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Solar-thermal combined energy systems for distributed small-scale applications

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A thesis submitted for the degree of
Doctor of Philosophy
June 2017

Statement of Originality

I hereby certify that, to the best of my knowledge, the content of this thesis is original and the results of my own work. All of the assistance I received and the sources I used have been properly acknowledged.

James Freeman

26 June 2017

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Publications

To date these are the publications that have been presented to the public domain in relation to the works herein.

- Freeman, J., Guarracino, I., Kalogirou, S. A., & Markides, C. N. (2017). A small-scale solar organic Rankine cycle combined heat and power system with integrated thermal energy storage. *Applied Thermal Engineering*, 127, 1543–1554.
- Freeman, J., Hellgardt, K., & Markides, C. N. (2017). Working fluid selection and electrical performance optimisation of a domestic solar-ORC combined heat and power system for year-round operation in the UK. *Applied Energy* 186(3), 291–303.
- Freeman, J., Hellgardt, K., & Markides, C. N. (2015). An assessment of solar-thermal collector designs for small-scale combined heating and power applications in the United Kingdom. *Heat Transfer Engineering*, 36(14–15), 1332–1347.
- Freeman, J., Hellgardt, K., & Markides, C. N. (2015). An assessment of solar-powered organic Rankine cycle systems for combined heating and power in UK domestic applications. *Applied Energy*, 138, 605–620.
- Freeman, J., Ramos, A., MacDowell, N., & Markides, C. N. An experimentally validated model of a solar-cooling system based on an ammonia-water diffusion-absorption cycle. *8th International Conference on Applied Energy*. October 8–11, 2016, Beijing, China.
- Guarracino, I., Freeman, J., Ekins-Daukes, N., & Markides, C. N. Performance assessment and comparison of solar ORC and hybrid PVT systems for the combined distributed generation of domestic heat and power. *12th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics*. July 11–13, 2016, Costa del Sol, Spain.
- Freeman, J., Guarracino, I., Kalogirou, S. A., & Markides, C. N. A small-scale solar organic Rankine cycle combined heat and power system with integrated thermal energy storage. *7th Heat Powered Cycles Conference*. June 26–29, 2016, Nottingham University, UK.

- Guarracino, I., Freeman, J., & Markides, C. N. Experimental evaluation of a 3-D dynamic solar-thermal collector model under time-varying environmental conditions. *29th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*. June 19–23, 2016, Portoro, Slovenia.
- Freeman, J., Hellgardt, K., & Markides, C. N. Optimisation of a domestic-scale solar-ORC heating and power system for maximum power output in the UK. *3rd International Seminar on ORC Power Systems*. October 12–14, Brussels, Belgium.
- Freeman, J., Hellgardt, K., & Markides, C. N. Working fluid selection and performance optimisation of a domestic-scale solar organic Rankine cycle combined heat and power system for year-round operation in the UK. *3rd International Sustainable Thermal Energy Management Conference*. July 7–8, 2015, Newcastle, UK
- Freeman, J., Hellgardt, K., & Markides, C. N. Towards the optimisation of a domestic-scale solar organic Rankine cycle system for combined heating and power provision in the United Kingdom. *2nd International Seminar on ORC Power Systems*. October 7–8, 2013, Rotterdam, Netherlands.
- Freeman, J., Hellgardt, K., & Markides, C. N. An assessment of solar-thermal collector designs for small-scale combined heating and power applications in the UK. *13th UK Heat Transfer Conference*. September 2–3, 2013, Imperial College London, UK.
- Freeman, J., Hellgardt, K., & Markides, C. N. An assessment of a solar-powered organic Rankine cycle system for combined heating and power in the domestic applications. *5th International Conference on Applied Energy*. July 1–4, 2013, University of Pretoria, SA.

Abstract

Solar power has the potential to provide a significant amount of the global energy requirement for power, heating, cooling and fresh water production. Solar thermal power systems based on steam Rankine cycles tend to be feasible for large-scale centralised power plants, with concentrating collectors generating temperatures of 400 °C or higher. This limits operation to locations that receive focusable direct beam radiation, uninterrupted by cloud cover. On the small scale, photovoltaic (PV) systems dominate the domestic solar market, with solar to electrical efficiencies of around 15%. Compared to PV, solar thermal systems are able to make use of a larger proportion of the solar resource through recovery of useful heat for thermal processes. In this thesis the potential of a small-scale solar combined heat and power (S-CHP) system based on an Organic Rankine Cycle (ORC) is investigated, for the combined provision of heating and power for domestic applications. The system consists of a solar collector array of total area equivalent to that available on the roof of a typical UK home, an ORC engine featuring a generalised positive-displacement expander, and a hot water storage tank. Water heating is provided through the precooling of the superheated exhaust vapour from the ORC expander, and an optional further contribution from the solar collector array.

A thermo-economic model is developed in order to evaluate the performance and cost of the S-CHP system under various configurations and climatic conditions. Concentrating and non-concentrating solar collector designs suitable for small-scale applications are compared by way of annual maximum work assessments based on reversible and endoreversible analyses. The thermo-economic model is used to perform a parametric analysis of the S-CHP system to determine the optimum operational settings, and to simulate the system performance over an annual period. For the system simulated with non-concentrating collectors, electrical outputs equivalent to 19% and 33% of the typical household annual demand are predicted in the climates of London (UK) and Larnaca (Cyprus) respectively. In addition, the ability to provide up to 28% (London) and 58% (Larnaca) of annual hot water demand through recovery of rejected heat from the ORC engine is also predicted. A hot-water prioritisation mode is also identified for which hot water output is increased, at the expense of a decrease in electrical output. A cost analysis is performed in order to compare the system with other solar heat and power systems. The whole system capital cost is €9,700 – €10,500 (£8,000 – 8,800). The cost per unit average electrical output is higher than that for PV-based systems, however when compared to hybrid

PV-thermal (PVT) systems, the total capital cost is found to be lower, and the potential to match local demand by varying the proportional outputs of electricity and hot water is seen as an additional advantage for the S-CHP system.

Two experimental apparatus were designed and constructed for the purpose of evaluating the S-CHP system model. The first was an outdoor solar collector testing facility to measure solar collector efficiencies under a range of operating conditions. Steady-state tests were performed on an evacuated tube collector at fluid inlet temperatures up to 112 °C. This exceeds the normal range of operation in water-heating applications but is closer to the optimal temperatures found in the maximum work analysis. The results were used to provide a more reliable model of collector performance at higher temperatures for the purposes of the S-CHP system modelling. Dynamic tests were also performed in which the collector's response to a step change in solar irradiance was measured. The results were used to determine the collector's effective thermal capacity, and to evaluate a range of dynamic solar collector models. The experimentally obtained thermal capacity value was found to provide improved predictions of dynamic collector behaviour when implemented into a lumped dynamic model, resulting in a $\sim 35\%$ reduction in modelling error; while conventional estimation methods for the effective thermal capacity were found to result in significant under-predictions of the collector's thermal response time and an over-prediction of the daily energy yield.

An experimental ORC apparatus was also designed and constructed in order to evaluate the performance of small-scale ORC components. The system featured a 1 kW_e scroll expander and the working fluid R245fa. Thermal input was provided by an 18 kW_{th} electrical oil heater. Measurements of electrical power output were taken over a range of thermal input settings and working fluid mass flow-rates, while expander rotational speed was also adjusted by varying the electrical load. The experimental results were compared to the predictions from the ORC system model and the performance of the various system components were investigated. The ORC system model was found to over-predict the gross thermal efficiency by 30–90%, attributed to off-design operation of the expander during the experiments and to thermal losses from components such as the heat exchanger, expander and pipework that were not included in the model. A maximum gross thermal efficiency of 5.3% and an expander isentropic efficiency of 49% were obtained during testing.

Acknowledgements

This work was supported by the UK Engineering and Physical Sciences Research Council (EPSRC) [grant number EP/K502856/1]. Data supporting this thesis can be obtained on request from cep-lab@imperial.ac.uk.

I would like to express my sincerest thanks to my supervisors, Dr. Christos Markides and Prof. Klaus Hellgardt, for the faith they have shown in me and for the support and expertise they have provided over the last four years.

My thanks also to the members of the Clean Energy Processes Group and other individuals who have provided valuable help and advice. In particular, the following fantastic people: Ilaria Guarracino, Aly Taleb, Niccolò Le Brun, Oyeniya Oyewumni, Richard Mathie, Ivan Zadrazil, Paul Sapin, Ajay Gupta, María Herrando, Jae Sik An, Alex Charogiannis, Christoph Kirmse, Alba Ramos, Martin White, Steven Lecompte, Marianna Chatzopoulou, Roberto Ibarra, Solomos Georgiou, Alex Mellor, Soteris Kalogirou, Paul Fennell, Jerry Heng, Steve Owen, Richard Wallington, Andrew Macey, Sarah Payne, Louise Le Cornu, Sara McCallum, Robert Edwards, Michael Reid, Jonathan Werner, Joseph Juan and Marc Giménez.

Finally, I would like to gratefully acknowledge the DTA studentship awarded to me by the Department of Chemical Engineering at Imperial College London, without which I would not have been able to embark on this research.

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Nomenclature

Acronyms

ARC	Absorption refrigeration cycle	PVT	Photovoltaic-thermal
CCGT	Combined cycle gas turbine	S-CHP	Solar combined heating and power
CCHP	Combined cooling, heating and power	SIC	Specific investment cost
CHP	Combined heat and power	VCC	Vapour compression (refrigeration) cycle
DAR	Diffusion absorption refrigeration	Variables	
DVG	Direct vapour generation	A	Area, [m ²]
PCM	Phase change material	C	Heat capacity, [J K ⁻¹]
EFPC	Evacuated flat plate collector	c	Specific heat capacity, [J kg ⁻¹ K ⁻¹]
ERC	Ejector refrigeration cycle	c_1	First-order coefficient of the collector efficiency, [W m ⁻² K ⁻¹]
ETC	Evacuated tube collector	c_2	Second-order coefficient of the collector efficiency, [W m ⁻² K ⁻²]
FIT	Feed-in tariff	c_3	Wind speed dependent heat loss coefficient, [J m ⁻³ K ⁻¹]
FPC	Flat plate collector	c_4	Sky temperature dependent heat loss coefficient, [J m ⁻³ K ⁻¹]
LCOE	Levelised cost of energy/electricity	c_5	Effective thermal capacity, [J m ⁻² K ⁻¹]
LMTD	Log-mean temperature difference		
ORC	Organic Rankine cycle		
PEC	Purchased equipment cost		
PTC	Parabolic trough collector		

c_6	Wind dependent zero loss efficiency, [s m ⁻¹]	p	Pressure, [Pa]
		p	Weighting factor, [-]
D	Demand provision, [%]	Q	Heat, [J]
D	Diameter, [m]	$\dot{Q}$	Heating rate, [W]
d	Discount rate, [-]	r_p	Pressure ratio, [-]
E_L	Longwave irradiance, [W m ⁻²]	r_v	Volume ratio, [-]
F	Dimensionless factor, [-]	$\dot{S}$	Entropy flow-rate, [W K ⁻¹]
f	Friction factor, [-]	s	Specific entropy, [J kg ⁻¹ K ⁻¹]
F'	Collector efficiency factor [-]	T	Temperature, [°C]
G	Solar Irradiance, [W m ⁻²]	t	Time, [s]
H	Height, [m]	U	Overall heat transfer coefficient, [W m ⁻² K ⁻¹]
$\dot{H}$	Enthalpy flow-rate, [W]	u	Speed, [m s ⁻¹]
h	Specific enthalpy, [J kg ⁻¹]	V	Volume, [m ³]
h	Local heat transfer coefficient, [W m ⁻² K ⁻¹]	$\dot{V}$	Volumetric flow-rate, [m ³ s ⁻¹]
h	hour angle, [rad]	W	Work, [W]
i	Inflation rate, [-]	$\dot{W}$	Power, [W]
J	Cost, [€]	X	Exergy, [J]
K_θ	Incident angle modifier, [-]	$\dot{X}$	Exergy flow-rate, [W]
L	Latitude, [°] or [rad]	Z_s	Surface azimuth angle, [rad]
L	Length, [m]	Greek Symbols	
m	Mass, [kg]	α	Absorption coefficient, [-]
$\dot{m}$	Mass flow-rate, [kg s ⁻¹]	β	Inclination angle, [°] or [rad]
n_p	Payback period, [years]	β	Influence coefficient, [-]
N_{rot}	Rotational speed, [s ⁻¹]	δ	Declination angle, [rad]

δ	Thickness, [m]	4	ORC expander outlet/condenser inlet
ε	Effectiveness, [-]		
ε	Emissivity, [-]	a	Air
η	Efficiency, [-]	ab	Absorber
θ	Incidence angle, [°] or [rad]	ann	Annual
λ	Thermal conductivity, [W m ⁻¹ K ⁻¹]	aper	Aperture
μ	Dynamic viscosity [Pa s]	app	Approach
ν	Degree of thermodynamic perfection, [-]	atm	Atmospheric
ρ	Density, [kg m ⁻³]	aux	Auxiliary
σ	Stefan-Boltzmann constant, [W m ⁻² K ⁻⁴]	ava	Available
τ	Time constant [s]	b	Beam
$(\tau\alpha)$	Transmittance-absorptance product [-]	b	Bubble point
Φ	Solar zenith angle, [rad]	cap	Capital

Dimensionless Variables

Nu	Nusselt number, $\frac{hD}{\lambda}$
Pr	Prandtl number, $\frac{c_p\mu}{\lambda}$
Re	Reynolds number, $\frac{\rho u D}{\mu}$

Subscripts

1	ORC condenser outlet/pump inlet
2	ORC pump outlet/evaporator inlet
3	ORC evaporator outlet/expander inlet

coil	Heating coil in hot water tank
con	Conduction
c	Critical
d	Dew point
d	Diffuse
e	Electrical
eff	Effective
enrv	Endoreversible
ex	Exergy

exp	Expander	pin	Pinch
exp	Experimental	p	Constant pressure process
f	Fluid	pp	Pump
fill	Tank fill	rev	Reversible
g	Generator	r	Refrigerant working fluid
gl	Glazing	sh	Superheat
gro	Gross	sol-e	Solar to electrical
h	Heat carrier fluid	sol	Solar
h	Horizontal	s	Isentropic process
hw	hot water	sc	Solar collector
hx	Heat exchanger	sub	Subcool
in	Inlet	sup	Supply
fan	Fan	surf	Surface
int	Internal	sw	Swept
load	Domestic hot water load	tank	Hot water storage tank
lo	Longitudinal	t	Tilted
loss	Thermal or exergy loss	tr	Transverse
m	Motor	use	Used
net	Net	vol	Volumetric
n	Node or element	wt	Weighted-component
OM	Operations and maintenance	w	Water
out	Outlet		

Chapter 1

Introduction

1.1 Background

The last two and half centuries of human development has seen our species become almost entirely dependent on fossil fuels for its energy needs. Coal, oil and gas are finite reserves that we have chosen to extract at an increasing rate in order to keep up with our growing demand for energy, knowing that this cannot possibly be sustainable in the long term. The peak oil theory proposed by M. King Hubbert in 1956 predicted that petroleum production in the USA would follow a bell-shaped curve with a peak around 1970 [1]. When these national predictions appeared to be correct the theory was applied to global production, and a global peak was predicted to occur before the end of the 20th century. At the time of writing in the year 2016, global oil production continues to increase thanks in part to the exploitation of so-called “unconventional” reserves resulting from technological, economic and political factors not originally considered in the Hubbert’s Peak model [2]. With regard to the long-term trend experts are still divided on whether peak oil is imminent, several decades away or has in fact already passed; many suggesting now that the production curve will not follow a symmetrical trend but rather that the decline in oil production will be more gradual and undulating [3].

An inevitable side-effect of fossil fuel combustion is the emission of so-called greenhouse gases, chiefly carbon dioxide, which unless captured at source are released into the atmosphere.

The strong consensus amongst the international scientific community is that such emissions are responsible for the comparatively rapid increase in temperature of the Earth's climate system observed over the last century, during which the average land and ocean surface temperature has risen by $0.85\text{ }^{\circ}\text{C}$ [4]. The United Nations Framework Convention on Climate Change (UNFCCC) recently adopted a set of ambitious targets to mitigate global warming [5]. To date, the Paris Agreement of 2015 has been signed by 193 nations, of which 115 have ratified. The specific aims of the Agreement are as follows:

- holding the increase in the global average temperature to well below $2\text{ }^{\circ}\text{C}$ above pre-industrial levels and to pursue efforts to limit the temperature increase to $1.5\text{ }^{\circ}\text{C}$ above pre-industrial levels, recognizing that this would significantly reduce the risks and impacts of climate change;
- increasing the ability to adapt to the adverse impacts of climate change and foster climate resilience and low greenhouse gas emissions development, in a manner that does not threaten food production;
- making finance flows consistent with a pathway towards low greenhouse gas emissions and climate-resilient development.

The emphasis in the second and third points are on maintaining development. Our planet's human population has increased from less than 2 billion at the start of the 20th century to an estimated 7.4 billion today, of which more than 6 billion live in less developed countries [6]. Energy consumption is integral to living standards, and countries with the lowest per-capita energy use also tend to be those with the lowest human development index [7]. Meanwhile less developed countries also tend to be abundant in renewable energy resources such as solar, wind and geothermal.

1.1.1 Solar heat and power in the UK

The onus is on the developed world to lead the way in emissions reduction through technological developments. This year, in light of the Paris Agreement, the International Energy

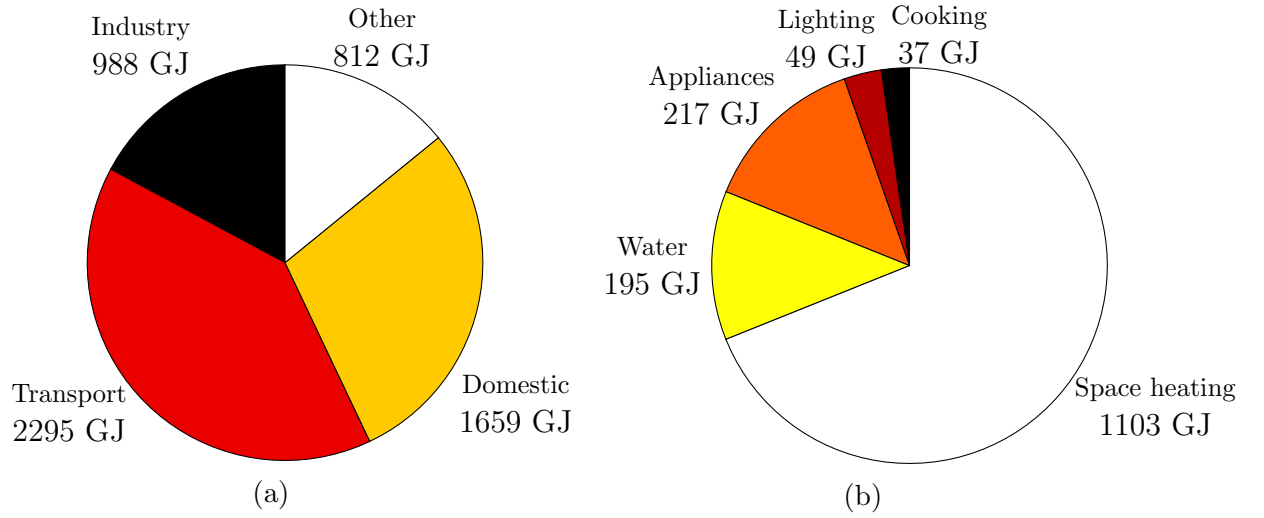


Figure 1.1: (a) Annual energy consumption in the UK in 2015 by sector; and (b) domestic energy consumption by end use [9].

Agency (IEA) updated its roadmap for emissions reduction over the next four decades [8]. In this roadmap it is predicted that end-use fuel and energy efficiency will provide the largest proportion of the emissions reduction, contributing 38% of the overall target, while renewables are expected to provide a further 32%. A particular emphasis in the report is placed on the role of energy use in buildings.

In the United Kingdom, between a quarter and 30% of the total CO₂ emissions are associated with domestic energy use [10, 11]. An end-use breakdown of total domestic sector energy consumption in the UK is shown in Fig. 1.1b. Approximately two-thirds is consumed for space heating, with the remaining one-third split between water heating, electrical appliances, lighting and cooking [9, 12]. This breakdown leads to an approximate heat-to-power consumption ratio of 5:1. Therefore it is necessary to consider and to promote advantageous options for the provision of both of these important energy components. Systems and technologies that are capable of providing both heat and power typically involve the burning of fossil fuels in prime movers, which are capable of converting a certain fraction of the consumed chemical energy in the fossil fuel into electrical power. The combustion of the fuel converts the chemical energy into thermal energy at a high temperature, which is then converted to mechanical energy by the prime mover and eventually electrical energy by a generator. These *cogeneration*, or combined heat and power (CHP), systems then also collect a significant fraction of the heat that is not

converted to electricity for direct use. Large-scale centralised power stations may be located close to industrial sites with a significant demand for high-temperature process heat, such as chemical processing plants and refineries. In small-scale distributed CHP applications such as in residential and commercial buildings, heat is usually recovered at low-temperature to provide space-heating or hot water. Internal combustion engine systems for distributed CHP typically have electrical efficiencies of 30% plus the ability to recover a further 50–55% of the primary energy as heat [13, 14].

Heat pumps are an alternative to thermally-driven CHP systems that provide heating or cooling from an available electricity source. In this case, electricity provided from high-efficiency centralised power stations is favoured, with domestic heat pump units delivering heat while being powered by the grid. For a decarbonised future with fewer combined cycle gas turbine (CCGT) plant, but also a lower heat to electricity demand ratio due to improved insulation of the housing stock, comparisons between CHP and heat pump solutions do not lead to a clear winner in terms of performance [11].

Beyond fossil-fuel CHP, heat and power can also be provided by non-fossil-fuel primary energy sources, such as solar-based systems. As of the end of 2015 the United Kingdom has an installed capacity of 30 GW of renewable energy, of which the largest portion (14 GW) is made up of on-shore and off-shore wind. Solar photovoltaics (PV) have seen a significant increase from <100 MW in 2011 to 11 GW in 2016 (while over the same period costs of PV systems have approximately halved), with the result that the UK now has the sixth largest installed capacity of solar PV in the world [15, 16]. Meanwhile, as of 2015 the installed capacity of solar water heating in the UK is 543 MW_{th}, corresponding to $\sim 780,000$ m² [17]. In per capita terms this is low compared to the rest of Europe (8.1 kW_{th} per 1000 capita, compared to 564.9 kW_{th} per 1000 capita in Cyprus), and the growth in the UK solar thermal market slowed since 2011, in part due to the more favourable market incentives for PV.

The UK has a modest solar resource compared to countries of subtropical latitudes. The global horizontal solar irradiation received annually in London is typically between 1,000–1,100 kWh m⁻² [18], equivalent to approximately 120 W m⁻² on average. This is a little

over half of that received in parts of southern Europe and North Africa, yet it is sufficient to reduce meaningfully the household demand for heat and power. Studies by the Energy Saving Trust have shown that a 2 kW_p PV system (approximately 15 m² array size if monocrystalline silicon) has the potential to provide around half of the annual energy consumed by a typical home in the UK, while a well-installed solar hot-water system (typical array size 3–4 m²) can provide around half of the annual energy required for water heating [19, 20]. The prospect of solar-based cogeneration of heat and power is restricted by the amount of roof space available to accommodate both PV and thermal arrays. The Energy Saving Trust provided a solar energy sizing guide [21] that lists the typical representative roof areas of a number of property-types in the UK, the smallest of which is a period mid-terrace or end terrace house with a typical suitable roof area of 16.5 m² and the largest a detached bungalow with a typical suitable roof area of 31 m². PV and solar thermal collector arrays installed side-by-side are an inefficient use of such limited space. One alternative is hybrid PV-thermal (PVT) technology that incorporates the PV module for generating electricity and the thermal absorber for heating water into the same panel [22, 23]. The flow of water also provides cooling of the PV module thereby increasing its electrical efficiency; however in some cases the PV coverage area is slightly reduced to provide access to the plumbing connections. Thus PVT panels have a similar overall solar to electrical efficiency to PV-only panels, while their thermal efficiency is considerably lower than an equivalent-sized conventional solar thermal collector due to the presence of the PV layer. Combined heat and power systems based on solar thermal arrays coupled with thermodynamic heat engines present a possible further alternative to a side-by-side PV and solar hot water systems. A potential advantage of such systems is greater flexibility, with the ability to vary the proportions of electricity and heating to suit demand, and to make use of cheap thermal energy storage as an alternative to battery storage for PV, currently at costs upward of $\sim 170 \text{ €/kWh}_e$ [24–26]. The potential success of solar thermal-based CHP systems for the domestic market is dependent on the development of smaller, cheaper heat engines capable of running efficiently at low temperatures.

1.1.2 Solar heat engines

The concept of the solar-powered heat engine dates back to 19th century France, when the mathematician Augustin Mouchot first connected a Rankine cycle engine to a steam boiler whose external surface was heated by sunlight concentrated by metal reflectors. Shortly after, a number of solar engine concepts were pioneered by the Swedish-American engineer, and inventor of the Ericsson cycle, John Ericsson, including a hot-air engine in which the cylinder was located along the focal-line of a parabolic trough collector. At the turn of the century, Willsie and Boyle were the first to conceive a solar engine design based on a low temperature Rankine cycle using ammonia, and later sulphur dioxide, as the working fluid. In the latter system, water was used as the heat carrier fluid that was circulated through a non-concentrating collector to reach temperatures of 82 °C before evaporating the working fluid in a separate vessel via a heat exchanger. The engine was claimed to produce 15 hp (11 kW) from a collector area of 93 m² [27].

The modern large (mega-Watt) scale concentrating solar thermal power (CSP) plant has been in existence since the late 1960s [28] and is now built to largely standardised designs, producing electricity at levelised costs in the region of 0.15–0.3 USD/kWh [29]. Parabolic-trough collector (PTC) power plants based on steam Rankine cycles represent around 80% of the world's 6 GW installed CSP capacity. These plants typically operate at temperatures of around 400 °C and incorporate steam turbines as the prime mover. Synthetic heat transfer oils are often used as the heat carrier fluid to be circulated through the collector array, and then used to raise superheated steam at around 100 bar. Several large modular PTC facilities exist in the Californian desert and the south of Spain, such as the Andasol, Solnova and SEGS plants. These larger plants tend to consist of multiple 30 or 50 MW units [30]. More recently, PTC plants using molten salt heat transfer fluids have been investigated for higher temperature operation. At present the 5 MW Archimede Integrated Solar Combined-Cycle (ISCC) plant in Sicily is the only demonstration plant using PTC with molten salts, and operates at solar-field outlet temperatures up to 550 °C [31, 32]. Other types of large-scale CSP plant based around the steam Rankine cycle are solar power towers, with cycle conditions in the region

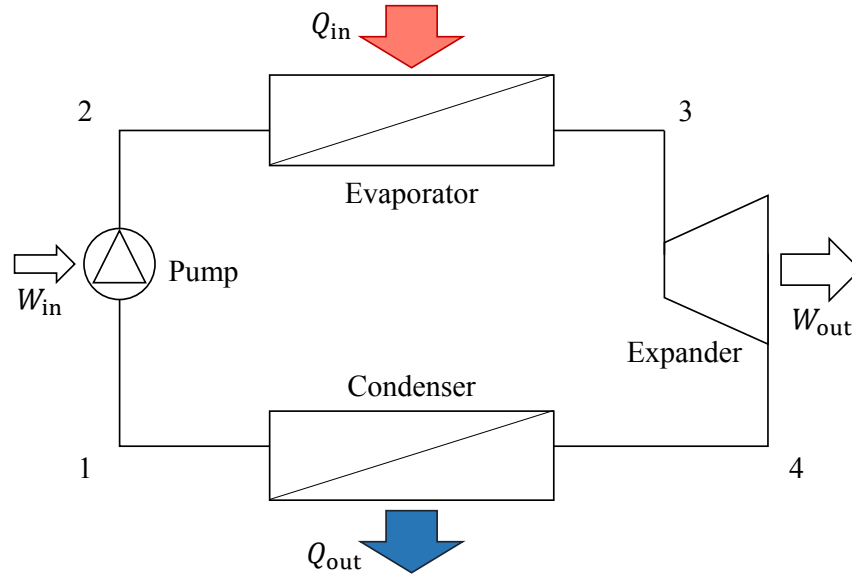


Figure 1.2: Schematic diagram showing the processes of a basic Rankine cycle.

400–600 °C, 100–160 bar and single plant capacities up to 130 MW; and plants using linear-Fresnel collectors, typically ~ 50 MW and raising steam at 55 bar, 270 °C. Parabolic dish systems coupled with centrally-mounted Stirling engines tend to be limited in size to < 20 m dish diameter and ~ 25 kW electrical power capacities. Such systems operate at temperatures of up to 800 °C, and exhibit efficiencies of up to 30%, compared to 15–20% for PTC plant. However at present few commercial dish-Stirling systems exist due to their high capital cost compared with other CSP technologies [30, 33, 34].

1.1.3 Organic Rankine cycle systems

The Organic Rankine Cycle (ORC) engine is one of a number of technologies being explored as an alternative to PV for the transformation of solar radiation into electrical energy at lower temperatures than typically required for steam Rankine cycles [35–39].

