool. Relative humidity must be maintained at less than 100% to avoid condensation forming on cold surfaces (Carbon Trust Swimming Pool Guide, 2008), and this is achieved either by diluting the pool hall air with outdoor air by using a push-pull type system, or by directly dehumidifying the air by a refrigerant based dehumidifier (Gonzalez Miguel, 2014).

The type of humidity control used has implications for the space heating of the zone. In the case of a push-pull type system, outdoor air must be heated before it is inserted into the pool hall. In some cases, this is achieved by using a heating coil, while other systems will make use of an air-to-air heat exchanger to recover some of the heat from the extracted moist air inside the pool hall, resulting in a comparative reduction in space heating energy demand.

4: Heat Pump

While the UK has made great progress in decarbonising it's electricity generation, heating is still mainly provided by combusting natural gas. In 2016, over 40% of the UKs domestic and non-domestic total energy demand came from heating (Wang, et al, 2020), while 63% of non-domestic properties are heated by gas (UK Government Heat and Buildings Strategy). With the UK Government target of net-zero by 2050, there will have to be a fundamental change in the way properties are heated.

Heat pumps have emerged as the most promising technology to achieve heating decarbonisation. They are powered by electricity, and therefore do not produce CO_2 emissions at the point of use, and as power generation is increasingly decarbonised, they have the potential to be a net-zero emission way of providing heat (National Grid ESO Carbon Intensity Dashboard).

Compared with gas boiler and direct electric heating, heat pumps are also vastly more efficient, producing multiple units of thermal energy from one unit of electrical input.

4.1: Thermodynamics

A simple heat pump consists of four main components: an evaporator, a condenser, a compressor and an expander. A working fluid, or refrigerant, is cycled through the system.

In the evaporator, the refrigerant absorbs latent heat from the cold source. This is achieved by supplying the refrigerant to the evaporator and a lower temperature than the ambient source. At this point, the refrigerant changes phase from saturated liquid to saturated vapour.

The refrigerant is then compressed to high pressure and fed to the condenser as a superheated vapour. The energy input to the compressor is electrical work.

At the condenser, the heat absorbed at the evaporation stage flows into the warm environment, condensing back to a saturated liquid. The refrigerant is then cycled back to the evaporator by an expansion where the cycle can be repeated.

This is illustrated in the following figures, showing how these basic components are configured together, and a T-s diagram of an ideal vapour-compression cycle..

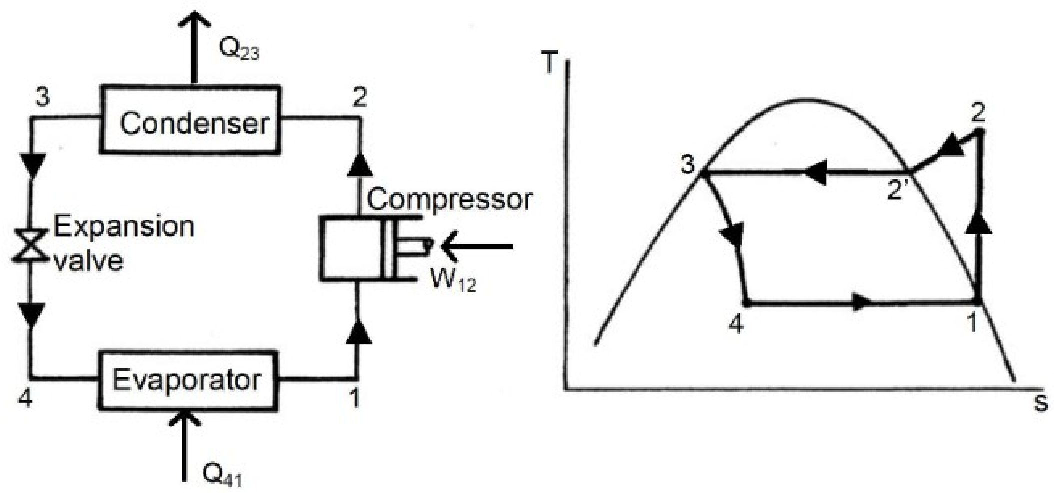


Figure 6: Ideal vapour compression cycle with T-s diagram (Tuohy, 2014)

In reality, the compression is irreversible due to fluid friction, and so the condenser is required to do extra work. The liquid leaving the compressor should also be subcooled to increase the refrigeration effect, and to ensure that there is no damage to the expansion valve from vapour left over in the liquid from the condenser. A more realistic configuration is shown in figure 7. This incorporates an intermediate heat exchanger, subcooling the liquid from the condenser, while raising the temperature of the vapour from the evaporator to ensure that the compressor is receiving dry vapour.

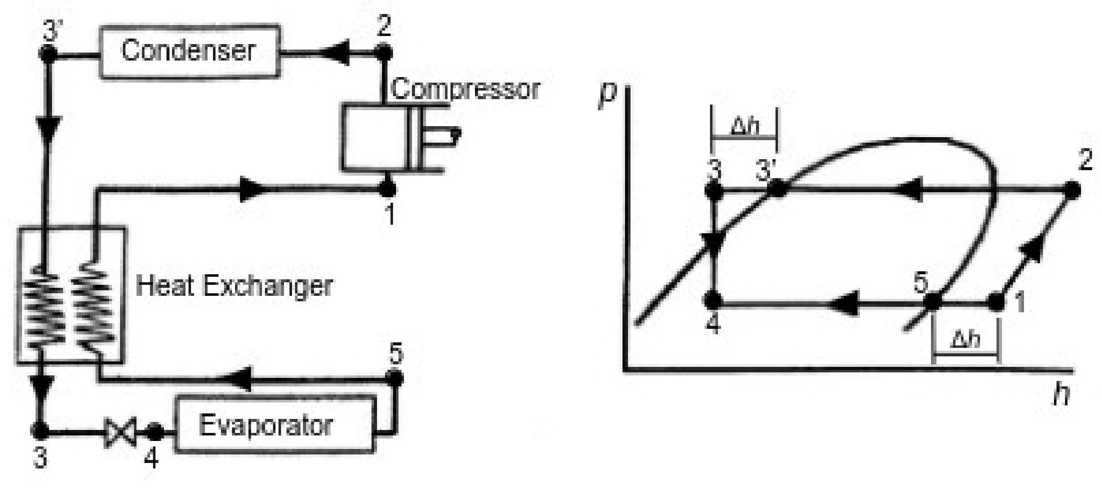


Figure 7: Vapour compression cycle with intermediate heat exchange and p-h Diagram (Tuohy, 2014)

4.2: Performance

As discussed, one of the most attractive features of a heat pump is it's high efficiency, with multiple units of thermal output gained from one unit of work input. Heat pump performance is calculated by relating the thermal energy output from the condenser to the work input to the compressor. This is the Coefficient of Performance (COP), and can be calculated from equation:

$$COP_{HP}=\left|\frac{Q_2}{W}\right|=1+\left|\frac{Q_1}{W}\right|$$

Q_2 represents the heat output from the condenser to the target, W represents the work input to the compressor, and Q_1 represents the heat absorbed in the evaporator.

For practical heating systems, a specific value of Q_2 must be achieved depending on the application. The amount of heat which can be absorbed, Q_1, can be subject to variations depending on the type of source used. Air source heat pumps, for example, experience a substantial variation in ambient temperature on both an hourly and seasonal scale. This has an effect on performance, as the amount of work input, W, required to achieve a sufficient amount of heat to the warm space, Q_2, will vary depending on the ambient temperature. So, for heat pump systems which rely on a constantly changing ambient source, COP will also vary. Likewise, a minimal temperature difference between evaporating and condensing temperatures requires less work input to the compressor to achieve the desired Q_2. This increases COP, and is illustrated in the equation:

$$W=\dot{m}\cdot C_p\cdot\Delta T$$

Where $\dot{m}$ is the mass flow of the refrigerant, C_p is the specific heat capacity of the refrigerant, and ΔT is the temperature difference between the evaporating and condensing temperature.

The maximum theoretical COP of a heat pump, known as the Carnot efficiency, can be easily determined if the absolute temperature of the source and supply are known. This is described in the following equation:

$$COP_{Carnot}=\frac{T_{hot}}{(T_{hot}-T_{cold})}$$

So, the ambient source used for the heat pump has a large effect on it's performance over time, and the type of heat pump used for any particular application should be well considered.

4.3: Heat Sources & Applications

Heat pumps can use a variety of different heat sources, as it is possible to absorb latent heat at relatively low temperatures. Air source heat pumps are a common type of installation for domestic properties, with thermal capacities ranging from 4 kW to 12 kW. Wu (2009) identifies a second type of heat pump, the ground source heat pump (GSHP). A GSHP encompasses many different types of heat source within its classification. Water source heat pumps are sometimes classified as a separate type of heat pump. In practice, the type of heat pump used and the type of heat source to be exploited depends on the demand to be met, the climate and the physical environment accessible around the installation.

4.3.1: Air Source Heat Pump (ASHP)

Air source heat pumps absorb heat from outdoor air by forcing outdoor air over an evaporator. At the condensing side, the heat is transferred to the heating distribution pipework, to be used in underfloor heating or radiators (Carroll, Chesser, Lyons, 2020).

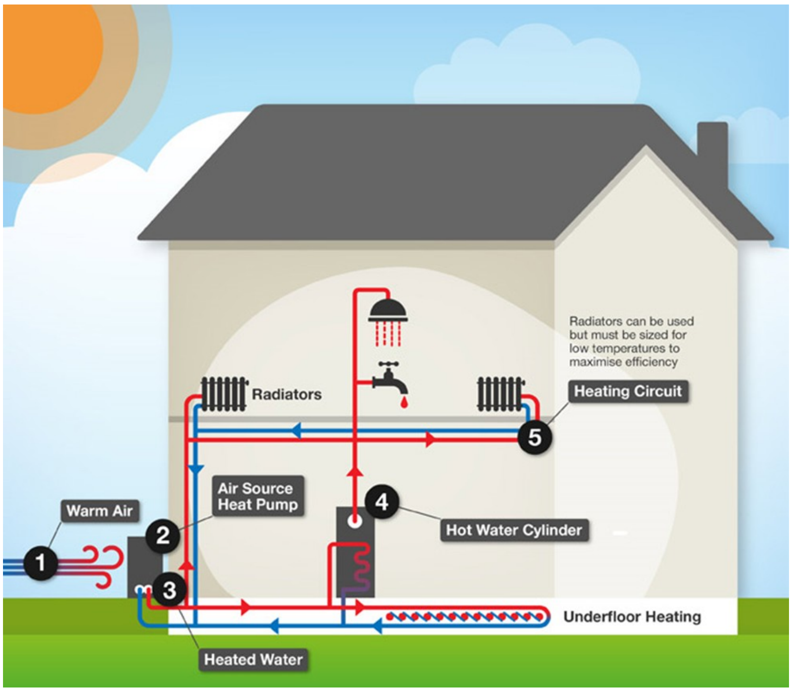


Figure 8: Schematic of an air source heat pump system in a domestic setting (energy.nl)

ASHPs experience a high variation in ambient temperature, and as such their performance is constantly changing on a daily and seasonal basis. During the winter, it may be necessary for the heat pump to defrost which further impacts performance, particularly when humidity is high (ASHRAE HVAC Systems & Equipment, 2020)

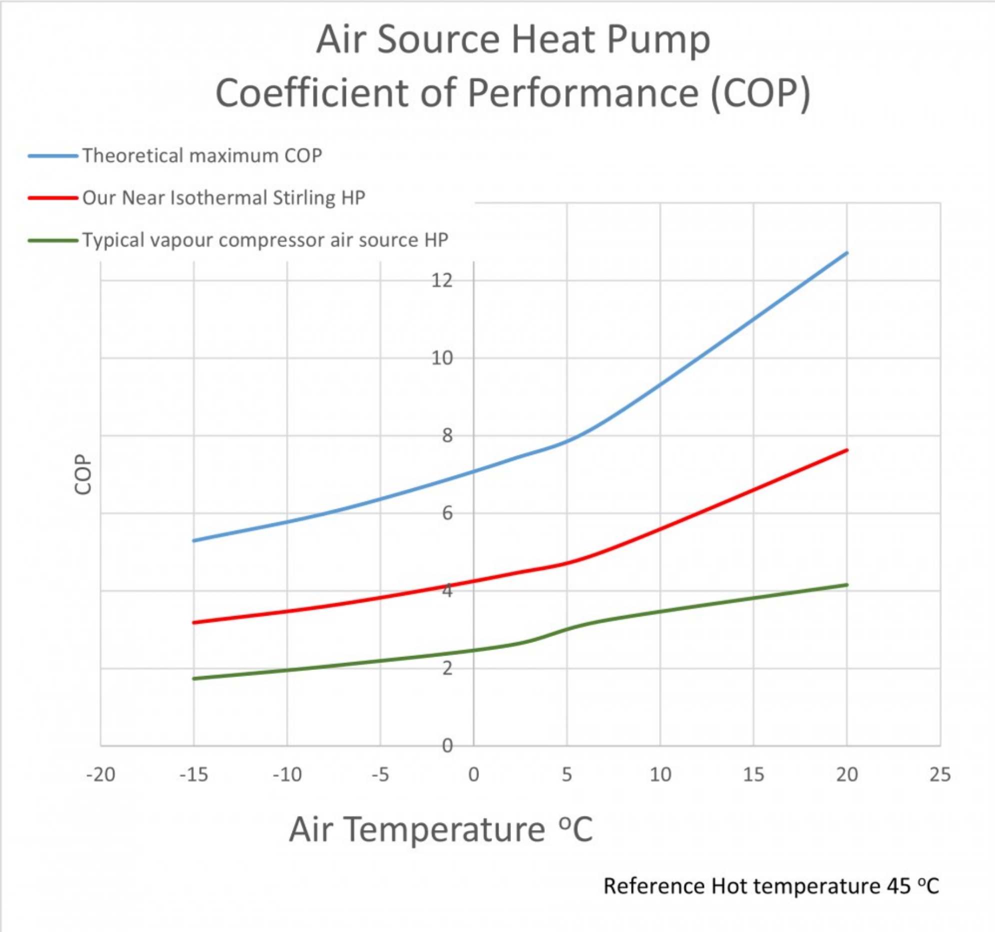


Figure 9: Typical ASHP performance with varying ambient conditions (fluidmechanics.co.uk)

ASHPs are commonly used in residential and light commercial settings (ASHRAE), due to the fact that air is an easily accessible source of heat for the majority of properties. ASHP equipment can be installed easily compared to GSHP systems, as they can be installed close to the desired heating zone, limiting the need for extensive pipework or extensive groundworks. Many off-the-shelf systems are available on the market. A residential property in the UK may commonly install an ASHP with thermal capacity of 4 kW to 12 kW depending on the size of the property (heat-pumps.org.uk).

4.3.2: Ground Source Heat Pump (GSHP)

Ground source heat pumps encompass three distinct heat sources, as categorised by ASHRAE. This classification includes heat pumps which make use of ground water, surface water and directly connected to the ground by way of a heat exchanger (ASHRAE HVAC Systems & Equipment, 2020).

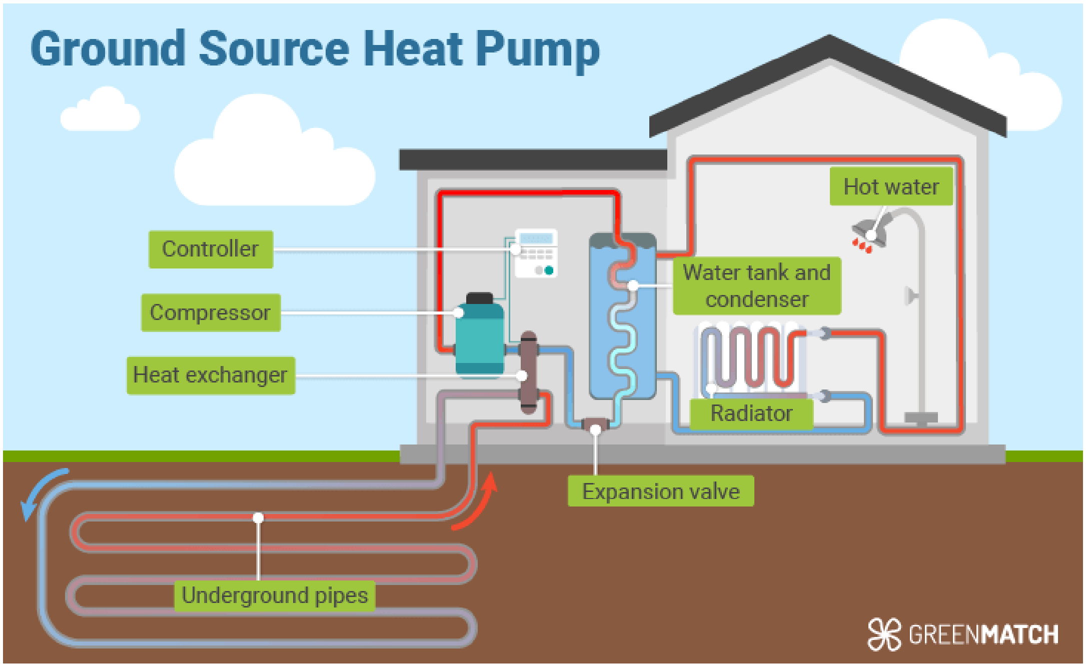


Figure 10: Schematic of a ground source heat pump system in a domestic setting (greenmatch.co.uk)

An advantage of GSHP systems is that the heat source is subject to much less variation in seasonal temperature changes when compared to ASHP systems (Schibuola, Scarpa, 2015), however the type of source used is greatly dependent on the location of the site and characteristics of the source.