An ORC engine has the same operating principle as a conventional Rankine cycle equivalent, but uses an organic compound instead of water as the working fluid. Historically the cycle has been employed for the conversion of heat from low-to-medium temperature sources (e.g. geothermal, biomass combustion, process waste heat), with systems being operational, often with little need for maintenance, over 2–3 decades. The four processes of the basic Rankine

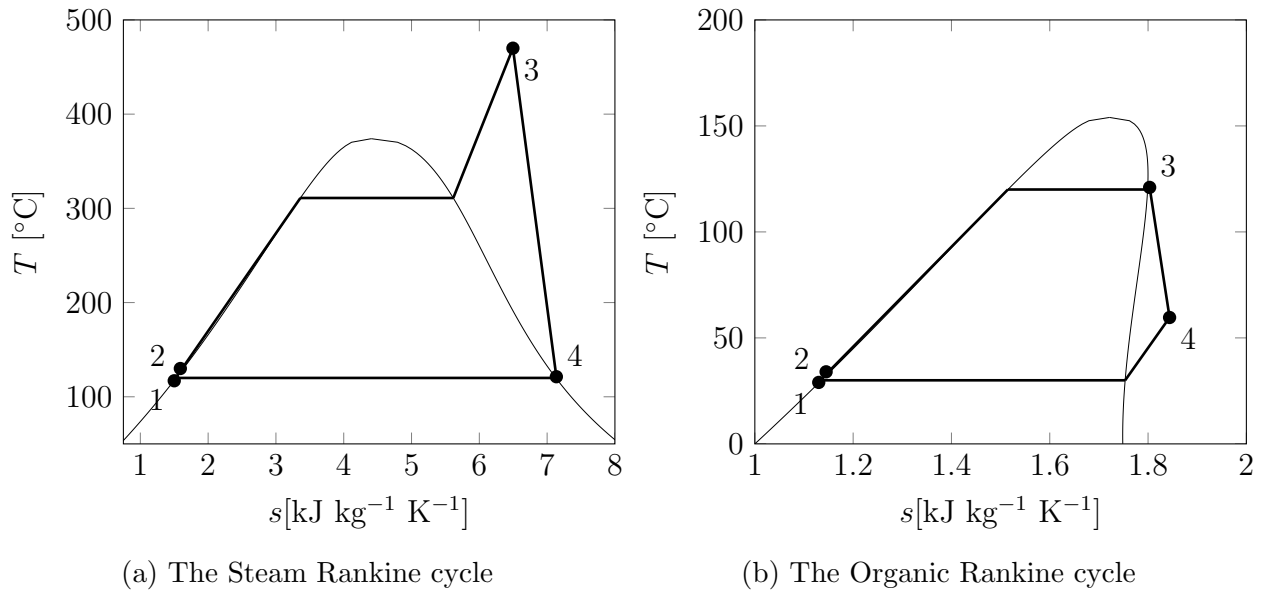


Figure 1.3: $T-s$ diagrams showing the basic steam Rankine cycle and the basic organic Rankine cycle (ORC).

cycle are shown in Fig. 1.2. The working fluid is pumped as a liquid (1–2) before being heated and evaporated at high pressure (2–3). The vapour is then expanded to the lower pressure (3–4), thereby generating shaft work, before being condensed back to its original low pressure liquid state (4–1). The steam Rankine cycle and organic Rankine cycle are compared on a temperature-entropy ($T-s$) diagram in Fig. 1.3. The negative slope of the saturated vapour curve for water means that superheating is often required for a steam Rankine cycle in order to avoid the formation of liquid droplets during the expansion process (often multiple expansion stages with reheat are also required). For organic fluids, the shape of the saturated vapour curve tends to vertical or positive and thus superheating is often not required prior to expansion, making the cycle suitable for low temperature applications. However, the enthalpy of evaporation for water is large compared to organic fluids, and thus ORCs require higher working fluid mass flow-rates. Vapour exiting the expander at State 4 is often superheated for so-called “dry” working fluids. The energy used to precool this vapour prior to condensation can be reclaimed to preheat the liquid leaving the pump prior to evaporation. This requires an additional heat exchanger (called a recuperator) and thus recuperative cycles tend to have higher efficiencies but also higher costs. Alternatively the reclaimed energy from the superheated vapour can be used to provide low temperature heat for an additional thermal process, presenting further possibilities for how ORC systems can be configured in cogeneration applications.

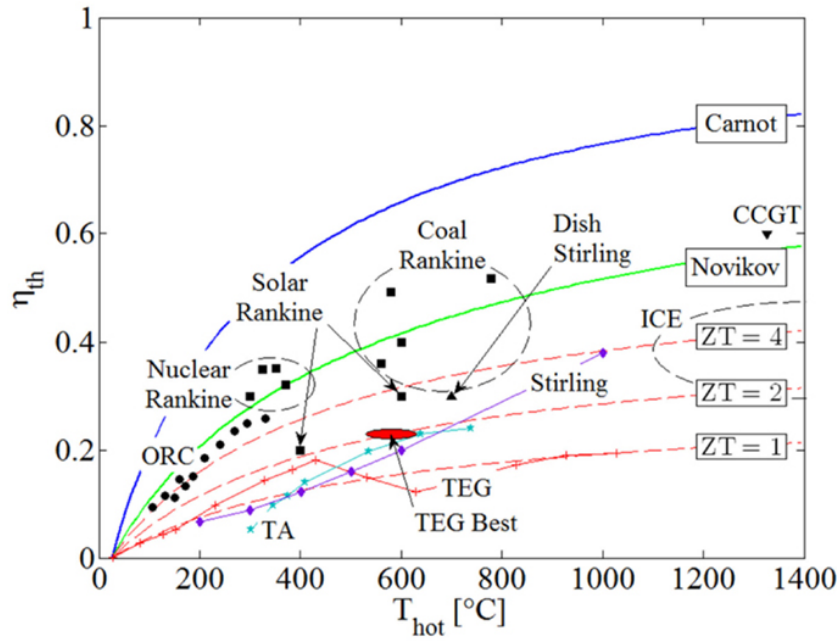


Figure 1.4: A comparison of thermodynamic heat engines, thermoacoustic engine (TAE) and thermoelectric generator (TEG) efficiencies over a range of a heat source temperatures [35]. The solid blue line and solid green line show the theoretical Carnot and Novikov efficiencies, respectively, while the dashed red lines indicate various theoretical efficiencies for TEGs based on thermoelectric figure-of-merits: $ZT = 1, 2$ and 4 .

The ORC technology is well established worldwide with several commercial systems in operation. Typical sizes range from the order of a few kW to 10 MW (rated electrical power output) for a wide range of working fluids and operating temperatures [40]. The majority of these systems incorporate turbines as the expansion device, which tend to operate at high rotational speeds and are best suited to larger scale systems with higher shaft output powers (> 10 kW). Scaling down turbomachines for smaller-scale applications presents non-trivial challenges and can be expensive and complex. Instead, positive-displacement machines may be used, which can operate at lower rotational speeds. For example, rotary screw expanders are used by some commercial manufacturers with power outputs as low as 30 kW [41]. Other non-commercial ORC systems have used adapted scroll-compressors run in reverse as the expansion device for power outputs less than 10 kW [38, 42], while more recently commercial scroll expanders dedicated for ORC applications as small as 1 kW have entered the market [43].

Markides [35] compared the efficiency of several technologies for heat to power conversion over a range of heat source temperatures (see Fig. 1.4). The comparison, based on existing installations, included ORCs, steam-Rankine cycles for CSP and large-scale nuclear power con-

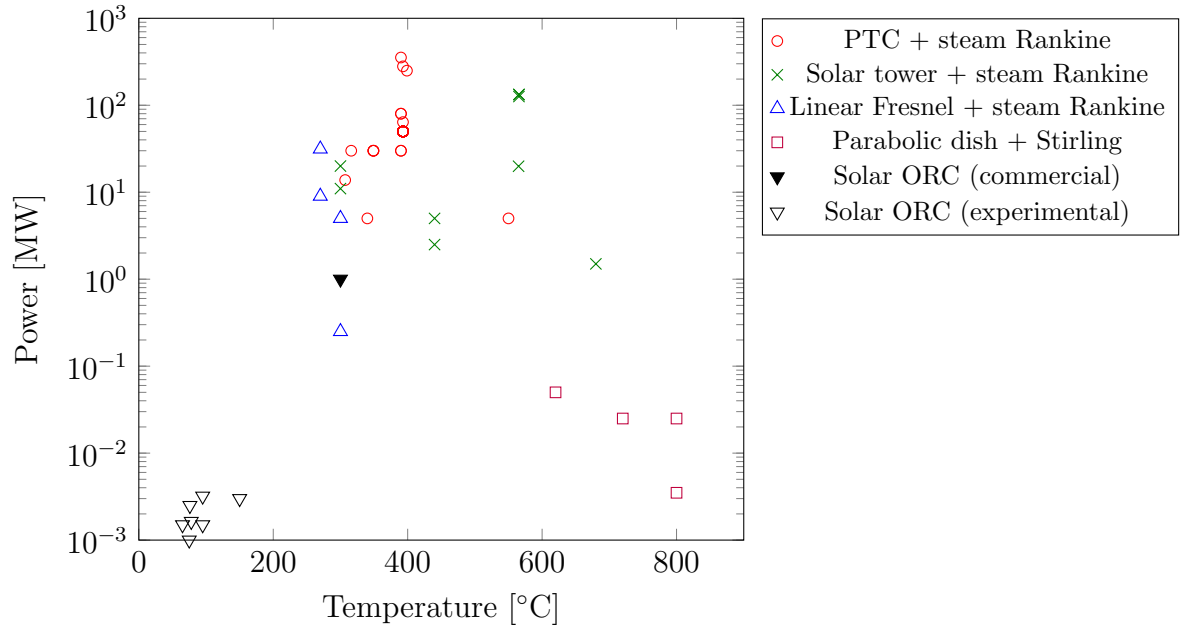


Figure 1.5: Rated electrical power outputs and solar heat source temperatures of various existing commercial solar power plant and experimental solar ORC systems reported in the literature [30, 34, 38, 42, 44–48].

version, gas phase engines including Stirling cycles, travelling wave thermoacoustic engines (TAEs) and thermoelectric generators (TEGs) based on the Seebeck effect. At temperatures below 400 °C, ORC engines were observed to out-perform Stirling engines, steam-Rankine engines and TAEs, while TEGs were considered a long way from achieving the theoretical best performance that might make them competitive at very low power outputs (< 100 W). Furthermore, existing low-temperature ORC systems were found to achieve performance efficiencies close to the endoreversible Novikov efficiency for maximum power output at the corresponding heat source temperatures.

Figure 1.5 shows a range of existing commercial plant and experimental/demonstration systems based on various solar power technologies, compared in terms of heat source (solar collector) temperatures and nominal generation capacities. To date, there are few working examples of low-temperature solar power systems based on the organic Rankine cycle. The 1 MW_e concentrating solar power (CSP) plant in Saguaro, Arizona (latitude 33 °N, insolation 2100–2200 kWh m⁻² yr⁻¹) is the only example of a commercial solar-ORC plant, with a PTC field area in excess of 10,000 m² [45]. Heat transfer fluid temperatures exiting the solar field of up to 300 °C are used to evaporate the pentane working fluid at an evaporation pressure of 22 bar

and the overall design efficiency of the plant is 12.1%. A small number of experimental solar-ORC systems are also described in the literature. These plants tend to be far smaller ($1-3 \text{ kW}_e$) with collector operating temperatures in the range $50-150^\circ\text{C}$ and reported efficiencies in the range $3-7\%$. These include a rural electrification project in Lesotho, Africa (latitude 29°S , insolation $2000-2100 \text{ kWh m}^{-2} \text{ yr}^{-1}$), with a PTC field of area 70 m^2 , a 3 kW_e scroll expander and the working fluid R245fa [38, 49]; an experimental system in Tianjin, China (latitude 39°N , insolation $1300-1400 \text{ kWh m}^{-2} \text{ yr}^{-1}$) with a 44 m^2 array of non-concentrating flat plate collectors (FPC) and evacuated tube collectors (ETC) and an ORC featuring a rolling piston expander and the working fluid R245fa; and an experimental system in Athens, Greece (latitude 38°N , insolation $1800-1900 \text{ kWh m}^{-2} \text{ yr}^{-1}$) with an ETC array area of 88 m^2 [42] and an ORC with a 2.5 kW scroll expander and the working fluid R134a. A single example was also found of an experimental system, located in Athens, featuring an array of concentrating PVT collectors delivering a thermal input at temperatures of up to 95°C to an ORC with a 3 kW scroll expander and the working fluid R404a [46]. The experimental systems mentioned above were tested in geographical locations with climates of moderate to high solar resource, and with solar collector arrays somewhat larger than that which could be accommodated on the roof of a domestic dwelling. To the best of the author's knowledge, there has at present been no detailed experimental evaluation of a solar-ORC system in a location as far north as London, UK (latitude 52°N , insolation $1000-1100 \text{ kWh m}^{-2} \text{ yr}^{-1}$).¹

1.1.4 Cogeneration and polygeneration

As noted in the previous section, the ORC is well suited to operation in a range of applications, through the selection of an appropriate working fluid for the desired temperature range. This also makes ORC systems potentially useful for integration with other thermal processes in cogeneration and polygeneration systems. A number of commercial ORC units are available

¹A single example of an experimental UK-based solar ORC system was found in the literature review, in which an ETC array was used in combination with a gas boiler as the heat source to a 1.5 kW ORC featuring a micro-turbine expander [47]. The system was tested in rural Leicestershire and demonstrated a cycle efficiency of 7.5% with the working fluid HFE-301, however the array size and the proportion of the total ORC thermal input provided by the solar collectors were not given by the authors.

on the market for integration with internal combustion engine CHP units in order to generate additional power from the waste heat exhaust gas or cooling water streams [41, 50], with the remaining heat rejected from the ORC unit used to provide space heating or pool heating at low temperature. Other studies have been performed into ORC-based cogeneration systems in which the heat rejected from the ORC is used to provide domestic hot water [51, 52]. Wieland *et al.* [53] considered a number of possible configurations for an ORC plant to provide both power and heating via a district heating network. This included options to provide water heating from the primary heat carrier fluid (downstream of the ORC evaporator) and also from the rejected heat by the ORC, by way of multiple heat exchangers. Where heating demand was low, increased power production through the use of a recuperative cycle was preferred.

The ORC has also been investigated for integration into combined cooling and power (CCP) and combined cooling heating and power (CCHP) systems based on vapour compression refrigeration cycles (VCC), vapour absorption refrigeration cycles (ARC) and ejector refrigeration cycles (ERC). One well-known configuration is that originally proposed by Goswami [54, 55] for a combined ORC and ARC using ammonia-water as the working fluid mixture. The ammonia is boiled out of solution at temperatures $> 100\text{ }^{\circ}\text{C}$ and expanded to a very low temperature ($\sim -20\text{ }^{\circ}\text{C}$) in a turbine producing work. The low temperature ammonia exhaust vapour is then used to provide cooling before being absorbed back into solution. Alternative configurations for ORC combined with ARC include using the rejected heat from the ORC condenser as the heat input to the generator/desorber of the ARC. Al Sulaiman *et al.* [56] investigated such a configuration for a solar CCHP configuration in which octane was considered as the ORC working fluid for the ORC and LiBr-water for a single-effect ARC. Thermal input was provided by a PTC array at temperatures of $327\text{ }^{\circ}\text{C}$, while heat was rejected at $92\text{ }^{\circ}\text{C}$ providing thermal input to the ARC.

An example of a combined ORC and VCC is the system proposed by Kim and Perez-Blanco [57], in which both cycles share the same working fluid, with the ORC and VCC vapour streams combining to reject heat in a common condenser heat exchanger. Isobutane was highlighted as a potential working fluid for such a cycle due to its high critical temperature. The design heat source temperature for the ORC was $150\text{ }^{\circ}\text{C}$, while the VCC evaporation temper-

ature was 5 °C. The common condensation temperature for both cycles was chosen as 40 °C. Dai *et al.* [58] investigated a configuration for a CCP system in which an ejector was located in between the ORC expander and condenser. The ejector entrains the secondary vapour flow from the cooling cycle which mixes with the primary flow from the ORC. The mixed stream is compressed by the ejector and then passes through the condenser before splitting again to undergo the remaining processes of the two cycles. Thus the ejector replaces the compressor in the refrigeration cycle (but not the pump in the Rankine cycle, which is still required). The refrigerant R123 was used as the working fluid for the CCP cycle, with a design heat source temperature of 150 °C, and a cooling cycle evaporating temperature of -10 °C. The common condensation temperature was 20 °C.

The diffusion absorption refrigeration cycle (DAR) is a mature technology that has received recent renewed interest for use in solar cooling systems. DAR cycles exhibit a lower coefficient of performance (COP, ratio of cooling output to thermal input) and require higher heat-source temperatures compared to other absorption technologies; however, they also have considerably lower capital costs (0.6 €/W_c). Also, unlike other absorption technologies, DAR systems do not require a pump or compressor for fluid circulation or pressurization, which allows silent operation without the need for an electrical input. This feature also makes these systems promising for off-grid cooling solutions in rural regions where the electricity grid is unreliable or non-existent. DAR systems can be adapted to work at low temperatures with low-cost solar-thermal collectors. The conventional use of an ammonia-water working-fluid mixture requires the heat-source operating temperature to be in the region 150–200 °C [59], however, alternative working-fluid mixtures such as LiNO₃-NH₃ and NaSCN-NH₃ are presently under investigation to lower this range down to 85–110 °C while achieving COP values in the range 0.25–0.50 [60].

Manolakos *et al.* [61–63] conducted multiple studies into the use of the ORC in a thermally-powered reverse osmosis (RO) desalination system, in which the RO high pressure pump was directly driven by the ORC expander (a reversed scroll compressor). The system was also experimentally investigated with the thermal input to the ORC delivered by a non-concentrating ETC array [42]. Delgado-Torres and García-Rodríguez [64, 65] investigated the possibility of us-

ing a 100 kW ORC system powered by a PTC array to provide power and also thermal input for a distillation-type desalination process by allowing an increase of the condensation temperature from 35 °C to 115 °C (considering ORC evaporation temperatures in the range 230–400 °C). A range of ORC working fluids were investigated and both direct vapour generation (DVG) and liquid heat transfer fluid circuit arrangements were considered for the integration of the PTC with the ORC. The same authors also investigated a solar ORC desalination system based on RO, whereby the ORC provided both input shaft power to drive the RO high pressure pump and also pre-heating of the sea-water feed using rejected heat from the cycle, with the intention of improving the membrane flux [66]. Increasing the pre-heating of the feed by increasing the ORC condensation temperature was found to be detrimental to maintaining high pumping power for the RO unit, and therefore offered no benefit to the overall RO performance.

1.1.5 Expanders for ORC systems

The expander is one of the most critical components for consideration in the design of a small-scale ORC system. Mago *et al.* [67] performed an evaluation of exergy destruction in a 50 kW non-regenerative ORC and found that the expander accounted for the second largest contribution to overall exergy losses in the cycle (21%), after the evaporator (77%). Expander design is seen to be one of the most potentially fruitful areas for development to yield improvements in performance and cost in small-scale ORC systems and thus a great deal of the literature concerns the design and modelling of this component. Positive-displacement (also known as *volumetric*) expanders are distinguished from turbine expanders in their principle of operation, which is to use the energy of expansion directly in the displacement of the prime mover. A turbine expander on the other hand converts enthalpy to kinetic energy and then to mechanical work. Badr *et al.* [68] noted the following advantages of positive displacement expanders in small-scale, low-temperature heat engines: (i) higher efficiencies, (ii) no requirement for speed-reducing gearboxes; (iii) good off-design performance; (iv) favourable performance with wet vapours; (v) simpler control-system requirements (vi) lower cost; (vii) smaller size/compactness; (viii) simplicity of manufacture.

A number of authors have developed modelling frameworks for the analysis of loss mechanisms in positive-displacement expanders in order to map their performance over a range of operating conditions. When plotting the basic Rankine cycle on a temperature-entropy (T - s) diagram, the negative gradient of the expansion process line (between States 3 and 4 in Fig. 1.3) denotes the isentropic efficiency of the expansion. According to Ng *et al.* [69], isentropic efficiency losses include those due to flow irreversibility and to leakage through clearance gaps between the moving components of the expander (e.g between the piston and cylinder wall in a reciprocating expander). Other losses that affect the *overall* efficiency of the expander (defined as the ratio of shaft work output to fluid enthalpy drop) are internal losses due to the mismatch between the imposed cycle pressure ratio and the so-called “built-in” expander volume ratio, and external mechanical efficiency losses resulting from shaft friction. Thermal losses to the environment may also be significant, although often adiabatic conditions are assumed for expander modelling.

Badr *et al.* [70] presented a model for the calculation of efficiency losses in multi-vane expanders due to internal leakage and the effect on the overall delivered shaft power. The most significant contribution to internal leakage losses was found to be leakages past the tips of the vanes, contributing an estimated 16–50% of the overall losses in specific work output, relative to a case with no internal leakage. The highest internal leakage rates were found to occur at low rotational speeds when the centrifugal force upon the vanes was insufficient to maintain contact with the stator cylinder wall. By eliminating this particular leakage path it was predicted that a 3× improvement in specific work output could be achieved at lower rotational speeds of 1500 RPM, while at higher rotational speeds of 4500 RPM, a 25% improvement in specific work output was predicted.

Ng *et al.* [69] developed a thermodynamic model to analyse the performance of screw expanders and applied the model to five screw expanders for which experimental data was available. The major causes of overall expander efficiency losses were found to be the so-called “diagram efficiency” losses (i.e. due to thermodynamic irreversibility and fluid leakage). It was also found that losses due to the mismatch between the system-specific volume ratio and the expander built-in volume ratio became significant as the extent of the mismatch became larger,

and that the highest overall efficiency (up to 85%) was obtained when the imposed volume ratio was equal to the built-in volume ratio. Mechanical friction losses were found to contribute a relatively small proportion of the overall efficiency losses.

McKenna *et al.* [71, 72] developed a model for a reciprocating piston expander in an ORC, in which the instantaneous work output during the expansion stroke was calculated by performing a first-law energy balance on the cylinder. Equations describing the mass flow rate through the inlet and exhaust valves were developed first using ideal gas laws and then modified to take account of real gas behaviour of the working fluid. Heat exchange with the walls of the cylinder was also considered for a known wall temperature. Piston ring frictional losses as a function of stroke length and rotational speed, and also leakage through the valves, were included based on empirical measurements from an experimental facility.

Lemort *et al.* [73] developed a semi-empirical model of a prototype scroll expander for which a set of 8 expander characterisation parameters were identified. These included the swept volume, built-in volume ratio, leakage area, supply port cross-sectional area, friction losses between scrolls and in the bearings (lumped together as a single “mechanical loss torque” parameter) and heat transfer coefficients describing thermal losses in the supply and exhaust pipes. The parameter values were experimentally obtained from a prototype scroll expander operating in an ORC system with R134a as the working fluid. By providing the supply and exhaust pressures, fluid inlet temperature, and rotational speed as input variables, the model was used to predict the mechanical shaft power, exhaust temperature and mass flow-rate.

1.1.6 Thermal energy storage (TES) for solar systems

An important design-challenge for the successful operation of solar-thermal power systems is the ability to deal with time-varying incident radiation intensity. Thermal energy storage (TES) solutions can provide buffering for stable operation on a timescale of minutes (using small fluid buffer vessels) to hours (using large tanks of molten salts or packed beds of solid materials).

Conventional TES solutions for large-scale CSP systems featuring steam-Rankine power

plants typically use indirect two-tank systems, providing storage for periods of 6–12 hours operation at full capacity. In this arrangement, a molten-salt mixture is pumped, during the charging cycle, from a cold-storage tank to a hot-storage tank via a heat exchanger where it is heated by a heat-transfer oil from the solar field. In order to deliver superheated steam at 100 bar, the temperatures of the salts in the hot- and cold-storage tanks need to be in the region of 390 °C and 300 °C, respectively [74]. Chacartegui *et al.* [75] investigated two TES configurations for use in a 5-MW_e CSP-ORC system with a design solar-array outlet temperature of 400 °C. These were an indirect two-tank configuration, similar to that described above, and a direct two-tank configuration in which the TES media (a eutectic mixture of sodium-nitrate and potassium-nitrate salts) was also used as the heat-transfer fluid to be circulated through the solar field. The direct-storage configuration was found to require a smaller tank volume and a lower investment cost, while the addition of TES was found to reduce the system's cost of electricity.

Single-tank thermocline systems are sometimes considered as a lower-cost alternative to two-tank systems, and low-cost filler materials such as quartzite and granite particles are often included to provide a higher specific heat capacity and to reduce the required storage volume and cost of additional heat-transfer fluid [76, 77]. The 1 MWe ORC-CSP demonstration plant in Saguaro, Arizona (described in Section 1.1.3) was originally installed without TES, however, a thermocline TES with gravel as the filler material has been proposed for a future retrofit [78]. A similar single-tank packed bed TES concept was also proposed by Quoilin *et al.* [38] for the 3 kW rural solar-ORC system in Lesotho, Africa.

A number of investigators have proposed TES solutions for solar systems using solid-liquid phase change materials (PCMs). Such materials are chosen to absorb a significant amount of heat as they undergo the melting process (from solid to liquid), over a narrower temperature range than would be achieved by sensible TES in an equivalent volume. PCMs are broadly divided into a wide range of organic compounds (such as waxes, fatty acids and glycol-based compounds) and inorganic salt-hydrates. Commercially available PCMs for thermal storage at low-to-medium temperatures have volumetric latent heats in the range of 30–80 kWh m⁻³ [79]. By comparison the energy density of sensible storage solutions depends on the usable temper-

ature difference as the storage medium is charged and discharged. A hot water store heated from 50 °C to 80 °C ($\Delta T = 30$ K) can provide an energy storage density of 35 kWh m⁻³; however, reducing the usable temperature difference to 10 K results in a lower storage density of 12 kWh m⁻³. PCM-TES solutions described in the literature often feature embedded heat exchangers [80, 81], or the PCM encapsulated in small containers, immersed within larger tanks to enhance the thermal storage capacity and heat transfer over a given temperature range [82]. Naghavi *et al.* [83] proposed the incorporation of PCMs into the manifold of an evacuated-tube heat-pipe solar collector and compared its simulated performance to a typical collector with a hot-water storage tank arrangement. The PCM system used paraffin wax with a melting temperature of 64 °C, and was found to offer a larger useful daily energy output compared to the conventional hot-water system that was found to suffer significant thermodynamic losses due to destratification over the diurnal storage period.

Casati *et al.* [84] presented a range of a TES concepts for use in solar ORC systems. The authors noted that the ability to use working fluids with a “dry” (positive-gradient) vapour-saturation curve in these systems was highly favourable for solutions that consider thermal storage via direct storage of the working fluid. If the working fluid is stored under pressure as a saturated liquid, vapour can be generated by allowing the fluid to expand isenthalpically (known as flashing), and then fed to the expander. For very dry fluids, expansion from a saturated liquid to a saturated or superheated vapour state can be achieved with a relatively small drop in pressure. Working fluid storage for steam generation has been used historically for various process applications. Steam accumulators are used in a number of linear Fresnel CSP plant such as the Puerto Errado Thermosolar powerplants in Spain [30]. A noted limitation of basic *sliding-pressure* steam accumulators, in which the water working fluid is stored in two-phase (liquid-vapour) thermodynamic equilibrium, is the associated drop in pressure of fluid in the vessel as steam is discharged. Steinman and Eck [85] presented a concept for *constant-pressure* storage using encapsulated PCMs in order to achieve higher energy storage densities under isothermal conditions. Thus temperature (and hence pressure) of the saturated steam or vapour supply can be maintained. This concept has been recently employed with success in prototype steam generator systems [86], and has been proposed for use in a refrigerant-based

solar-ORC system by Jing *et al.* [87], in order to provide a constant vapour quality to the expander under varying solar conditions.

Other energy storage solutions considered for use in solar-thermal systems are thermo-chemical storage technologies that make use of either sorption processes or reversible chemical reactions such as hydration or carbonation of metal oxides. At present, solar applications of thermo-chemical storage have predominantly been of the sorption-storage type when working with charging temperatures ≤ 150 °C, and deliver reported energy densities of 100–600 kWh/m³. Chemical-reaction energy storage meanwhile requires the careful selection of a suitable reaction to match the temperature of the specific application, and tends to be considered in higher temperature applications where charging temperatures ≥ 400 °C can be achieved [88–91].

1.2 Thesis Overview

In this thesis, a model is developed of a small-scale system for the combined provision of heat (hot water) and electrical power from solar thermal energy, and to develop experimental methods for evaluating this model. The modelled system features a solar thermal collector array, an organic Rankine cycle engine and a hot water tank. The model will be used to assess the performance and cost of the system in a domestic setting, and the results from the model will be compared to alternative technologies for heat and power provision, including PV and solar hot water systems based on side-by-side configurations occupying the same total array area. An array area representative of a typical roof space on a single domestic residence will be used as the starting point for the study, and this will be taken to be 15 m². The experimental work will focus on the evaluation of the solar collector and ORC sub-models. An outdoor apparatus will be presented for the steady-state and dynamic performance testing of solar collectors under real solar conditions; and a laboratory test-bench will also be presented for the testing of small-scale ORC components.

The thesis is structured as follows. In Chapter 2, the maximum work potential of a

range of solar collectors will be investigated using reversible and endoreversible analyses. The analysis is performed for a range of steady-state solar irradiance conditions and is then extended to an annual analysis for a northern and a southern European location using climate data for London, UK and Larnaca, Cyprus, respectively. Based on the results of the analysis, suitable collector designs for power generation in the two locations are identified. Non-concentrating evacuated flat plate collectors (EFPC) are found to offer the highest potential for London, whereas a concentrating parabolic trough collector shows a higher annual potential in Cyprus when implemented with a single-axis or two-axis tracking system at the expense of increased system cost.

In Chapter 3 the detailed thermo-economic model of the solar combined heat and power (S-CHP) system is presented. The model is described as a thermo-economic model because it is used to model the thermal processes of the system in which solar energy is converted to useful heat and power, and also the system cost based on the sizing of the individual system components. In Chapter 4 the model is used to provide a parametric analysis of the system, in which the system operational parameters and component sizes are optimised for a set of representative climate conditions. Operational modes for electricity and hot water prioritisation are identified and an annual simulation is also performed to determine the electrical and hot-water provision potential in the UK and Cyprus climates. System cost and performance is compared to those for other solar-based heat and power systems occupying the same total array area, including a side-by-side PV and hot-water system, and a PV-thermal hybrid system. Exergy destruction in the system components is examined which forms the basis for suggesting improved configurations. Consideration is given to thermal energy storage using the solar heat transfer fluid and also solid-liquid phase change materials.