GSHPs are often bespoke systems, designed to suit a particular location or source type, and so are typically employed in larger multi-user or district type schemes. Unlike an ASHP where the main costs are in the heat pump equipment itself, GSHP systems have additional costs of necessary groundworks to reach the heat source and extensive piping networks to connect the source with the end user (Kensa Heat Pumps).

4.3.3: Water Source Heat Pump (WSHP)

Where a site is close to a body of water such as a lake, river or sea, surface water may be an attractive heat source. Sea water, in the case of the Citadel leisure centre, which is located near a tidal river mouth, is a promising source due to its lower freezing temperature and high thermal inertia (Schibuola, Scarpa, 2015).

4.3.4: Closed Loop System

A closed loop system makes use of an intermediate heat exchange, with a loop of liquid submerged in the water body. Heat from the surface water is absorbed in the loop and transferred to the evaporator where the second heat transfer takes place. Closed loop systems may be constructed in a horizontal, vertical or flat arrangement, if the source is shallow, within the water body (Wu, 2009).

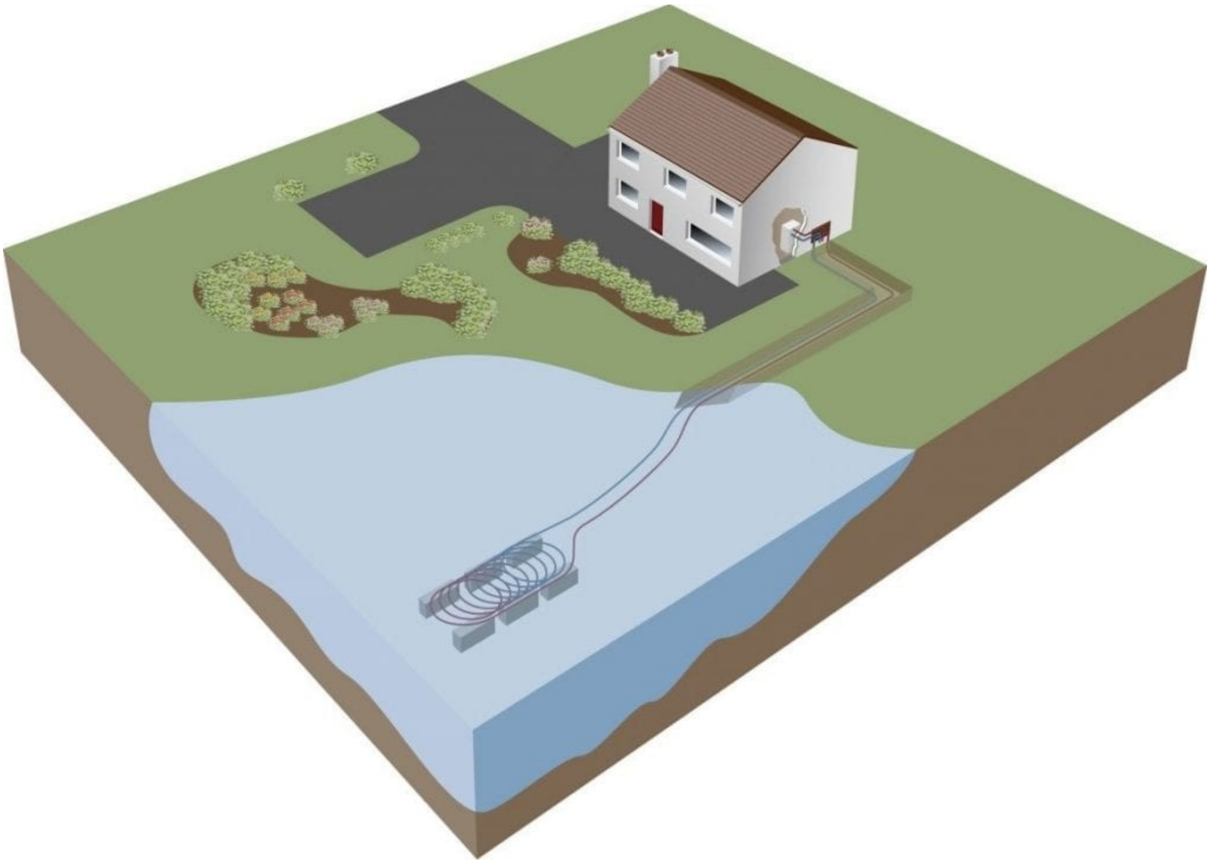


Figure 11: Schematic of a water source heat pump with intermediate closed loop (Kensa Heat Pumps)

4.3.5: Open Loop System

In an open loop system, surface water flows directly over the evaporator in a direct heat exchange between the ambient temperature of the surface water and the refrigerant. This type of system is widely used, however may be limited by local environmental regulation with regards to water abstraction (Wu, 2009). Open loop systems are more efficient than closed loop systems as there is no intermediate heat exchange required.

Figure 12: Abstraction equipment for an open loop water source heat pump (ACR Journal)

Open loop systems require a number of components to bring water to the heat pump and to protect equipment and the environment. Water is pumped from the water body to the heat pump, and then returned to the source. The system must also be designed to prevent the entry of debris, wildlife and other biological organisms. This is achieved by incorporating filtration or screening to the water intake system (Mitchell, Spitler, 2013).

For systems using seawater, there is a risk of corrosion and fouling within the heat exchanger piping, which may degrade performance over time. Titanium is considered to be the most suitable material to use for this application. Copper alloys can also be suitable, but have a limited lifespan if local flow velocities exceed 1-3 m/s. Biological fouling of the inner surfaces of the heat exchanger pipework also poses a risk to overall system performance. This may include algae or other biological organisms such as molluscs, which can increase thermal resistance, and so adequate biocide or cleaning measures must be implemented (Mitchell, Spitler, 2013).

4.4: Environmental Considerations

While the Scottish Government are supportive of the installation of WSHPs, as demonstrated by its Water Source Heat Pump Heat Maps (WSHPHM), the construction of such a system must be consented to by the Scottish Environmental Protection Agency (SEPA) through a CAR authorisation. A WSHP installation will involve the abstraction of water from the source, construction of necessary infrastructure, and cause changes to the temperature of the receiving water body (SEPA Guidance for WSHPs). The level of water abstracted by any installation will affect the level of consent required from SEPA.

For water discharged back into the water body, adjacent to the point of abstraction, SEPA guidance considers there to be no significant thermal impact if the water is within 3°C of the ambient temperature of the source.

4.5: Case Studies

4.5.1: Queens Quay, Clydebank

Scotland's largest WSHP system is located in Clydebank, with two large 2.6 MW_{th} heat pumps servicing a district heating network. The system also incorporates a 15 MW gas boiler which can be operated when the district is at peak demand, which occurs for 25 hours each year (Star Refrigeration).

Figure 13: Heat pump plant room at Queen's Quay, Clydebank (Star Refrigeration)

4.5.2: Crystal Leisure Centre, Dudley

As part of a renovation, the Crystal Leisure Centre in Dudley, West Midlands, had a closed loop WSHP system, drawing ground water from boreholes, installed in 2020. Four 60 kW_{th} heat pumps, combined to give a total capacity of 240 kW_{th}, were installed to supply water at 50°C to a 2000 L buffer tank. Since installation, the local authority are expected to save on the running costs of the centre and receive subsidy through the Non-Domestic Renewable Heat Incentive to cover the capital cost of the system (Cotswold Energy Website, Dudley Metropolitan Borough Council).

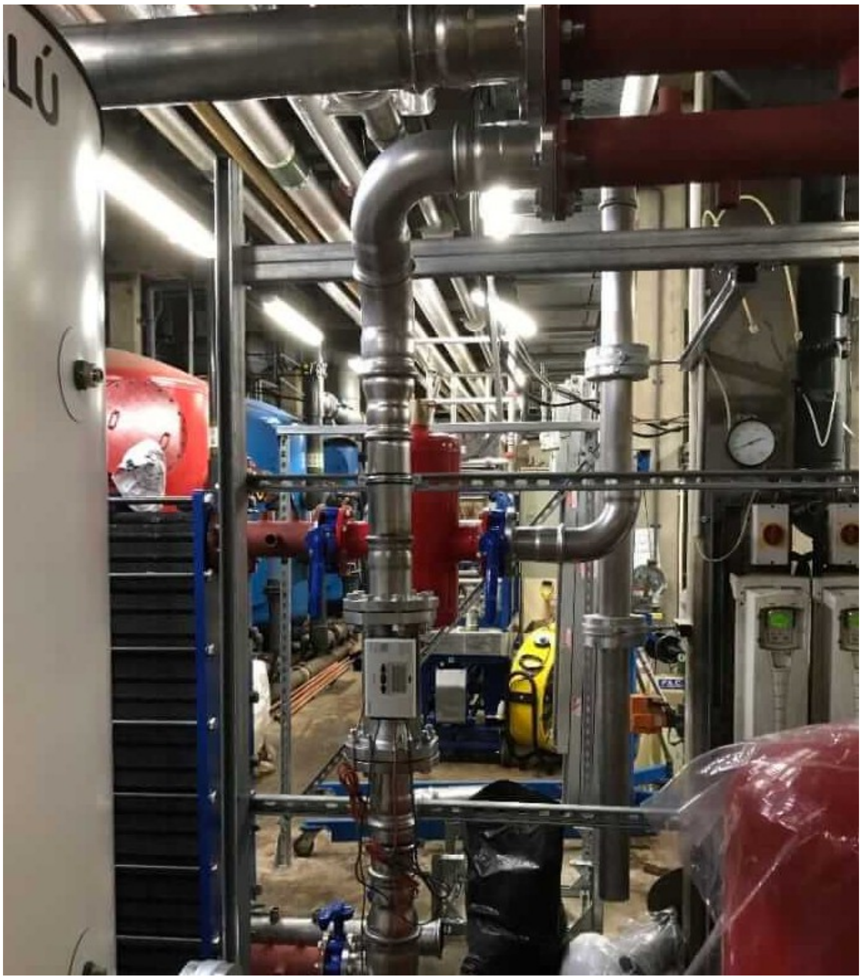


Figure 14: Heat pump plant room at Crystal Leisure Centre, Dudley (Cotswold Energy)

5: Methodology

This report will now set out the methods required to determine the feasibility of using a river source heat pump in the context of the Citadel leisure centre. The methods outlined in this section will be used to analyse the characteristics of the site's demand, the behaviour of the river which will be used as a heat source, the suitability of adjacent land for the construction of equipment and the criteria for the type of heat pump to be employed.

5.1: Demand Split

Indoor swimming pools are characterised by high heating loads as a result of water evaporation and fresh air ventilation; therefore, it is necessary to better understand how the demands of the indoor pool area fit within the overall demands of the building.

5.1.1: Demand Profile

South Ayrshire Council have provided a dataset of monitored energy use from May 2023 to May 2024. This includes total electricity and gas usage in kWh for the measured period, as well as monitored energy usage at a half-hourly resolution.

5.1.2: Daily Profile Analysis

The maximum demand over the monitored year is determined from the data, and the day which records the peak demand is further analysed to identify the half-hourly period in which the demand is greatest. Historic weather data from this point in time is used to analyse the heating load of the building.

5.1.3: Citadel Zones

Architectural plan drawings and sections of the Citadel are used to determine the different use zones within the building. The scale drawings are imported to AutoCAD where measurements are taken. Surface area for walls, partitions and glazed areas are taken from the drawing, and these are assigned U-values consistent with typical building materials for a leisure centre of this age.

5.1.4: Pool Zone Details

The entire pool zone of the building consists of a 1130 m² pool hall containing three swimming pools. The main pool has a water surface area of 345 m², the auxiliary pool has a water surface area of 135 m², and the training pool has a water surface area of 75 m². Much of the surfaces facing the outdoor environment are glazed, with 293 m² of glazed surface area and 68 m² of opaque surface area. The rest of the zone boundary faces other zones internal to the building. This includes changing facilities and an internal corridor. A plan drawing of the pool area, and a simplified isometric diagram illustrating the extent and outer facing boundaries of the zone are shown in figures 15 & 16.

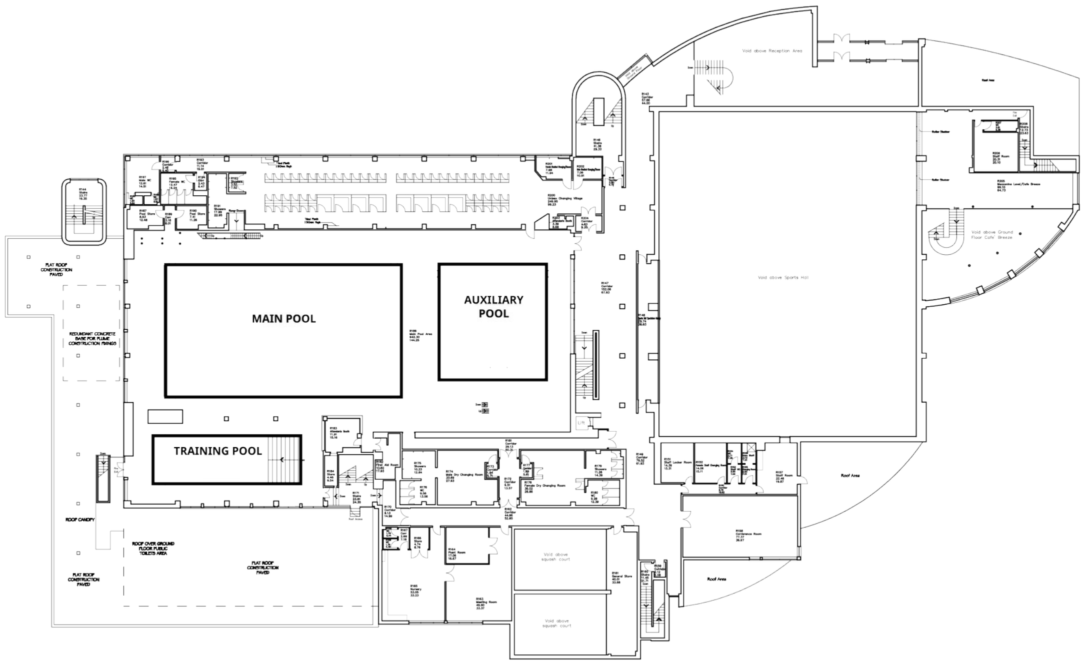


Figure 15: Site plan drawing of Citadel at pool hall level with pool areas highlighted (South Ayrshire Council)

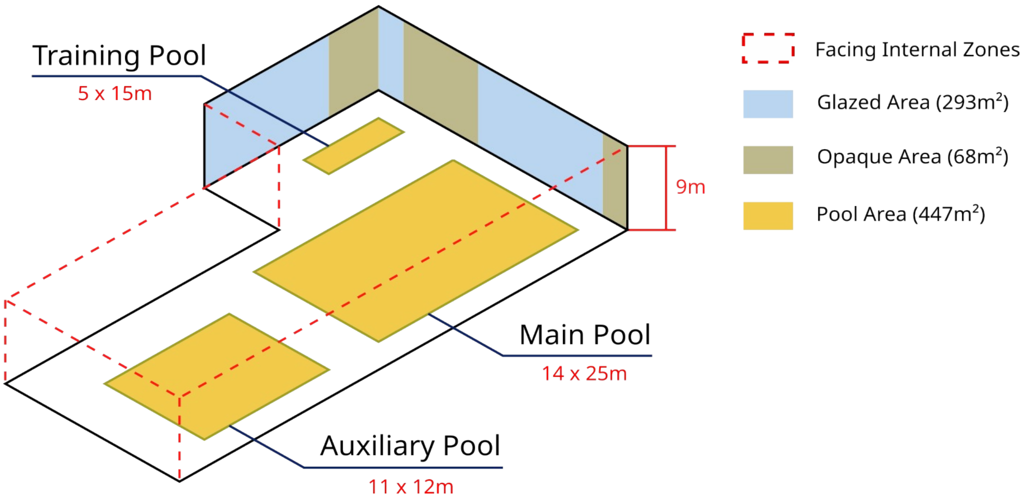


Figure 16: Simplified isometric view of the pool hall area with boundaries

The operational hours of the building are shown in table 1.

Table 1: Citadel opening hours

Day Type	Opening Time	Closing Time
Weekday	05:45	22:00
Weekend	08:30	17:30

5.1.5: Pool Zone Losses

Indoor pools exhibit multiple complex heat losses when compared with conventional building types. A model of an indoor pool has to take into account the requirements for ventilation to reduce humidity, the effects of user behaviour on water evaporation, high heating loads to maintain thermal comfort, and the constant refilling of water to maintain pool volume. Figure 17 illustrates the different types of heat loss observed with an indoor pool.

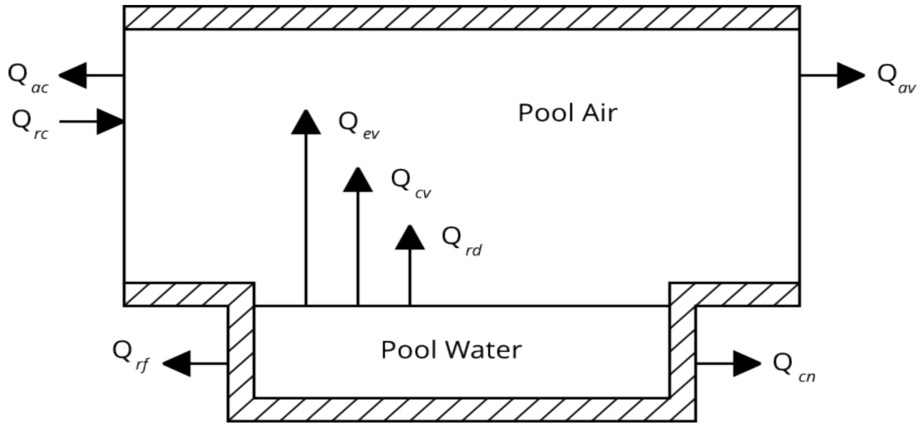


Figure 17: Illustration of heat losses within an indoor pool area

According to Mančić, et al (2014) around 30% of an indoor pool's energy demand comes from the heating of pool water, while around 45% of demand comes from ventilation losses. In the case of the Citadel, a numerical model will be employed to determine the split in demand.

The pool zone is divided into two regions to be assessed: the water body of the pool itself, which will be referred to as the 'pool water', and the surrounding air in the pool enclosure, which will be referred to as the 'pool air'. The 'pool zone' will refer to the entire zone which includes the pool water and pool air. The total heat loss of the pool zone is expressed by the following equation:

$$Q_{TOT} = Q_{PW} + Q_{PA}$$

Each region of the pool zone exhibits multiple separate heat losses, as illustrated in figure 18, which will be described in the following sections. This will be computed for an operational day in winter corresponding to the year of the demand dataset used in this report.

5.2: Pool Air Heat Losses

The total heat loss of the pool, Q_{PA}, area is described in the following equation. Q_{av} represents the energy needed to heat fresh outdoor air to the require temperatire and Q_{ac} represents conductive losses through the building fabric. Losses from uninteded air infiltration will not be considered, in line with plans to make the building wind and watertight. This report will model a push-pull type ventiliation system with heat recovery as illustrated in the review section.

$$Q_{PA}=Q_{av}+Q_{ac}$$

5.2.1: Ventilation Losses

The PWTAG code of practice recommends >10 l/s, or 36 m³/h, of fresh air ventilation per m² of pool hall area. This is needed to control the humidity levels within the pool hall, as discussed in the review section. This represents a large energy flow from the zone. Warm humid air is ejected from the pool hall and replaced by cooler outdoor air which needs to be heated to the correct temperature. The energy required to heat the cooler outdoor air to the required temperature is expressed in the following equation:

$$Q_{av}=\dot{V}\cdot\rho_o\cdot(h_i-h_o)$$

Where $\dot{V}$ is the fresh air ventilation flowrate, ρ_o is the density of the outside air, h_i is the enthalpy of the inside air and h_o is the enthalpy of the outdoor air.

5.2.2: Heat Recovery: LMTD Method

The air handling unit (AHU) at the Citadel makes use of a counterflow plate heat exchanger (PHE) as an energy efficiency measure. The cooler outdoor air interfaces with the warm exhaust air, where it is heated to a higher temperature, decreasing its humidity. The amount of heat recovered, Q_{rc}, to the outdoor air is expressed by the following equation:

$$Q_{rc}=U\cdot A\cdot\Delta T$$

Where U is the overall heat transfer coefficient (W/m²K), A is the surface area of the heat exchanger (m²), and ΔT is an appropriate mean temperature difference (°C); in this case the log mean temperature difference ($LM\Delta T$). Figure 18 illustrates the energy flow between two fluids in a counterflow arrangement. Each fluid has a mass flow rate, $\dot{m}$, a specific heat capacity, C_p, an inlet temperature, T_i, and a discharge temperature, T_o. Properties of the fluids are taken at film temperature, which is the mean temperature between inlet and discharge temperature.

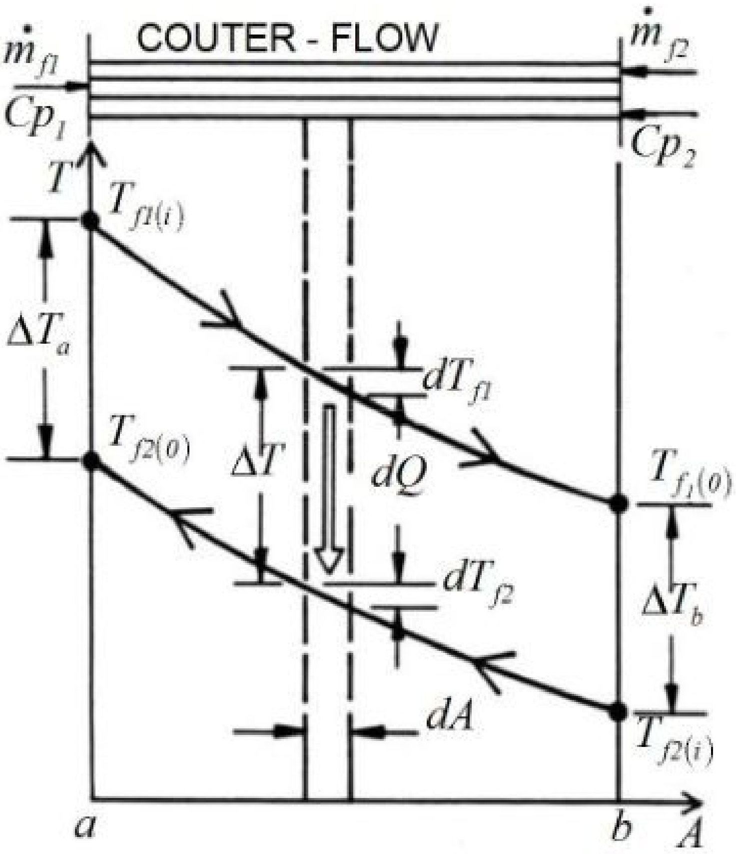


Figure 18: Diagram illustrating the exchange of heat between two fluids in counterflow (Tuohy, 2008)

The *LMΔT* is determined by analysing the inlet and discharge temperatures for each fluid. ΔT_a is the temperature difference between the inlet temperature of fluid '1' and the discharge temperature of fluid '2'. Conversely, ΔT_b is the temperature difference between the discharge temperature of fluid '1' and the inlet temperature of fluid '2'. The *LMΔT* is determined from the equation:

$$\Delta T_{lm} = \frac{\Delta T_b - \Delta T_a}{\ln(\Delta T_b / \Delta T_a)}$$

The overall heat transfer coefficient, *U*, is determined by analysing the convective heat transfer coefficients of the fluids at film temperature, as shown in the following equations.

$$\frac{1}{U} = \frac{1}{h_{f1}} + \frac{1}{h_{f2}}$$

The air in the ventilation system is assumed to be turbulent, and the flow is forced through the system by fans. Therefore, the convective heat transfer coefficient, *h*, for each fluid in the heat exchanger can be calculated by:

$$Nu = \frac{hl}{k}$$

and,

$$Nu = 0.0225 \cdot Re^{0.8} \cdot Pr^{0.33}$$

The Reynolds and Prandtl numbers, *Re* and *Pr*, are obtained for each fluid at film temperature, *l* is the characteristic dimension of the duct through which the fluid is flowing, and *k* is the thermal conductivity of the fluid.

5.2.3: Heat Recovery: Effectiveness Method

Alternatively, if the inlet temperatures of the outdoor air and exhaust air are known, and there is information available from the manufacturer regarding heat exchanger performance at test conditions, then it is possible to calculate the outlet temperatures of both fluids with the effectiveness method.

The effectiveness, ε, of the heat exchanger is the actual heat exchanged between the fluids divided by the theoretical maximum.

$$\varepsilon = \frac{Q}{Q_{max}}$$

The theoretical maximum, Q_{max}, is calculated using the following equation.

$$Q_{max} = C_{min} \cdot \left(T_{hot(i)} - T_{cold(i)} \right)$$

Where C_{min} is the strength of the lowest strength fluid in the system.

$$C_{min/max} = \dot{m} \cdot C_p$$

Once the effectiveness of the heat exchanger is calculated at test conditions, the value of Q_{max} at the desired ambient condition can be calculated. Then, the value of Q can be determined, and the outlet temperature for each fluid can be calculated separately by rearranging:

$$Q=\dot{m}\cdot C_p\cdot\Delta T$$

5.2.4: Exhaust Air Recirculation

The cooler the temperature of the outdoor air, the more the air will dehumidify as it is heated in the PHE. As pool hall areas require to be maintained at a certain humidity level, the outdoor air stream must have an acceptable moisture content before it is heated for supply. Figure 19 shows the effect on the humidity of air at 4°C and 93% humidity is heated to 28°C. While the moisture content remains the same, the increase of temperature decreases the humidity to below 20%, which would be unsuitable for supply to the pool hall.

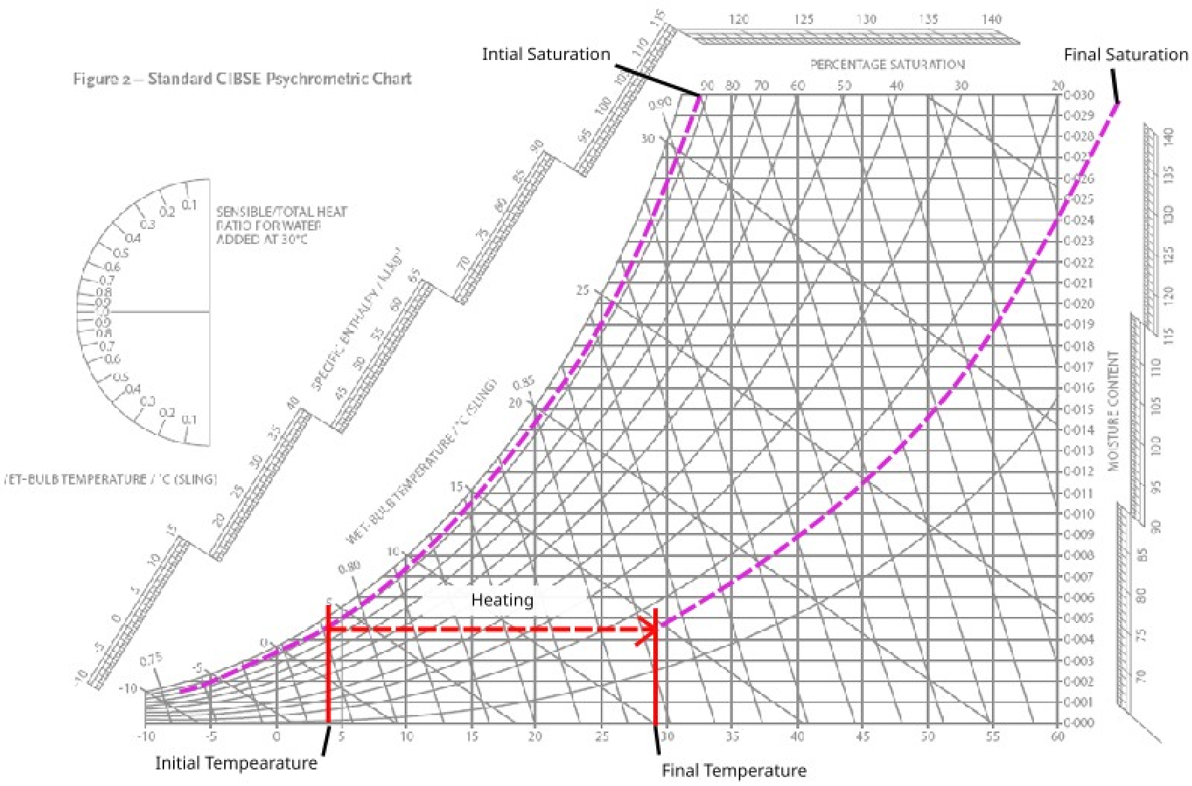


Figure 19: Effect of heating on the moisture content of air, plotted on a psychrometric chart

To control humidity, AHUs recirculate a proportion of exhaust air and mix this with the outdoor air stream. The proportion of outdoor air and exhaust air is controlled by dampers within the unit, and this air mix controls the humidity of the supply air.

The lower the mixture of outdoor air, the less energy is required to heat the mixed air back to the desired temperature. However, the higher the mixture of outdoor air, the lower the moisture content of the mixed air will be, therefore it is important to have the correct ratio of mixed to outdoor air to achieve the required humidity within the pool hall. This is described in the following equations (CIBSE, 2012):

$$\dot{m}_m \cdot h_m = \dot{m}_A \cdot h_A + \dot{m}_B \cdot h_B$$

Where $\dot{m}_m$ is the mass flow rate of the mixed air stream, and $\dot{m}_A$ and $\dot{m}_B$ are the two air streams to be mixed (kg/h). The enthalpy of each stream, h, is also analysed (kJ/kg). A similar equation can be used to calculate the moisture content, x, of the mixed air stream if the moisture content of the two air streams to be mixed are known (kg/kg).

$$\dot{m}_m \cdot x_m = \dot{m}_A \cdot x_A + \dot{m}_B \cdot x_B$$

The mass flowrate can be calculated by multiplying the volume flow rate of the air stream (m^3/h) by the density of the air in the stream (kg/m^3).

$$\dot{m} = \dot{V} \cdot \rho$$

If the temperatures of the two airstreams are known, the temperature of the mixed air stream (°C) can be calculated.

$$\dot{m}_m \cdot T_m = \dot{m}_A \cdot T_A + \dot{m}_B \cdot T_B$$

The properties of the mixed air may also be analysed by using a psychrometric chart, where a straight line is drawn between the properties of each air stream, and the properties of the mixed air lie on this line.

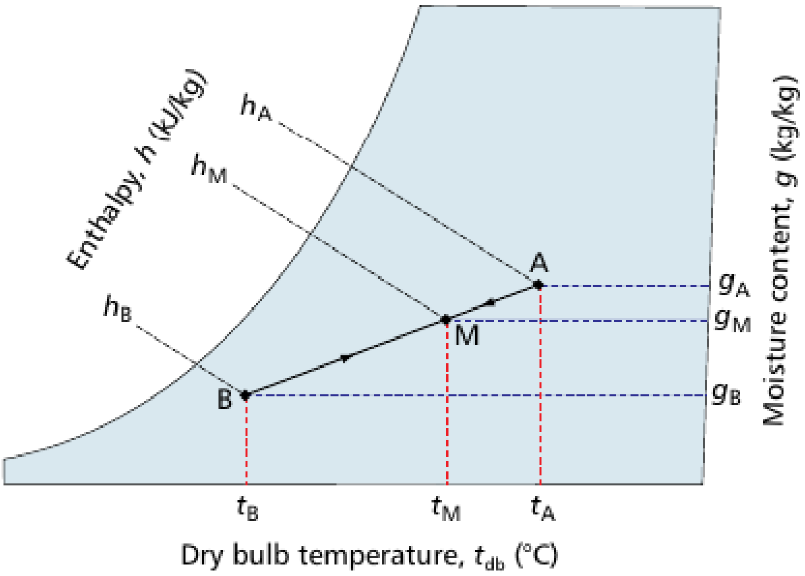


Figure 20: Properties of mixed air, plotted on a psycrhometric chart (CIBSE)

The PWTAG Code of Practice recommends that, where air recirculation is possible, at least 30% of the supply air must come from a fresh source. This air mix proportion will be used to determine the properties of the mixed air during a winter operation scenario.

The net heat loss, or the amount of heat required to bring the mixed air to 29°C, is determined numerically. The enthalpy of the mixed air is calculated by analysing the sensible and latent heat in the mixed air.

$$h_m = h_a + (x_m \cdot h_w)$$

Where h_a is the specific enthalpy of dry air in the mixed air stream, h_w is the specific enthalpy of water vapour in the mixed air stream and x_m is the moisture content of the mixed air, as determined from the equations in section 5.4.2. h_a and h_w are calculated from the following equations.

$$h_a = c_{pa} \cdot T_m$$

$$h_w = c_{pw} \cdot (T_m + h_{we})$$

Where T_m (°C) is the temperature of the mixed air, c_{pa} (kJ/kg°C) is the specific heat of air at constant pressure, c_{pw} (kJ/kg°C) is the specific heat of water vapour at constant pressure, and h_{we} (kJ/kg) is the evaporation heat of water at 0°C (Engineering Toolbox).