In Chapter 5 two experiments are presented for the testing of non-concentrating solar collectors under steady-state and dynamic conditions. A detailed description of the outdoor testing apparatus is provided that was constructed on the roof of the university building in London, as well as a second apparatus constructed in Cyprus. Results from the experiments are used to evaluate the solar collector sub-model of the S-CHP system. For the dynamic performance characterisation, alternative methodologies in the relevant European Standards

are discussed and compared.

In Chapter 6, a 1 kWe (nominal) experimental ORC system is described. The system was designed for the testing and characterisation of various system components, including small scale positive-displacement expanders. Preliminary results from the system with a 1 kW scroll expander and the working fluid R245fa are presented. The results are used to evaluate the ORC sub-component model in the S-CHP system. Reasons for deviations between the predicted performance and the achieved performance in preliminary results are discussed and improvements to the test-bench are suggested.

The thesis is concluded with a summary of the major findings and outcomes, and suggestions for future work.

Chapter 2

Maximum Work Assessment of Solar Thermal Collectors

This chapter focuses on suitable solar thermal collectors for use in a small-scale combined heat and power systems. Concentrating and non-concentrating collector designs are compared by way of annual maximum work assessments based on reversible and endoreversible analyses using climate data for two geographical regions: one of rich solar resource (Larnaca, Cyprus), the other of low-to-moderate solar resource (London, UK). It is found that non-concentrating collectors show a wide range of annual maximum work outputs, up to a highest of $67 \text{ kWh m}^{-2} \text{ yr}^{-1}$ (endoreversible) for the UK and $146 \text{ kWh m}^{-2} \text{ yr}^{-1}$ for Cyprus. Scaled to an array size of 15 m^2 , representing a typical domestic roof space, this is 1008 kWh yr^{-1} (115 W average) for London and 2196 kWh yr^{-1} (250 W average) for Cyprus. This represents 30% of the electrical demand in a typical UK household and 44% of the demand in a typical Cyprus household. This is also equivalent to an annual solar to electrical efficiency of 6.3% and 7.4% respectively, which compares to annual efficiencies of 12–15% for monocrystalline PV. The use of a concentrating parabolic trough collector array with single-axis tracking does not provide an annual performance advantage for the UK where $\sim 50\%$ of solar irradiance received is diffuse; but shows a further 20% increased maximum work output for Cyprus relative to the highest performing stationary non-concentrating collector. The choice of mains water or air as the cooling medium

for heat rejection from the power cycle (modelled using their representative seasonal temperatures) is found to make a relatively small difference to the annual power output of between 3–11%, and is most significant for collectors with the lowest optimum operating temperature.

2.1 Introduction

The maximum power production from a solar-powered heat engine is a trade-off between the thermal efficiency of the heat engine, which increases at higher source temperatures; and the efficiency of the collector, which decreases as its temperature becomes significantly higher than its surroundings. Maximum work production can be evaluated by way of a second-law “exergy” analysis. A number of other studies have featured second-law analyses of solar thermal systems in order to identify improvements in energy utilisation. Farahat *et al.* [92] and Xiaowu and Ben [93], for example, examined solar collector design parameters, in particular the absorber plate dimensions, in order to maximise the exergy efficiency of a solar hot-water system. Jafarkazemi and Ahmadifard [94] evaluated the first-law and second-law efficiencies of a flat-plate collector as a function of inlet temperature and mass flow-rate, again highlighting potential considerations for improved design of the absorber plate.

Endoreversible thermodynamic analyses have been employed to assess the optimal design and operating conditions of solar-powered heat engines [95–98]. A fundamental characteristic of these approaches concerns the assumption of a reversible Carnot engine operating between internal heat addition and heat rejection temperatures experienced by the working fluid as this undergoes the cycle that are different from the external hot and cold temperatures from/to which the heat is taken/rejected. Although they attempt to consider this irreversibility due to heat transfer across the external and internal temperature differences, these analyses are otherwise ideal. Thus, the power output from a real engine will always deviate from (and if the internal temperature in the theoretical cycle is close to the operating conditions of the real engine, be lower than) endoreversible maximum power predictions, due to other forms of irreversibility (e.g. associated with heat losses, friction etc.). Nonetheless, these relatively

simple endoreversible analyses do allow important performance-related parameters such as the efficiency, specific power (ratio of power to solar collector area) or power output per unit cost (assuming cost of plant proportional to total heat transfer area) to be maximised through optimisation of system parameters such as fluid temperatures and heat exchanger surface areas.

The efficiency of an endoreversible cycle at maximum power output is that found by Novikov [99]¹ and is always lower than the Carnot efficiency. As noted by De Vos [102], engines designed for maximum efficiency (approaching the Carnot efficiency) will have the lowest possible fuel costs, while engines designed for maximum power (approaching the Novikov efficiency) will recoup their capital costs the fastest. Thus for fossil-fuel power plants the design efficiency tends to be in between the Carnot and Novikov efficiency and the economically optimal point depends on the ratio of fixed costs to variable costs (costs that vary with output). Therefore endoreversible analyses based on the Novikov efficiency are very useful in the study of renewable energy systems, where fuel costs are free. The endoreversible analysis in which the Novikov efficiency is used to derive an expression for maximum power (as it is later in this chapter) is based on an assumption of heat transfer into and out of the cycle by conduction. Thus, when applied to a solar engine, this analysis is suitable when the heat source is the fluid stream leaving the collector which imparts heat to the cycle via a heat exchanger downstream of the collector. However, if the heat source is taken as the surface of the sun itself, and heat transfer into the cycle is dominated by radiation then, as noted by Sahin [95], the maximum power for the endoreversible engine is in fact higher than that predicted by the Novikov efficiency due to the fourth-order relation between temperature and heat transfer. An endoreversible engine called a Stefan-Boltzmann engine was presented for such a case by De Vos [102, 103], in which heat transfer into and out of the cycle is modelled by Stefan-Boltzmann laws, rather than the linear transport equations assumed for the Novikov efficiency.

In this chapter the thermodynamic analysis is performed downstream of the collector, and thus the Novikov model is used. We approach the subject with the aim of determining the maximum potential for power generation from a heat engine receiving thermal energy from a stream of hot fluid leaving a solar thermal collector. The exergy content of the fluid stream

¹Often cited to Chambadal [100], and later work by Curzon and Ahlborn [101]

is related to the solar collector efficiency, and as part of the analysis we consider a range of collector types using manufacturers data to model their performance. The potential success of a system for the domestic market is increased by the current availability of low-cost collectors in an already mature market. Therefore, the internal exergy destruction in the collector as a result of its design is not evaluated in detail, but rather the maximisation of exergy in the fluid stream leaving the collector is considered in order to compare the suitability of existing available products.

2.2 Modelling Methodology

2.2.1 Solar collector efficiency

The useful energy output from a solar thermal collector can be predicted as a function of the local climatic conditions. The widely used expression by Perers and Bale [104] was developed for so-called “quasi-dynamic” testing of solar collectors under real, time-varying climatic conditions, and then developed for solar thermal system modelling in the TRNSYS energy simulation software package [105]:

$$\begin{aligned} \frac{\dot{Q}_u}{A_{sc}} = & F'(\tau\alpha)_n K_{\theta b} G_b + F'(\tau\alpha)_n K_{\theta d} G_d - c_6 u_a G - c_1 (\bar{T}_{sc} - T_a) \\ & - c_2 (\bar{T}_{sc} - T_a)^2 - c_3 u_w (\bar{T}_{sc} - T_a) + c_4 (E_L - \sigma T_a^4) - c_5 \frac{d\bar{T}_{sc}}{dt}. \end{aligned} \quad (2.1)$$

In line with studies of solar thermal systems featuring high-efficiency glazed collectors in the literature, the collectors in the present analysis will be modelled according to the following assumptions: (i) the wind-speed and long-wave irradiance dependent terms for high-efficiency glazed collectors are less significant than for low-efficiency unglazed collectors and therefore may be neglected, as is the standard approach for evacuated collectors [87, 104, 106, 107]; (ii) the response time of the collector is small compared to the hourly time-interval for the climate data used in the present analysis [108]. Thus the associated terms in Eq. 2.1 are omitted,

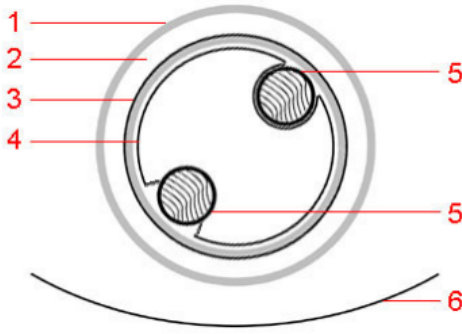
resulting in the following simplified expression:

$$\frac{\dot{Q}_u}{A_{sc}} = \eta_0 (K_{\theta b} G_b + K_{\theta d} G_d) - c_1 (\bar{T}_{sc} - T_a) - c_2 (\bar{T}_{sc} - T_a)^2, \quad (2.2)$$

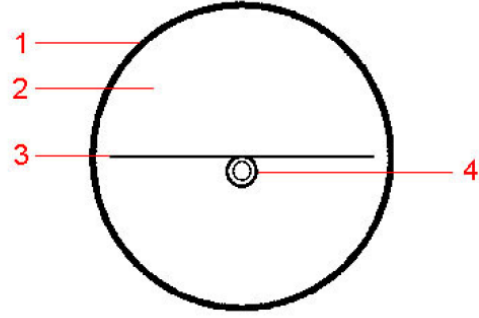
where η_0 is the zero-loss optical efficiency, c_1 and c_2 are heat loss coefficients and $K_{\theta b}$ and $K_{\theta d}$ are incident angle modifier (IAM) factors applied to the zero-loss efficiency for the beam and diffuse components of the solar irradiance, respectively.

Each of the parameters in Eq. 2.2 are determined experimentally as part of the Solar Keymark certification scheme², for which the testing procedures are described in European Standard EN 12975-2 [106, 109]. Solar Keymark test reports for a wide range of collectors are available on the website of the Institute for Solar Technology (SPF) [110]. The majority of collectors on the database are flat plate collectors (FPC). A sample of 20 FPC models were chosen for the present analysis. Of the 63 evacuated tube collector (ETC) models listed on the database (excluding duplicates under different trade names), 19 are direct-flow (DF) type, and 44 are heat pipe (HP) type. The DF collectors can further be segregated into three main sub-types: U-tube (6 collectors), coaxial (4 collectors) and U-tube collectors with compound parabolic concentrators (CPC, 9 collectors); where U-tube and coaxial refers to the flow and return copper tubing arrangement. As shown in Fig. 2.1, the U-tube collectors incorporate a circular fin design, which allows solar radiation to be absorbed at incidence angles $> 90^\circ$. Thus these collectors lend themselves well to configurations with reflectors and some of the models feature CPC devices. The coaxial designs on the the other hand use flat fin arrangements, without reflecting devices. Likewise, the heat pipe collectors can also be segregated into flat fin (10 collectors), and circular fin (34 collectors) sub-categories. The collectors with flat fin designs (both DF and HP type) use single-wall glazing where the whole interior core of the glass tube is evacuated; while circular-fin collectors feature a double-walled glazing arrangement where only the cavity in between the glazing layers is evacuated. The latter tends to be cheaper to manufacture, because a glass to metal seal is not required to maintain the vacuum.

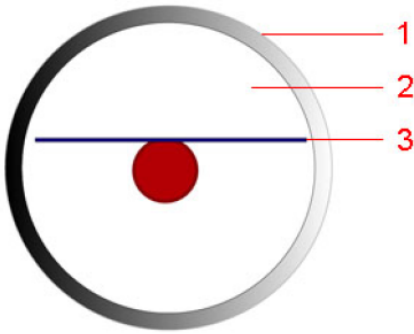
²The one exception is $K_{\theta, d}$, which for the majority of collectors examined in this chapter was not reported in their Solar Keymark test certificate, see Table 2.3



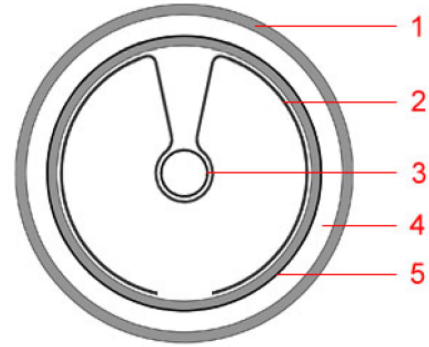
(a) Direct-flow U-tube collector with CPC reflector: 1) glazing; 2) vacuum; 3) absorber; 4) heat-conducting metal sheet; 5) U-tube; 6) CPC reflector.



(b) Coaxial direct-flow collector: 1) glazing; 2) vacuum; 3) absorber; 4) coaxial tube.



(c) Heat pipe collector with flat absorber arrangement: 1) glazing; 2) vacuum; 3) absorber.



(d) Heat pipe collector with circular absorber arrangement: 1) glazing; 2) heat-conducting metal sheet; 3) heat pipe; 4) vacuum; 5) absorber.

Figure 2.1: Various evacuated tube collector design configurations with annotated features described in the relevant captions. Figures taken from Ref [110].

The SPF database contains a single test report for a parabolic trough collector (PTC), and a single report for an evacuated flat plate collector (EFPC). EFPCs are a comparatively new technology and not widely available on the market, although are cited as a promising future technology for obtaining high solar energy conversion efficiency at elevated temperatures [107, 111, 112]. Commercialisation of EFPC modules has been held back in the past by the problem of maintaining a vacuum over long periods as collector materials undergo different rates of thermal expansion and contraction, but a number of companies have recently introduced novel designs for the collector edge seals to extend performance lifetime [113–116]. The PTC included on the database is an NEP Polytrough 1800, with a single-axis tracking system and an aperture area of 18.45 m^{-2} , while the EFPC is an SRB Energy UHV (Ultra High Vacuum). Unlike most other EFPCs and FPCs, the SRB collector is glazed on its under-side as well as its top side

which allows it to receive incident radiation on both sides of the absorber plate. Therefore this collector is suitable for use with reflecting devices, and the database contains performance data for this collector in two configurations: with and without a compound parabolic concentrator (CPC). A second EFPC included in the analysis was the TVP Solar MT-Power, for which performance data was available in a Solar Keymark test report by the University of Stuttgart [117].

2.2.2 Maximum work analysis

This analysis will consider the maximum power that can be produced from the stream of hot fluid leaving the collector for a given set of environmental (solar irradiance and ambient air temperature) conditions. In an exergy analysis the maximum power, or exergy content, of a fluid stream is calculated as the integral of the power produced by an infinite number of infinitesimal Carnot engines operating between the fluid stream and a heat sink at the reference environmental temperature. Each engine extracts an amount of heat from the fluid stream until its temperature is reduced to that of the environment (referred to as the “dead space” temperature), T_0 . This analysis leads to the following expression:

$$\begin{aligned}\dot{X}_{\text{rev}} &= \int_{T_0}^{T_1} \left(1 - \frac{T_0}{T}\right) \dot{m} c_p \cdot dT = \dot{m} c_p \left[T_1 - T_0 - T_0 \cdot \ln \left(\frac{T_1}{T_0} \right) \right] \\ &= (\dot{H}_1 - \dot{H}_0) - T_0 (\dot{S}_1 - \dot{S}_0).\end{aligned}\tag{2.3}$$

For a solar collector, the increase in fluid flow exergy between the inlet and outlet is a function of the flow-rate and the mean collector temperature. In this analysis, it is assumed that the system is a closed-loop in which the stream of fluid is returned to the inlet of the collector after heat is extracted by the Carnot engines, as shown in Fig. 2.2. An additional term is also required to account for the exergy destruction through the collector due to pressure

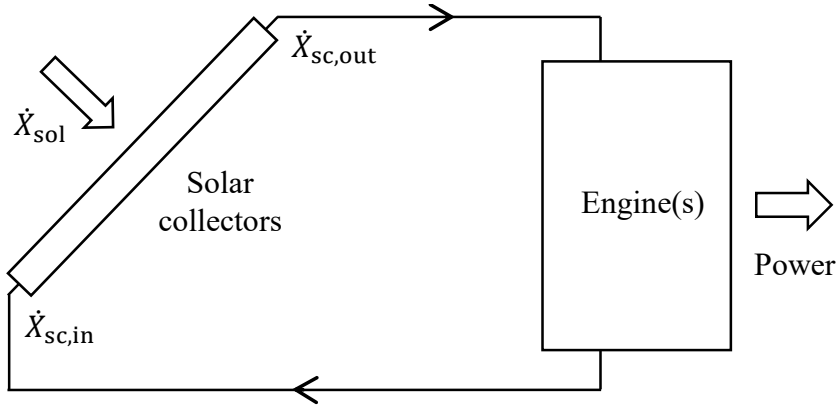


Figure 2.2: Schematic diagram of the notional closed-loop system considered in the maximum work analysis.

drop, which is also a function of the flow-rate, resulting in Eq 2.4:

$$\begin{aligned} \Delta \dot{X}_{f,rev} &= \dot{X}_{sc,out} - \dot{X}_{sc,in} = \dot{m} c_p \left[T_{sc,out} - T_{sc,in} - T_0 \cdot \ln \left(\frac{T_{sc,out}}{T_{sc,in}} \right) \right] + \frac{\dot{m}}{\rho} (p_{sc,out} - p_{sc,in}) \\ &= \left(\dot{H}_{sc,out} - \dot{H}_{sc,in} \right) - T_0 \left(\dot{S}_{sc,out} - \dot{S}_{sc,in} \right) + \frac{\dot{m}}{\rho} (p_{sc,out} - p_{sc,in}) . \end{aligned} \quad (2.4)$$

In a real solar collector system, the inlet temperature and the fluid mass flow-rate are two degrees of freedom that can be optimised in order to maximise the exergy imparted to the fluid stream flowing through the collector. In Section 2.3.1, an optimisation of these two variables will be performed for a range of collectors and the resulting maximum solar to electrical efficiency will be compared. This is defined as the maximum work divided by the solar irradiation incident on the collector surface:

$$\eta_{sol-e} = \frac{\Delta \dot{X}_{f,rev}}{G \cdot A_{sc}} . \quad (2.5)$$

The solar collector exergy efficiency will also be evaluated, using the expression:

$$\eta_{ex} = \frac{\Delta \dot{X}_{f,rev}}{\dot{X}_{sol,rev}} . \quad (2.6)$$

where $\dot{X}_{sol,rev}$ is defined by Petela [118] as:

$$\dot{X}_{sol,rev} = \left[1 + \frac{1}{3} \left(\frac{T_0}{T_{sun}} \right)^4 - \frac{4T_0}{3T_{sun}} \right] G A_{sc} , \quad (2.7)$$

and where $T_{\text{sun}} = 5,778 \text{ K}$ is the sun's effective black-body temperature. For typical ambient air temperatures in the range of $0 - 30 \text{ }^\circ\text{C}$, Eq. 2.7 can be approximated as $\dot{X}_{\text{sol,rev}} \approx 0.93 \cdot GA_{\text{sc}}$.

The inherent assumption of full reversibility in the process of energy conversion to power downstream of the collector in Eq. 2.3, carries an additional implicit assumption in the use of the Carnot cycle (and associated efficiency expression) across the entire temperature difference available between the external heat source at $T_{\text{sc,out}}$ and external heat sink at T_0 . This is not possible in a practical system, since it requires that heat be transferred into and out of the cycle across a zero temperature difference. In an actual heat engine, during the heat addition and heat rejection processes the temperatures of the working fluid internal to the heat engine will be different (lower and higher, respectively) than the external temperatures of heat source and sink streams. The irreversibility introduced due to the heat transfer across these finite temperature differences will result in an exergy loss and a reduction in maximum work/power from the prediction of Eq. 2.3. Therefore, in neglecting thermal irreversibilities, this expression results in an ideal maximum rather than a practical expectation of the power that can be generated by a solar-powered heat engine.

For the reasons given above, a fully reversible analysis is oversimplified and leads to results that are disconnected from practical experience of relevant systems (significant overestimates of actual efficiency and power). In order to improve on these estimates, one must include some information on the irreversibilities, due to heat transfer and otherwise. A complete analysis accounting for all loss mechanisms is the best way to understand and predict reliably the performance of the system being analysed, however this is a complex and time consuming task, because one must first identify and then account accurately for all forms of irreversibility in the system. As mentioned previously in Section 2.1, an intermediate approach between these two extremes (fully reversible vs. accounting for all irreversibilities) is an endoreversible analysis, which is a simple way of obtaining a slightly improved prediction of realisable performance over a reversible analysis. In this approach the heat transfer into and out of the cycle are modelled explicitly as functions of the temperatures of the heat source/sink and the cycle, while the cycle is still taken as internally (i.e. otherwise) reversible. The endoreversible approach then considers the compromise between the increased heat transfer allowed by higher temperature

differences at the boundaries between the cycle and the heat source and sink, and the increased internal cycle efficiency when the same temperature differences are decreased. The main result from this analysis is obtained by identifying the (inner) cycle temperatures that maximise the work/power output from the cycle, given fixed external temperature. This leads to the so-called Novikov efficiency relation $1 - \sqrt{T_0}/\sqrt{T_1}$, in place of the Carnot efficiency $1 - T_0/T_1$ [99].

A limitation that one must always consider when performing an endoreversible analysis arises from its focus only on the main thermal irreversibility (between the cycle and the external source/sink) in obtaining the maximum work output. This approach will be valid as long as the external irreversibilities related to heat transfer are actually the dominant mode of exergy loss in the heat engine under consideration. A possible way to interpret an endoreversible analysis is that of a “first order” irreversibility analysis, which only focuses on the main thermal irreversibility, assuming that this is dominant.

Returning to our system, if the Carnot efficiency is replaced by the Novikov efficiency, we obtain a revised expression to Eq. 2.3:

$$\begin{aligned}\dot{X}_{\text{enrv}} &= \int_{T_0}^{T_1} \left(1 - \frac{\sqrt{T_0}}{\sqrt{T}}\right) \dot{m}c_p \cdot dT = \dot{m}c_p \left(T_1 + T_0 - 2\sqrt{T_1 T_0}\right) \\ &= \left(\dot{H}_1 - \dot{H}_2\right) \left(1 - \frac{2}{1 + \sqrt{T_{\text{sc,out}}/T_0}}\right),\end{aligned}\tag{2.8}$$

which describes the maximum efficiency of a “semi-ideal” endoreversible heat engine. For this endoreversible cycle it is assumed that the internal hot temperature T'_1 and cold temperature T'_0 of the cycle are lower and higher respectively than the external temperatures of the source (T_1) and sink (T_0), and the Novikov efficiency is then that of a reversible Carnot cycle operating between T'_1 and T'_0 . Equation 2.4 can also be revised to give the expression for endoreversible exergy used by the collector:

$$\begin{aligned}
\Delta \dot{X}_{f,env} &= \dot{m} c_p \left[T_{sc,out} - T_{sc,in} - 2\sqrt{T_0} \left(\sqrt{T_{sc,out}} - \sqrt{T_{sc,in}} \right) \right] + \frac{\dot{m}}{\rho} (p_{sc,out} - p_{sc,in}) \\
&= \left(\dot{H}_{sc,out} - \dot{H}_{sc,in} \right) - \dot{m} \left[2c_p \sqrt{T_0} \left(\sqrt{T_{sc,out}} - \sqrt{T_{sc,in}} \right) + \frac{(p_{sc,out} - p_{sc,in})}{\rho} \right].
\end{aligned} \tag{2.9}$$

The result of evaluating the exergy flow using an endoreversible analysis rather than the fully-reversible analysis presented earlier is a lower but more realistic value of maximum power production, representing a practical maximum of what might be achieved for an optimally designed system. In the results section of this chapter, the calculated exergy outputs from both analyses are compared. The solar to electrical efficiency for the endoreversible analysis will be evaluated using a similar approach to Eq. 2.5 for the reversible analysis. However the solar collector exergy efficiency for the endoreversible case,

$$\eta_{ex} = \frac{\Delta \dot{X}_{f,env}}{\dot{X}_{sol,env}}, \tag{2.10}$$

will be evaluated by considering a special case of a Stefan-Boltzmann engine known as a Müser engine in which irreversible radiation transfer is assumed from the source while reversible heat conduction is assumed to the sink [119]. For the analysis in this chapter we will consider the specific case of a Müser engine with a concentrator achieving full solar concentration, described by De Vos [102], for which the maximum terrestrial solar energy conversion can be evaluated as follows:

$$\dot{X}_{sol,env} = \left(1 - \frac{T_0}{T_{ab}} \right) \left(1 - \frac{T_{ab}^4}{T_{sun}^4} \right) G A_{sc}, \tag{2.11}$$

where T_{ab} is the temperature of an absorber surface, optimised for maximum power at each time instant so that:

$$4T_{ab}^5 - 3T_0 T_{ab}^4 - T_{sun}^4 T_0 = 0. \tag{2.12}$$

For typical ambient air temperatures in the range of 0–30 °C, Eq. 2.11 can be approximated as $\dot{X}_{sol,env} \approx 0.85 \cdot G A_{sc}$.

2.2.3 Annual assessment using solar irradiance data

An annual assessment of the maximum power generation capability in a UK setting is performed for each collector type and compared to a southern European location: Cyprus. Hourly weather data for London (latitude 51.2° , longitude -0.2°) and Larnaca (latitude 34.9° , longitude 33.6°) are used for this analysis, available from the International Weather for Energy Calculations (IWECC) published by ASHRAE [120]. The data is extracted from 18 years of meteorological measurements (from 1982 to 1999), such that the annual data series is assembled from twelve typical meteorological months selected from the long-term time series [121]. Hourly-averaged data is considered suitable for the present analysis, for which it will be assumed that the collectors perform under quasi-steady state conditions at each time-instant (neglecting inertial effects due to the thermal capacity of the collector). However, it is recognised that time-averaging results in a loss of information, particularly for a location such as the UK, which undergoes short time-period variations in solar irradiance due to cloud cover. As part of the annual assessment, the hourly solar irradiance and ambient temperature data are used to optimise the collector temperature and flow-rate for each hour of the year to provide maximum exergy conversion at each time-instant. The total maximum work output over the annual period

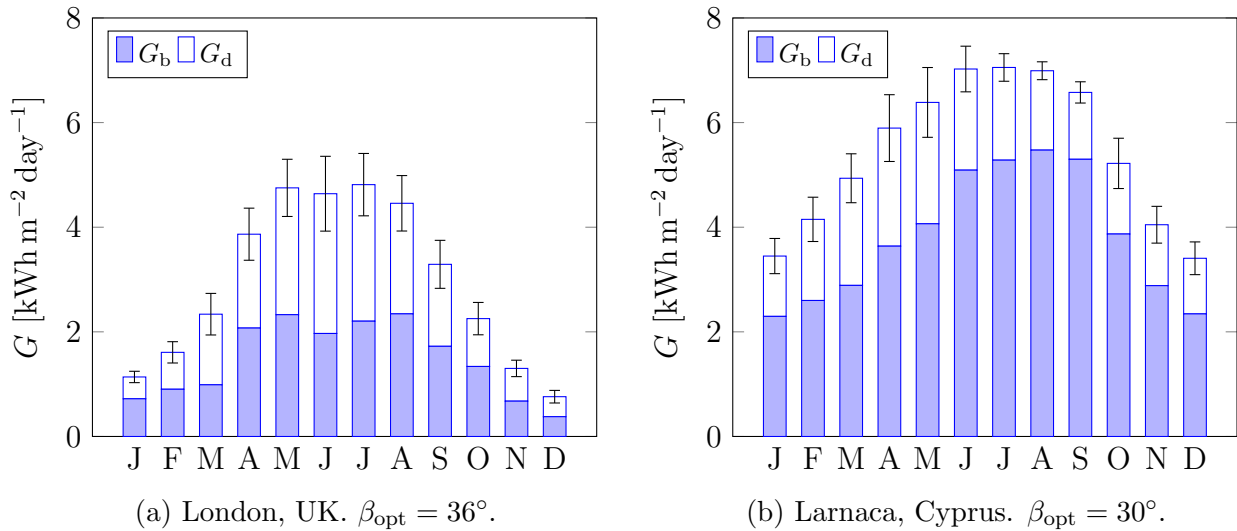


Figure 2.3: Mean daily global solar irradiation on optimally inclined plane, shown as the sum of the mean direct beam and mean diffuse irradiation components. Error bars denote the standard deviation of the daily global irradiation. β_{opt} is determined by way of an annual optimisation to maximise the irradiation received on the tilted plane (see Appendix A.)

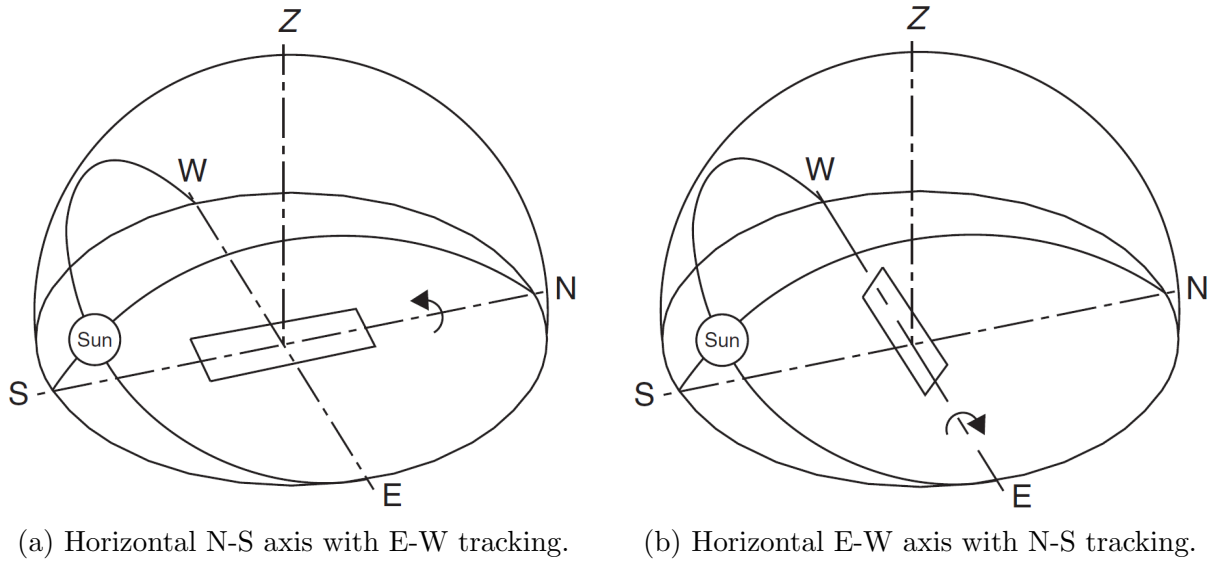


Figure 2.4: Single axis tracking geometries for line-focussing parabolic trough collectors [122].

will then be compared for the two locations and for each collector.