5.2.5: Fabric Losses

In addition to the overall losses from the ventilation system, heat lost to conduction through the fabric of the building itself must be considered. Glazed and opaque surface areas facing the outside environment are considered. For the purposes of this report, any surfaces facing adjacent internal zones are not considered. Overall fabric losses are expected to be much smaller than ventilation losses, and so any losses to adjacent building zones will be insignificant to the total heat loss of the pool zone. The heat lost to conduction will be determined by:

$$Q_{ac}=\sum U_f \cdot A_f \cdot (T_i - T_o)$$

Where U_f is the U-value, or heat transfer coefficient, of the fabric, A_f is the area of the fabric, T_i is the temperature of the air inside the zone, and T_o is the temperature of the air in the external environment.

5.3: Pool Water Heat Losses

As illustrated in figure 17, the pool water exhibits multiple complex heat losses. In addition to the usual convective, conductive and radiative heat losses, heat losses from water evaporation and the circulation of fresh water in the pool must also be modelled. The total heat loss from the pool water is expressed in the following equation:

$$Q_{PW}=Q_{ev}+Q_{cv}+Q_{cn}+Q_{rd}+Q_{rf}$$

Q_{ev} represents the heat lost by the evaporation of water from the pool surface, Q_{cv} represents the convective heat lost by the movement of air over the pool surface, Q_{cn} represents the heat lost to conduction through the pool floor, Q_{rd} represents to heat lost through long wave radiation between the pool water and the roof surface of the pool enclosure, and Q_{rf} represents the heat lost through the constant refilling of the pool with fresh water.

5.3.1: Evaporative Losses

Evaporation causes the largest heat loss from the pool water (Mančić et al, 2014). The conversion of liquid pool water to vapour results in a transfer of water from the pool to the surrounding environment. The heat lost from the pool is related to the transfer of mass from the pool water to the pool hall, as illustrated in the following equation:

$$Q_{ev} = \dot{m} \cdot l \cdot A_p$$

Where $\dot{m}$ is the mass flow rate of the water leaving the pool, l is the latent heat of the vapourisation of water, and A_p is the surface area of the pool water. The latent heat of vapourisation is obtained from the relevant table (Engineering Toolbox) and is dependent on the temperature of the pool water. The following figure shows the latent heat of vapourisation for the pool water at the water temperatures recommended by the PWTAG code of practice.

Table 2: Latent heat of vapourisation in recommended pool temperature range

Pool Temperature (°C)	Latent Heat of Vapourisation (Wh/kg)
27	676.9
28	676.3
29	675.6

During pool operational hours, the mass flow rate of water leaving the pool, $\dot{m}$, is dependent on the number of occupants. This is due to the increased turbulence of the water in the pool as its occupants disturb the water while swimming. González Miguel (2014) proposes an empirical method to determine this value, with a factor for the number of occupants.

$$\dot{m} = (16 + 133n)(X_w - \varphi X_a)$$

Where n is the occupancy rate (swimmers/hr m²), φ is the relative humidity, X_w is the humidity ratio at saturation at the temperature of the water and X_a is the humidity ratio at saturation at the temperature of the air in the pool hall, expressed in kilogram of water vapour per kilogram of dry air (kg_w/kg_a).

As there is no data available on occupancy, a range of occupancy rates will be assessed to test the extent to which occupancy affects the evaporation losses from the pool. Based on a safe maximum occupancy of one swimmer per 3 m² of pool area, set out by the Health & Safety Executive (HSG179 Managing Health & Safety in Swimming Pools), the combined maximum occupancy of all pool areas in the Citadel is 223 swimmers, which gives an occupancy rate of 0.5. Table 3 compares occupancy rates with the number of swimmers using the Citadel pool areas.

Table 3: Occupancy rate and number of swimmers for the Citadel pool area

Occupancy Rate (n)	Number of Swimmers
0.1	45
0.2	89
0.3	134
0.4	179
0.5	223

The PWTAG code of practice recommends a relative humidity within the pool hall of 60% and must be no less than 50% and no greater than 70%. For the purposes of this report, a relative humidity of 60% will be assumed.

The humidity ratios are identified using a psychrometric chart. The following table shows values of X_w and X_a at the recommended water and air temperatures.

Table 4: Humidity ratios at saturation for recommended water and air temperatures

Pool Temp. (°C)	X_w (kg_w/kg_a)	Air Temp. (°C)	X_a (kg_w/kg_a)
27	0.0227	28	0.0223
28	0.0223	29	0.0218
29	0.0218	30	0.0214

Outside of operational hours, pool water evaporation is minimised by use of a thermal insulation cover. This method will analyse heat losses at peak operational times, therefore heat loss during the use of a thermal insulation cover will not be considered.

5.3.2: Convective Losses

The pool hall is maintained at a similar temperature to the pool water, and PWTAG guidance recommends that air movement be minimised at the pool surface for comfort reasons. In this case, it can be assumed that the air movement localised at the pool surface is minimal and therefore convective losses are insignificant.

5.3.3: Conductive Losses

Heat is also transferred from the pool water through the pool container to the surrounding environment. In outdoor pools this may be dependent on the temperature of the soil (Govaer, Zami, 1981). In the case of the Citadel, where the pool is situated on the first floor of the building, the surrounding temperature will be slightly lower than that of the pool, therefore conductive losses will be minimal and will be assumed to be insignificant.

5.3.4: Radiative Losses

Further heat losses from the pool water are caused by long wave radiation, transferring heat from the pool surface to the surface of the ceiling in the pool hall. This is calculated from the following equation:

$$Q_{rd} = A_p \cdot \varepsilon_w \cdot \sigma \cdot \left((T_p + 273)^4 - (T_{sur} + 273)^4 \right)$$

Where A_p is the surface area of the pool water, ε_w is the emissivity of the pool water, σ is the Stefan-Boltzmann constant, T_p is the temperature of the pool water, and T_{sur} is the temperature of the ceiling surface.

5.3.5: Refilling Losses

The effects of evaporation, combined with the stochastic behaviour of occupants and other processes within the pool, result in a loss of pool volume. This water must be replaced to compensate, and this is achieved by refilling the pool with fresh water. The temperature difference between the pool water and the replacement water results in a further heat loss. This is expressed by the equation:

$$Q_{rf} = c_p \cdot \dot{m}_{rf} \cdot (T_p - T_{rf})$$

Where c_p is the specific heat of water, $\dot{m}_{rf}$ is the mass flow rate of the refilling water, T_p is the temperature of the pool water, and T_{rf} is the temperature of the refilling water. There are no official guidelines or recommendations for mass flow rate of water refill in the UK, therefore this report will consider a refill rate to match that of evaporative losses.

5.4: Final Demand Breakdown

The total heat loss of the pool zone will be calculated for the peak hour of each target day and this will be subtracted from the peak monitored gas demand. This will allow the demand split between the pool zone and the rest of the building to be analysed further.

5.4.1: Comparison of Pool Water Heating and Pool Air Heating

A further comparison will be made between the energy demand of the pool water and the pool air. Both combined are expected to be a large proportion of the overall building demand, however it is useful to compare the two when considering the type of heat pump that will be used.

5.5: River Characteristics

5.5.1: Location

The water intake for the heat pump is proposed to be located close to the Citadel at the mouth of the River Ayr as it discharges into the Firth of Clyde.

5.5.2: River Temperature

To properly analyse the potential heat pump performance, it is necessary to obtain real measured data, or to directly measure the water temperature over a sufficient period using data loggers. Obtaining real temperature data over a year will help to determine the COP of any new heat pump system. Specific data for the proposed site is limited, and annual temperature variation is affected by water depth. To demonstrate the methodology, average sea surface temperature data from the Clyde estuary is used (Marine Scotland), figure 21. Surface temperatures vary between around 7°C in winter to around 15°C in the summer.

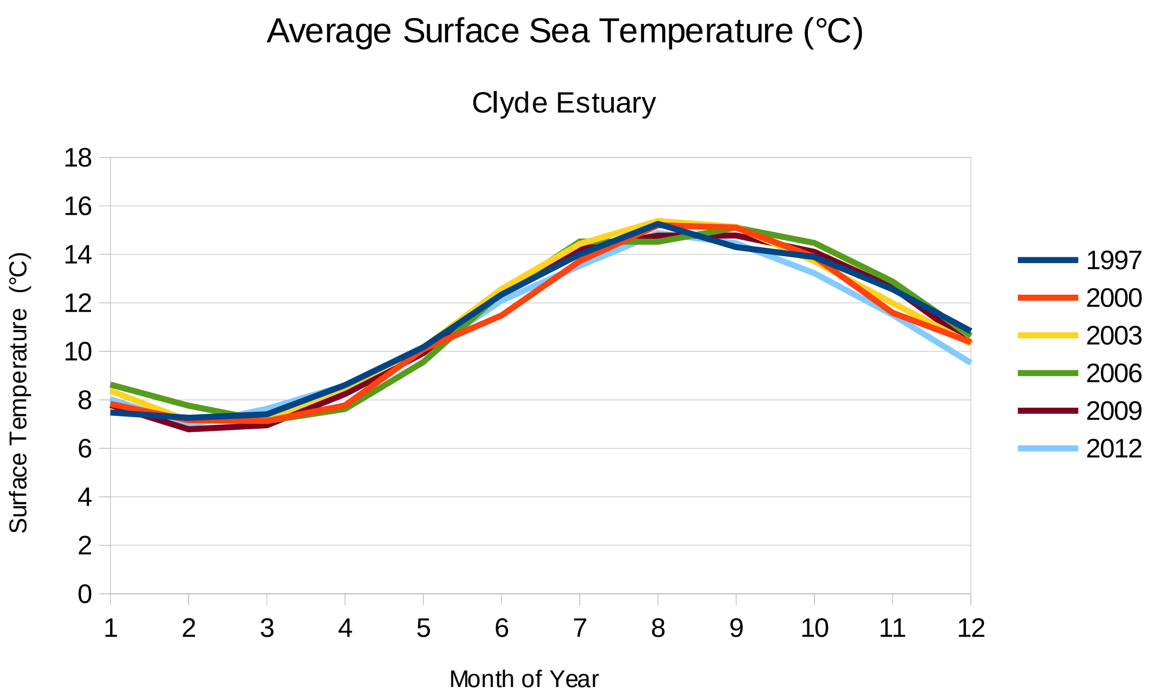
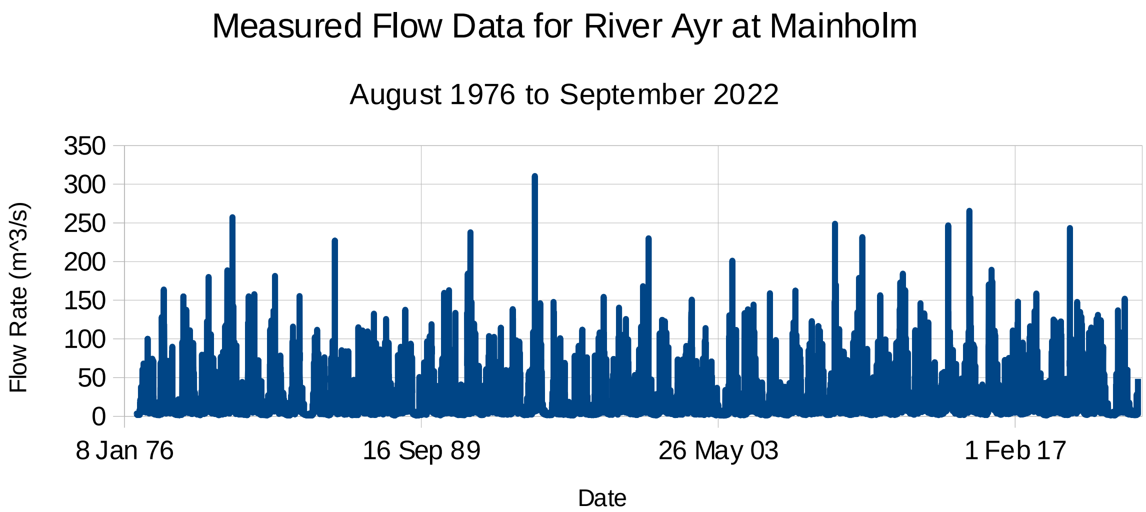


Figure 21: Average surface sea temperature for Clyde Estuary between 1997 and 2012 (Marine Scotland)

5.5.3: Flow Rate

The flow rate of the river is an important factor to consider when assessing the feasibility of a river source heat pump system.

River flow data at the exact location of the Citadel is not available. The closest station is maintained by the UK National River Flow Archive at Mainholm. This station is around 3km from the proposed site (figure 22), which is measured using satelite map data. The dataset contains daily flow rate in m^3/s from August 1976 to September 2022. Figure 23 shows the variation in river flow rate in this date range.



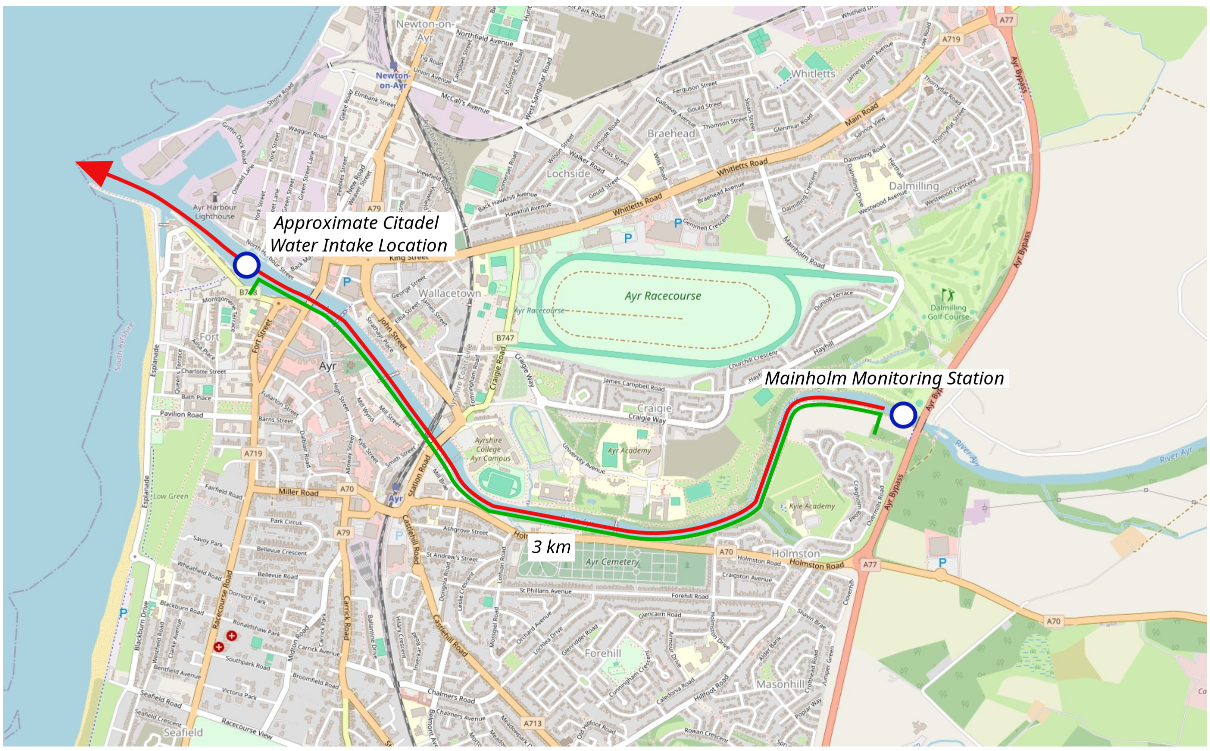


Figure 22: Location of Mainholm monitoring station, relative to proposed water intake location (OpenStreetMap)

As can be seen from figure 24, there is a high variation in river flow rate. The exact location of the river used for the Citadel system will experience large volumes of tidal water entering from the Clyde Estuary, and so without accurate data, it is assumed that the energy in the flow of the waterbody will be sufficient for the heating demand of the site.

5.5.4: Water Abstraction

The abstraction of inland surface water is regulated and authorised by the Scottish Environmental Protection Agency (SEPA). The level of water abstraction by any scheme determines the level of Controlled Activities Regulation (CAR) authorisation required. Abstraction levels greater than 2000 m^3/day require a complex license.

The level of abstraction should be suited to the demands of the site, which can be calculated by rearranging the following equation:

$$Q = \dot{V} \cdot \rho \cdot c_p \cdot \Delta T$$

Where Q is the heating demand (W), $\dot{V}$ is the volume flow rate of the river water (m^3/s), ρ is the density of the water (kg/m^3), c_p is the specific heat capacity of the water (J/kg°C), and ΔT is the temperature difference of water entering the heat pump and being discharged back into the watercourse (°C). ΔT will be taken as 2°C as this is a common temperature difference for water source heat pumps as found from a review of similar systems and SEPA guidance, which allows for the return of water within 3°C of river ambient temperature.

5.6: Site Selection

5.6.1: Site Survey

During the site visit with South Ayrshire Council, a preliminary site survey was undertaken to identify potential areas at which a plant building or equipment could realistically be constructed.

5.6.2: Geographical Analysis

In addition to the site survey, map data from OpenStreetMap and Google Earth is used to determine promising locations for any new heat pump system and associated plant building and pipe routing. Existing built-on areas are identified, as are future sites identified from planning documents. Figure 24 shows the location of existing and future built-on areas, and figure 25 illustrates potential corridors for the routing of pipework and other ancillaries.

Figure 24: Citadel location showing adjacent land uses

Figure 25: Unoccupied space for pipework routing

The Citadel is located in an area of high residential development, with existing properties lining the adjacent quay wall and a large number of apartments immediately to the north and west of the site. Other land uses include green space and private vehicle parking. Immediately to the south-east of the site is a historic wall of Cromwell's fort, from which the Citadel gets its name, which is located in a green corridor with a pedestrian and cycle route. Any siting of plant equipment should take into account these features.

There have been plans for housing development along South Harbour Street for some time, with the most recent plans being withdrawn in 2019 (South Ayrshire Council). However, this site remains included in SACs Strategic Housing Investment Plan (SHIP) and so it should be expected that this land is to be developed in future and would therefore be unsuitable for the siting of any plant buildings or equipment.

The existing housing features surface private vehicle parking which could provide a corridor for pipework to follow from the quay wall to any new plant equipment. This would cause minimal disruption from construction, and is conveniently placed to provide the shortest distance between the Citadel and any water abstraction equipment constructed in the quay wall.

5.7: Heat Pump

5.7.1: Flow Temperature

This report will consider two levels of output flow temperature from any potential heat pump system. Low temperature will refer to 40-55°C output, and high temperature will refer to 70-80°C output.

A low temperature and high temperature system may have the same thermal capacity; however a low temperature system may achieve higher efficiencies as the temperature difference between the evaporator and the condenser is minimised.

A high temperature system has the benefit of easily integrating with the existing distribution system for space heating and hot water. An output of 70-80°C will allow for the building to use the existing pipework and radiators. However, as it is expected that most of the demand will come from pool water and pool air heating, it may be more beneficial in the long-term to employ a low temperature heat pump for the pool zone, taking advantage of the higher COP of such a system.

Therefore, there may be some trade-offs to consider. A low temperature system servicing the entire site will require a substantial replacement of radiators within the conventional building zones such as the gym, reception and foyer, and a high temperature system will have a lower efficiency and therefore higher energy costs.

5.7.2: Thermal Capacity

The thermal capacity of the heat pump will be selected based on the maximum demand measured from the demand data, and from analysis of the heat losses from the pool zone. For the rest of the building, the conventional space heating demand may be higher if there are any instances of ambient temperatures colder than that experienced between the dates monitored. For the pool zone, the heat losses are variable depending on ambient conditions and pool occupancy. Any new heat pump system must be capable of operating under a wider range of conditions which may include extreme temperatures in winter, or an increase in patronage. The thermal capacity of the system must be carefully considered to allow for additional capacity in the system.

5.7.3: Design Option

This report will consider three design options for the heat pump system. The first option will be one high temperature heat pump servicing all requirements for pool water heating, pool air heating, rest-of-building space heating and sanitary hot water. This option will be suited to using existing distribution systems and radiators.

The second option will be one low temperature heat pump to service the pool water and pool air heating, with a smaller capacity high temperature system servicing the rest-of-building requirements. This allows the conventional space heating and DHW distribution systems to be used, but as most of the demand is expected to come from the pool zone, a low temperature system may have the benefit of higher efficiency, as the temperature difference between source and supply for the bulk of the demand is reduced. In terms of equipment, only the pool water and pool air heat exchangers will have to be replaced.

The third option will retain the use of the gas boiler system for rest-of-building duty, while the pool zone will be supplied by a low temperature heat pump. Again, this is expected to cover the majority of heating demand for the building, and while this may not offer the same level of decarbonisation, it may present a lower cost option and allow the gas boiler to be replaced by a different type of system in future. For example, an air source heat pump (ASHP) which may be more suited for this use, be more affordable, and therefore more attractive to the facility operator.

5.7.4: Heat Pump Performance

The performance of each design option will be analysed. This will allow a comparison of each scenario in terms of electrical energy use, and thus cost of energy and carbon savings compared with the existing base case where heating is supplied by a gas boiler system. This report will consider two methods of determining the performance of each system: the coefficient of performance (COP), and the seasonal performance factor (SPF).

Coefficient of performance (COP) is a commonly used measure of heat pump performance. COP compares the amount of heat provided by the system with the amount of electrical energy required to produce this heat, as discussed in the review section. Heat pumps often come with a rated COP at test conditions, however in reality, where the heat pump is operating in variable conditions, the COP will also vary. Temperature data from the river Ayr will be used to calculate the difference between the varying source temperature and target supply temperature for each design option.

Calculating the COP over a year of operation allows for the seasonal performance factor (SPF) to be calculated. This is simply the amount of thermal energy generated over a year of operation, compared with the electrical input over a year of operation.

A simple COP model will be created by calculating the Carnot COP for each system type. The system efficiency will be assumed to be 60%, and this will allow for an estimation of comparative COP between the systems (industrialheatpumps.nl). A more detailed analysis of system COP will require a more detailed design of the heat pump system itself, which is out with the scope of this report.

5.8: Carbon Emissions

Carbon emissions for each heat pump design option will be calculated using conversion factors, published by the UK Government each year. The conversion factors relate to the carbon intensity of electricity generation and natural gas usage in terms of carbon dioxide equivalent per kilowatt hour of energy (kg CO_2e/kWh) consumed. For 2023, these factors are 0.18293 kg CO_2e/kWh for natural gas and 0.22499 kg CO_2e/kWh for electricity (ITP Energised). Total energy used in a year for each system design option will be calculated using the COP and the monitored demand from 2023/2024. The overall CO_2 equivalent emissions per kWh of energy used will be calculated for each case and compared to the base case, existing heating system.

5.9: Energy Cost

Like the calculation of CO_2 emissions, energy cost will be based on the total energy used for each system design option using the on the cost of electricity and natural gas per kilowatt hour in 2023. Energy prices will be set at 28.62p per kWh for electricity and 7.42p per kWh for natural gas (WSP). The total price of energy for one year, based on the half-hourly data and the theoretical COP of each design option, will be calculated and compared with the base case.

6: Results

6.1: Citadel Demand Profile

Figure 26 shows the monitored gas demand of the building between May 2023 and May 2024. This represents all energy demand for pool water heating, pool air heating, rest-of-building space heating and sanitary hot water heating.

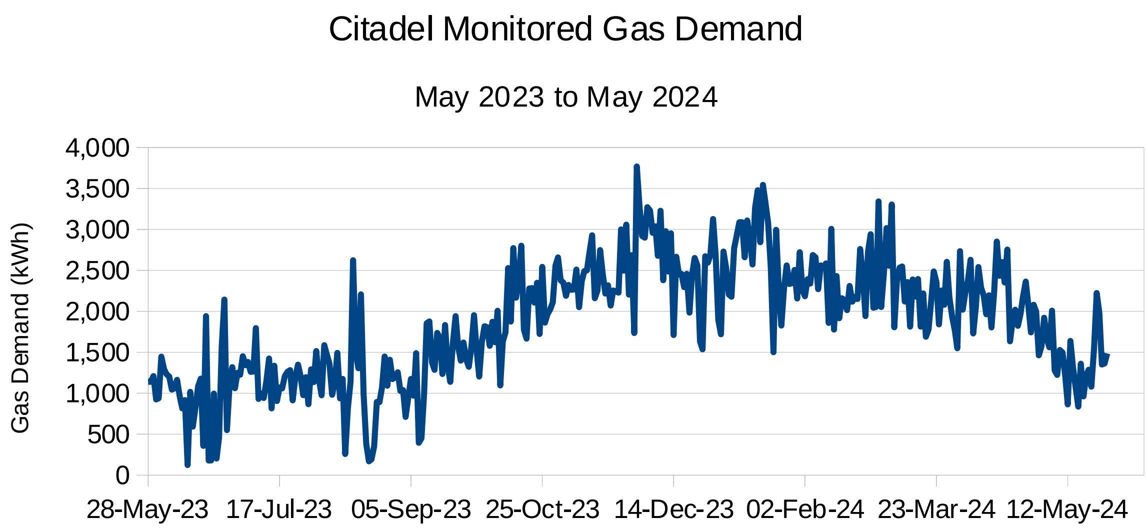


Figure 26: Annual gas demand profile for Citadel 2023/2024

The profile exhibits a seasonal variation in demand, with the highest demand observed in the winter months. The values relate to the daily total demand in kWh. At half-hourly resolution, the peak demand for the year is recorded between 05:30 and 06:00 on the 17th of October 2023, with 224kWh recorded during this time.

6.2: Winter Scenario

As identified in section 6.1, the peak demand for the year during operational hours is recorded on the 17th of October 2023 between 05:30 and 06:00. At this time, the pool is opened to the public, and it can be assumed that occupancy is low at this early time in the morning. Historical weather data, obtained from Weather Underground, shows that the air temperature at this time was 4°C and the relative humidity was 93%. Figure 27 shows the half-hourly monitored gas demand for that day.

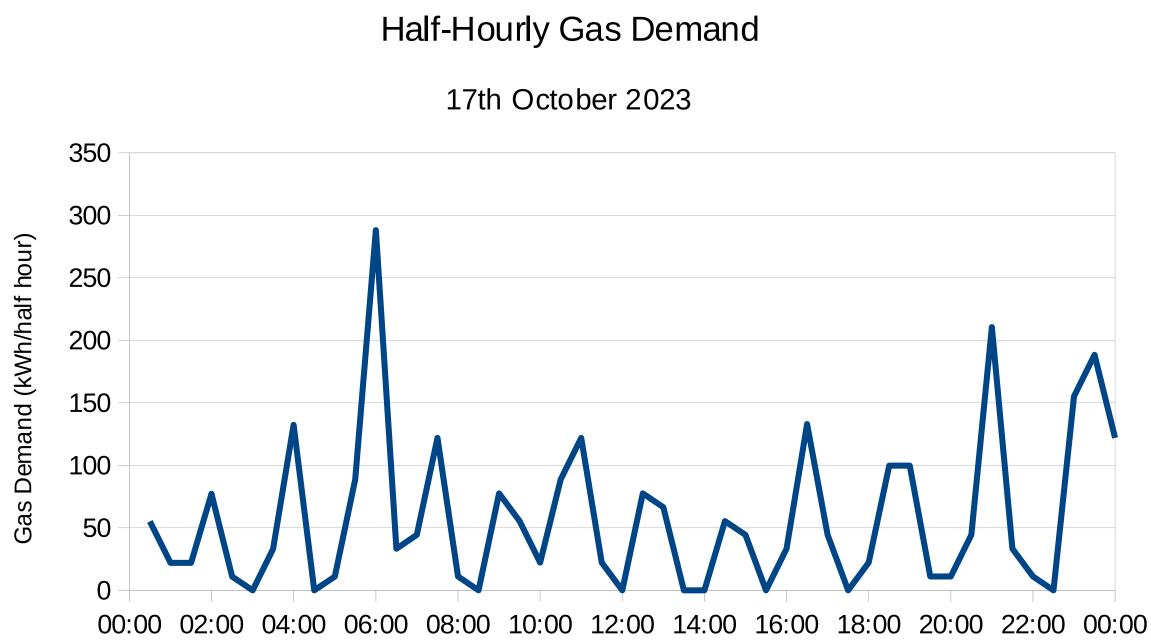


Figure 27: Demand profile for 17/10/23

The demand is measured in kWh for half hour intervals. The peak demand is recorded as 288 kWh over the half hour, so the peak demand is taken as 576 kW.

Table 5: Air properties at target time (from Weather Underground)

Date	Time	Air Temperature (°C)	Humidity (%)
17/10/2023	05:30 - 06:00	4	93

6.3: Pool Air Losses

The air handling unit (AHU) at the Citadel is a Dantherm manufactured system which uses fans to convey exhaust air from the pool hall and fresh air from the exterior of the building. The unit uses a counterflow heat exchanger and a bypass to control the temperature and humidity of the air supplied back to the pool hall. During the site visit, the AHU was inspected. Figure 28 shows the nameplate of the unit, showing its stated air volume flow rate.

Figure 28: Citadel AHU nameplate

The energy required to maintain the pool air at a temperature of 30°C is now determined based on the properties of the outdoor air between 05:30 and 06:00 on the 17th of October 2023. An excel spreadsheet using the equations described in section 5.4 is used to model the two processes undergone by the outdoor air. Firstly, the air is passed through the counterflow heat exchanger, increasing its temperature. Secondly, the heated outdoor air is mixed with a proportion of recirculated air. The enthalpy difference between the indoor air and the mixed air stream determines the net heat loss from the pool hall area.

6.3.1: Heat Exchange

The first step in calculating the heating load of the pool hall in the winter scenario is to determine the temperature gained by the outdoor air during the winter scenario as it passes through the counterflow heat exchanger. The first step is to determine the effectiveness of the heat exchanger using the manufacturers data regarding heat exchanger performance at test conditions (figure 29).

PT Counterflow exchanger		coat.7035/coat.7035	282 Pa
Type: **PCF-K-270-2617-GE-T-B-500-C-SM-RML**		plate material: **Alu-coated**	
Winter data:	Power: **376.56** kW	Efficiency dry/wet: **76 / 91.1** %	
Supply air: **35280** m³/h	Pressure drop: **282** Pa	Return air: **37044** m³/h	Pressure drop: **333** Pa
Inlet: **-5** °C **100** %		Inlet: **30** °C **60** %	
Outlet: **26.9** °C **11** %	ER-class: **H1**	Outlet: **14.5** °C **100** %	
freezing temperature: **-99.9 °C**			
Access.: 2x sloped drain pan galvanized/powder coated 2x adapter for inside mounted damper acting motor 2x hinged service door with EMKA + door open holder plate heat exchanger also splitted in the deep bypass on supply air		1x servo motor SIEMENS GDB161.1E 5 Nm IP54, 24V, 0..10 V fitted on recirculation damper recirculating air damper fitted on plate heat exchanger differential pressure switch for bypass control inside fitted recuperator section dismanteled/numbered during transport 1x servo motor SIEMENS GIB161.1E 35 Nm IP54, 24V, 0..10 V fitted on bypass damper	

Figure 29: Existing heat exchanger data sheet

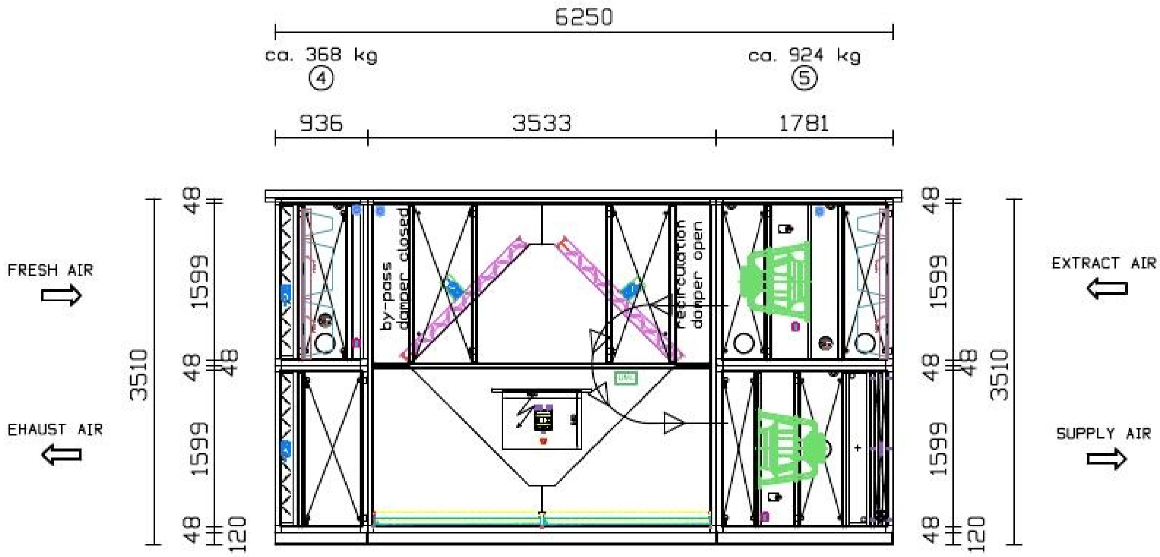


Figure 30: Side elevation section view of Citadel AHU

In the data sheet, the power, Q (376.56 kW), refers to the amount of heat transferred between the hot and cold fluids at test conditions. The theoretical maximum heat flow, Q_{max}, can be determined by analysing the strength of each fluid, C_{min} and C_{max}, and the difference between the inlet temperatures of the hot and cold fluids. Note that C_{min} is whichever fluid has the lowest strength. In this case it is the hot fluid.

$$Q_{max} = C_{min} \cdot \left(T_{hot(i)} - T_{cold(i)}\right) \ \& \ C_{min/max} = \dot{m} \cdot C_p$$

$$Q_{max} = 11783.1 \cdot (30 - (-5)) = 412.4\, kW$$

The effectiveness of the heat exchanger, ε, can now be calculated by comparing the actual heat flow at test conditions with the theoretical maximum heat flow.