The collector array is assumed to be orientated due south in both locations, as both locations are at latitudes north of the maximum solar declination angle (23.45°). For the analysis of non-tracking collectors where a fixed orientation and inclination angle is maintained throughout the year, a tilt angle optimisation is performed, to maximise the annual global (direct + diffuse) irradiation on the collector plane. The optimal inclination angle for each location is then used for the annual maximum power analysis. The respective optimum inclination angles are found to be 36° for London and 30° for Larnaca. The daily direct-beam and diffuse irradiance received on the tilted plane (plotted as averages for each month of the year) is shown in Fig. 2.3. The equations for the calculation of beam and diffuse irradiance on the tilted plane are included in Appendix A. For the line-focussing PTC, an ability to track the sun on at least one axis is required. A comparison will be performed between the following: (i) a PTC with its axis aligned horizontally north-south, with continuous tracking of the sun east to west over the course of the day (illustrated in Fig. 2.4a); (ii) a PTC with its axis aligned horizontally east-west, with continuous tracking along the north-south plane (illustrated in Fig. 2.4b); (iii) a PTC with perfect continuous tracking along two axes. The latter is not a common configuration for PTC systems, and can result in shadowing complications where array area is limited. In the calculations it is assumed that through continuous adjustment the solar incident angle is

equal to zero at all times for two-axis tracking, and minimised for single-axis tracking.

2.2.4 Heat rejection and cold sink temperature

When maximising the efficiency of the solar-driven heat engine it is thermodynamically desirable to reject heat at the lowest possible temperature within the limitations of the available resources. Mains cold water is thermodynamically preferable as the cooling medium as it tends to be available at a lower temperature than ambient air during summer when solar potential is highest, although if large quantities are required the associated cost may be prohibitive. It is also an oversimplification to assume that mains water is available at a constant temperature throughout the year. In reality, mains water temperature usually exhibits a seasonal fluctuation, the extent of which varies geographically. The average mains water temperature in the UK as reported by Murphy [123] is 10 °C with a seasonal variation of ± 2.6 °C, although in some UK regions mains water temperatures can reach local extremes of 4 °C in the winter and 25 °C in the summer according to the advisory body Water-UK [124]. Another factor affecting domestic cold water supply temperatures is the use of a cold water storage tank. A study of domestic hot water use in UK homes by the Energy Saving Trust [125] reported a mean cold water supply temperature of 13.4 °C among 38 properties without cold water storage tanks and 16.2 °C among 68 cases with cold water storage tanks. The seasonal and geographical variation in cold water temperatures was not reported in the aforementioned study. For Cyprus, information on mains water supply temperatures is less widely available. Drinking water is supplied directly from reservoirs while cold water for other domestic purposes is often stored locally to dwellings in cold water tanks.

As part of the current analysis, air and water are compared as heat rejection media. The air temperature is assumed to vary with the local ambient condition, whereas that of water is modelled as a sinusoidal oscillation over the annual period about a 10 °C mean and with a 2.6 °C amplitude as reported by Murphy [123]. The peaks and troughs are set to coincide approximately with those of the local air temperature (peak in late July and trough in late January). A plot of monthly mean air and mains water temperatures is shown in Fig 2.5. For Cyprus, information on local mains water temperatures was not available, and instead the

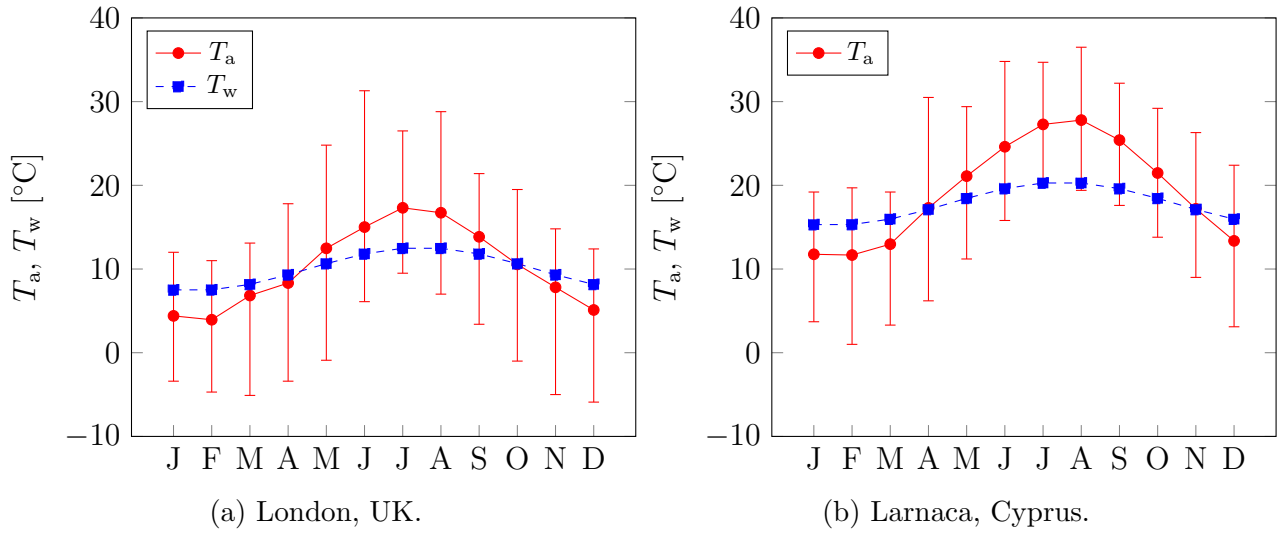


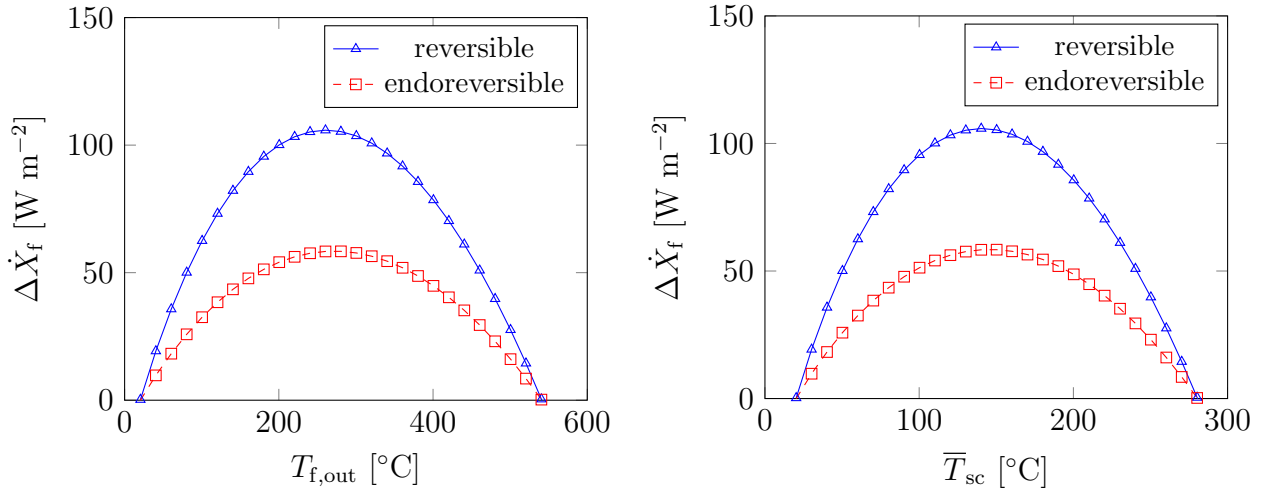
Figure 2.5: Monthly mean air temperatures for London and Larnaca (solid red lines), with vertical bars denoting monthly minimum and maximum values. The dashed blue lines show representative mains water temperatures for the two locations (for Larnaca the closest available data was for Athens, Greece).

water temperatures shown in Fig 2.5b are based on the annual mean mains water temperature reported for Athens [126], with the same amplitude of annual variation about the mean as shown for the UK. Despite warmer average air temperatures than the UK, mains water in Cyprus is supplied from dams set high in the mountains, and therefore it is expected that supply temperatures may be significantly cooler than the average air temperature condition in Larnaca [127].

2.3 Results and Discussion

2.3.1 Reversible vs. endoreversible maximum work analysis

In this section, the relationship between the collector operating parameters and the reversible and endoreversible maximum power output is investigated. The operating parameters of interest are the fluid mass flow-rate and the collector temperature. The analysis is presented for an example ETC collector (the Thermomax DF-100) in Figs 2.6–2.8, and the fixed parameters used to model the collector are listed in Table 2.3. Results are presented as the increase in fluid flow exergy $\Delta\dot{X}_f$ experienced by the fluid stream flowing through the collector, per m^2 of

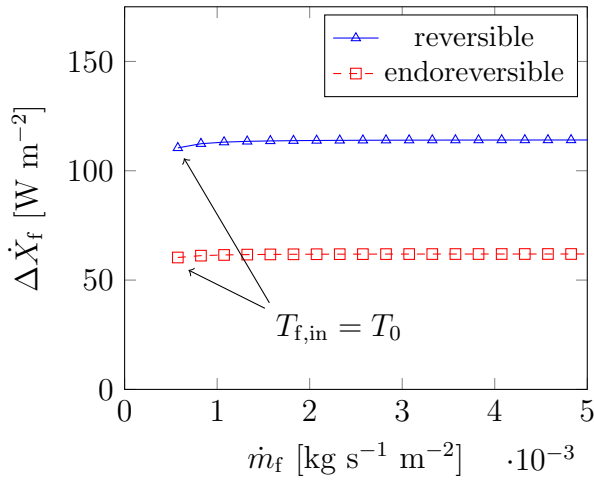


(a) Exergy use as a function of the collector outlet temperature. (b) Exergy use as a function of mean collector temperature.

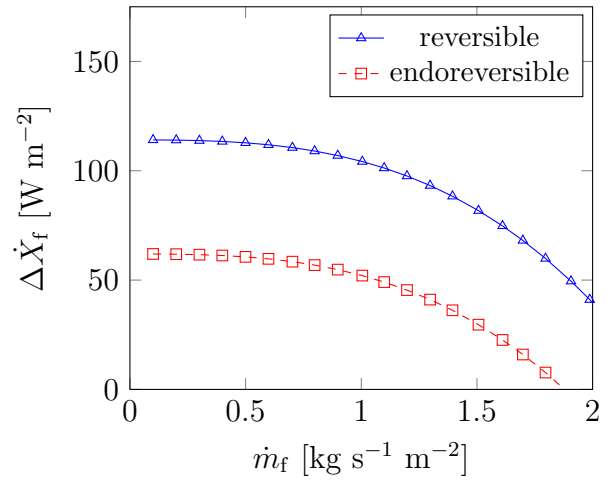
Figure 2.6: Solar collector reversible and endoreversible exergy use plotted as a function of collector temperature, for fixed fluid inlet temperature $T_{f,in} = 20\text{ }^{\circ}\text{C}$ and $G = 1000\text{ W m}^{-2}$. Presented for an example ETC-DF collector.

collector aperture area. Fluid mass flow rates are also presented per m^{-2} of collector area. The collector fluid is modelled as water, with a c_p of $4.2\text{ kJ kg}^{-1}\text{ K}^{-1}$; the temperature dependence of which is neglected in the model. A fixed default value of $20\text{ }^{\circ}\text{C}$ is chosen for the environmental temperature.

The reversible and endoreversible exergy increase for the example collector are compared in Fig. 2.6 for a steady-state irradiance of 1000 W m^{-2} . The collector inlet temperature is set to the same value as the environmental “dead space” temperature ($20\text{ }^{\circ}\text{C}$), and the collector outlet temperature is varied by adjusting the mass flow-rate. As the mass flow-rate is increased, the fluid temperature rise in the collector decreases and $T_{f,out}$ approaches $T_{f,in}$. Thus, high values of $T_{f,out}$ correspond to low values of $\dot{m}_f$. For a fixed value of $T_{f,in}$, higher values of $T_{f,out}$ result in a higher mean collector temperature, $\bar{T}_{sc}$, as shown in Fig. 2.6b. When the collector temperature is sufficiently high, the collector’s efficiency (given by Eq. 2.2) becomes negative due to increased thermal losses to the environment, resulting in a net exergy loss by the fluid stream in the collector. The mean collector temperature at which the net exergy gain becomes negative corresponds very closely to the collector stagnation temperature, reported as $286\text{ }^{\circ}\text{C}$ in the Keymark test report for the collector used in the analysis[128].



(a) Low mass flow-rate range.



(b) High mass flow-rate range.

Figure 2.7: Solar collector reversible and endoreversible exergy use plotted as a function of mass flow rate through the collector, for fixed $\bar{T}_{sc} = 140$ °C and $G = 1000$ W m⁻². Presented for an example ETC-DF collector.

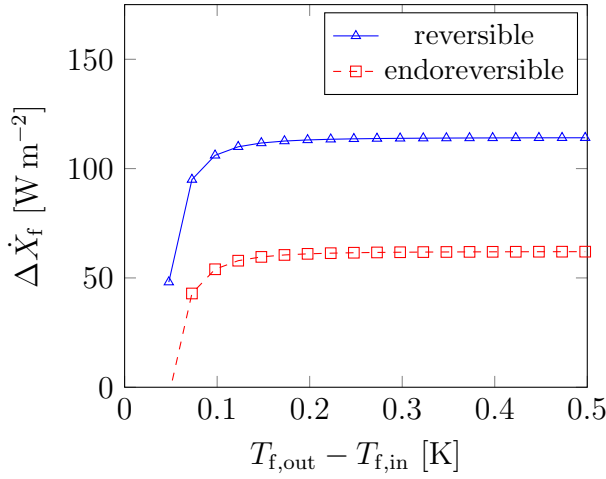
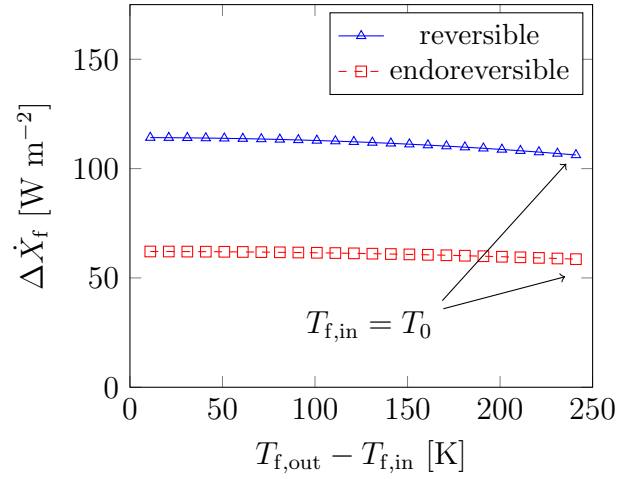
(a) Small ΔT_f (corresponding to large $\dot{m}_f$)(b) Large ΔT_f (corresponding to small $\dot{m}_f$)

Figure 2.8: Solar collector reversible and endoreversible exergy use plotted against fluid temperature rise through the collector as a function of mass flow-rate, for fixed $\bar{T}_{sc} = 140$ °C and $G = 1000$ W m⁻². Presented for an example ETC-DF collector.

In Figs. 2.7 and 2.8, the mean collector temperature is fixed to the optimum value corresponding to maximum exergy increase in Fig. 2.6b (140 °C), and the fluid mass flow-rate is varied. Low mass flow-rates (Fig. 2.7a) correspond to a large temperature difference between the collector inlet and outlet (Fig. 2.8b), and vice-versa. The lower limit shown for the mass flow-rate is the value at which $T_{f,in} = T_0$, and corresponds to the largest ΔT_f of 240 K. For this scenario the reversible exergy gain is 106 W m⁻² and the endoreversible exergy gain is 58 W m⁻². As the flow-rate is increased, there is a slight increase in exergy gain, reaching a

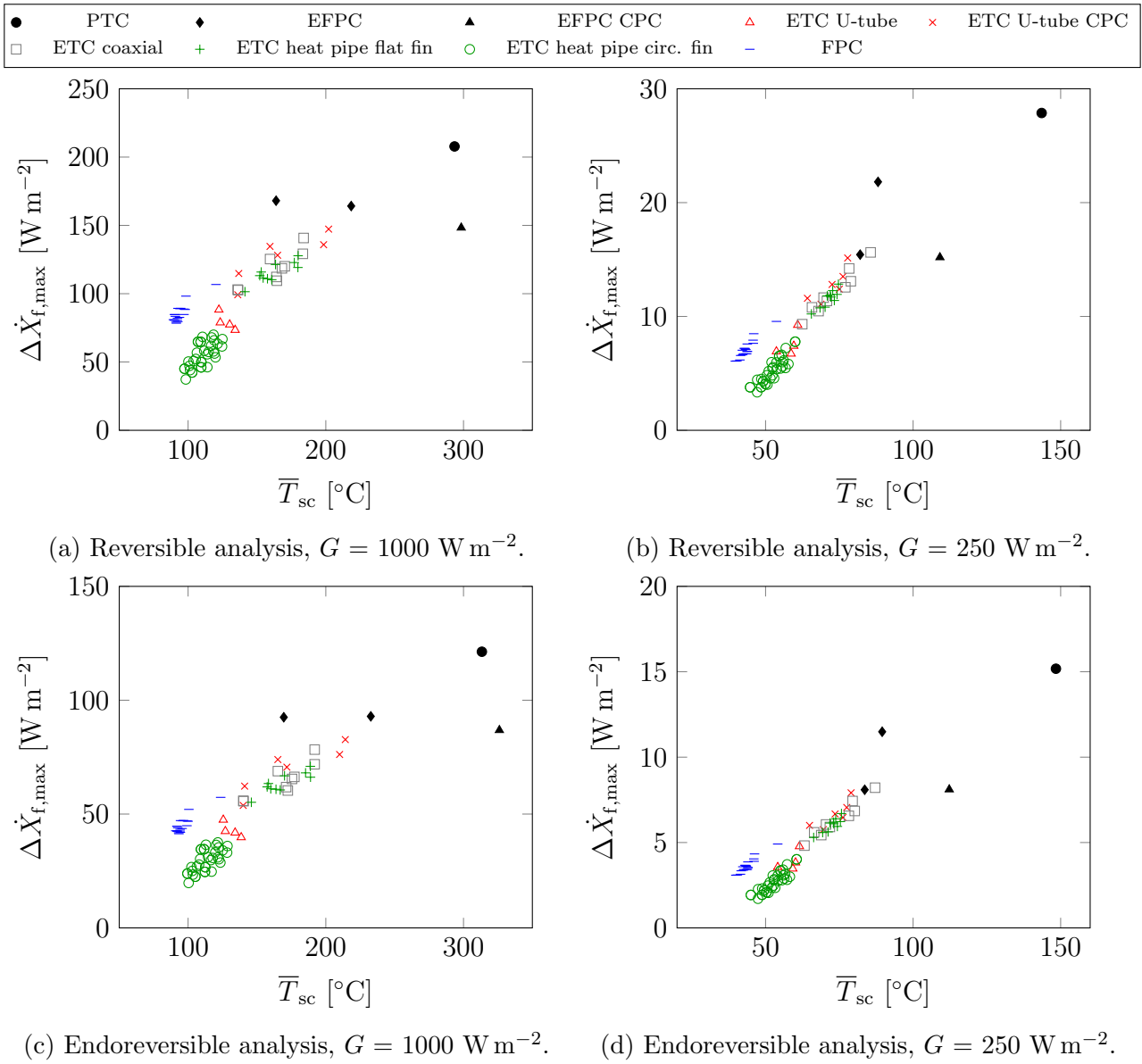


Figure 2.9: Solar collector maximum exergy use plotted against optimal collector temperature for various collector types; for peak solar and annual-average solar conditions, and using the fully-reversible and endoreversible analysis.

maximum of 114 W m^{-2} for the reversible analysis (62 W m^{-2} for the endoreversible analysis) at $\dot{m}_f = 0.02 \text{ kg s}^{-1} \text{ m}^{-2}$ and $\Delta T_f = 5 \text{ K}$. Only a very small variation ($< 2\%$) in exergy gain is observed over the range $0.001 - 0.5 \text{ kg s}^{-1} \text{ m}^{-2}$. At higher mass flow-rates ($> 0.5 \text{ kg s}^{-1} \text{ m}^{-2}$) the exergy destruction associated with pressure drop in the collector becomes significant, and eventually results in a net exergy loss. If the exergy destruction associated with pressure drop were omitted from Eqs. 2.3 and 2.8 then the exergy gained by the fluid would reach its maximum as $\dot{m}_f$ becomes infinite and ΔT_f approaches zero. It is found that the exergy gained by the fluid stream in this case would be equal to $\eta_{\text{Carnot}} \cdot \eta_{sc} \cdot G$ in the reversible analysis, or $\eta_{\text{Novikov}} \cdot \eta_{sc} \cdot G$

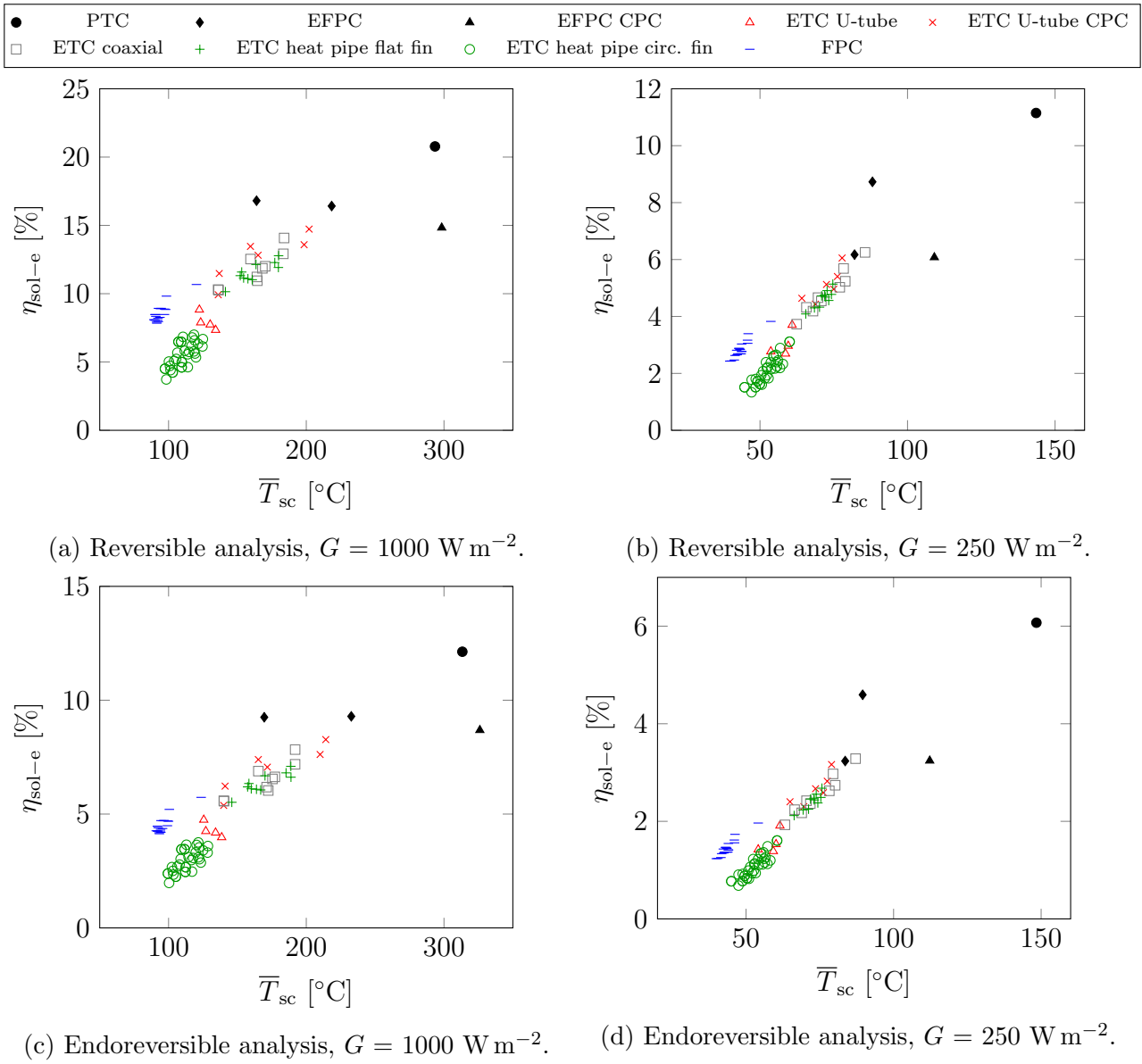


Figure 2.10: Maximum solar to work efficiency plotted against optimal collector temperature for various collector types; for peak solar and annual-average solar conditions, and using the fully-reversible and endoreversible analysis.

in the endoreversible analysis.

The reversible and endoreversible maximum power for a sample of 87 collector designs are plotted in Fig 2.9, for a peak irradiance condition of 1000 W m^{-2} (1 Sun) and for a lower irradiance condition of 250 W m^{-2} , which is the annual average solar irradiance received during daylight hours in the UK on the optimally-tilted (36°) plane. The maximum exergy gain of the fluid passing through the collector is plotted against the optimised mean collector temperature. The collectors are grouped according to their design characteristics and the results are reported

Table 2.1: Reversible and endoreversible maximum power conditions for various solar collector designs (shown as the sample mean for each collector type), for $G = 1000 \text{ W m}^{-2}$ and $T_a = 20^\circ\text{C}$.

	$\Delta\dot{X}_{f,\max}$ W m^{-2}	$\overline{T}_{sc}$ $^\circ\text{C}$	$\dot{m}_f$ $\text{kg}/(\text{s m}^2)$	ΔT_f K	$\eta_{\text{sol-e}}$ $\%$	η_{ex} $\%$
<i>Reversible analysis</i>						
PTC	207.8	293.5	0.031	3.3	20.8	22.3
EFPC	166.1	191.2	0.016	7.7	16.6	17.8
EFPC CPC	148.3	298.4	0.007	11.0	14.6	15.9
ETC DF - Utube	79.5	127.5	0.010	7.5	8.0	8.5
ETC DF - Utube CPC	126.6	166.3	0.015	6.2	12.7	13.6
ETC DF - coaxial	117.9	162.9	0.010	8.4	11.8	12.7
ETC HP - flat fin	115.4	162.0	0.011	7.9	11.5	12.4
ETC HP - circular fin	55.1	110.4	0.015	3.9	5.5	5.9
FPC	85.6	94.3	0.015	8.2	8.6	9.2
<i>Endoreversible analysis</i>						
PTC	121.3	313.3	0.027	3.7	12.1	14.2
EFPC	92.7	201.1	0.013	9.1	9.3	10.9
EFPC CPC	86.8	326.0	0.006	12.6	8.7	10.2
ETC DF - Utube	42.9	131.3	0.008	9.2	4.3	5.0
ETC DF - Utube CPC	69.9	173.7	0.012	7.6	7.0	8.2
ETC DF - coaxial	64.9	169.5	0.008	10.2	6.5	7.6
ETC HP - flat fin	63.5	168.5	0.009	9.3	6.4	7.5
ETC HP - circular fin	29.5	113.1	0.012	4.7	3.0	3.5
FPC	45.3	96.3	0.012	10.0	4.5	5.3

in Tables 2.1 and 2.2 as the mean values for each group. In Fig. 2.10, the maximum power is shown as a fraction of the solar irradiance (the maximum solar-to-electrical efficiency).

The following general trends are observed from the results: (i) for the range of collectors examined, higher maximum exergy efficiency generally corresponds to higher optimal operating temperatures; (ii) the reversible and endoreversible maximum power in all cases are found to be very close to (within 1% of) the maximum values of $\eta_{\text{Carnot}} \cdot \eta_{sc} \cdot G$ and $\eta_{\text{Novikov}} \cdot \eta_{sc} \cdot G$ respectively, indicating that for most collectors the pressure drop has only a small effect on the net exergy gain at the optimal flow-rate; (iii) under peak irradiance conditions, the endoreversible maximum power is 41–47% lower than the fully-reversible maximum power for the range of collectors; (iv) under low irradiance conditions this difference is slightly larger (47–49%); (v) the optimum collector temperature for endoreversible maximum power is higher than the optimum temperature for fully-reversible maximum power, and the difference is found

Table 2.2: Reversible and endoreversible maximum power conditions for various solar collector designs (shown as the sample mean for each collector type), for $G = 250 \text{ W m}^{-2}$ and $T_a = 20^\circ\text{C}$.

	$\Delta\dot{X}_{f,\max}$ W m^{-2}	$\overline{T}_{sc}$ $^\circ\text{C}$	$\dot{m}_f$ $\text{kg}/(\text{s m}^2)$	ΔT_f K	$\eta_{\text{sol-e}}$ $\%$	η_{ex} $\%$
<i>Reversible analysis</i>						
PTC	27.9	143.6	0.014	1.6	11.1	12.0
EFPC	18.6	85.1	0.006	4.0	7.5	8.0
EFPC CPC	15.2	109.1	0.003	4.8	6.1	6.5
ETC DF - Utube	7.6	58.2	0.004	4.4	3.0	3.3
ETC DF - Utube CPC	12.8	72.4	0.006	3.5	5.1	5.5
ETC DF - coaxial	12.1	72.9	0.004	4.8	4.8	5.2
ETC HP - flat fin	11.6	71.4	0.005	4.0	4.6	5.0
ETC HP - circular fin	5.2	52.3	0.006	2.1	2.1	2.2
FPC	7.1	3.6	0.006	4.7	2.8	3.1
<i>Endoreversible analysis</i>						
PTC	15.2	148.4	0.019	1.1	6.1	7.1
EFPC	9.8	86.6	0.005	4.8	3.9	4.6
EFPC CPC	8.1	112.2	0.003	3.5	3.2	3.8
ETC DF - Utube	3.9	58.8	0.003	5.5	1.6	1.8
ETC DF - Utube CPC	6.6	73.5	0.004	2.8	2.7	3.1
ETC DF - coaxial	6.3	74	0.003	6.0	2.5	3.0
ETC HP - flat fin	6.0	72.4	0.004	4.8	2.4	2.8
ETC HP - circular fin	2.7	52.7	0.005	2.5	1.1	1.3
FPC	3.6	43.8	0.004	6.0	1.5	1.7

to be more significant at higher irradiance levels.