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{412.4}{376.6} = 0.913$$

It is now possible to determine the heat flow at winter scenario conditions of 4°C outdoor air temperature. The theoretical maximum heat flow, Q_{max}, is calculated as before. Table 6 shows all relevant properties of the air during the winter scenario. Note that the volume flow rate reflects the ricirculating a 70% proportion of the exhaust air, which will be discussed in more detail in the following section.

Table 6: Air properties at winter scenario conditions

Air Inlet Temp. (°C)	Density (kg/m³)	Volume Flow Rate (m³/h)	Mass Flow Rate (kg/s)	Specific Heat Capacity (J/kg K)	Fluid Strength (C)
4	1.273	10584	3.743	1006	3765
30	1.164	10584	3.422	1007	3446

$$Q_{max} = C_{min} \cdot \left(T_{hot(i)} - T_{cold(i)}\right) = 3446 \cdot (30 - 4) = 89.6\, kW$$

The real heat flow, *Q*, at these conditions, based on the effectiveness of the heat exchanger, is now calculated.

$$Q = \varepsilon \cdot Q_{max} = 0.913 \cdot 89.6 = 81.8\, kW$$

Now, it is possible to determine the outlet temperature of each fluid.

$$Q=\dot{m}\cdot C_p\cdot\Delta T$$

Where, for the cold fluid:

$$T_{cold(o)}=T_{cold(i)}+(\frac{Q}{\dot{m}\cdot C_p})=4+(\frac{81811}{3.743\cdot 1006})=25.7\,°C$$

And, for the hot fluid:

$$T_{hot(o)}=T_{hot(i)}-(\frac{Q}{\dot{m}\cdot C_p})=30-(\frac{81811}{3.422\cdot 1007})=6.3\,°C$$

6.3.2: Air Mixing

Now that the air has passed through the heat exchanger, it is mixed with the recirculated air. The PWTAG code of practice states that 30% of air delivered to the pool hall must come from fresh sources. The mixing of air serves to further minimise the need for heating, and to ensure the pool air is supplied at the correct humidity level.

Now that the fresh outdoor air has been heated to 25.7°C, it's properties can be plotted on a psychrometric chart, along with the properties of the exhaust air (figure 31). The moisture content of the fresh air remains constant during heating. The indoor air is assumed to be exhausted at 30°C and 70% humidity, which is the maximum recommended in the PWTAG code of practice.

A line intersecting the two points is plotted on the chart, and the temperature of the mixed air stream is determined by using the equations described in section 5.4.3. Table 7 shows the relevant properties of the heated fresh air and the recirculated exhaust air. The total volume flow rate delivered to the pool hall is 35280 m³/h.

Table 7: Air properties of fresh and exhaust air streams

Air Stream	Temp. (°C)	Moisture Content (kg/kg)	Volume Flow Rate (m³/h)	Air Mix Proportion (%)	Density of Air (kg/m³)	Mass Flow Rate (kg/h)
Fresh	26	0.0177	10584	30	1.164	12489
Exhaust	30	0.0047	24696	70	1.18	28746

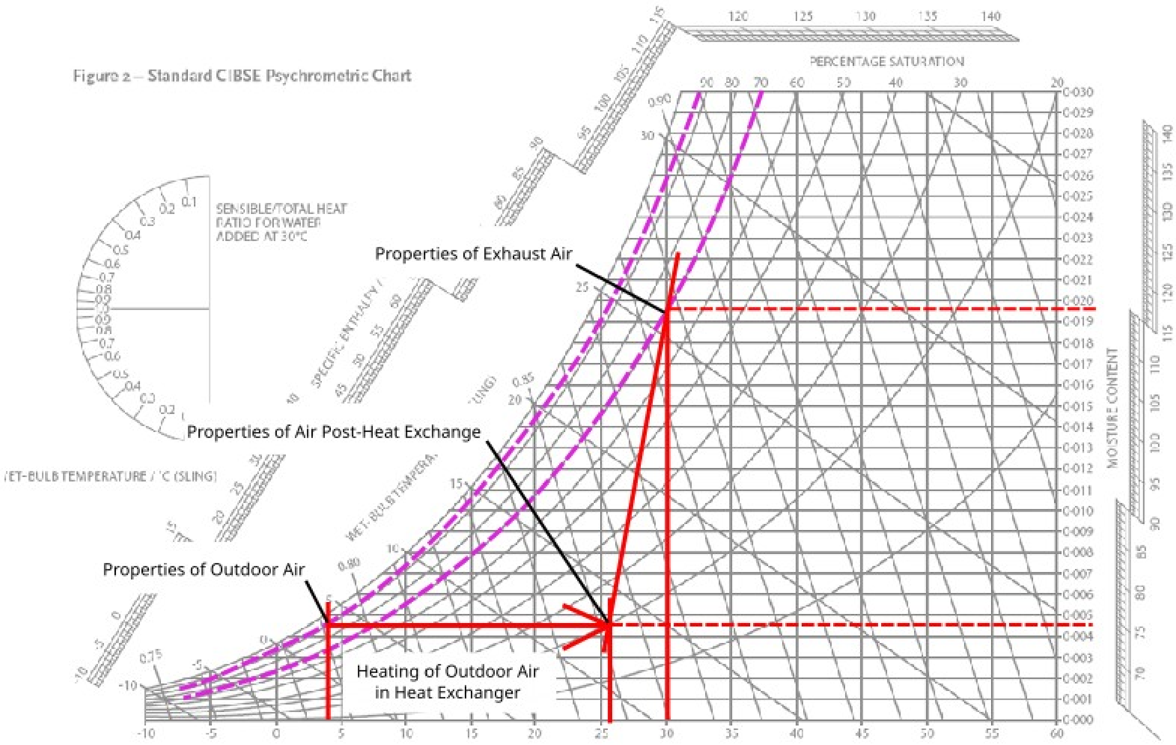


Figure 31: Plot of air streams on a psychrometric chart

The temperature of the mixed air stream is determined by rearranging the equation described in section 5.4.3.

$$T_m = \frac{\dot{m}_A \cdot T_A + \dot{m}_B \cdot T_B}{\dot{m}_m}$$

The mass flow rate of the mixed stream is the combined mass flow rate of the fresh air and exhaust air streams.

$$\dot{m}_m = \dot{m}_{fresh} + \dot{m}_{exhaust} = 12489 + 28746 = 41235\, kg/h$$

And so,

$$T_m = \frac{12489 \cdot 26 + 24696 \cdot 30}{41235} = 28.8\,°C$$

This is now plotted on the psychrometric chart (figure 32), where the dashed blue line is plotted upwards from the mixed air temperature. The point of intersection with the red line indicates the moisture content and therefore saturation of the mixed air stream. The line is then projected along to 30°C to represent the required heating for supply, and the saturation is observed as around 60% which is ideally within the recommended range as set out by PWTAG.

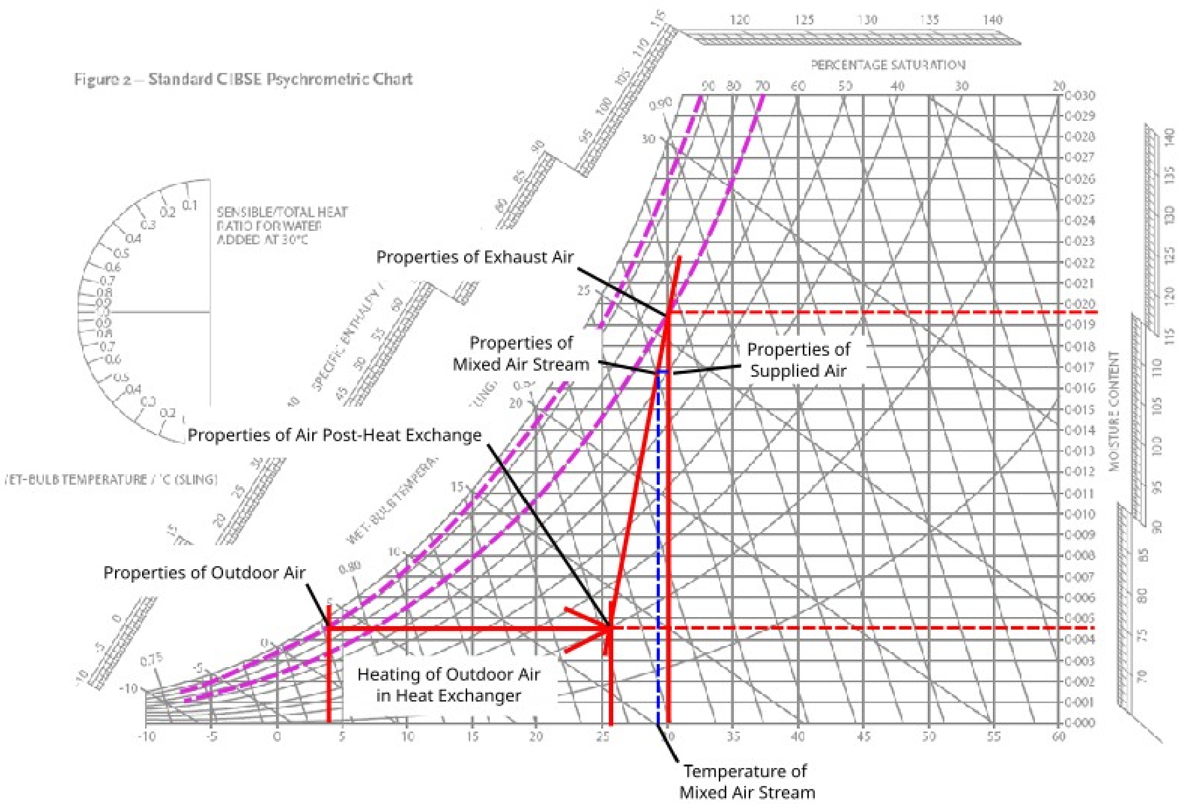


Figure 32: Plot of mixed air stream on a psychrometric chart

6.3.3: Ventilation Losses

The net heat loss from ventilation, or the amount of energy required to heat the mixed air stream from 28.8°C to 30°C, can now be determined numerically by analysing the enthalpy difference between the mixed air and the indoor pool hall air.

The enthalpy of the mixed air is calculated by analysing the sensible and latent heat in the mixed air.

$$h_m = h_a + (x_m \cdot h_w)$$

Where h_a (kJ/kg) is the specific enthalpy of dry air in the mixed air stream, h_w (kJ/kg) is the specific enthalpy of water vapour in the mixed air stream and x_m is the moisture content of the mixed air, as determined from the equations in section 5.4.2. h_a and h_w are calculated from the following equations.

$$h_a = c_{pa} \cdot T_m$$

$$h_w = c_{pw} \cdot (T_m + h_{we})$$

Where T_m (°C) is the temperature of the mixed air, c_{pa} (kJ/kg°C) is the specific heat of air at constant pressure, c_{pw} (kJ/kg°C) is the specific heat of water vapour at constant pressure, and h_{we} (kJ/kg) is the evaporation heat of water at 0°C (Engineering Toolbox). This process is repeated for the exhaust air, and the energy required to heat the mixed air to the required temperature is calculated by the following described in section 5.4.1.

For the mixed air stream,

$$h_{a(m)} = c_{pa(m)} \cdot T_m = 1006 \cdot 28.8 = 28961.2\, J/kg$$

and,

$$h_{w(m)} = c_{pw(m)} \cdot (T_m + h_{we}) = 1.864 \cdot (28.8 + 2501) = 2554.7\, J/kg$$

Therefore, the enthalpy of the supplied mixed air is,

$$h_m = h_{a(m)} + (x_m \cdot h_{w(m)}) = 28961.2 + (0.0138 \cdot 2554) = 28996.4\, J/kg = 29\, kJ/kg$$

Likewise, for the indoor air,

$$h_{a(i)} = c_{pa(i)} \cdot T_i = 1006 \cdot 30 = 30180\, J/kg$$

and,

$$h_{w(i)} = c_{pw(i)} \cdot (T_i + h_{we}) = 1.864 \cdot (30 + 2501) = 2556.9\, J/kg$$

Therefore, the enthalpy of the pool hall air is,

$$h_i = h_{a(i)} + (x_i \cdot h_{w(i)}) = 30180 + (0.0138 \cdot 2556.9) = 30215.2\, J/kg = 30.2\, kJ/kg$$

The total volume flow rate is 35280 m³/h, and so the energy required to heat the air to 30°C is calculated.

$$Q_{av} = \dot{V}_m \cdot \rho_m \cdot (h_i - h_m) = 35280 \cdot 1.269 \cdot (30.2 - 28.3) = 85283\, W = 85.3\, kW$$

6.3.4: Fabric Losses

A site visit and a review of technical documents shared by the council confirmed the fabric types found in the pool hall. The external walls are cavity walls, consisting of two layers of brick either side of an air gap. The windows within the pool hall are double glazed, and the roof is uninsulated deck (South Ayrshire Council). Table 8 summarises the building fabrics, associated U-values and surface areas (designingbuildings.co.uk, Green and Heritage Roofing).

Table 8: U-values and surface area of construction types in the pool zone

Material	U-Value (W/m²K)	Area (m²)
Double Glazing	2.5	293
Brick Cavity Wall	1.5	68
Roof Deck	2.5	1130

The heat loss through the building fabric of the pool is therefore calculated using the equation in section 5.4.4, where the temperature inside the pool hall is 30°C and the ambient air temperature is 4°C.

$$Q_{ac}=((2.5\cdot 293)+(1.5\cdot 68)+(2.5\cdot 1130))\cdot(30-4)$$

$$Q_{ac}=95.2\,kW$$

6.3.5: Total Pool Air Losses

Table 9: Summary of heat loss from the pool air

Heat Loss Description	Heat Loss Value (kW)
Ventilation losses	85
Losses through building fabric	95
TOTAL	180

6.4: Pool Water Losses

6.4.1: Evaporation Losses

The evaporative losses from the pool are highly sensitive to the number of swimmers. The Citadel does not keep record of the exact number of swimmers in the pools at any given time, so the exact occupancy rate at the time of the winter scenario is unknown. Figure 33 shows the effect of occupancy rate on evaporative losses, calculated using the equations presented in section 5.5.1.

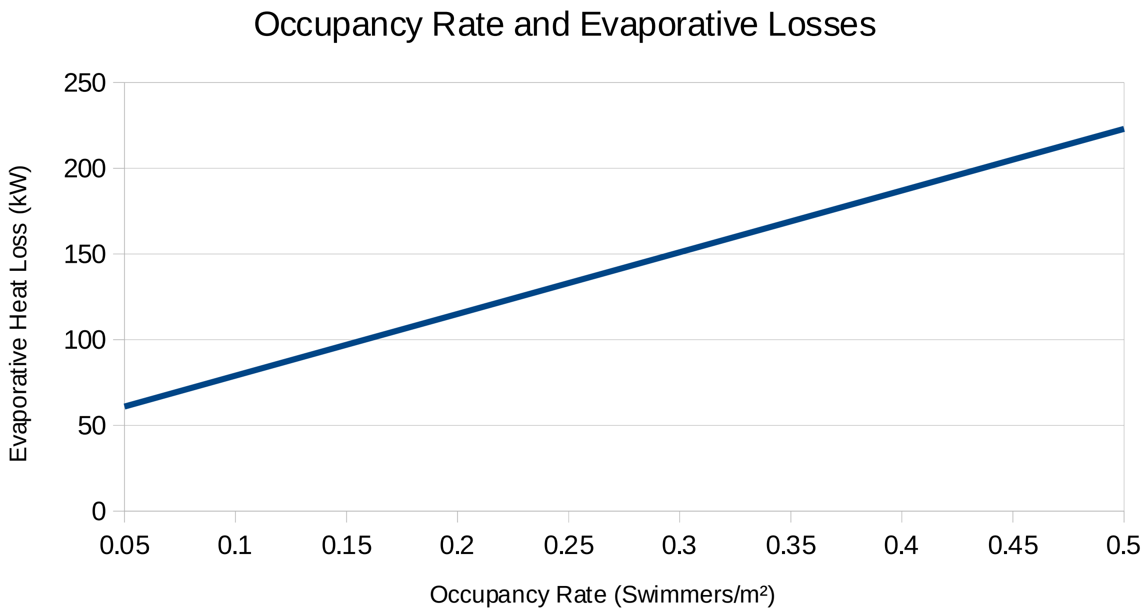


Figure 33: Evaporative heat loss and occupancy rate

Despite this exact value being unknown, a reasonable assumption to make is that there were a minimal number of swimmers using the pool at this early time in the morning. An occupancy rate of 0.05 gives 22 swimmers in the pool, and this is considered to be more realistic for morning patronage.

The minimal occupancy rate results in a mass transfer of water from the pool through evaporation of 0.203 kg/h, which results in heat lost to evaporation of 63.2 kW.

6.4.2: Radiation

The heat transfer between the pool water surface and the roof is determined by the equation described in section 5.5.4.

$$Q_{rd} = A_p \cdot \varepsilon_w \cdot \sigma \cdot \left((T_p + 273)^4 - (T_{sur} + 273)^4 \right)$$

The combined area of the three pools, A_p, is 447 m², the emissivity of water, ε_w, is 1, the Stefan-Boltzmann constant, σ, is 5.67 x 10^{-8} W/m² K⁴, the temperature of the pool, T_p, is 29°C, and the temperature of the roof, T_{sur}, is 30°C.

$$Q_{rd}=447\cdot1\cdot5.67\,x\,10^{-8}\cdot\left((29+273)^4-(30+273)^4\right)=-2806\,W$$

This represents a small gain in heat from the surface of the roof to the water of 2.8 kW.

6.4.3: Refill Losses

The heat loss from the refilling of fresh water to compensate for the loss of water volume due to evaporation is calculated using the equation described in section 5.5.5.

$$Q_{rf}=c_p\cdot\dot{m}_{rf}\cdot(T_p-T_{rf})$$

The specific heat of water, c_p, is 4200 J/kg°C, the mass flow rate of the water is equal to the mass flow of water evaporated from the surface, which was calculated as 0.203 kg/h. The temperature of the pool, T_p, is 29°C. The temperature of the refilling water, T_{rf}, is 17°C.

$$Q_{rf}=4200\cdot0.203\cdot(29-17)=10231\,W=10.2\,kW$$

6.4.4: Total Pool Water Losses

Table 10: Summary of heat loss from the pool water

Heat Loss Description	Heat Loss Value (kW)
Evaporation	63
Radiation	-3
Refilling	10
TOTAL	70

6.5: Demand Split

Based on the methodology, the combined heating demand for the entire pool zone is determined to be 250 kW. When compared with the peak gas demand of the building, 576 kW, the pool zone accounts for 43% of the entire demand of the facility during the winter scenario. Figure 34 illustrates the demand split for the facility.

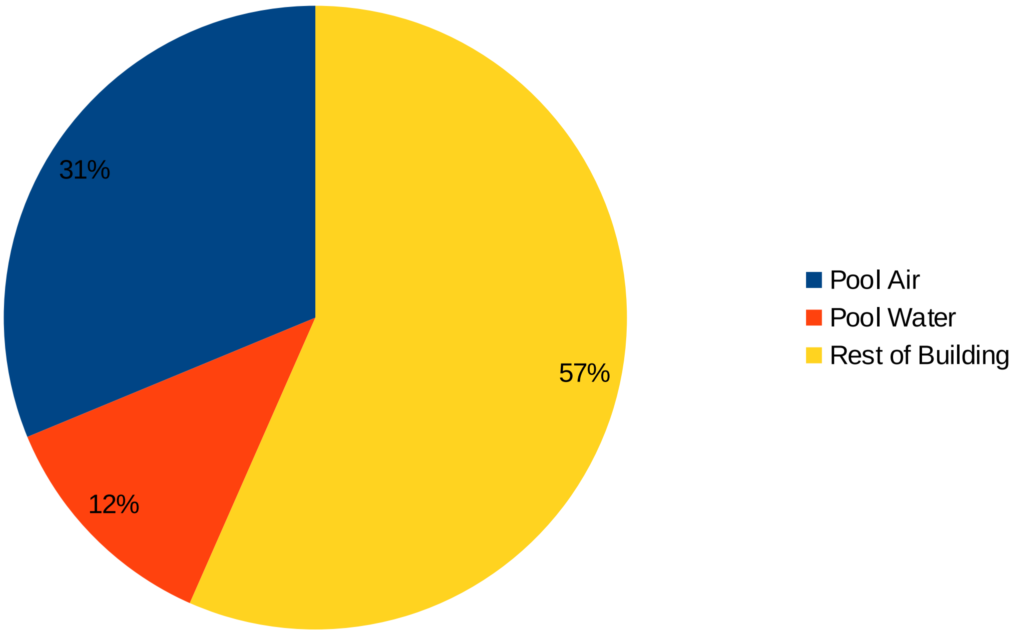


Figure 34: Demand split of the building

The winter scenario case provides a snapshot of the building demand during a time in the morning when the building is opening to the public, and so conventional space heating demand will be larger at this time as the building is brought up to temperature. Pool occupancy is also expected to be low at this time, and occupancy has a large effect on demand as illustrated in section 6.4.1. Therefore, there may be times when the overall demands of the pool hall take a larger share of the entire building demand.

6.6: Heat Pump Sizing

The total thermal capacity of the heat pump system must be large enough to cover the peak demand of the site and have enough reserve capacity to allow for periods of extreme weather conditions and high pool occupancy. It may also be beneficial to retain the existing gas boiler heating system as a redundancy measure.

The maximum evaporative losses from occupancy were calculated to be around 220 kW, and the low ambient temperatures resulted in a heat loss of 180 kW. This would give a total demand of 410 kW. To allow for colder ambient temperatures, an additional 25% capacity will be proposed to cover the demands of the pool zone, resulting in a required thermal capacity of approximately 500 kW_{th}.

For the rest of the building, the peak demand was determined to be 326 kW. Likewise, an additional capacity of 25% will be added for headroom, resulting in a thermal capacity of approximately 400 kW_{th}. The total thermal capacity of the heat pump system will be 900 kW_{th}.

6.7: River Water Abstraction Level

The amount of water to be abstracted from the river to meet the thermal capacity of the heat pump can now be determined. Only a proportion of the total flow of the river can be captured, and the flow rate of river water required for the heat pump can be calculated by rearranging the equation described in section 5.5. *Q* is the thermal capacity of the heat pump, the density of water is 1000 kg/m^3, the specific heat capacity of water is 4200 J/kg°C, and the temperature difference between water abstracted from the river and returned from the heat pump is 2°C.

$$\dot{V} = \frac{Q}{\rho \cdot c_p \cdot \Delta T}$$

$$\dot{m} = \frac{900000}{1000 \cdot 4200 \cdot 2} = 0.107\, m^3/s = 385.7\, m^3/h = 9257.1\, m^3/day$$

As the required level of abstraction is greater than 2000 m^3/day, the heat pump system will require a complex license level of authorisation from SEPA.

6.8: Heat Pump Configuration

With the characteristics of the site's demand determined, a number of different options for the heat pump system can be examined. These options will take into account the existing mechanical systems within the building, operational procedures for the pool hall, and cost of energy for each design option. In all options, the gas boilers are retained as back-up for heat pump downtime or for times of large demand.

6.8.1: Design Option 1 (DO1)

The first design option (Figure 35) is one 900 kW_{th} heat pump providing a flow temperature of 80°C. The main advantage of this system is that it can be easily integrated with the existing distribution and heating systems, including the radiators for conventional space heating, and the heat exchangers used for pool water and air heating. In the figure, 'PZ' represents the heating systems in the pool zone, and 'RoB' represents the space heating and sanitary hot water systems for the rest of the building.

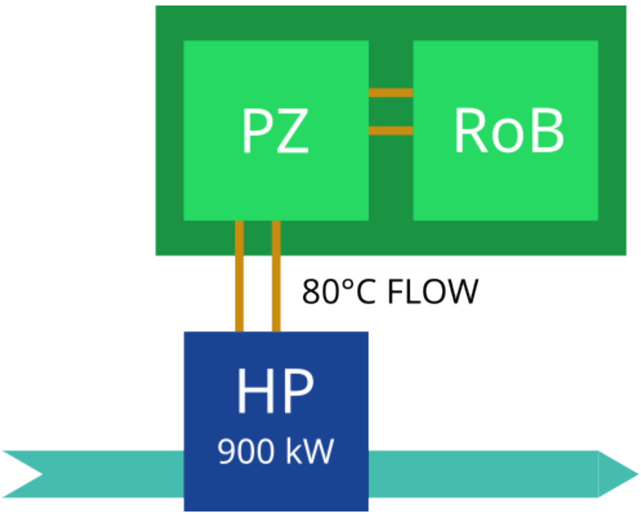


Figure 35: Design Option 1 schematic

6.8.2: Design Option 2 (DO2)

The second design option (Figure 36) consists of two separate heat pumps. The first heat pump serves the pool water and air heating demand, with a thermal capacity of 500 kW_{th}. The second heat pump serves the space heating and sanitary hot water demand for the rest of the building, with a thermal capacity of 400 kW_{th}. The 500 kW_{th} heat pump will supply water to the pool heating systems at 50°C, while the 400 kW_{th} heat pump will supply water at 80°C. This serves to shift a large portion of the demand to a low flow temperature, taking advantage of the higher energy efficiency of a low temperature heat pump while continuing to supply the rest of the building with 80°C heating water.

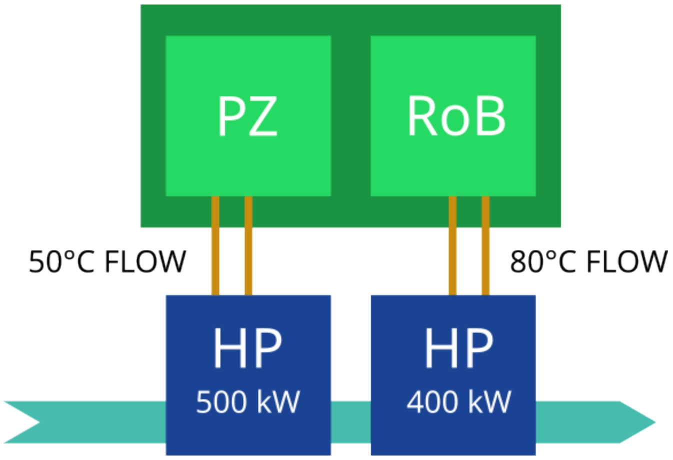


Figure 36: Design Option 2 schematic

Supplying the conventional heating systems at high temperature removes the need to upgrade the radiators and other systems within the main building which is costly.

The existing pool water heating system is showing signs of wear and tear with age. Several of the pumps are non-operational, some of the pipework has been temporarily routed through modular polymer piping and the various concrete plinths support frames for the pumps and heat exchangers are in poor condition. Therefore, as there will be some future cost in replacing equipment in the plant room, there may be an opportunity to make the system fit for use with a low temperature distribution without excessive cost of upgrading existing functional equipment.

Likewise, the replacement of heat exchangers for the pool water and air heating systems to accommodate a low temperature distribution system would be minimal in comparison to the replacement of radiators and other equipment for the conventional space heating and hot water system.

6.8.3: Design Option 3 (DO3)

The third design option (Figure 37) is a hybrid system which retains the gas boilers for conventional heating but employs a 500 kW_{th} thermal capacity heat pump with a 50°C flow temperature for all heating in the pool zone. While this does not fully decarbonise heating supply, it does shift the pool zone to heat pump technology, and as this is a considerable portion of the total demand, this would allow for considerable carbon savings at a lower cost than installing a complete system for the entire building. This would also allow the gas system to be phased out with alternative technology in the future, such as an air-source heat pump with solar thermal. Much like design option 2, the heating systems for the pool zone are more suitable for the adoption of a 50°C flow temperature.

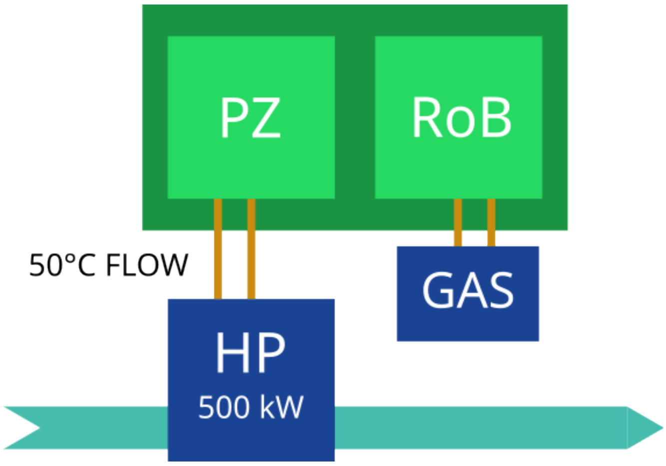


Figure 37: Design Option 3 schematic

6.9: Heat Pump Performance

The coefficient of performance (COP) for each system design option is calculated using an excel spreadsheet. The monthly sea temperatures from 2012 shown in section 5.8.2 are used to determine the average Carnot COP for each month in the year. A realistic value is obtained by using 60% of the Carnot value, as illustrated in the following equation.

$$COP_{Carnot}=\frac{T_{hot}}{(T_{hot}-T_{cold})}\cdot 60\,\%$$

Where T_{hot} is the target flow temperature for the heat pump (K), and T_{cold} is the temperature of the river water (K). The COP for each system over the course of the year is plotted for comparison (Figure 38).

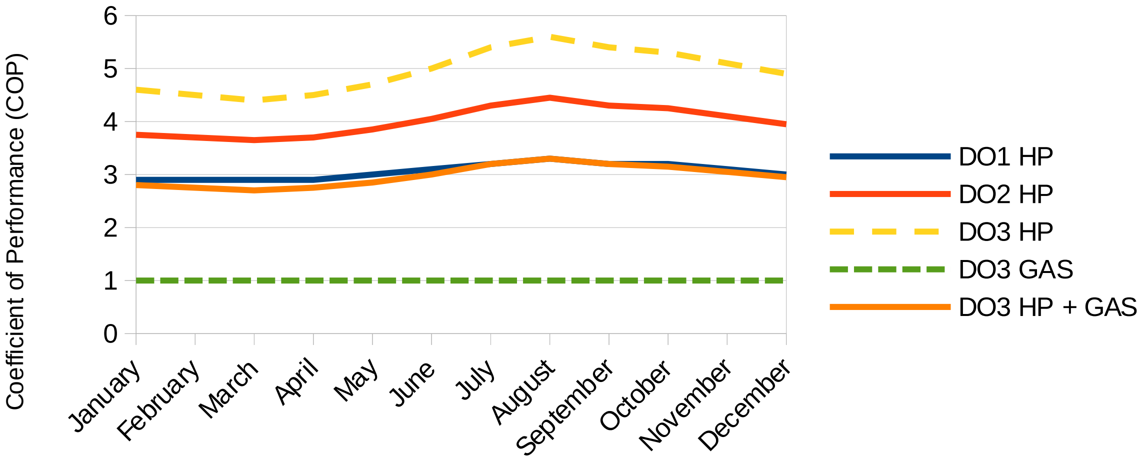


Figure 38: Seasonal COP for design options

As expected, the low temperature heat pumps exhibit a higher COP compared with the high temperature heat pump. For design option 2, which makes use of separate high temperature and low temperature heat pumps, the average COP of the two systems is plotted. For design option 3 (DO3), the efficiency of the heat pump and gas boiler are plotted separately, illustrating the COP of the low temperature heat pump itself. The gas boiler is considered to be 100% efficient, as discussions with the mechanical engineer on site confirmed this to be the case.

The combined COP of the low temperature heat pump and boiler of DO3 exhibits a similar COP to the single 850 kW high temperature heat pump of DO1. The separate high and low temperature heat pumps of DO2 offer the best efficiency of the three options, peaking at 4.5 in August. The total units of energy used for heating in each design option are shown in table 11.

Table 11: Energy use for each design option

	Base (kWh)	DO1 (kWh)	DO2 (kWh)	DO3 (kWh)
Electricity	-	226927	173208	70854
Gas	687013	-	-	343507
Total	**687013**	**226927**	**173208**	**414360**

The base case assumes a boiler efficiency of 100% and is taken directly from the real monitored data provided by SAC. Design option 2, as expected, uses the least units of energy of all three design options with a 75% reduction in energy use for heating. Design option 3 results in the least reduction in energy use for the year as a significant portion of the heating is still provided by the gas boiler system, however, this still represents a 40% reduction in energy use for heating.

6.10: Energy Cost

Table 12 shows how the energy use of each system design option translates to cost of energy for heating, based on 2023 prices. Due to the higher cost of electricity when compared with gas, any heat pump system must vastly reduce energy required for heating. This is illustrated by design option 1, which presents a 27% increase in energy costs despite it's 67% reduction in energy usage. Design option 2 presents marginal cost savings, however this will depend on the comparative cost of gas and electricity year on year. Design option 3 presents the best cost savings due to retaining gas for a portion of the heating required.

Table 12: Annual energy cost (based on 2023 prices)

	p/kWh	Base (£)	DO1 (£)	DO2 (£)	DO3 (£)
Electricity	28.62	-	64946.36	49572.14	20278.31
Gas	7.42	50976.39	-	-	25488.19
TOTAL		**50976.39**	**64946.36**	**49572.14**	**45766.50**
Change (±%)			**+27%**	**-3%**	**-10%**

6.11: Carbon Emissions

Based on 2024 conversion factors (CF), table 13 shows the carbon savings compared to base case for each design option. All options result in a significant decrease in CO_2 emissions.