The collector that shows the highest maximum exergy conversion under both high and low irradiance conditions is the PTC. In the fully reversible analysis for peak irradiance conditions, the PTC delivers a maximum 207.8 W m^{-2} exergy increase to the fluid stream at an optimal operating temperature of 293.5°C , corresponding to a maximum solar to electrical efficiency of 20.8%. Meanwhile in the endoreversible analysis the maximum exergy conversion by the PTC is 121.3 W m^{-2} at an optimal temperature of 313.3°C , corresponding to a solar to electrical efficiency of 12.1%. For the low irradiance condition the maximum exergy conversion, solar to electrical efficiency and corresponding optimal temperature are found to be considerably lower. For the reversible analysis these are 27.9 W m^{-2} , 11.1% and 143.6°C respectively, while for the endoreversible analysis these are 15.2 W m^{-2} , 6.1% and 148.4°C . The relative performance of the PTC compared to the non-concentrating collectors under low irradiance conditions does

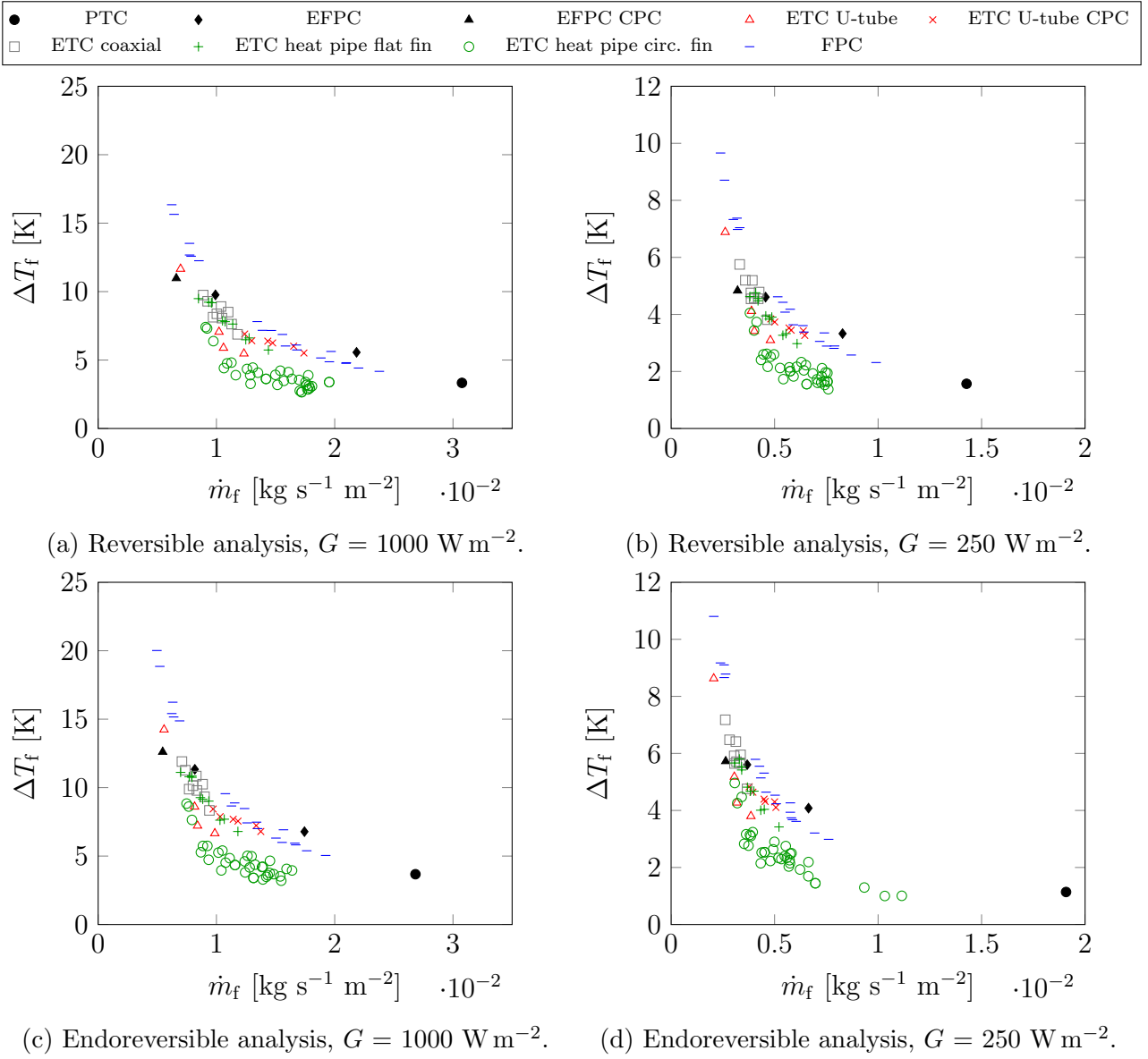


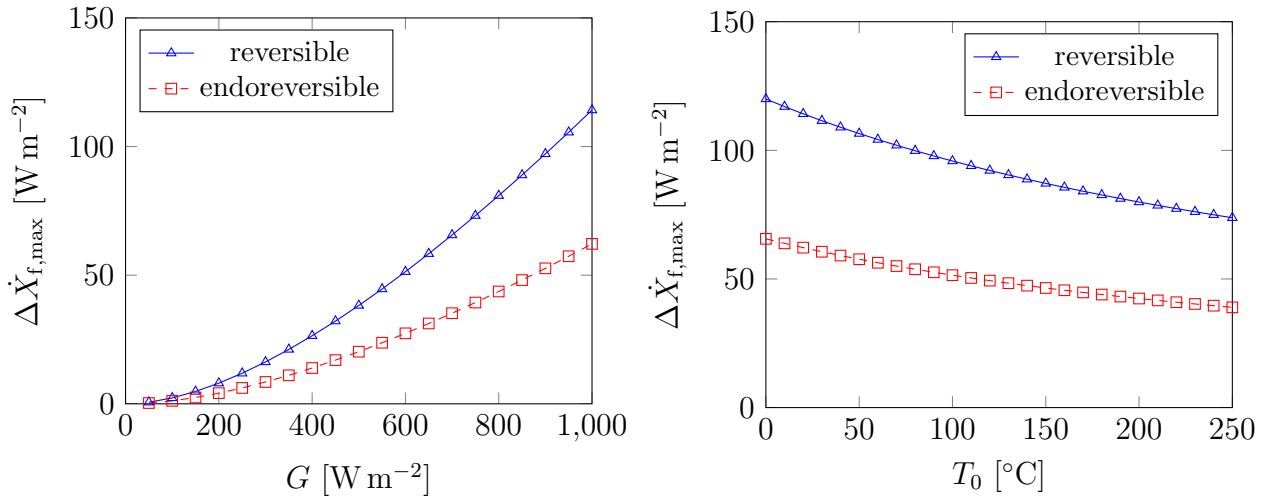
Figure 2.11: Fluid temperature rise plotted against mass flow-rate for maximum power conditions under peak solar and annual-average solar conditions, and determined using the fully-reversible and endoreversible analysis.

not account for the effect of scattering, and assumes that all irradiance is uni-directional and usable for a line-focussing concentration system. In Section 2.3.2, consideration will be given to beam and diffuse irradiance components.

Both EFPC collectors show high maximum exergy conversion under high irradiance conditions compared to the other non-concentrating collectors, although there is a significant difference in optimum operating temperature between the two EFPC collectors considered in the analysis. Under low irradiance conditions, the same two collectors show a significant divergence

in maximum exergy conversion. This is due to the difference in values of the first-order heat loss coefficient (c_1), which leads to a decrease in collector efficiency according to Eq. 2.2 at low irradiance as well as at high collector temperatures. A wide range of first and second-law efficiencies are shown among the ETC designs. Direct-flow (DF) and heat pipe (HP) collectors that incorporate a flat-fin absorber design show higher efficiencies than those with a circular fin (excluding those with a CPC reflecting device). However, this analysis does not consider the effect of non-perpendicular incidence angles for which circular fins may offer an advantage (this will be taken into account in the annual analysis). The poorest efficiency is found for the circular-fin heat pipe collectors, which are in most cases also the most cost-effective due to their simple manufacturing process. The FPC collectors show higher first and second-law efficiency than the circular-fin ETC-HP collectors due to their higher gross-area efficiency (defined as the ratio of the aperture area to the gross module area), but at a lower optimal temperature. At high irradiance the results for the ETC and FPC collector types show a strong grouping among the different design variants. The standard deviation of the maximum power values under peak irradiance is 14% among the U-tube ETC designs, 21% among the coaxial ETC designs, 17% among the heat-pipe ETC designs and 8% among the FPC designs. The grouping of the results becomes less distinct for the lower irradiance analysis, with more considerable overlap between the performance of the different collector types. For the PTC and EFPC collectors, the smaller sample size means that the results are less applicable generally to a wider range of collectors of the same type. In particular EFPC collectors are a relatively new technology compared to ETC and FPC collectors at present and the few market-ready models that exist are based on widely varying design and manufacturing processes; hence a significant variation in their performance is expected.

It is instructive to compare the temperatures and flow-rates determined for maximum exergy with the recommended operational ranges specified by the manufacturers, while noting that the collector temperatures shown in Figs. 2.9 and 2.10 and the flow-rates shown in Fig. 2.11 represent optimum values predicted from an idealised thermodynamic analysis. Taking the Thermomax DF-100 ETC as an example, the maximum exergy under peak irradiance (1000 W m^{-2}) corresponds to an optimum collector temperature of 143°C (reversible analysis)



(a) Maximum exergy use as a function of solar irradiance. (b) Maximum exergy use as a function of cold sink temperature.

Figure 2.12: Solar collector maximum exergy use plotted against (a) varying solar irradiance, for fixed $T_0 = 20^{\circ}\text{C}$; and (b) varying cold sink temperature, for fixed $G = 1000 \text{ W m}^{-2}$. Shown for an example ETC-DF collector and determined using the fully-reversible and endoreversible analysis.

or 148°C (endoreversible analysis). This is higher than the typical operating temperature range for this type of collector, which according to the manufacturers test report are $< 90^{\circ}\text{C}$ [128]. For the annual average irradiance condition (250 W m^{-2}), the optimum collector temperature for the Thermomax DF-100 collector is $67\text{--}68^{\circ}\text{C}$, which is well within the typical operating range for the solar collector. The range of mass flow-rates corresponding to maximum exergy at high and low irradiance ($0.006\text{--}0.018 \text{ kg s}^{-1} \text{ m}^{-2}$) is found to match the manufacturer's recommended flow rate range ($0.008\text{--}0.019 \text{ kg s}^{-1} \text{ m}^{-2}$) relatively well.

Under the low irradiance condition, it is found that for most of the collectors investigated the optimal flow-rate for maximum power is below the lower limit recommended by the manufacturer. However it is also found that the exergy converted by the collector is relatively insensitive to variation of the flow-rate over a wide range of values, as was demonstrated in Fig. 2.7. Thus in all cases it is found that when the manufacturers' recommended flow-rate range is applied as a constraint in the optimisation the maximum power changes by less than 1%. Moreover, for the majority of collectors only a small variation in exergy use (typically less than 5%) is observed across the entire recommended flow-rate range.

Figure 2.12 (a) and (b) show plots of solar collector maximum exergy against solar irradiance and cold sink temperature respectively. It is important to note the nonlinearity, i.e. increasing gradient of exergy output with irradiance. This has important implications for the suitability of using time-averaged irradiance data to predict the maximum annual work output, because the exergy output will be disproportionately higher under peak irradiance conditions than average irradiance conditions. This implies that the use of higher resolution data will result in higher predictions for the total maximum work that can be delivered for a particular collector.

2.3.2 Annual assessment

In this section, the results from the annual assessment will be presented. As described in Section 2.2.3 the annual assessment methodology involves the optimisation of collector temperature and flow-rate for each hour of the year to provide maximum exergy conversion under varying irradiance and ambient temperature conditions. The annual assessment is performed for a selection of collectors representative of each type: PTC, EFPC, ETC-DF, ETC-HP and FPC. The collectors chosen are those whose maximum power and optimum operating temperature under peak irradiance are close to the mean values for the respective collector types presented in Table 2.1. The design and performance parameters used to model each collector are listed in Table 2.3. Results from the annual assessment in the climates of London (UK) and Larnaca (Cyprus) are shown in Table 2.4.

2.3.3 Comparison between UK and Cyprus

The geographic location is found to have a large influence on the maximum work potential from the solar collectors. The total annual (beam + diffuse) irradiation received on the optimally tilted plane is $1074 \text{ kWh m}^{-2} \text{ yr}^{-1}$ in London compared to $1983 \text{ m}^{-2} \text{ yr}^{-1}$ in Larnaca, which constitutes a difference of 85% in solar resource. Furthermore, Fig. 2.3 shows that there is a significant difference in the *quality* of the solar resource: only 50% of the irradiation re-

Table 2.3: Solar collector properties used in the annual maximum work assessment.

Collector type	Manufacturer & model	$\eta_0^{(i)}$ (-)	$c_1^{(i)}$ (-)	$c_2^{(i)}$ (-)	A_{aper} (m ²)	A_{gro} (m ²)	$K_{\theta_{\text{b}},\text{tr}}^{(ii)}$ (-)	$K_{\theta_{\text{b}},\text{lo}}^{(ii)}$ (-)	$K_{\theta_{\text{d}}}^{(iii)}$ (-)
PTC	NEP Polytrough 1800	0.69	0.36	0.001	18.45	21.78	0.93	-	0
EFPC	TVP Solar MT-Power	0.76	0.51	0.007	1.84	1.95	0.91	0.91	0.97
ETC DF:									
- Utube	AMK-Collectra OWR 20	0.78	1.95	0.006	1.98	3.46	1.52	0.95	n/a
- Utube CPC	De Dietrich Power 10	0.76	1.42	0.003	1.14	1.45	0.96	0.97	n/a
- coaxial	Thermomax DF 100 20	0.77	1.43	0.006	2.15	2.83	1.05	0.99	0.88
ETC HP:									
- flat fin	Enertech Enersol HP65-20	0.71	1.25	0.005	2.27	2.90	0.99	0.94	n/a
- circular fin	Dallinger Sonnenpower 22	0.62	1.33	0.010	2.07	3.64	1.5	0.93	n/a
FPC	Caotec CAO TLP	0.80	3.80	0.007	1.82	2.10	0.94	0.94	n/a

⁽ⁱ⁾ Efficiency curve coefficients related to aperture area.

⁽ⁱⁱ⁾ $K_{\theta_{\text{b}},\text{tr}}$ and $K_{\theta_{\text{b}},\text{lo}}$ values reported for incident angles of 50°. Full IAM curves available from Refs. [110, 117, 128]

⁽ⁱⁱⁱ⁾ Where stated “n/a”, a default value of $K_{\theta_{\text{d}}} = 1$ is assumed.

ceived annually in London is direct-beam radiation, with the remainder diffuse radiation that is scattered by clouds and atmospheric particulates (ground-reflected irradiation is omitted in this analysis); while for Larnaca 70% of the annual irradiation is direct-beam. The Larnaca solar resource also undergoes less variation on a daily and seasonal timescale, with a standard deviation of 0.2 kWh m⁻² day⁻¹ (3%) between the months of June to August compared to 1.8 kWh m⁻² day⁻¹ (39%) over the same period for London.

For each of the collectors assessed, the maximum annual work output determined in the UK analysis is less than half that determined in the Cyprus analysis. The difference is most significant for the concentrating PTC, which requires direct-beam irradiance in order to operate and therefore is of limited potential in the London UK setting where the diffuse irradiance fraction is significantly larger. In the Cyprus assessment, the PTC collector with east-west (single-axis) tracking is found to have a maximum work potential of 308 kWh m⁻² yr⁻¹ (reversible) or 175 kWh m⁻² yr⁻¹ (endoreversible). This is 19% higher than that for the best non-concentrating collector, the EFPC. For the UK on the other hand, the EFPC is found to have a 22% higher maximum work potential than the PTC with single-axis tracking (125 kWh m⁻² yr⁻¹ reversible, and 67 kWh m⁻² yr⁻¹ endoreversible). Expressing the maximum work as an annual-averaged maximum power in Watts for a 15 m² collector array area (representative of a typical domestic roof-space) results in 528 W (reversible) and 299 W (endoreversible) for the PTC in Cyprus;

Table 2.4: Annual reversible and endoreversible maximum total exergy conversion for various solar collector designs, and the corresponding annual average collector mean operating temperature.

	London				Larnaca			
	$X_{\max}$ kWh m ⁻²	$\bar{T}_{\text{sc,avg}}$ °C	$\eta_{\text{sol-e}}$ %	η_{ex} %	$X_{\max}$ kWh m ⁻²	$\bar{T}_{\text{sc,avg}}$ °C	$\eta_{\text{sol-e}}$ %	η_{ex} %
<i>Reversible analysis</i>								
PTC - full tracking	136.9	115.3	10.7	11.5	377.4	188.7	15.4	16.4
PTC - E-W tracking	97.5	99.6	8.8	9.4	308.4	174.4	14.0	15.1
PTC - N-S tracking	96.3	95.1	8.8	9.4	275.2	158.3	13.3	14.3
EFPC	124.8	65.7	11.6	12.5	270.1	100.8	13.6	14.6
ETC DF - U-tube	98.8	47.7	9.2	9.9	229.0	80.9	11.5	12.4
ETC DF - U-tube CPC	100.5	55.8	9.4	10.0	227.6	92.8	11.5	12.3
ETC DF - coaxial	92.3	49.9	8.6	9.2	215.1	83.5	10.8	11.6
ETC HP - flat fin	95.6	54.1	8.9	9.5	216.1	88.8	10.9	11.7
ETC HP - circular fin	72.4	44.6	6.7	7.2	164.9	73.7	8.3	8.9
FPC	56.5	33.3	5.3	5.6	133.7	57.5	6.7	7.2
<i>Endoreversible analysis</i>								
PTC - full tracking	77.6	120.4	6.1	7.1	215.4	198.1	8.8	10.3
PTC - E-W tracking	54.9	103.5	5.0	5.8	174.9	182.5	8.0	9.4
PTC - N-S tracking	54.4	98.8	5.0	5.8	156.4	165.5	7.6	8.9
EFPC	67.2	66.9	6.3	7.3	146.4	103.1	7.4	8.7
ETC DF - U-tube	52.5	48.4	4.9	5.7	122.5	82.4	6.2	7.3
ETC DF - U-tube CPC	53.9	56.9	5.0	5.9	123.2	95.1	6.2	7.3
ETC DF - coaxial	49.2	50.7	4.6	5.4	115.5	85.2	5.8	6.8
ETC HP - flat fin	51.2	55.0	4.8	5.6	116.6	90.7	5.9	6.9
ETC HP - circular fin	38.2	45.2	3.6	4.2	87.5	74.7	4.4	5.2
FPC	29.5	33.6	2.7	3.2	70.2	58.1	3.5	4.2

and 214 W (reversible) and 115 W (endoreversible) for the EFPC in the UK. The average power values are obtained by dividing the annual maximum work in kWh by the total number of hours in the year. This calculation does not exclude non-daylight hours, which account for approximately 47% of the year.

2.3.4 Comparison between collector designs and tracking arrangement

The monthly endoreversible exergy output for the concentrating PTC (E-W tracking variant) and non-concentrating EFPC are compared in Fig. 2.13 for Cyprus and the UK. For Cyprus, the

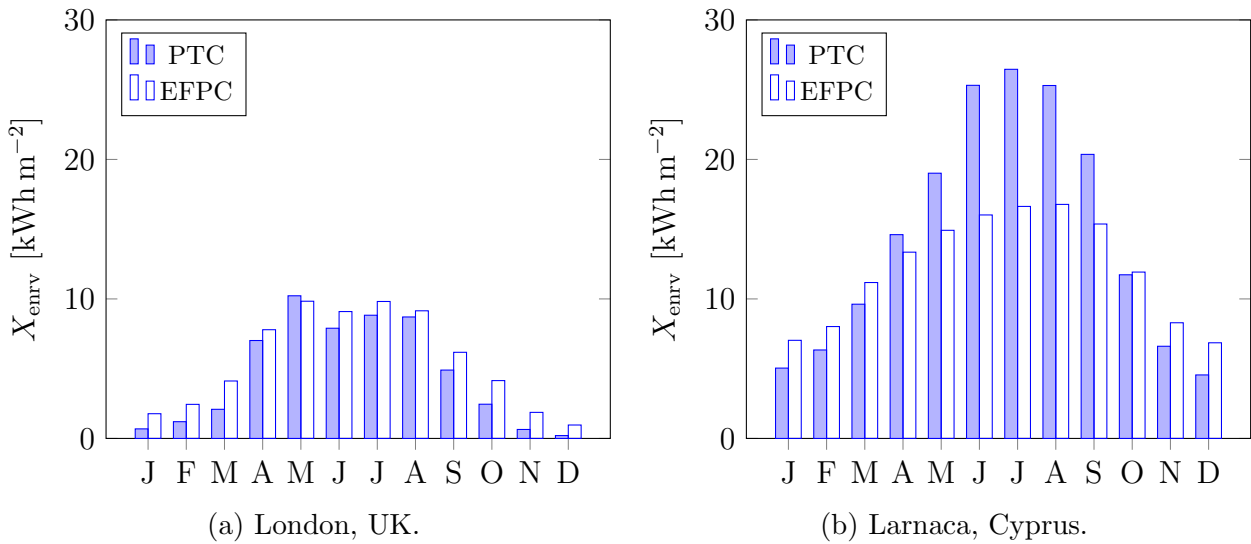


Figure 2.13: Endoreversible maximum work shown for each month of the year for the concentrating PTC (single-axis, E-W tracking) and non-concentrating EFPC.

PTC shows a 41% higher exergy output than the EFPC during the period April–September, while for the October–March period the EFPC exergy output is 21% higher than the PTC. For the UK, the EFPC has a year-round advantage due to its ability to use diffuse irradiance; delivering approximately double the exergy output of the PTC in winter and $\sim 10\%$ more in summer. The choice of single-axis tracking configuration is found to make only a small ($\sim 1\%$) difference in the total annual exergy output from the PTC in the UK, although in Cyprus the E-W tracking configuration results in a $\sim 12\%$ improvement relative to the N-S tracking arrangement. Full two-axis tracking on the other hand yields a more significant improvement for the UK ($\sim 40\%$) than for Cyprus ($\sim 20\%$) relative to single-axis tracking; although the complexity and cost of implementation is usually prohibitive for PTC systems.

In both the UK and the Cyprus analysis, the EFPC is found to deliver a higher annual maximum work output than the next-best non-tracking collector by 20–25%. The performance advantage can be attributed to the high gross-area efficiency (relative to the ETCs) and the low first-order heat loss coefficient (relative to the FPC). The next-best collectors in the annual analysis are found to be the direct-flow ETCs that incorporate a U-tube with a circular fin. The difference in annual exergy output between the U-tube ETC with and without the CPC reflector is found to be far smaller for the annual assessment than for the peak direct normal irradiance assessment in Table 2.1 and Fig. 2.9. The reason for this can be attributed to how

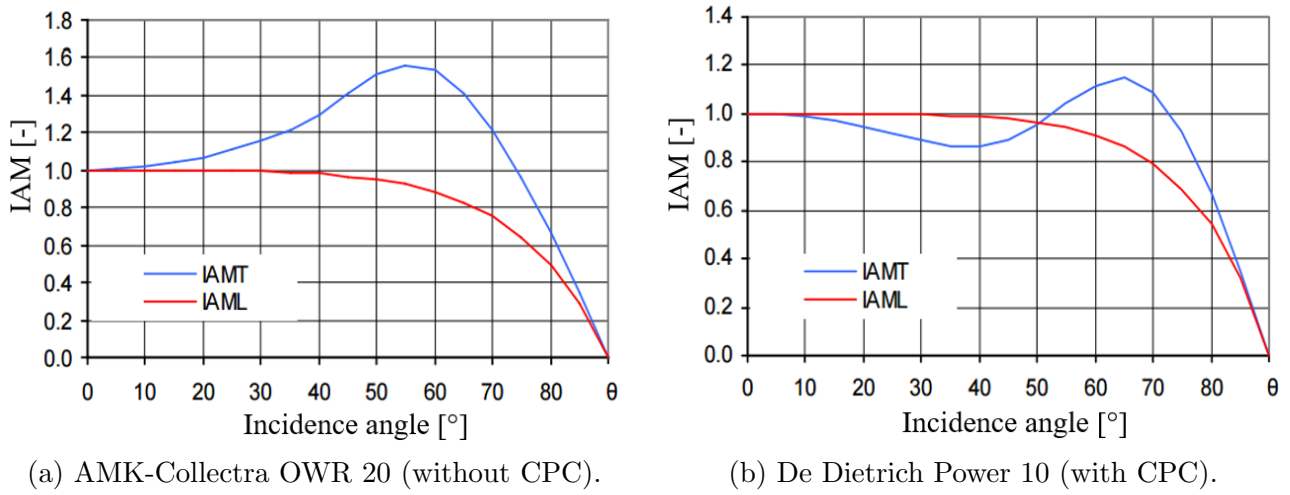


Figure 2.14: Transversal (IAMT, blue) and longitudinal (IAML, red) incident angle modifier plotted against transversal and longitudinal components of the solar incident angle, for (a) U-tube direct flow ETC without CPC reflector and (b) U-tube direct flow ETC with CPC reflector. Taken from Ref. [110].

the collectors perform under non-perpendicular solar incidence angles according to their direct-radiation incidence angle modifier (IAM), introduced as $K_{\theta,b}$ in Eq. 2.2. For evacuated tube collectors, manufacturers typically provide IAM data measured separately along the transversal and longitudinal symmetry planes (see Fig. 2.14). ETCs often experience a peak in IAM at incidence angles between $40-70^\circ$ along the transversal plane due to beneficial reflections from neighbouring tubes, which can be further enhanced for U-tube collectors due to their circular fin [129, 130]. At larger incidence angles, neighbouring tubes begin to shade each other and the IAM decreases. The shape of the IAM curve is therefore dependent on the tube spacing, which may be larger for collectors with CPC reflectors. The IAM is also dependent on the performance of the CPC itself which is often most beneficial at perpendicular incidence angles. As a result of these competing factors the transversal-plane IAM for the CPC collector in Fig. 2.14b shows two peaks, at 0° and 65° , with a slight dip between $20-50^\circ$.

The maximum annual solar to electrical efficiency for the EFPC, determined using endoreversible analysis, is 6.3% in London and 7.4% in Larnaca. This can be considered a realistic target for a well-designed solar heat engine system, and can be compared with typical annual efficiencies for monocrystalline silicon (mono-Si) PV of $12-15\%^3$. By comparison the maximum

³Based on a typical nominal efficiencies of $13-17\%$ under standard testing conditions (STC), and assuming that the annual efficiency is $\sim 90\%$ of the nominal efficiency [19, 131]

annual efficiency for the PTC collector with single-axis tracking is higher than the fixed EFPC in Larnaca (8.0%, corrected for an increased solar radiation exergy incident on the tracking plane), and lower in London (5.0%). The ETCs meanwhile have slightly lower maximum efficiencies than the EFPC, between 4.4–6.2% in Larnaca, and 3.6–5.0% in London, while the FPC has the poorest solar to electrical efficiency over the annual period ($< 4\%$ in Larnaca and $< 3\%$ in London).

2.3.5 Practical considerations for part-load operation

As noted earlier in Section 2.1, endoreversible analyses tend to result in performance predictions that are a realistic indicator of what can be achieved for well-designed systems. Based on the present analysis, achieving an annual work output that is close to the calculated endoreversible maximum would require the ability to modulate the collector's temperature to obtain the maximum power-point under variable solar irradiance and ambient temperature conditions by modulating the load (i.e. the rate of energy extraction from the fluid stream by the engine). This has practical implications concerning the cost and complexity of the system if designed for variable part-load operation, and further considerations for efficiency losses in engine components such as the pump and expander when not operating at their intended design-point. It is therefore instructive to consider a revised maximum annual work output that can be delivered by a collector to an engine operating close to its design-point, albeit for a reduced number of operating hours.

2.3.6 Choice of heat rejection medium

The choice of air or mains water as the heat rejection medium has a less significant influence on the annual maximum work output. For the UK analysis, the annual mean air temperature is 10.2°C , which is very similar to the annual mean mains water temperature (10°C , [123]). Furthermore, from Fig. 2.12 the cold sink temperature was found to have a smaller effect on exergy output over the range of annual variation than solar irradiance, however it was assumed

that the cold sink was also the ambient air temperature to which thermal energy is lost from the solar collector. Mains water, when compared to air as the cold sink, is found to result in a 3–11% increase in the annual maximum work output across the range of collector types, with the more significant improvement occurring in summer when air temperatures are higher. In the winter months there is a small advantage shown with air as the cold sink. The largest relative benefit in annual work output is for the collector with the lowest optimum operating temperature (the FPC). This analysis however does not include a consideration of the parasitic exergy destruction associated with the pumping of the heat rejection medium.