Table 13: Annual carbon emissions (based on 2024 conversion factors)

	CF (kg CO_2e/kWh)	Base (kg CO_2e/kWh)	DO1 (kg CO_2e/kWh)	DO2 (kg CO_2e/kWh)	DO3 (kg CO_2e/kWh)
Electricity	0.22499	-	51056	38970	15941
Gas	0.18293	125675	-	-	62838
TOTAL		**125675**	**51056**	**38970**	**78779**
Change (±%)			**-59%**	**-69%**	**-37%**

The reduction in energy usage afforded by a heat pump is the main driver in CO_2 emission reductions. As things stand, electricity production is considered to be more carbon intensive than natural gas production, however, with the trend towards decarbonisation of electricity supply (Department for Energy Security and Net Zero, 2023), the carbon savings of electric heat pumps will continue to increase.

6.12: System Comparison

To summarise the performance of each system design option, figure 39 plots the percentage change from base case for each scenario in terms of energy use, cost and carbon emissions. A reduction is represented by a negative percentage value, and an increase is represented by a positive percentage value.

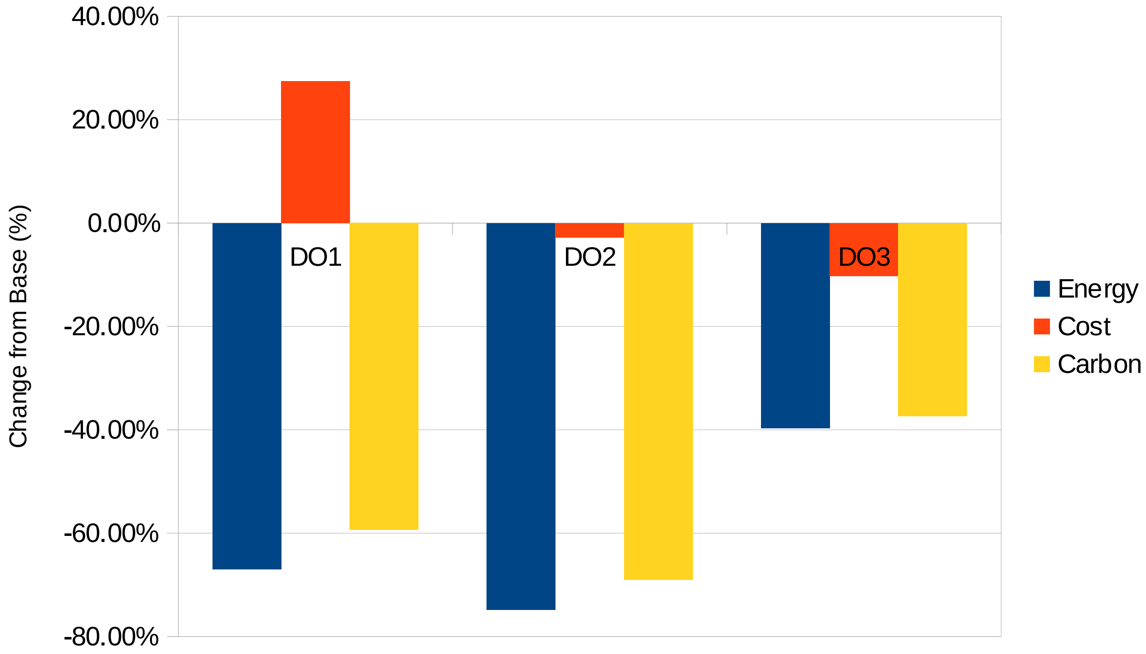


Figure 39: System comparison

All options exhibit a large reduction in both energy use and CO_2 emissions, but the current high cost of electricity compared to gas makes DO1 to appear a less attractive proposition. Each design option is a trade-off between decarbonisation goals and cost. In discussions with council employees, there was a concern that a low flow temperature will increase the time it will take to heat the pool from a complete refill from two days to four days, however it was noted that this procedure has not been carried out for at least ten years. While a high flow temperature results in a greater cost of energy, there is an opportunity to expand this system to other SAC buildings close by which would also be unsuitable for conversion of pipework and radiators to take a lower flow temperature. Design option 3 represents an intermediate step in building decarbonisation, but still delivers substantial reductions in energy use and CO_2 emissions. The existing gas boiler system itself is highly efficient and modern, and therefore it may make sense to utilise it for as long as possible until it is required to be replaced, at which point another solution can be found for the rest-of-building demand. Table 14 summarises the advantages and disadvantages of each system design option.

Table 14: Design option advantages and disadvantages

Design Option 1	Design Option 2	Design Option 3
+ No need for existing heating systems to be modified + No change in time required to heat pool from start-up + High flow temperatures can be utilised for heating nearby legacy buildings	+ Offers highest efficiency of complete HP options + Offers largest reduction in CO_2 emissions relative to base + Largest reduction in overall energy use compared to base	+ Largest reduction in energy costs relative to base + Shifts majority of demand to low carbon heating with minimal capital cost in the short term + Offers an immediate reduction in energy costs at current wholesale prices
- Increased cost of energy at current wholesale prices - Lowest efficiency of complete HP options	- Requires modification to existing pool zone heating systems - No immediate substantial reduction to energy costs at current wholesale prices	- Further decarbonisation will require investment in future - Requires modification to existing pool zone heating systems

7: Conclusions

The main purpose of this report was to assess the feasibility of decarbonising the heating supply of the Citadel leisure centre by installing a river source heat pump system. The results gained from the developed methodology allow for some conclusions to be drawn.

7.1: Energy in River

Firstly, there is an abundance of exploitable heat in the river Ayr. From the data obtained, it is found that abstracting around 10% of the volume flow rate of the River Ayr can provide enough energy to provide heating to the pool water, pool air, conventional space heating and sanitary hot water for the Citadel.

7.2: Demand Split & Design Options

Characterisation of the demand of the building through analysis of the heat losses in the pool zone confirmed that the pool water and pool air heating combined were the dominant energy demands of the entire building. The pool zone demands were also found to be highly sensitive to occupant levels within the pool itself.

As it is not possible, for cost reasons, to replace heat emitters and pipework to make compatible the conventional space heating demands, a high flow temperature is required from any future heat pump system. The pool zone is a different situation. The equipment in the pool water heating plant room is showing its age, and so there is an opportunity to replace the pipework and plate heat exchangers, of which there are three, to make compatible for a lower flow temperature. Similarly, the ventilation system requires the replacement of just one heating coil to allow for a low temperature system to be installed. Therefore, the dominant demand can make use of the higher efficiencies provided by a 50°C flow temperature from the heat pump.

7.3: Suitability of Site

The Citadel was found to be ideally located for a river source heat pump system. Not only is the centre close to the water body, but there is a clear path from the building to the quay wall which would limit disruption in the case of excavation to lay pipework. Depending on the size of the chosen system, there are multiple options for the construction of the plant buildings which house the heat pump, whether this be located on the quay wall itself or located next to the building in the existing car park.

7.4: Heat Pump Performance

In all cases, addition of a heat pump results in a large reduction in energy use and CO_2 emissions, even in the case where the gas boiler is retained for conventional space heating and sanitary hot water demands. Design option 2, in particular, performs the best of the three options in terms of decarbonisation. For a complete heat pump system, efficiency must be maximised to make any real cost savings compared to the existing situation. Design option 3, where a heat pump is used only for the pool zone, is therefore attractive in terms of cost.

7.5: Future Work

Throughout the development of the methodology and engagement with stakeholders involved with SAC and the operation of the Citadel, some areas of future work and research related to this project were identified.

7.5.1: Localised River Monitoring

The river characteristics used in the methodology were obtained from different sources. The river flow data was obtained from a monitoring station around 3 km from the proposed site, while the water temperatures used were averages recorded in the wider Clyde Estuary. A lack of real data for the characteristics of the river flow at the proposed site was a limiting factor in the analysis of the energy available in the river. Any future study would benefit from a localised monitoring of the river at the proposed site using data loggers.

7.5.2: Fabric-First Approach

This report was mainly concerned with the performance of a heat pump system within the context of the existing building with planned limited fabric improvements to improve wind and water tightness. Further energy efficiency measures relating the building fabric could be investigated first to reduce the energy demands of the building.

7.5.3: Heat Pump Detailed Design

An investigation into the detail design of any future heat pump system will be required beyond any feasibility study. This report has indicated that such a system is feasible but is ultimately limited in scope. Creation of more detailed design concepts for each heat pump system would allow for an analysis of capital cost of construction and future operational costs. Other factors, such as the choice of refrigerant used in the heat pump, may also be analysed.

7.5.4: District Heating Applications

This report was closely focused on the feasibility of providing heating for the Citadel building only, but some discussions indicated an interest in using a river source heat pump as part of a wider district heating scheme for the area, including the Citadel. An analysis of the heating demands for adjacent domestic properties and public buildings could be carried out, and the implementation of a higher capacity heat pump system could be advantageous for SAC decarbonisation plans.

Bibliography

ACR Journal. (2021). *CIAT heat pumps harness river power.* Available at: https://www.acrjournal.uk/heat-pumps/ciat-heat-pumps-harness-river-power/ [Accessed 26/05/2024].

American Society of Heating, R.A.-C.E. (2020). *2020 ASHRAE handbook : hvac systems and equipment.* SI edition.. Atlanta: Ashrae.

Ayrshire Growth Deal. (2023). *Ayrshire Energy Masterplan.* Available at: https://www.ayrshiregrowthdeal.co.uk/ayrshire-energy-masterplan/ [Accessed 22/05/2024].

Carbon Trust. (2008). *Swimming Pool Guide*

Carroll, P., Chesser, M., Lyons, P. (2020). *Air Source Heat Pumps field studies: A systematic literature review.* Renewable and Sustainable Energy Reviews 134, 110275. https://doi.org/10.1016/j.rser.2020.110275

CIBSE. (2012). *KS20 Practical Psychrometry : KS20.* London: CIBSE Chartered Institute of Building Services Eng.

CIBSE. (2013). *PC01 CIBSE Psychrometric Chart (-10 to +60 degree C). London: CIBSE Chartered Institute of Building Services Eng.*

Costwold Energy Group. (2023). *Crystal Leisure Centre: Open Loop (Borehole) Water Source Heat Pump.* Available at: https://cotswold.energy/case-study/crystal-leisure-centre/ [Accessed 02/06/2024].

Department for Energy Security and Net Zero. (2023). *Greenhouse gas reporting: conversion factors 2023*. Available at: https://www.gov.uk/government/publications/greenhouse-gas-reporting-conversion-factors-2023 [Accessed 30/07/2024]

Designing Buildings. (2023). *U-Values.* Available at: https://www.designingbuildings.co.uk/wiki/U-values [Accessed 12/07/2024]

Dudley Metropolitan Borough Council. (2021). *New green heating system for Crystal leisure centre*. Available at: https://councilnews.dudley.gov.uk/news/new-green-heating-system-for-crystal-leisure-centre [Accessed 02/06/2024].

Fluid Mechanics Ltd. (2020). *Air Source Heat Pump Coefficient of Performance.* Available at: https://www.fluidmechanics.co.uk/near-isothermal-stirling-heat-pump-technology/ [Accessed 07/06/2024].

Gonzalez Miguel, J. (2014). *Thermal Systems for Indoor Pools Utilising Heat Recovery from Exhaust Air.*

Govaer, D. & Zarmi, Yair. *(1981). Analytical evaluation of direct solar heating of swimming pools.* Solar Energy. 27. 529-533. 10.1016/0038-092X(81)90047-5.

Green and Heritage Roofing. (2023). *U-Values Explained.* Available at: https://www.greenandheritage.uk/roofing-tips/u-values-explained/ [Accessed 12/07/2024]

GreenMatch. (2024). *Ground Source Heat Pumps in the UK: A Complete Guide.* Available at: https://www.greenmatch.co.uk/ground-source-heat-pump [Accessed 03/06/2024].

Health and Safety Executive. (2018). *HSG179 Health and safety in swimming pools.*

Heat Pumps UK. (2023). *What Size Air Source Heat Pump Do I Need For My Home?.* Available at: https://heat-pumps.org.uk/what-size-air-source-heat-pump-do-i-need-for-my-home/ [Accessed 02/06/2024].

Industrial Heat Pumps NL. (2024). *Coefficient of Performance.* Available at: https://industrialheatpumps.nl/english/operating_principle/cop_heat_pump/ [Accessed 03/06/2024].

ITP Energised. (2023). *New UK Grid Emissions Factors 2023.* Available at: https://www.itpenergised.com/new-uk-grid-emissions-factors-2023/ [Accessed 29/07/2024].

Kensa Heat Pumps. (2024). *Air Source vs. Ground Source Heat Pumps.* Available at: https://www.kensaheatpumps.com/air-source-vs-ground-source-heat-pumps/ [Accessed 06/06/2024].

Kensa Heat Pumps. (2024). *What is a Water Source Heat Pump?.* Available at: https://www.kensaheatpumps.com/water-source-heat-pump/ [Accessed 06/06/2024].

Mančić, M.V., Živković, D.S., Milosavljević, P.M., Todorović, M.N. (2014). *Mathematical modelling and simulation of the thermal performance of a solar heated indoor swimming pool.* Thermal Science 18, 999–1010.

Marine Scotland. (2013). *Monthly average sea surface temperature for 13 Scottish Sea Areas.* Available at: https://data.marine.gov.scot/dataset/scottish-coastal-observatory-data/resource/7d3483aa-c8ad-4652-91fc-b0ca84d496b7#{} [Accessed 10/07/2024].

Mitchell, M.S., Spitler, J.D. (2013). *Open-loop direct surface water cooling and surface water heat pump systems—A review*. HVAC&R Research 19, 125–140. https://doi.org/10.1080/10789669.2012.747374

National Grid ESO. (2024). *Carbon Intensity Dashboard.* Available at: https://www.nationalgrideso.com/future-energy/our-progress-towards-net-zero/carbon-intensity-dashboard [Accessed 20/06/2024].

National River Flow Archive. (2023). *83006 - Ayr at Mainholm.* Available at: https://nrfa.ceh.ac.uk/data/station/info/83006 [Accessed 24/06/2024].

Pool Water Treatment Advisory Group. (2021). *Code of Practice: The Management and Treatment of Swimming Pool Water*

Ricardo. (2023). *South Ayrshire Council LHEES Technical Report*

Schibuola, L., Scarpa, M. (2016). *Experimental analysis of the performances of a surface water source heat pump.* Energy and Buildings 113, 182–188. https://doi.org/10.1016/j.enbuild.2015.12.048

Scottish Environment Protection Agency. (2011). *The Water Environment (Controlled Activities) (Scotland) Regulations 2011 (as amended): A Practical Guide.*

Scottish Environment Protection Agency. (2024). *Water Source Heat Pump Guidance.*

Scottish Government. (2022). *Local Heat and Energy Efficiency Strategies and Delivery Plans Guidance*

South Ayrshire Council. (2019). *Planning Application Documents: 19/00971/APP | Erection of flatted development | Former Ice Factory South Harbour Street Ayr South Ayrshire KA7 1JB.* Available at: https://publicaccess.south-ayrshire.gov.uk/online-applications/applicationDetails.do?activeTab=documents&keyVal=Q06GFKBDKFM00 [Accessed 25/07/2024].

South Ayrshire Council. (2023). *Citadel Net-Zero Retrofit Feasibility Study.* Unpublished internal document.

South Ayrshire Council. (2023). *Strategic Housing Investment Plan 2024/25 - 28/29.*

South Ayrshire Council. (2024). *Evaluation of Low/Zero Carbon Technologies.* Unpublished internal document.

Star Refrigeration. (2020). *Star Renewable Energy provide Queens Quay with the largest water source heat pump for district heating in Scotland.* Available at: https://www.star-ref.co.uk/wp-content/uploads/2020/04/queens-quay-water-source-heat-pump.pdf [Accessed 08/06/2024].

The Engineering ToolBox. (2004). *Moist Air - Enthalpy* . Available at: https://www.engineeringtoolbox.com/enthalpy-moist-air-d_683.html [Accessed 11/07/2024].

The Engineering ToolBox. (2010). *Water - Heat of Vaporization vs. Temperature*. Available at: https://www.engineeringtoolbox.com/water-properties-d_1573.html [Accessed 18/07/2024].

Tim Smith Heat Exchangers. (2019). *Swimming Pool Heating.* Available at: https://www.tlsheatexchangers.co.uk/heat-exchangers/swimming-pool-heating/ [Accessed 15/06/2024].

Tuohy, P. (2008). *Energy Systems Analysis.*

UK Government. (2021). *Heat and Buildings Strategy.*

Wang, Y., Wang, J., He, W. (2022). *Development of efficient, flexible and affordable heat pumps for supporting heat and power decarbonisation in the UK and beyond: Review and perspectives.* Renewable and Sustainable Energy Reviews 154, 111747. https://doi.org/10.1016/j.rser.2021.111747

Wu, R. (2009). *Energy Efficiency Technologies – Air Source Heat Pump vs. Ground Source Heat Pump.* Journal of Sustainable Development, Vol 2, No. 2.