2.4 Summary

A method was presented for calculating the maximum exergy output from a solar thermal collector in order to assess the potential for power generation from a suitable heat engine operating downstream of the collector, based on both reversible and endoreversible thermodynamic analyses. The evaluation of the collector exergy output using an endoreversible analysis based on the Novikov efficiency results in collector exergy output values that are 50–60% lower than those obtained from a fully reversible analysis based on the Carnot efficiency. The calculation procedure was extended to an annual simulation using climate data for London (UK) and Larnaca (Cyprus), to determine the total annual exergy deliverable from a range of existing small-scale solar collector modules (concentrating and non-concentrating) and to determine their optimum operating temperatures and flow-rates for exergy maximisation. The results show a large degree of variation across the range of solar thermal collectors. Based on the endoreversible analysis the maximum annual work production from the range of non-concentrating collectors is found to be 29–67 kWh yr⁻¹ m⁻² for London, which scaled-up to a 15 m² domestic rooftop array size, is equivalent to 442–1008 kWh yr⁻¹ and an average power of 50–115 W. This is also equivalent to 13–31% of the average UK household demand for electricity. The concentrating parabolic trough collector (PTC, featuring single-axis or two-axis tracking systems) is shown to have the potential to deliver a higher maximum exergy flow under equal direct irradiance conditions than all of the non-concentrating collectors studied. However, over the range of an-

nual conditions for London that feature prolonged periods of diffuse scattering by cloud cover, the total annual work potential of the PTC with single-axis tracking is found to be 22% lower than the highest performing of the non-concentrating collectors.

An important finding is the extent of the error introduced by using averaged data for the annual work output calculations from the solar heat engine. Due to the non-linearity of power output with solar irradiance, the annual calculations require sufficiently high-resolution data in which short-duration peaks in solar irradiance are represented accurately. This is especially relevant for the UK, where direct sunlight is highly intermittent due to cloud cover.

The ranges of temperatures and flow-rates required for maximum work production are in some cases found to be outside of the typical range of operation as specified by the manufacturers. This is to be expected, as the collectors are in most cases specified for hot-water production at temperatures lower than the optimum for power generation. Analysis of test reports for non-concentrating collectors shows that these products are typically tested for operational temperatures below 90 °C. However, optimal mean temperatures for maximum power generation under peak irradiance conditions are found to be in the range 110-170 °C for the non-concentrating evacuated tube collectors. However, further analysis has shown that exergy output from the collectors deviates only slightly from the optimal value over a wide range of mass flow-rates, providing that the mean collector temperature is maintained at its optimal value. The findings have shown that the best-performing ETC collectors designed for domestic water heating applications deliver up to 85% of the annual exergy output of EFPC collectors designed for higher-temperature process-heat applications, while the poorest deliver around 60%.

In the London endoreversible analysis, annual solar to electrical efficiencies of up to 6.3% were predicted for the EFPC, while in Larnaca up to 8% was predicted for the PTC. This is a realistic target for a well-designed solar thermal power system and can be compared to reported annual efficiencies of 12–15% for monocrystalline (mono-Si) PV. Solar-thermal collectors experience a more significant decrease in efficiency than PV under low-irradiance conditions. Thus when solar to electrical efficiencies are compared to PV under peak irradiance

conditions the results are far closer, with 12.1% obtained for the PTC and 9.3% obtained for the EFPC, compared to 15–17% for mono-Si PV. Solar thermal collectors also have an advantage that the thermal output can be used to provide water or space heating instead of, or in addition to, electrical power. For these reasons, it is considered interesting to further explore the potential of a solar combined heat and power system based on solar thermal collectors. In the following chapter, an S-CHP system based on an organic Rankine cycle engine will be introduced and its performance will be simulated and compared with both EFPC and ETC collectors.

Chapter 3

Solar Combined Heat and Power (S-CHP) System Model

In this chapter, a thermo-economic model is presented of a small-scale solar combined heat and power (S-CHP) system based on an Organic Rankine Cycle (ORC), in order to investigate the potential of this technology for the combined provision of heating and power. The system model consists of a solar collector array, an ORC engine, and a hot water storage tank. The solar collector array is limited to a total area of 15 m^2 , equivalent to that available on the roof of a typical UK home. The ORC sub-model features a generalised positive-displacement expander and a water-cooled condenser. Water heating is provided through the precooling of the superheated exhaust vapour from the ORC expander, and an optional further contribution from the solar collector array. The full equations for each of the component sub-models are presented, as well as a discussion of the control strategy for the various modes of operation. The results of a market analysis are also used to develop cost models for each of the components which will form the basis of an installed cost estimation for the whole system.

3.1 Model Description

The S-CHP system model was written in MATLAB. The main component sub-models are a solar collector array, an ORC engine and a hot water tank, as shown in Fig. 3.1. The model consists of three fluid circuits: (i) a solar collector fluid circuit; (ii) an ORC working fluid circuit, and (iii) a domestic hot water (DHW) circuit, which is an open-loop where cold water input is provided from the utility mains to the hot water tank where it is heated and then delivered for consumption in the household. Water heating is provided via precooling of the superheated vapour exiting the expander in a heat exchanger coil submerged in the hot water tank. Mains water is also used as the heat-rejection medium in the ORC condenser, some of which is then delivered as pre-heated feed to the hot water tank. The solar collector fluid is circulated in the collector array before flowing through the evaporator where it exchanges heat with the ORC working fluid. Collector fluid exits the evaporator at a lower temperature and then flows to a heat exchanger coil located in the hot water tank where it is cooled further before returning to the collector array. Two bypasses are included in the solar collector circuit:

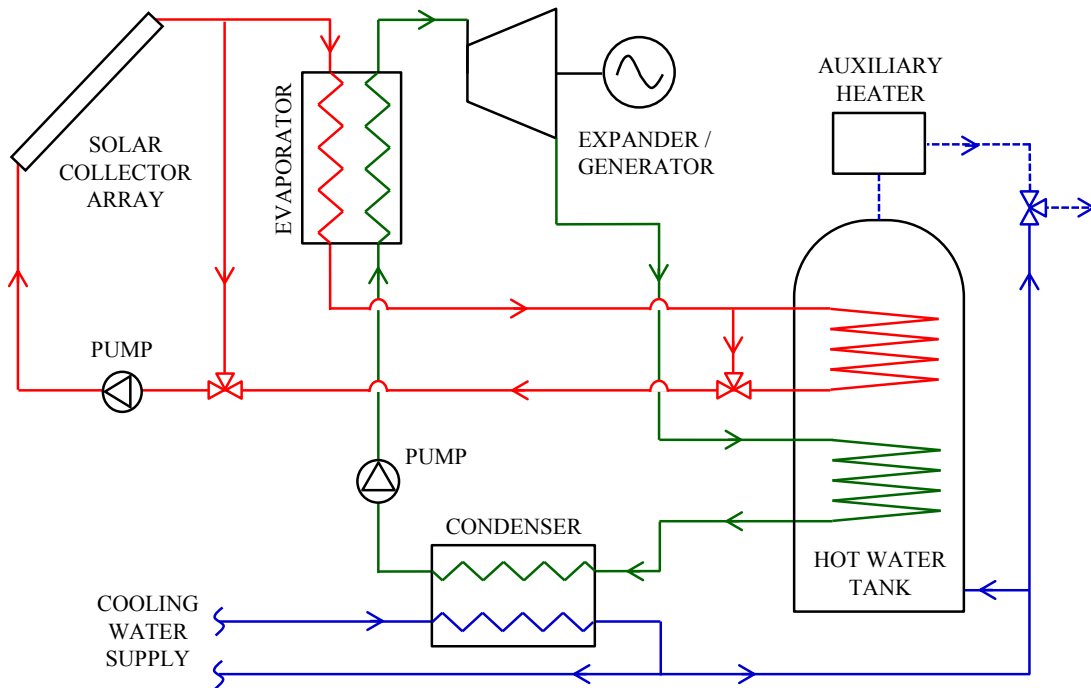


Figure 3.1: Schematic representation of the S-CHP system, consisting of the the solar collector circuit (shown in red), the ORC circuit (shown in green), and the domestic hot water circuit (shown in blue).

the first bypass allows the solar collector fluid to recirculate in the collector array until its temperature is sufficiently high to evaporate the ORC working fluid; while the second bypass allows a variable fraction of the collector fluid to bypass the coil in the hot water tank. Finally, an auxiliary heater and cold water mixing device are included at the outlet port of the hot water tank to control the final temperature of the hot water delivered for domestic consumption. It is assumed for simplicity in the model that all hot water is required at 60 °C.

The inputs to the model are the external climate conditions (including solar irradiance and ambient air temperatures), household electricity and hot water demand data and cold water supply temperatures. As for the previous chapter, the London climate data used in the annual simulations is the IWECC dataset published by ASHRAE [120]. The equations with dynamic terms are solved using an explicit Euler method, where the time-step interval for the simulation is determined by the stiffness of the numerical integration algorithm. For an individual 24-hour day simulation, a default time-step interval of 30 seconds is used such that the 24-hour period is divided into a set of 2,880 inputs and outputs.

Alternative configurations for the system will also be considered, including an air-cooled variant in which the water-cooled condenser in Fig. 3.1 is replaced with a fan-assisted air-cooled condenser. Consideration will also be given to the use of additional thermal storage in the solar fluid circuit in order to buffer variations solar thermal input.

3.2 Model Equations

3.2.1 Solar collector array

As in Chapter 2, the approach for modelling the solar thermal collector is based on the empirical efficiency curve. The collector's thermal capacity is also included as a time-derivative term in the collector energy balance. The useful energy output from the solar collector is calculated as a function of the incident solar irradiance, the mean collector temperature and the ambient air

temperature, and is a simplification of Eq. 2.2:

$$\frac{\dot{Q}_u}{A_{sc}} = \eta_0 (K_{\theta b} G_b + K_{\theta d} G_d) - c_1 (\bar{T}_{sc} - T_a) - c_2 (\bar{T}_{sc} - T_a)^2 - c_5 \frac{d\bar{T}_{sc}}{dt}, \quad (3.1)$$

where the solar collector parameters, η_0 , c_1 , c_2 , c_5 , $K_{\theta b}$, $K_{\theta d}$ are listed in Table 3.1). The energy balance is solved so that:

$$\dot{Q}_u = \dot{m}_h c_{p,h} (T_{sc,out} - T_{sc,in}). \quad (3.2)$$

In an earlier version of the model, the equations for the collector loop were solved using a quasi-steady numerical method in which the time-derivative term for the collector is omitted, and temperatures and heat flows in the loop remain in steady-state at each time-instant. A comparison of the quasi-steady and dynamic approaches to solar collector modelling is performed in Chapter 5. A comparison was also performed between the quasi-steady numerical method and a so-called “time-marching” numerical method, and is included in Appendix B.

In Chapter 2 it was found that the optimal collector temperature for maximum power under peak irradiance conditions is $> 100^\circ\text{C}$ for ETC collectors, and $> 200^\circ\text{C}$ for PTC and EFPC collectors. For safe operation at these temperatures, a suitable circulating fluid must be used. Synthetic oils such as DOWTHERM, THERMINOL and MARLOTHERM are commonly used as circulating fluids in medium temperature solar collector systems. For the subsequent analysis, unless otherwise stated, the circulating fluid will be modelled as DOWTHERM A, which was also modelled as a circulating fluid for EFPC collectors in Ref. [112]. The fluid thermophysical properties will be obtained as functions of fluid temperature, using polynomial equations derived from manufacturer’s literature [132].

By default the solar collector array is modelled as identical solar collector modules connected in parallel, such that the Eq. 3.1 can be applied to the entire collector array area. Where a large rise in temperature through the collector array is required, a series connection configuration of the collector modules may be preferable. This will be discussed further in Section 4.4.

Table 3.1: Solar collector parameters used for the S-CHP system simulations.

	EFPC	ETC
Manufacturer & model	TVP Solar MT-Power	Thermomax DF 100 20
η_0 , -	0.76	0.77
c_1 , W m ⁻² K ⁻¹	0.51	1.43
c_2 , W m ⁻² K ⁻²	0.007	0.006
c_5 , J m ⁻² K ⁻¹	7505	9300
A_{aper} , m ²	1.84	2.15
A_{gro} , m ²	1.95	2.83
$K_{\theta_{\text{b,tr}}}^{(i)}$, -	0.91	1.05
$K_{\theta_{\text{b,lo}}}^{(i)}$, -	0.91	0.99
$K_{\theta_{\text{d}}}$, -	0.97	0.88

⁽ⁱ⁾ $K_{\theta_{\text{b,tr}}}$ and $K_{\theta_{\text{b,lo}}}$ values reported for incident angles of 50°. Full IAM curves available from Refs. [110, 117, 128]

Solar fluid pump

The solar fluid pump will be modelled as a modern circulator pump, complying to the European directive defining ecodesign requirements for energy related products (ERP) [133]. Modern ERP circulators (such as the Grundfos UPS2) typically have design-point electrical (pump + motor) efficiencies of $\sim 35\%$, compared to pre-ERP directive circulators (such as the earlier Grundfos UPS), with design-point electrical efficiencies of $\sim 10\%$ [134]. The electrical power consumed by the pump is calculated as:

$$\dot{W}_{\text{pp,h}} = \frac{\dot{m}_{\text{h}} \Delta p}{\eta_{\text{pp,h}} \rho_{\text{f}}}, \quad (3.3)$$

where a fixed value of 35% is assumed for $\eta_{\text{pp,h}}$. The total pressure drop is comprised of the pressure losses in the evaporator and the solar collector array. A pressure drop of 50 kPa is assumed for the evaporator, based on the plate heat exchanger sized for the experimental system described in Chapter 6. This value will also be used as the design-point pressure drop for the sizing of the evaporator in Section 4.5. The pressure drop through the solar collector array meanwhile is calculated as a function of the flow-rate based on pressure drop data in the manufacturer's literature.

3.2.2 Organic Rankine Cycle

ORC pump

The input parameters for the ORC pump calculations are the working fluid flow-rate, the cycle evaporation and condensation temperatures (or pressures), and the isentropic efficiency of the pump:

$$\eta_{pp,orc} = \frac{h_{2s} - h_1}{h_2 - h_1}. \quad (3.4)$$

In Eq.3.4, h_1 and h_2 are the specific enthalpies at the inlet and outlet of the pump respectively (see Fig. 3.2), and h_{2s} is the specific enthalpy following an isentropic pumping process between p_1 (the condensation pressure) and p_2 (the evaporation pressure) starting at State 1. The fluid properties at State 1 are defined by the condensation pressure and the degree of subcooling, which are specified as input parameters to the ORC model. As will be shown in Chapter 6, subcooling is often necessary in real ORC systems prior to the pump inlet to prevent cavitation. However, the larger the degree of subcooling the larger the penalty in cycle performance. For the simulation results presented in the following chapter, it will be assumed that the fluid entering the pump is a saturated liquid. The rise in enthalpy of the working fluid through the pump is then calculated from the isentropic efficiency. A fixed value of $\eta_{pp,orc} = 65\%$ was chosen, which is within the range of values used in similar studies [135–138]. The input power required by the pump is then:

$$\dot{W}_{pp,orc} = \dot{m}_r (h_2 - h_1) / \eta_m \quad (3.5)$$

where a value of $\eta_m = 90\%$ taken as the combined electromechanical efficiency of the pump shaft and motor [139].

ORC evaporator

The heat addition process to the working fluid entering the evaporator at State 2 and exiting at State 3 (see Fig. 3.2) is assumed to be an isobaric process (zero pressure loss). For the

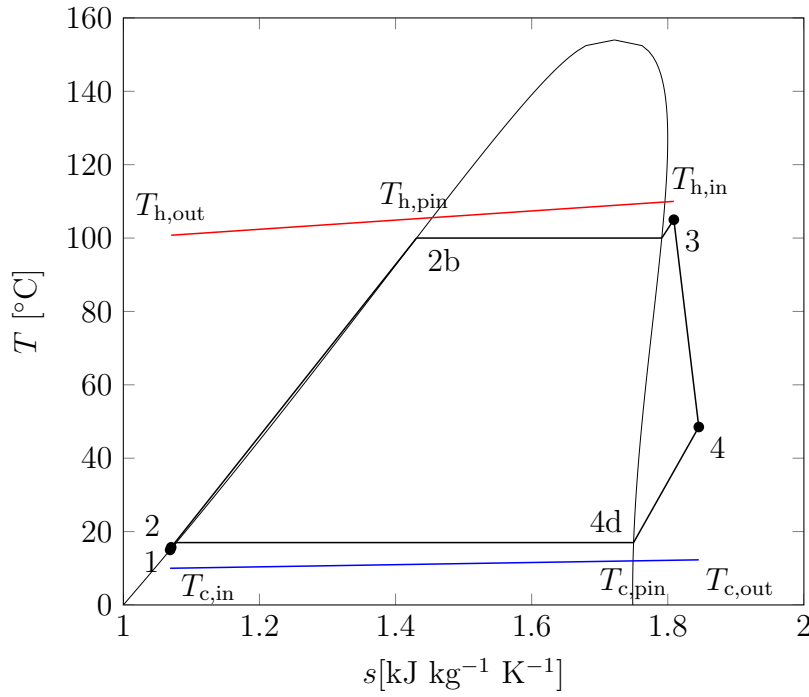


Figure 3.2: Temperature-entropy (T - s) plot of organic Rankine cycle with working fluid R245fa with $P_{\text{evap}} = 12$ bar, and $P_{\text{cond}} = 1.1$ bar. Also shown are the heat exchanger pinch points, as well as the heat source fluid stream process (high temperature line) and cold sink fluid stream process (low temperature line) that are external to the cycle. Also, Point ‘b’ marks the beginning of boiling/evaporation (bubble-point) in the evaporator and ‘d’ the beginning of condensation (dew-point) in the condenser, respectively.

nominal design condition of the evaporator heat exchanger, an approach temperature difference of $\Delta T_{\text{app}} = 5$ K is assumed between the heat source fluid inlet and the ORC working fluid outlet, such that the working fluid temperature at State 3 is calculated as:

$$T_3 = T_{h,\text{in}} - \Delta T_{\text{app}}. \quad (3.6)$$

The heating source fluid flow-rate must be sufficient that the temperature difference between the solar collector fluid and ORC working fluid is not less than the minimum “pinch” temperature difference at any point along the length of the heat exchanger. For an isothermal boiling process, a critical pinch point occurs at the point where the working fluid begins to boil (State 2b). At this point, it is ensured that the temperature difference between the heat source fluid stream and the working fluid stream is greater than or equal to a minimum value

of $\Delta T_{\text{pin}} = 5 \text{ K}$,

$$T_{\text{h,pin}} - T_{2\text{b}} \geq \Delta T_{\text{pin}} , \quad (3.7)$$

and therefore the minimum heat source temperature required for operation of the ORC is:

$$T_{\text{h,in,min}} = T_{\text{h,pin}} + \frac{\dot{m}_{\text{r}} (h_3 - h_{2\text{b}})}{\dot{m}_{\text{h}} c_{\text{p,h}}} \quad (3.8)$$

where thermal losses to the surroundings are neglected and the temperature of the heat source fluid entering the evaporator is assumed equal to the temperature of the fluid leaving the solar collector array ($T_{\text{h,in}} = T_{\text{sc,out}}$).

As is the common approach in the ORC literature, the heat exchanger surface area is calculated using the log-mean temperature difference (LMTD) method for a counterflow heat exchanger applied to the preheating, boiling, and superheating zones separately. For each zone:

$$\dot{Q} = \dot{m}_{\text{r}} (h_{\text{r,out}} - h_{\text{r,in}}) , \quad (3.9)$$

$$\dot{Q} = \dot{m}_{\text{h}} c_{\text{p}} (T_{\text{h,in}} - T_{\text{h,out}}) , \quad (3.10)$$

$$\dot{Q} = UA \Delta T_{\text{LMTD}} , \quad (3.11)$$

where

$$\Delta T_{\text{LMTD}} = \frac{\Delta T_1 - \Delta T_2}{\ln \left(\frac{\Delta T_1}{\Delta T_2} \right)} . \quad (3.12)$$

The equations for each zone are solved simultaneously, where the input variables are the refrigerant inlet enthalpy to the preheating zone (equal to the pump outlet enthalpy h_2) and the heat source inlet temperature $T_{\text{h,in}}$. The key parameter of interest is the heat exchanger surface area, which will provide the basis for the cost estimation. The overall heat transfer coefficients and surface areas for the preheating, boiling and superheating zones, are determined by way of an initial sizing calculation for a nominal design condition. The convection heat transfer coefficients in the heat source fluid and refrigerant streams are dependent on the flow characteristics and heat exchanger geometry, which also affect the fluid pressure drops and the

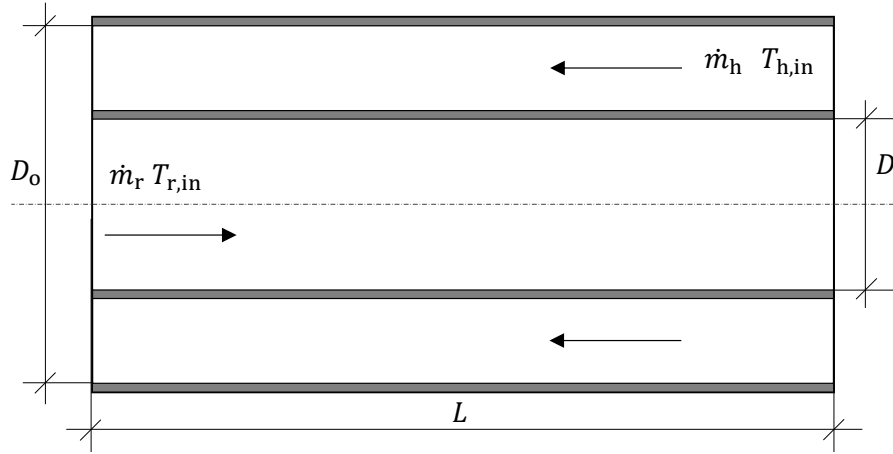


Figure 3.3: Counterflow concentric-tube heat exchanger configuration modelled for the ORC evaporator.

pumping power requirements.

In the experimental ORC system described in Chapter 6 of this thesis, brazed plate heat exchangers are used for the evaporator and condenser. However, for the heat transfer calculations in the S-CHP model, the evaporator is modelled as a counterflow concentric-tube heat exchanger, with the ORC working fluid flowing inside the inner tube and the solar heat-transfer fluid flowing in the annulus. A concentric tube is a simple configuration commonly used to determine approximate heat exchanger surface areas in ORC literature [87, 140]. A schematic representation of the heat exchanger is shown in Fig. 3.3. A finite element method is used to model the heat transfer processes. The heat exchanger length is divided into N finite elements, and the enthalpy and temperatures of the ORC working fluid and solar collector heat-transfer fluid for the $(n + 1)^{\text{th}}$ element are calculated from:

$$h_{r,n+1} = h_{r,n} + \frac{U\pi D_i N_{\text{tubes}} (T_{h,n} - T_{r,n})}{\dot{m}_h} \Delta L, \quad (3.13)$$

$$T_{h,n+1} = T_{h,n} - \frac{U\pi D_i N_{\text{tubes}} (T_{h,n} - T_{r,n})}{(\dot{m}c_p)_h} \Delta L, \quad (3.14)$$

$$\dot{m}_r (h_{r,n+1} - h_{r,n}) = -\dot{m}_h c_{p,h} (T_{h,n+1} - T_{h,n}), \quad (3.15)$$

where the total heat transfer coefficient between the fluid streams is $U = 1/(h_{c,i}^{-1} + h_{c,o}^{-1})$. Note that due to the counterflow arrangement of the heat exchanger, element $n = 1$ of the heat exchanger represents the working fluid inlet and the solar fluid outlet. Full details of the heat

transfer correlations used are provided in Appendix D. Single-phase heat transfer coefficients are calculated using Nusselt number correlations of Gnielinski [141], while for the refrigerant in the two-phase region, the local heat transfer coefficients are calculated using the Lockhart-Martinelli parameter approach described in Wang and Toubert [142]. Finally, the pressure drops in the evaporator heat exchangers are calculated using the Reynolds number/friction factor correlations in Incropera *et al* [141], and the tube dimensions are sized to achieve a total $\Delta P \approx 50$ kPa in the solar/heat source fluid stream.

ORC expander

The power output from the expander under nominal design conditions is calculated by using a specified isentropic efficiency:

$$\dot{W}_{\text{exp}} = \eta_{\text{exp}} \cdot \dot{m} \cdot (h_3 - h_{4s}) \quad (3.16)$$

where

$$\eta_{\text{exp}} = \frac{h_3 - h_4}{h_3 - h_{4s}}, \quad (3.17)$$

and h_{4s} is the specific enthalpy of the fluid after an isentropic expansion from State 3 to State 4s on the cycle condensation pressure isobar. For a low power-output system in the region of (~ 1 kW) such as this, positive-displacement expanders may be favoured over turbomachines (as discussed Chapter 1). Equation 3.16 assumes that the cycle expansion ratio is matched to the expander built-in volume ratio such that no over-expansion or under-expansion losses occur. Under such conditions, an isentropic efficiency of 75% is used in the model which is typical of those reported for small-scale positive-displacement machines [38, 68, 136, 143–147]. When the system is operating under off-design conditions (for example at solar irradiance levels higher than that for which the system has been sized), the imposed volume ratio across the expander will no longer match the built-in volume ratio and the expansion process will be subject to additional associated losses. In this case the following expression developed by Quoilin *et*

al. [143] is used to calculate the power output from the expander:

$$\dot{W}_{\text{exp}} = \eta_{\text{exp}} \cdot \dot{m}_r \cdot \left(h_3 - h_{\text{int},s} + r_{v,\text{exp}} \frac{p_{\text{int}} - p_4}{\rho_3} \right), \quad (3.18)$$

where $h_{\text{int},s}$ is the specific enthalpy of the fluid after an isentropic expansion from State 3 to an internal pressure p_{int} imposed by the built-in volume ratio of the expander $r_{v,\text{exp}}$. Finally the electrical power output from the system is calculated using a generator lumped electromechanical efficiency of 90% (representing electrical losses and also losses in the shaft and coupling to the expander [148]), and by subtracting the input power required for the system pumps:

$$\dot{W}_{\text{net}} = \eta_g \cdot \dot{W}_{\text{exp}} - \dot{W}_{\text{pp,orc}} - \dot{W}_{\text{pp,h}}. \quad (3.19)$$

Water-cooled condenser

For the parametric analysis, the default cooling water inlet temperature to the condenser is assumed to be 10 °C, which is representative of the annual average value for the UK [123]. For the annual analysis a sinusoidal variation in water temperature over the year will be assumed, with a mean of 10 °C and an amplitude of ± 2.6 °C with the maximum occurring in late July, as was also assumed for the annual analysis in Chapter 2. For the condenser sizing purposes it is assumed that the working fluid enters the condenser as a superheated vapour at State 4 and leaves as a saturated liquid at State 1, which was defined in Section 3.2.2. The flow-rate of cooling water must be sufficient such that the temperature difference between working fluid and cooling water at all points along the heat exchanger does not exceed the pre-defined pinch difference. For the condenser, the critical pinch-point is found to occur at the point at which the working fluid begins to condense, i.e. State 4d. At this point, the temperature difference between the working fluid and the cooling medium must be greater than or equal to a pinch difference of $\Delta T_{\text{pin}} = 5$ K, such that:

$$T_{4d} - T_{c,\text{pin}} \geq \Delta T_{\text{pin}}. \quad (3.20)$$

For the simulations in this chapter the cooling medium is modelled as mains water, and the mass flow-rate is calculated based on a nominal inlet temperature of 10 °C:

$$\dot{m}_{c,\min} = \frac{\dot{m}_r (h_{4d} - h_1)}{c_{p,c} (T_{c,\text{pin}} - T_{c,\text{in}})}, \quad (3.21)$$

The cooling water temperature at the condenser outlet has been raised above the mains temperature. Some of this cooling water may now be used as preheated feed water to top-up the hot water tank (as indicated by the blue line in Fig. 3.1), thereby reducing the energy consumed for domestic hot water heating. The limit for the amount of preheated water that can be used at any instant is determined by the hot water demand, in Eq. 3.28. It is assumed that any surplus cooling water is discharged to drain. For the condenser sizing and costing calculations, a similar finite-element method is applied as for the evaporator calculations in Section 3.2.2, with the condenser modelled as a counterflow concentric-tube heat exchanger. The Nusselt number correlation of Gnielinski is again used for single-phase heat transfer, while for two-phase heat transfer in the condensation zone the correlation of Shah is used [149]. The full set of heat transfer calculations are listed in Appendix D. For the water-cooled ORC model it is assumed that pressurised water is available from the local utility mains and therefore water pumping requirements are not considered in the net power calculation.

Air-cooled condenser

A comparison will also be made between water and ambient air as the heat rejection medium. In this case the above Eqs 3.20 and 3.21 for the condenser will also apply, with the cold stream inlet temperature varying with that of the ambient air condition and the specific heat capacity $c_{p,c}$ being that of air instead of water. For the calculation of the ORC net power output in the air-cooled case, the energy input for the fan will also be considered:

$$\dot{W}_{\text{fan}} = \frac{\dot{m}_c \Delta p}{\rho_c \eta_{\text{fan}} \eta_{\text{m}}}, \quad (3.22)$$

where a value of 100 Pa is assumed for the air-side pressure drop, and values of 65% and 90% are assumed for the respective efficiencies of the fan and motor¹. For sizing purposes, the heat exchanger surface area is calculated using equations for a cross-flow finned tube heat exchanger, given in Appendix D.

3.2.3 Control algorithms

The model uses a series of logic statements to determine whether certain parts of the system are active or inactive. In the solar collector circuit, the flow of fluid through the collector array is activated when there is an availability of solar irradiance $G > 0$, such that the pump does not operate at night without benefit:

$$\text{Solar pump status} = \begin{cases} \text{OFF} & \text{if } G = 0 \text{ ;} \\ \text{ON} & \text{otherwise .} \end{cases} \quad (3.23)$$

The flow of refrigerant in the ORC circuit is activated only when the temperature of the solar collector fluid at the collector's outlet $T_{\text{sc,out}}$ reaches the evaporator heat source temperature set-point $T_{\text{h,in,min}}$, as calculated from Eq. 3.8:

$$\text{ORC status} = \begin{cases} \text{OFF} & \text{if } T_{\text{sc,out}} < T_{\text{h,in,min}} \text{ ;} \\ \text{ON} & \text{otherwise .} \end{cases} \quad (3.24)$$

When the ORC is inactive, the solar collector fluid is returned to the collector for further heating providing $G > 0$. This bypassing continues to occur until $T_{\text{sc,out}} \geq T_{\text{h,in,min}}$ and the ORC engine is triggered to run.

3.2.4 Domestic hot water storage tank

A one-dimensional model is used for the hot water storage tank. The tank is assumed to consist of N fully mixed equal-volume segments that divide the vessel along its vertical axis.

¹Based on typical design-point values for Nuair axial flow fans [150].

Each segment is represented by a node n , where $n = 1$ is the bottom segment and $n = N$ is the top one. Depending on how the tank is configured, any of the nodes may contain a heat exchanger coil or a fluid inlet and/or outlet port. The heat exchanger coil is used for indirect heat transfer with a continuous flow of hotter or colder fluid. The total mass of fluid in the tank is assumed constant, therefore the total mass flow rate entering the tank through the inlet ports must equal the total mass exiting the tank from the outlet ports. Two basic modes of operation are considered for the tank: stratified charging/discharging and mixed charging/discharging. In the stratified mode a temperature gradient is preserved in the tank by ensuring that the heating is supplied via a port or heat exchanger coil to the top of the tank. Hot water is then extracted from the top of the tank, while replacement cold water is introduced at the bottom. In the mixed mode, heating is supplied to bottom of the tank, and mixing occurs between the layers as water is drawn from the top of the tank. The stratified tank model requires a larger number of N layers in order to accurately track the position of the thermocline as the tank is charged or discharged while the mixed tank model requires fewer layers and in many cases can be represented by a single lumped volume.

The default configuration for the hot water tank in the S-CHP system model is shown Fig 3.4. The tank effectively serves as pre-heater to an auxiliary heater down-stream of the tank. The auxiliary heater is assumed to be a direct-fired water heater that provides water at a supply temperature of 60 °C. The hot water tank will function as a heat sink to recover a portion of the thermal energy that is rejected from the ORC, and store it for later use as domestic hot water supply when required. For this purpose a stratified tank is useful because it allows the coldest water from the bottom of the tank to be available for heat rejection (maximising the Carnot efficiency) while storing heated water at the highest possible temperature at the top of the tank and minimising the requirement for downstream water heating by the auxiliary source. Thus cold water enters the tank from the bottom to replace hot water extracted from the top. The tank is modelled with $N = 2$ fully-mixed segments. The heat exchanger coil in the bottom segment provides low temperature water heating as a result of the precooling of the superheated ORC expander exhaust vapour flowing through the coil; while the heat exchanger coil in the top segment provides optional supplementary heating at higher temperature from

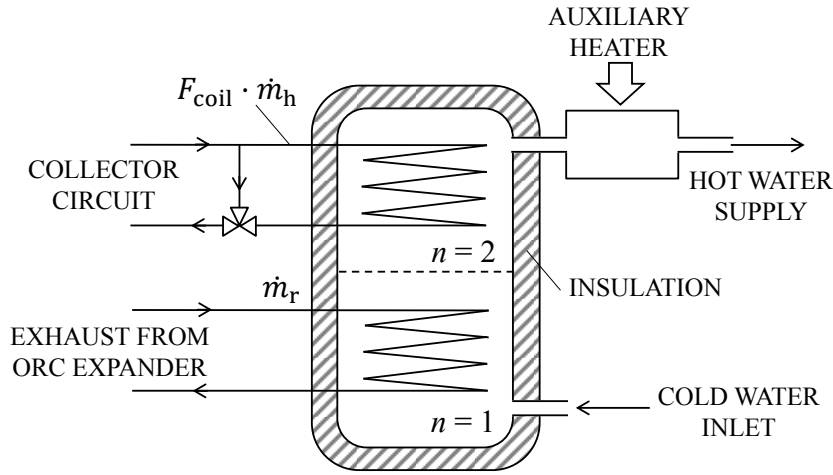


Figure 3.4: Schematic diagram of the hot water tank sub-model used in the S-CHP system model.

the solar collector fluid, at the expense of a reduced thermal input to the ORC.

The tank equations are based on those for the Multiport Store model developed for TRNSYS [151]. A further model input is the coil flow-factor, F_{coil} , which defines the proportion of the solar collector return flow (0–100%, fixed) that is diverted through the heat exchanger coil in the upper segment of the tank. By default the solar collector fluid leaving the ORC evaporator is sent back directly to the collector to be heated again. If some of this fluid is first diverted through the heat exchanger coil in the hot water tank there will be a further reduction in the fuel demand to heat the water conventionally. However this will also decrease the collector return temperature, and the available heating input to the ORC.

For the heat exchanger coils in the upper and lower segments of the tank, the heat transfer from the fluid circulating in the coil to the water in the tank is calculated using the effectiveness-NTU method:

$$\dot{Q}_{\text{coil},n} = \dot{m}_{\text{coil}} \cdot c_{p,w} \cdot \varepsilon (T_{\text{coil,in},n} - T_{\text{tank},n}) , \quad (3.25)$$

where the heat exchanger effectiveness is based on a approximate total surface area of 1 m^2 for each coil and an average heat transfer coefficient of $U = 200 \text{ W m}^{-2} \text{ K}^{-1}$ [22]. The system is by default set to prioritise electricity generation over water heating, therefore when the ORC engine is bypassed the hot water tank is also bypassed in order for the solar collector fluid to

come up to temperature as quickly as possible. When $T_{\text{sc,out}} \geq T_{\text{h,in,min}}$ the ORC engine is switched on but collector fluid leaving the ORC evaporator will be sent to the heating coil only if it is at a higher temperature than the water in the top segment of the tank. No heating is delivered if the tank temperature is above the maximum hot water storage temperature of 80 °C. This strategy can be summarised as:

$$\dot{m}_{\text{coil}(n=2)} = \begin{cases} F_{\text{coil}} \cdot \dot{m}_{\text{h}} & \text{if ORC status = ON;} \\ & \text{and } T_{\text{coil,in}(n=2)} \geq T_{\text{tank}(n=2)}; \\ & \text{and } T_{\text{tank}(n=2)} < 80 \text{ }^\circ\text{C}; \\ 0 & \text{otherwise.} \end{cases} \quad (3.26)$$

The energy balance for each segment of the hot water tank is as follows:

$$m_{\text{tank},n} c_{\text{p,w}} \frac{dT_{\text{tank},n}}{dt} = \dot{Q}_{\text{coil},n} + \dot{Q}_{\text{load},n} + \dot{Q}_{\text{con},n} - \dot{Q}_{\text{loss},n}. \quad (3.27)$$

The hot water demand profile is constructed from hourly UK data published in a study by the Energy Saving Trust [125], which has been converted from a volumetric to a mass flow-rate, $\dot{m}_{\text{load}}$, in kg/s. Hot water extraction for domestic consumption results in a thermal energy exchange between the nodes of the hot water tank, as cold water is fed in at the bottom to replace the water that is drawn-off from the top:

$$\dot{Q}_{\text{load},n} = \begin{cases} \dot{m}_{\text{load}} c_{\text{p,w}} (T_{\text{fill}} - T_{\text{tank},n}) & \text{if } n = 1; \\ \dot{m}_{\text{load}} c_{\text{p,w}} (T_{\text{tank},n-1} - T_{\text{tank},n}) & \text{otherwise.} \end{cases} \quad (3.28)$$

The temperature of the water that re-supplies the tank is either at the water mains temperature (set to $T_{\text{fill}} = T_{\text{cw,in}}$), or at a higher temperature as a result of utilising preheated cooling water leaving the ORC condenser. In order to ensure that the flow-rate of preheated cooling water from the condenser that is then used to re-supply the hot water tank is maximised within the limit imposed by the instantaneous household demand for hot water, the following

strategy is applied at each time-step:

$$T_{\text{fill}} = \begin{cases} \frac{\dot{m}_c}{\dot{m}_{\text{load}}} \cdot T_{c,\text{out}} + \left(1 - \frac{\dot{m}_c}{\dot{m}_{\text{load}}}\right) T_{c,\text{in}} & \text{if ORC status} = \text{ON;} \\ T_{c,\text{in}} & \text{otherwise.} \end{cases} \quad (3.29)$$

The heat transfer due to conduction between the layers of the store is calculated using Eq. 3.30:

$$\dot{Q}_{\text{con},n} = \frac{\lambda_{\text{eff}} \pi D_{\text{tank}}^2}{4H_n} (T_{\text{tank,adj}} - T_{\text{tank},n}) , \quad (3.30)$$

where $T_{\text{tank,adj}}$ is the temperature of the adjacent node and λ_{eff} is the effective vertical thermal conductivity. The latter is a lumped parameter that takes into account conduction in the water, the walls of the tank and the internal components such as the heat exchanger coils. The parameter is obtained experimentally and varies for different designs of tank. For the tank model a value of $1.8 \text{ W m}^{-1} \text{ K}^{-1}$ is chosen which is representative of a range of solar hot water cylinders [152]. Finally, the thermal energy loss through the wall of the tank to the ambient internal environment is given for each node as:

$$\dot{Q}_{\text{loss},n} = U_{\text{tank}} A_{\text{surf,tank},n} (T_{\text{tank},n} - T_{\text{int}}) , \quad (3.31)$$

where the indoor environmental temperature surrounding the tank T_{int} is assumed constant at 20°C . It is necessary to assume that the tank is well insulated in order to prevent significant thermal losses to the environment, and therefore an average U-value of $0.3 \text{ W m}^{-2} \text{ K}^{-1}$ is chosen for the simulations.

Auxiliary hot water demand

For the sake of simplicity, it is assumed that all auxiliary hot-water heating takes place downstream of the hot water tank. The demand for auxiliary water heating is then the energy required to bring the water from the tank temperature to the required domestic hot water

supply temperature, which is $T_{\text{sup}} = 60 \text{ }^{\circ}\text{C}$,

$$\dot{Q}_{\text{hw,aux}} = \dot{m}_{\text{load}} c_{p,w} (T_{\text{sup}} - T_{\text{tank,(n=2)}}) . \quad (3.32)$$

3.3 Exergy Analysis

The equations in this section are relevant for the whole system exergy analysis, results from which are presented in Section 4.8. In this analysis, the exergy destroyed in each component of the S-CHP system is tracked in order to identify the components that have the greatest capacity to limit the system's efficiency. The definition of exergy was defined in the previous chapter as the upper limit for the convertible work in cooling a stream of fluid down to the reference ambient temperature T_0 while extracting work reversibly:

$$\dot{X}_f = \dot{m}_f [(h_f - h_0) - T_0 (s_f - s_0)] , \quad (3.33)$$

where T_0 is taken here to be either the temperature of the cooling medium T_c or of the ambient air T_a , depending on which is lowest.

In the present work a number of definitions are used that follow the exergy topological method as used by Mago *et al.* [67] in a comparative study of regenerative and a non-regenerative ORC systems for waste heat recovery. The rate of exergy destroyed in each component $\dot{X}_{\text{loss}}$ is equal to the total exergy flowing into the component $\dot{X}_{\text{in}}$ minus the total exergy flowing out $\dot{X}_{\text{out}}$. Available exergy $\dot{X}_{\text{ava}}$ is the amount of exergy removed from the exergy source, which may be a fluid stream (in the case of a heat exchanger), radiation source (in the case of the solar collector) or prime-mover (in the case of a pump or compressor). Used exergy $\dot{X}_{\text{use}}$ is the amount of available exergy that is delivered as a useful output by the component. For an expander this will be shaft work whereas for a heat exchanger this will be the exergy increase experienced by the cold stream. The exergy efficiency is then defined as $\eta_{\text{ex}} = \dot{X}_{\text{use}}/\dot{X}_{\text{ava}}$. Additional indicators of interest are: (1) the degree of thermodynamic perfection $\nu = \dot{X}_{\text{out}}/\dot{X}_{\text{in}}$; and (2) the influence coefficient $\beta = \dot{X}_{\text{ava}}/\dot{X}_{\text{ava,sys}}$, which represent the proportion of exergy that is not destroyed

and the fraction of the available exergy of the entire system that is associated with a particular component, respectively.

For the solar collector array, the exergy flow associated with the solar irradiance incident on its surface represents the maximum portion of the radiation energy that can be converted to useful work, and is given in by Eq. 2.7 in Chapter 2.

3.4 System Cost Evaluation

Each of the main components of the S-CHP system will be estimated in order to obtain a total investment cost for the S-CHP system. The four main components of the ORC system, i.e. the pump, expander and two heat exchangers (evaporator and condenser), represent a significant proportion of the cost of the overall system. Approximate estimates of the fraction of the overall installation costs of ORC systems, when these are used in waste heat recovery applications, indicate that these four components account for approximately 50–70% of the overall installed system cost [153, 154]. In the present work, another significant cost is the solar thermal collector array. The cost of each of the major items will be correlated to one or more key sizing variables by way of a market analysis. The cost of the other ancillary items such as the tank, plumbing, working fluids and associated installation costs will also be considered in relation to the major cost items. The cost relations are summarised in Table 3.2 (presented in Euros) and discussed below.

3.4.1 ORC costs

Heat exchangers

The main factors that influence heat exchanger cost are the choice of fluids and the rate of heat transfer per unit log-mean temperature difference, $\dot{Q}/\Delta T_{LM}$. This forms the basis of the C-value method for approximate costing of heat exchangers described in Hewitt *et al.* [155]. Although tables of C-values for heat exchangers are widely available, the range of costs and duties in

these charts were found to be too high to be applicable to the small-scale system considered here. An alternative approach used in the ORC literature is to evaluate heat exchanger cost using correlations based simply on the heat exchanger surface area. Numerous studies in ORC literature [135, 138] cite the heat exchanger cost correlation of Quoilin [156]:

$$J_{\text{hx}} = 190 + 310 \cdot A_{\text{hx}}. \quad (3.34)$$

In order to test the suitability of this correlation for the range of heat exchanger sizes considered in the present study, catalogue listed prices were obtained for a range of small brazed-plate heat exchangers from different manufacturers. The data is plotted in Fig. 3.5, showing good agreement with the correlation of Quoilin [156] and confirming its suitability for the present study. It is noted that most of the listed price data available was for single unit purchases. Where data is available for the purchase of bulk quantities (the Alfa Laval catalogue lists discount prices for the smallest heat exchangers less than 1 m² heat exchange area when bought in quantities of 50 units or more) the price per item is found to be 25% lower than the single unit cost. Thus the single item price can be assumed to include a retail factor of at least 25%. The Quoilin correlation produces results that are closer to the single item price and thus appears to be representative of the market price rather than the production cost of the items. In the costing model Eq. 3.34 is used to return approximate costs for the evaporator and water-cooled condenser heat exchangers based on the heat transfer areas determined in the system sizing model.

ORC pump

Costs were obtained and compared for a range of positive displacement pumps, spanning a broad range of flow rates and operating pressures. Based on the approach of Bejan *et al.* [157] and Quoilin [156] pump cost can be related to the rated power of the pump. For this study the required motor input power was used, which unless otherwise stated was calculated as the maximum delivered hydraulic power divided by an assumed overall pump efficiency (including electrical, mechanical and internal irreversibility losses) of 65%. For some of the smaller vane

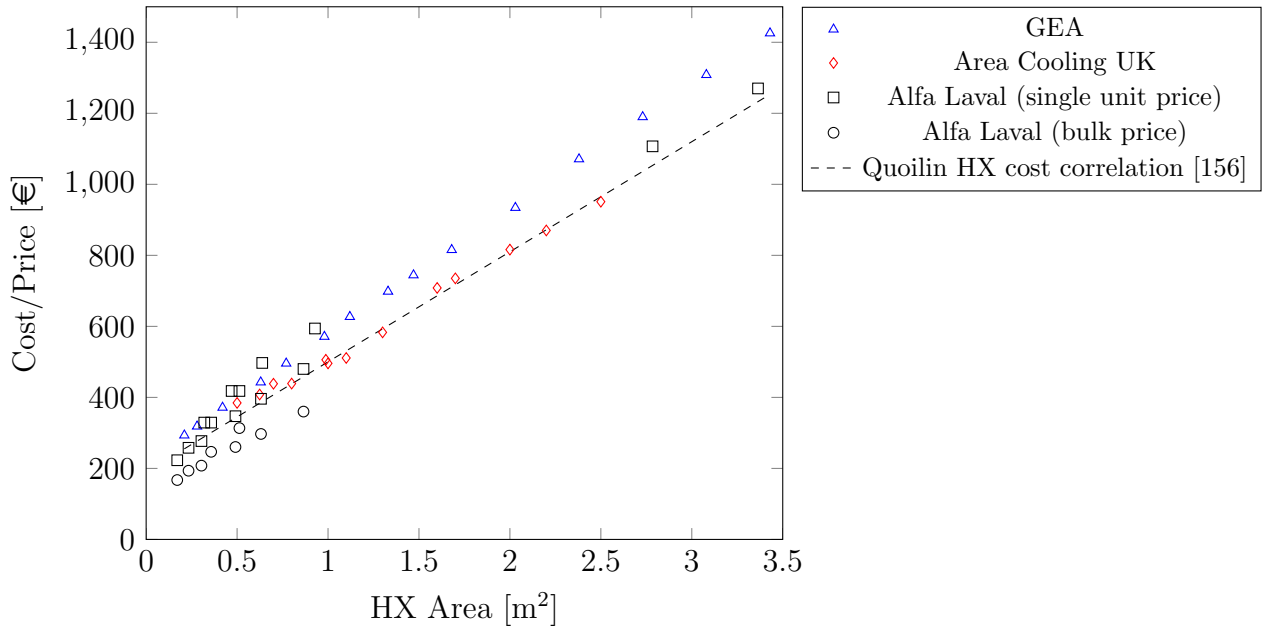


Figure 3.5: Brazed plate heat exchanger list prices compared with the cost correlation of Quoilin [156].

pumps investigated, the cost of the motor was not included in the list price. Where this was the case, the cost of the motor was assumed to be equal to the cost of the pump (thus the overall cost of the pump and motor was assumed to be double the standalone cost of the pump). The above assumptions were verified against the findings for other pumps where detailed information on motor rating and cost breakdown was available. As shown in Fig. 3.6, the prices obtained were found to vary widely among pumps depending on their intended application. Diaphragm pumps for metering and high-precision chemical dosing applications were found to have higher listed prices than diaphragm pumps for agricultural spraying applications and vane pumps for fluid-pressurisation and beverage making applications. The range of vane pumps were found to cover a lower rated power more applicable to this study, but were also found to have a lower range of maximum pressure ratings (4–18 bar) compared to the diaphragm pumps investigated (30–100 bar). For the cost analysis, only vane pumps with a minimum pressure rating of 8 bar were considered. A linear fit was applied to the vane pump prices plotted as a function of the motor input power. The fitted line is compared in Fig. 3.6 to the correlation of Quoilin [156] for ORC working fluid pumps. The latter correlation is based on a reference price of €900 for a 300 W positive displacement pump and then extended over a wider range of pump capacities using a scaling exponent of 0.25; and is found to provide a good fit for vane pumps in the range

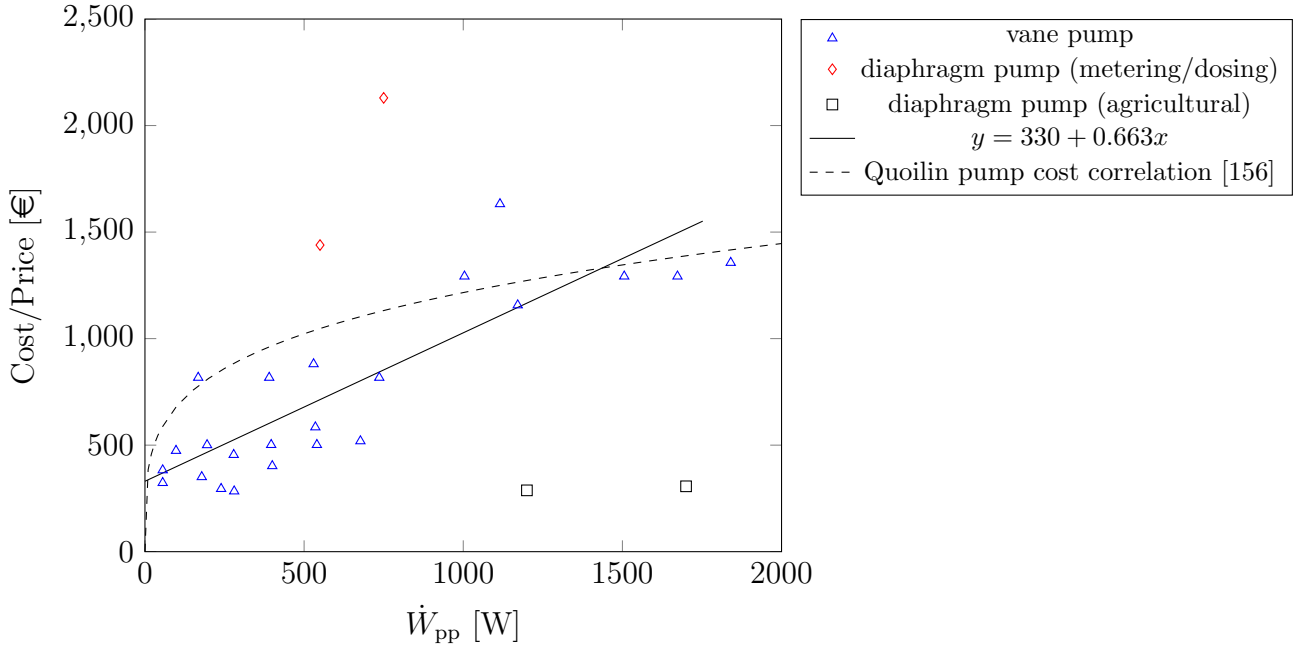


Figure 3.6: Small positive displacement pump prices plotted against rated input power. The linear fit for vane pumps (solid line) is compared to the pump cost correlation of Quoilin [156] (dashed line).

of 1–2 kW but over-predicts the mean price per W obtained for pumps < 1 kW. As the size of pump for the small-scale ORC system considered here is predicted to be considerably smaller than 1 kW, the linear fit obtained from the vane pump prices will be used instead, giving the following equation for estimated cost as a function of the pump input power:

$$J_{pp} = 330 + 0.663 \cdot \dot{W}_{pp}. \quad (3.35)$$

ORC expander

For the reasons listed in Section 1.1.3, the cost analysis for the expander will focus on positive-displacement displacement machines due to their suitability in small-scale applications. Recent studies have noted that positive-displacement expanders are not widely available at present as off-the-shelf items specifically for use in power generation [38, 158]. Instead, following the approach of Quoilin [156] a cost correlation will be based on the cost of suitable compressors with an equivalent swept volume. It should be noted here that the aforementioned study defines “swept volume in expander mode” as the swept volume in compressor mode divided by

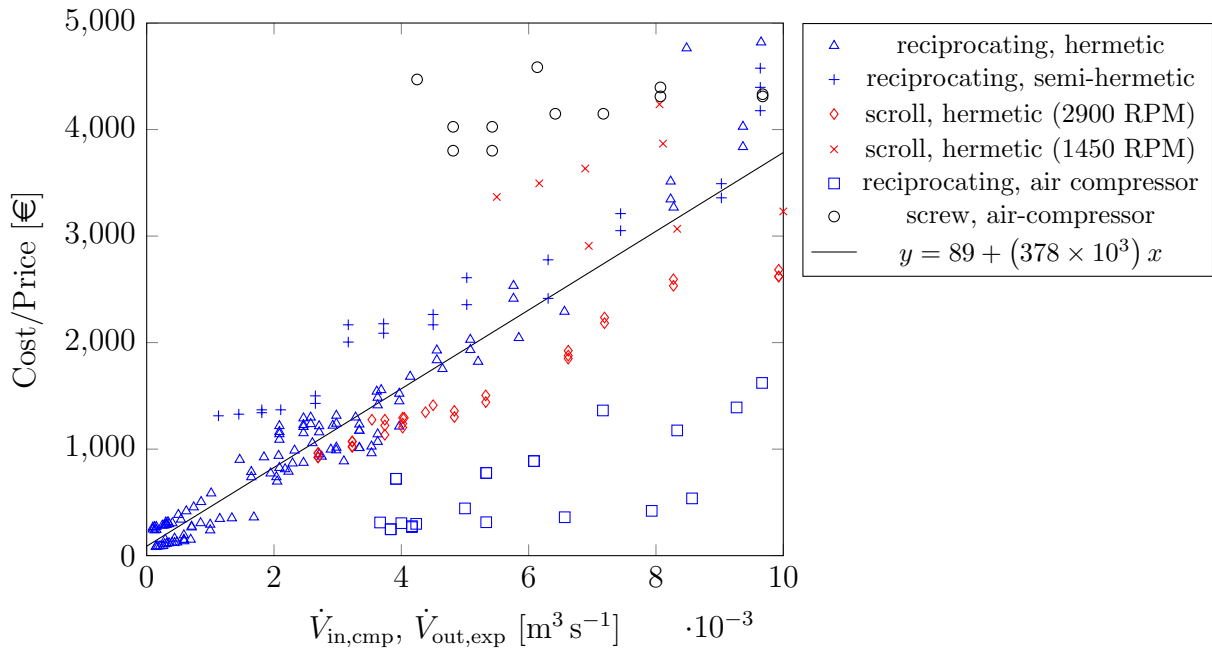


Figure 3.7: Small positive displacement compressor prices plotted against volumetric (displacement) flow rate.

the built-in volume ratio of the machine:

$$V_{\text{sw,exp}} = \frac{V_{\text{sw,cmp}}}{r_v} . \quad (3.36)$$

If the swept volume in compressor mode is representative of the displacement volume of the intake/suction stroke, then according to Eq. 3.36, the swept volume in expander mode also represents the volume of the intake stroke but does not include the volume of the expansion stroke. Rather than using this definition for the present cost analysis, the intake volumetric flow-rate in compressor mode will be assumed equivalent to the exhaust volumetric flow-rate in expander mode for the purpose of defining the size and cost of the machine. These two parameters are defined as follows:

$$\dot{V}_{\text{in,cmp}} = \frac{\dot{m}_r}{\rho_{\text{in,cmp}}} \approx V_{\text{sw,cmp}} \cdot N_{\text{rot,cmp}} , \quad (3.37)$$

$$\dot{V}_{\text{out,exp}} = \frac{\dot{m}_r}{\rho_{\text{out,exp}}} . \quad (3.38)$$

It will be shown later in Chapter 4 that the volumetric flow-rate defined by Eq. 3.39 is

expected to be less than $0.01 \text{ m}^3 \text{ s}^{-1}$ (600 L min^{-1}) for the proposed S-CHP system with a solar collector array sized for a domestic roof-space. Fig 3.7 shows that there is a wide variation in compressor costs in this flow-rate range depending primarily on application (reciprocating refrigeration compressors are found to have higher costs than reciprocating air compressors due to the requirement for hermetic sealing to prevent refrigerant leakage); and also depending on the design of the prime-mover. Reciprocating compressors are the most common type of refrigeration compressor at smaller flow-rates and cooling duties. It is noted in Fig. 3.7 that for the scroll compressors shown, those with lower rotational speeds are more expensive due to their larger swept volume and physical size. The advantage of the larger, more expensive compressors when used in a refrigeration cycle appears to be a higher coefficient of performance. Rotary screw-type air compressors show comparatively higher costs, but for the data obtained it was not possible to separate the cost of the compressor from the cost of the control system and other accessories. The trend line in Figure 3.4 is fitted only to the data for hermetic reciprocating and scroll-type refrigeration compressors because these are considered to be most representative of the cost of an equivalent expander for a refrigerant-based ORC system, and result in the following expression for expander cost:

$$J_{\text{exp}} = 89 + (378 \times 10^3) \dot{V}_{\text{out,exp}} . \quad (3.39)$$

3.4.2 Solar system costs

Approximate costs for the solar system components were obtained from a market survey of packaged solar systems and are listed in Table 3.2. According to the Energy Saving Trust, the typical installed cost of a solar water heating system in the UK is in the region of £3,000–5,000, ($\sim \text{€}3,500\text{--}6,000$) based on a typical collector array area of 3.6 m^2 [159]. The cost for such a system is considerably higher in the UK than in countries where solar hot water systems are sold in greater numbers. According to the UK Solar Trade Association [160], at present only $\sim 44\%$ of the installed cost of a solar hot water system in the UK is typically attributed to wholesale equipment costs, with the remainder attributed to customer acquisition, business

administration, physical installation, and VAT. An increase in the size of the UK market could see the total installed costs fall by as much as 30%. The cost of the collector array forms a significant fraction of the purchased equipment cost, but is highly variable depending on the type of collector, manufacturer and country of purchase. For the present study, a techno-economic comparison of the S-CHP system with standard ETC and high-performance EFPC collectors will be undertaken. This will take into account parameters such as the specific investment cost (SIC) and levelised cost of energy (LCOE) and will therefore require a representative cost for the two types of collector.

ETC market prices as low as as ¥30 (€4) per tube are reported in China, where the majority of the world's solar water heater (SWH) capacity are manufactured and installed [161, 162]. Based on typical collector areas for 20 and 30-tube ETC modules this is approximately 25 €/m², a value considerably lower than that found in UK retailer price-lists, which can be as high as £60 (€70) per tube, or 500 €/m². The EFPC meanwhile is cited as a promising technology for the near future but has not yet entered the wider solar-thermal market; and although Solar Keymark certified test data is available for the performance of these collectors very little information is available to allow an estimate of their anticipated sale price. A target cost-profile comparable with standard (glazed) flat plate collectors of €150–250 is the best estimate available at present [111], which is also comparable with the range of costs associated with ETCs in some other economic analyses in the literature [163, 164]². For the present study, the EFPC collector will be assigned a cost in the middle of the range quoted above (200 €/m²), while the ETC will be assigned a lower cost of 150 €/m² on account of there being a wider choice of manufacturers to choose from at present. By comparison, a small-scale commercial PTC system including solar tracking and drive system, can be expected to cost in the region of ~ 500 €/m² [166].

²Some studies report standard FPCs as having lower area-related capital costs than ETCs [164, 165], while Qiu *et al.* [162] report the opposite finding for China, due primarily to the low cost of ETC manufacturing cited above.

3.4.3 ORC and solar system ancillary item costs

The items described above form the major purchased equipment costs of the S-CHP system. Table 3.2 gives the full list of components included in the cost model along with the cost value assigned to the item, or the equation used to calculate the cost from one or more key sizing variables. Significant items on the list include the ORC working fluid cost, the cost of the air-cooled heat exchanger module used to reject heat in the air-cooled variant of the S-CHP system (based on the equivalent costs of small dry-air cooling tower units), and the hot water tank. A liquid receiver tank is also often included in ORC systems found in the literature [135, 158] in order to provide a steady head of liquid to the pump and protect against cavitation. Costs for items such as the supports, pipework and heat transfer fluid are obtained as a fraction of the collector array cost from price breakdowns of evacuated-tube solar hot water systems and then converted to an area-related cost.

Table 3.2: Summary of S-CHP system capital costs.

Item	Cost, €
<i>ORC main system component costs</i>	
Expander	$89 + 378 \times 10^{-3} \cdot \dot{V}_{\text{sw,exp}}$
Pump	$330 + 0.663 \cdot \dot{W}_{\text{pp,orc}}$
Heat exchangers	$190 + 310 \cdot A_{\text{hx}}$
Air-cooled heat exchanger [167]	$90 \cdot \dot{Q}_{\text{cond}}$
Generator [168]	520
Liquid receiver tank (5 L) [156]	110
Working fluid [169]	$30 \cdot M_{\text{fluid}}$
<i>Area-related solar-thermal system costs</i>	
Evacuated tube collector (ETC)	$150 \cdot A_{\text{sc}}$
Evacuated flat plate collector (EFPC)	$200 \cdot A_{\text{sc}}$
Circulation pump [170]	$83 + 0.64 \cdot \dot{W}_{\text{pp,sc}}$
Supporting framework [171]	$12 \cdot A_{\text{sc}}$
Pipework and valves [171]	$6 \cdot A_{\text{sc}}$
Heat transfer fluid [171]	$7.5 \cdot A_{\text{sc}}$
<i>Non area-related ancillary costs</i>	
Hot water tank ⁽ⁱ⁾ [167]	$5500 \cdot V_{\text{tank}}$
Expansion vessel (25 L) [172]	60
Instrumentation and controls [171]	300
Installation cost	$0.2 \cdot J_{\text{PEC}}$

⁽ⁱ⁾ Based on a Grade 2 specification, tested to 2.2 bar, 15 m maximum head.

All cost values are obtained from current list prices, or from cost relations given in the literature and updated to 2016 values where necessary. The installation (labour) cost is given as a percentage of the overall system component cost. Labour costs are estimated as 30% of purchase equipment costs (PEC) for medium sized ORC system that are assembled on site [135, 138]. However, as explained by Lukawski [154], for small-scale ORC plant assembled off-site as a packaged unit, on-site installation costs can fall dramatically to as low as 6% of PEC. For this study, an installation cost of 20% of PEC is chosen, which is comparable to value chosen by Herrando [23] for a domestic-scale CHP system based on PVT collectors that includes solar hot water and electricity generating sub-systems. Finally, annual operation and maintenance (O&M) costs will be estimated as 1% of the initial investment cost [122].

3.4.4 Electricity and hot water sub-system costs

The system capital costs and running costs can be approximately apportioned into “electrical sub-system costs” and “hot water sub-system costs”. This will have particular relevance for the calculation of the levelised costs of electricity and water heating in Section 4.6, and for Section 4.7 in which the installed cost of the system is compared with that of a PV-thermal hybrid (PVT) system (producing electricity and hot water), and a PV-only system (producing electricity but no hot water). The costs are divided in such a way that the costs of the S-CHP system associated with electricity provision are additional to that that would otherwise be required for a basic solar hot water system installation.

The approach for apportioning the costs is as follows: the full cost of the ORC components are designated as electrical sub-system costs, as well as 50% of the cost of the collector array and the associated fixings and plumbing items (on account of the array area being larger than that required for an equivalent hot water-only system). The remaining 50% of these costs are designated as hot water system costs. The control system cost, installation cost, and annual operation and maintenance (O & M) costs are also split evenly between the electricity and hot water sub-systems. Finally, the full cost of the solar fluid pump, heat transfer fluid, hot water tank and expansion vessel are designated as hot water sub-system costs.

3.4.5 Levelised cost analysis

A levelised cost analysis of the S-CHP system is undertaken in order to quantify the total economic savings over the system lifetime, relative to a “conventional” electricity and hot water system whereby electricity is purchased from the grid and hot water is provided from a boiler burning natural gas. The system will be assessed in terms of the levelised cost of energy (LCOE) it produces over a specified lifetime of N years, and also the time required for the cumulative fuel savings to equal the original investment, at which point the system is said to have achieved payback. The key economic and environmental parameters used in the analysis are summarised in Table 3.3. The LCOE is calculated separately for the electrical and thermal energy outputs as follows:

$$\text{LCOE} = \frac{J_{\text{cap}} + \sum_{n=1}^N J_{\text{OM}} (1+i)^{n-1} (1+d)^{-n}}{\sum_{n=1}^N Q (1+d)^{-n}} \quad (3.40)$$

where Q is the net annual production of either electrical or thermal energy, and J_{cap} and J_{OM} are the initial investment and annual O&M costs designated between the electricity and hot water sub-systems (see Section 3.4.4). The payback period n_p of the S-CHP system is found by calculating time required for the cumulative fuel savings, relative to the conventional system, to equal the initial investment. For this analysis, the annual cash flows associated with the S-CHP system must also include the additional cost of the auxiliary energy required to meet the fraction of the household electricity and hot water demand not provided by the S-CHP system:

$$n_p = \ln \left[\frac{J_{\text{cap}}(i-d)}{J_{\text{ann}}} \right] \cdot \ln \left[\frac{(1+i)}{(1+d)} \right]^{-1}, \quad (3.41)$$

where

$$J_{\text{ann}} = J_{\text{OM}} + J_{\text{aux}}. \quad (3.42)$$

The auxiliary hot water cost is obtained from the annual total $Q_{\text{hw,aux}}$ (see Eq. 3.32) converted to a primary fuel demand and an equivalent annual cost using the boiler efficiency and unit price of natural gas listed in Table 3.3. For the auxiliary electricity requirement it is assumed that all electrical energy produced over the year by S-CHP system is used on site and can be

Table 3.3: Economic parameters, utility prices, emissions factors and auxiliary system assumptions.

Parameter	Value	Reference
Electricity price (UK), €/kWh	0.2054	[174]
Natural gas price (UK), €/kWh	0.0636	[174]
Grid electricity equivalent emission, kgCO ₂ e/kWh _e	0.41205	[175]
Natural gas (gross CV) equivalent emission, kgCO ₂ e/kWh _{th}	0.18400	[175]
Domestic boiler efficiency (typical, UK), %	88	[176]
Annual household electricity demand (typical, UK), kWh yr ⁻¹	3,300	[173]
Annual household electricity demand (typical, Cyprus) ⁽ⁱ⁾ , kWh yr ⁻¹	4,973	[177]
Daily hot water consumption, L day ⁻¹	122	[125]
Discount rate for the S-CHP system, %	8	
Discount rate for the conventional system, %	2.5	
Inflation rate, %	3.2	

⁽ⁱ⁾ Cyprus electricity consumption includes air conditioning.

⁽ⁱⁱ⁾ The same domestic hot water consumption is assumed for UK and Cyprus.

subtracted from the total annual electricity demand, which is taken to be 3,300 kWh yr⁻¹ [173] for the UK. It is assumed that the remainder of this annual demand is met by purchasing electricity from the grid.

The discount rate d assumed in the calculations is 8% for the S-CHP system, based on a representative value for low-risk low-carbon technologies such as solar PV [23, 178–180]. For the conventional system, the lower risk is reflected in a lower discount rate, which following the approach of Herrando [23] is taken as the typical variable mortgage interest rate offered in the UK (2.5% as of March 2016 [181]). Inflation rate i is taken as 3.2% which was the average retail price index (RPI) between January 2010 and January 2016.

In Section 4.6, the S-CHP system will be simulated over an annual period using climate data for London (UK) and Larnaca (Cyprus). The purpose is primarily to assess the difference in electrical and thermal outputs, resulting from the difference in solar resource between the northern and southern European locations. However, it is noted that the economic case for the system may also be affected by market factors specific to the two localities. At the time of writing, electricity is slightly cheaper for domestic consumers in Cyprus than in the UK; however prior to 2014 electricity prices were lower for the UK. In order to simplify the analysis, the electricity price chosen for the calculation of the annual savings is the current average

for the 28 Member States of the European Union, weighted according to national domestic consumption [174]. Similarly, an EU weighted-average price for natural gas will be used for the calculation of the hot water savings, although it is noted that domestic water heating in Cyprus is usually provided by solar thermal systems with back up electrical immersion heaters [182], and that most homes are not served by a mains gas network.

3.5 Summary

A thermo-economic model has been presented in this chapter which will be used to investigate the performance and cost of a domestic-scale solar combined heat and power (S-CHP) system featuring an organic Rankine cycle (ORC) engine for electricity generation. Equations were presented for the sub-system models of the solar collector array, organic Rankine cycle engine components and domestic hot water storage tank, and for the various control algorithms that govern the system operation. Methodologies were also presented for the exergy analysis and levelised cost analysis of the system. The market analysis presented in this chapter showed a wide range of costs for some of the system components, in particular non-concentrating solar thermal collectors and positive-displacement expanders for ORC systems (based on available cost data for equivalent-sized reciprocating, screw and scroll compressors).

In the following chapter the model will be used to optimise the system under various design conditions, to simulate its performance over an annual period in a northern European and southern European climate and to provide an installed cost comparison with other established solar heat and power technologies.

Chapter 4

Thermo-economic Analysis of the S-CHP System

This chapter begins with a parametric analysis in which the key performance parameters of the S-CHP system are optimised for two different non-concentrating collector arrays (one a low-cost ETC array, the other a high-performance EFPC array). The electrical and hot water generation profiles of the S-CHP system over a diurnal operating period are then examined and some practical improvements to the system are discussed. Next, a comparison of air and water as the heat rejection medium is undertaken, followed by an investigation into the influence of the collector array configuration on the heat source temperatures, and a discussion of the system cost analysis. Results are then presented from an annual simulation, including: (1) the annual electrical power output that can be achieved from the system for a fixed 15 m² area of collector array, equivalent to that available on the roof of a typical UK home; (2) the cost per unit generating capacity; (3) the additional thermal output achievable and percentage of water-heating demand that can be met; (4) the system running costs, payback time and levelised cost of electricity. Finally an examination of the exergy destruction in each of the system components is performed using the results from the annual simulation.

4.1 Parametric Analysis

A parametric analysis was undertaken to examine the influence of the following variables on the electrical output from the system: (1) cycle evaporation temperature, T_{evap} ; (2) cycle condensation temperature, T_{cond} ; (3) ORC working fluid (refrigerant) flow-rate, $\dot{m}_r$; (4) degree of superheating at the expander inlet, ΔT_{sh} (also affecting the flow-rate and temperature glide in the solar/heat source fluid stream), and (5) fraction of solar fluid flow used for supplemental heating of the hot water tank, F_{coil} . Prior to this analysis, a preliminary screening of pure working fluids from the NIST REFPROP database [183] was undertaken in order to select a suitable fluid for the ORC system. The assessment was based on thermodynamic and environmental considerations. Of the shortlisted fluids that showed the best thermodynamic performance over the range of temperatures and pressures considered for the study, the majority were chlorfluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC), with non-zero ozone depletion potential (ODP). The hydrofluorocarbons (HFC) R245fa and R245ca, and the hydrofluoroolefin (HFO) R1233zd(E) were also found to show favourable thermodynamic performance. Due to its present wider availability, R245fa was used for the analysis and is also the fluid selected for the experimental system described in Chapter 6. A full description of the working fluid assessment is included in Appendix C.

The parametric analysis was performed for two variants of the S-CHP system: one with an array of EFPC collectors and one with an array of ETC collectors. The array size is limited to 15 m^2 , which as discussed in Chapter 1 is representative of the typical size of a single-residence home in the UK [21]. Simulations of the system were performed over a 24-hour operating period. The solar irradiance and air temperature datasets used for the simulation were obtained using a Davis Vantage Pro 2 weather station installed on the roof of a house in London. High resolution climate data is preferred to hourly-averaged data for simulations of solar systems in climates that experience intermittent irradiance conditions. Fig. 4.1 show power spectral density plots for solar irradiance data on a clear day and an intermittently cloudy day respectively, sampled at a frequency of 1 Hz. For the cloudy day the plot shows that useful information is contained in the signal at frequencies as high as 0.2 Hz. By comparison, the plot for the clear day shows

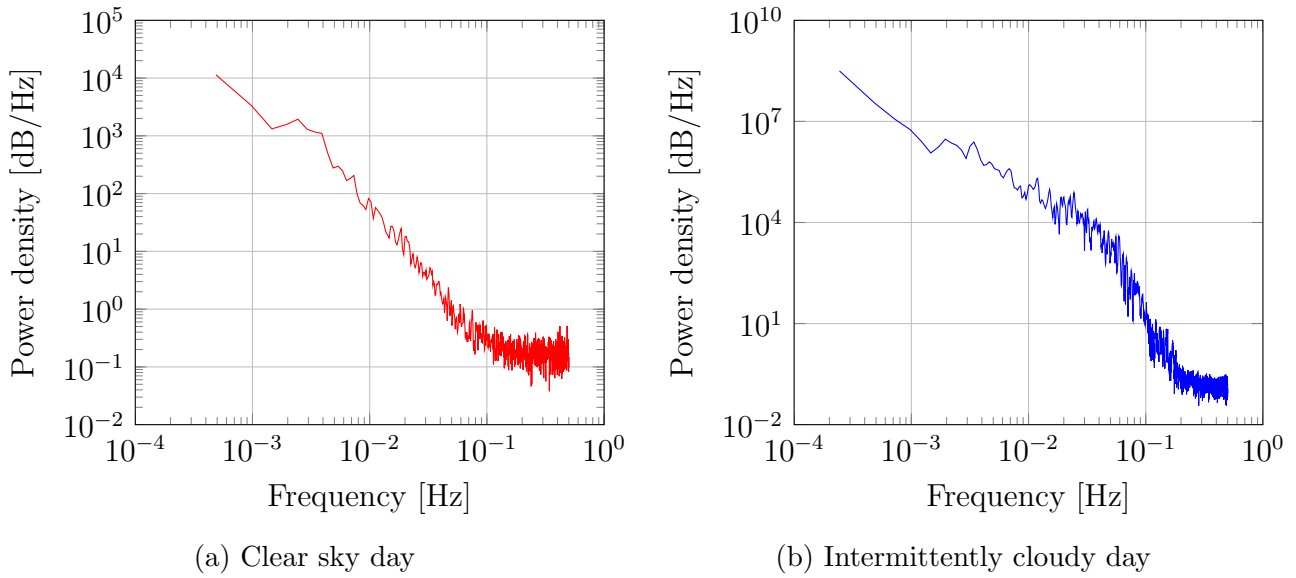
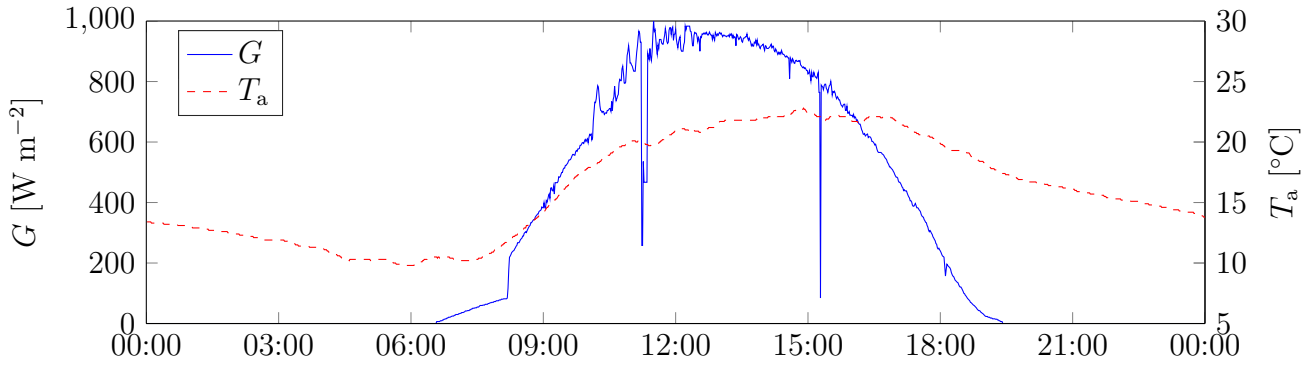


Figure 4.1: Power spectral density plots of solar irradiance data obtained for a clear sky day and a intermittently cloudy day.

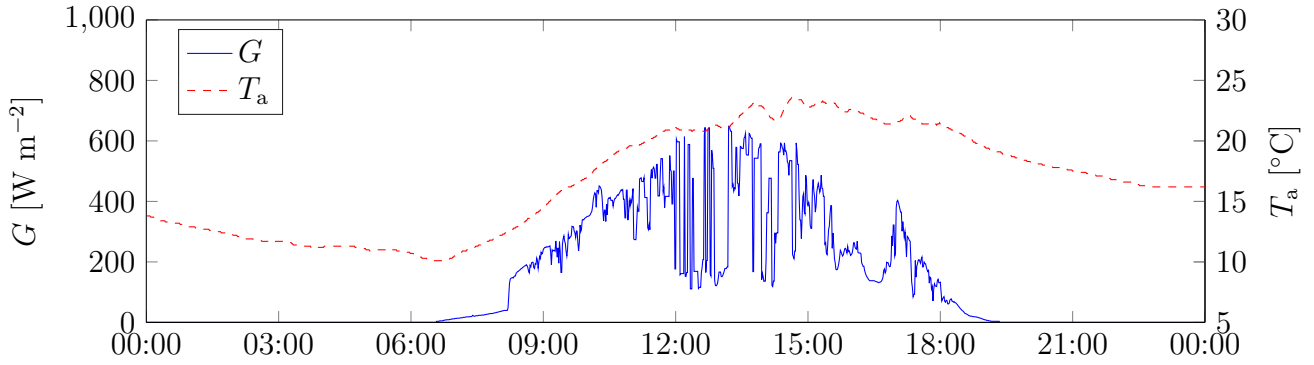
less useful information at frequencies higher than 0.05 Hz. This indicates that ideally a sample rate of 0.5 Hz or higher is required in order to capture all of the useful information in the signal for the cloudy day. Due to limitations of the weather station logging system, a sample rate of 1 per minute was the highest available for the data obtained over periods of several days at a time, but nonetheless this is a significant improvement compared to hourly averaged data. The profiles used for the parametric analysis are shown in Fig. 4.2, and were extracted from a dataset obtained during the month of September 2015. The two diurnal profiles were chosen to represent a “clear-sky” day and an intermittently cloudy day. The irradiance data in the two diurnal datasets has been corrected give a total daily insolation representative of that received by a south-facing surface inclined at an angle of approximately $30-40^\circ$ on a “peak solar” and an “annual average” day in the UK¹.

The objective of these diurnal simulations is to examine the system operating parameters and to understand their influence on the electrical performance of the system, with a view towards maximising the total daily work output and determining the “maximum work” settings. It follows therefore that the diverted fraction of solar collector fluid flow sent to the hot water tank F_{coil} was initially set to zero, thus prioritising power generation. In each simulation run,

¹ A brief analysis of hourly irradiance values for London over a whole year (Ref. [120]) shows that the mean daily irradiance varies between $7-333 \text{ W/m}^2$ with an annual average of 132 W/m^2 .



(a) Day profile 1: clear-sky day. Mean irradiance 293 W m^{-2} , mean temperature 16.1°C , maximum irradiance 1000 W m^{-2} , maximum temperature 22.8°C



(b) Day profile 2: intermittently cloudy day. Mean irradiance 132 W m^{-2} , mean temperature 16.7°C , maximum irradiance 648 W m^{-2} , maximum temperature 23.6°C

Figure 4.2: Daily irradiance and air temperature profiles

a single parameter was adjusted while the others were held constant. Figure 4.3 shows the sensitivity of the system net electrical power output (shown as the daily average value) to the individual operating parameters.

When simulated with the “clear-sky” solar irradiance profile the EFPC system was found to produce a maximum net electrical work output of 6.5 kWh day^{-1} ($271 \text{ W}_{\text{e,avg}}$) at an optimum ORC evaporation temperature of 130°C with 28 K of superheating; while the ETC system was found to produce a maximum net work output of 4.3 kWh day^{-1} ($180 \text{ W}_{\text{e,avg}}$) at an optimum evaporation temperature of 105°C with 14 K of superheating. When simulated with the intermittent “cloudy” irradiance profile, the EFPC system maximum work output was 2.0 kWh day^{-1} ($84 \text{ W}_{\text{e,avg}}$) and the ETC system maximum was 1.1 kWh day^{-1} ($45 \text{ W}_{\text{e,avg}}$), for optimal evaporation temperatures of 85°C . and 65°C , and with 18 K and 12 K of superheating respectively. The results were obtained with the cycle condensation temperature set to the

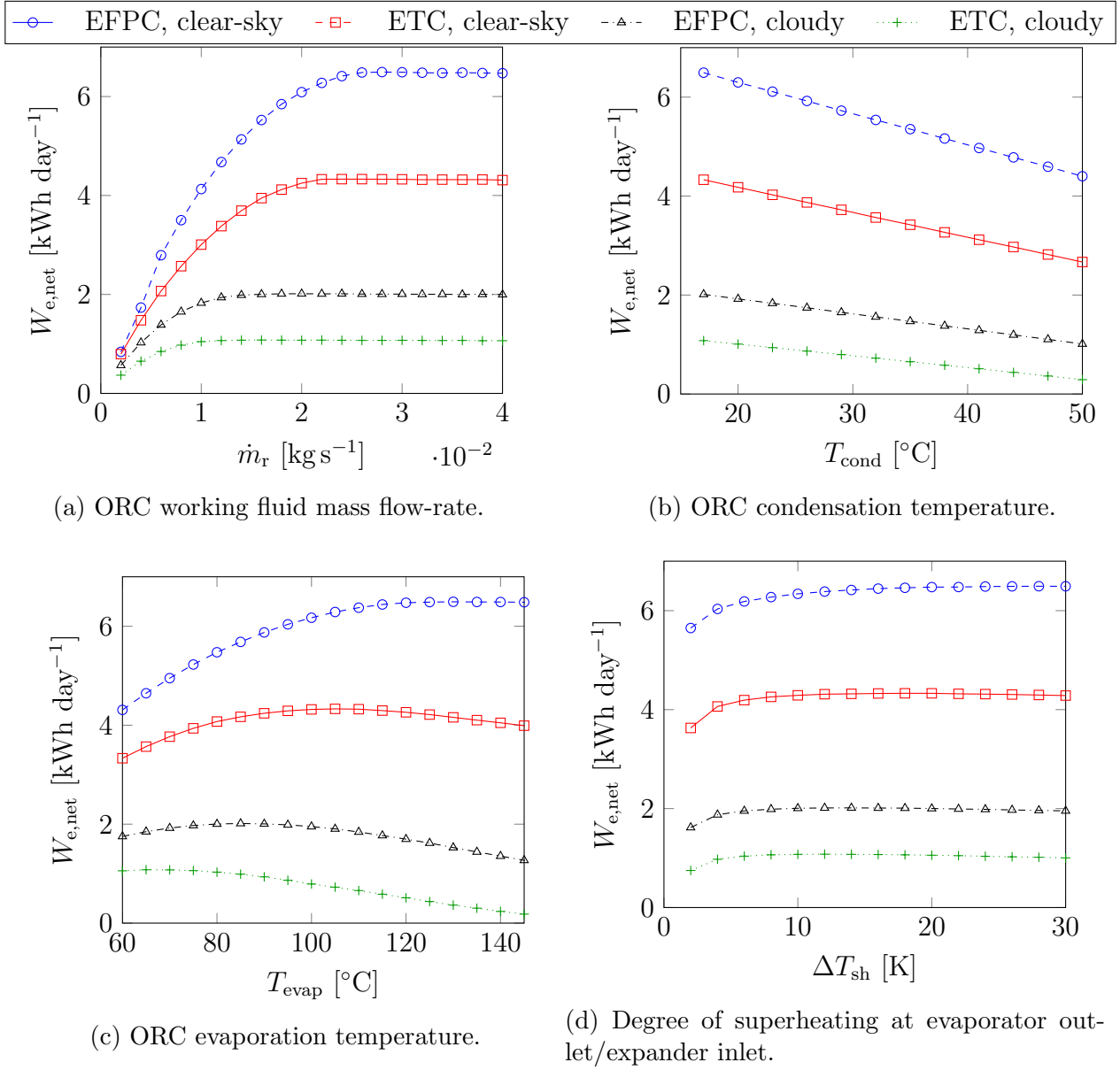


Figure 4.3: Parametric analysis of ORC electrical power output for the S-CHP system with EFPC and ETC collector arrays and for clear-sky and intermittently cloudy climatic conditions. Individual system operating parameters were varied as indicated, while all other parameters were set to maximum power settings as reported in Table 4.1.

Table 4.1: Electricity and hot water provision for Day profile 1 (clear-sky) and Day profile 2 (intermittently cloudy)

	EFPC system		ETC system	
	Day profile 1 (clear sky)	Day profile 2 (cloudy)	Day profile 1 (clear sky)	Day type 2 (cloudy)
$\dot{m}_r$, kg s ⁻¹	0.028	0.020	0.026	0.016
$\dot{m}_h$, kg s ⁻¹	0.071	0.130	0.118	0.135
$\dot{m}_w$, kg s ⁻¹	0.651	0.465	0.604	0.372
T_{evap} , °C	130	85	105	65
T_{cond} , °C	17	17	17	17
ΔT_{sh} (nominal), K	28	14	18	12
$\dot{W}_{\text{exp}}$ (nominal), W	1376	615	1006	361
$\dot{W}_{\text{net}}$ (nominal), W	1156	515	847	297
$r_{v,\text{exp}}$ (nominal), -	23.5	8.3	13.6	4.9
<i>Electricity prioritisation</i>				
F_{coil}	0	0	0	0
W_{elec} , kWh day ⁻¹	6.5	2.0	4.3	1.1
Q_{hw} , kWh day ⁻¹	5.0	2.0	3.3	1.3
$\eta_{\text{sol-e}}$, %	6.2	4.2	4.1	2.3
$\dot{W}_{\text{net,avg}}$, W	271	84	180	45
Hours of ORC operation	5.6	3.9	5.1	3.6
D_{elec} , %	71.8	22.3	47.9	11.9
D_{hw} , %	70.4	27.7	47.1	19.7
$T_{\text{hw,supply,max}}$, °C	50.4	25.9	36.9	21.0
$T_{\text{hw,supply,min}}$, °C	39.6	21.7	29.9	18.6
<i>Hot water prioritisation</i>				
F_{coil}	1	1	1	1
W_{elec} , kWh day ⁻¹	6.2	1.6	3.9	0.8
Q_{hw} , kWh day ⁻¹	6.9	6.4	6.4	5.1
$\eta_{\text{sol-e}}$, %	5.9	3.3	3.7	1.7
$\dot{W}_{\text{net,avg}}$, W	258	64	164	33
Hours of ORC operation	5.3	3.0	4.6	2.7
D_{elec} , %	68.6	17.1	43.5	8.8
D_{hw} , %	98.4	90.9	91.3	72.5
$T_{\text{hw,supply,max}}$, °C	80.1	77.9	80.1	61.0
$T_{\text{hw,supply,min}}$, °C	53.0	41.8	42.9	31.4

lowest practically achievable value of 17 °C (allowing for 10 °C cooling water temperature, with a 5 K condenser pinch and a 2 K temperature glide between cooling water inlet and the pinch-point — see Fig. 3.2). The full set of optimal operating parameters for each of the four cases (EFPC clear-sky, EFPC cloudy, ETC clear-sky and ETC cloudy) are presented in Table 4.1.

The mean irradiance during daylight hours on the clear-sky day is 548 W m⁻², while on the intermittently cloudy day it is 248 W m⁻². The corresponding solar to electrical efficiency for the S-CHP system with the EFPC collector is 6.2% on the clear-sky day and 4.2% on the cloudy day, while for the ETC it is 4.1% on the clear-sky day and 2.3% on the cloudy day. This

can be compared to the range of values predicted for the representative collector types in the endoreversible analysis in Chapter 2. In Tables 2.1 and 2.2, the endoreversible efficiency for the EFPC was 9.3% under peak irradiance conditions ($G = 1000 \text{ W m}^{-2}$) and 3.9% under low irradiance conditions ($G = 250 \text{ W m}^{-2}$), while for the ETC (DF, coaxial) it was 6.5% under peak irradiance conditions and 2.5% under low irradiance conditions. Thus based on the mean daily irradiance levels in the present analysis, the obtained efficiencies are broadly within the expected ranges. By further comparison, monocrystalline (mono-Si) PV can be expected to perform with nominal solar to electrical efficiencies of $\sim 15\%$ under peak solar conditions and modules can be expected to perform to within 90% of their nominal efficiency at irradiance values as low as 250 W m^{-2} [19, 131]. A further comparison with the endoreversible analysis from Chapter 2 and with PV will be made later in this chapter for the results of the annual performance analysis.

Working fluid flow-rate

The working fluid mass flow-rate was the parameter shown to have the largest impact on the daily net work output. The work output increases with working fluid flow-rate up to a threshold value (0.028 kg s^{-1} for the EFPC system in the “clear-sky” day simulation), beyond which a further increase in flow-rate no longer has an impact. As will be shown in Section 4.2, higher mass flow-rates correspond to higher instantaneous power output from the expander but lead to a more intermittent performance from the ORC (shorter periods of continuous operation) when the solar heat input to the cycle is limited.

Condensation temperature

The condensation temperature has the next most significant impact on the daily net work output, with both the instantaneous power output and the daily net work output decreasing linearly with increasing condensation temperature. As shown in Fig. 4.4, lower condensation temperatures require a higher mass flow-rate of the cold sink fluid stream, which in the present analysis is water. It is assumed here that pressurised water is available from the local utility

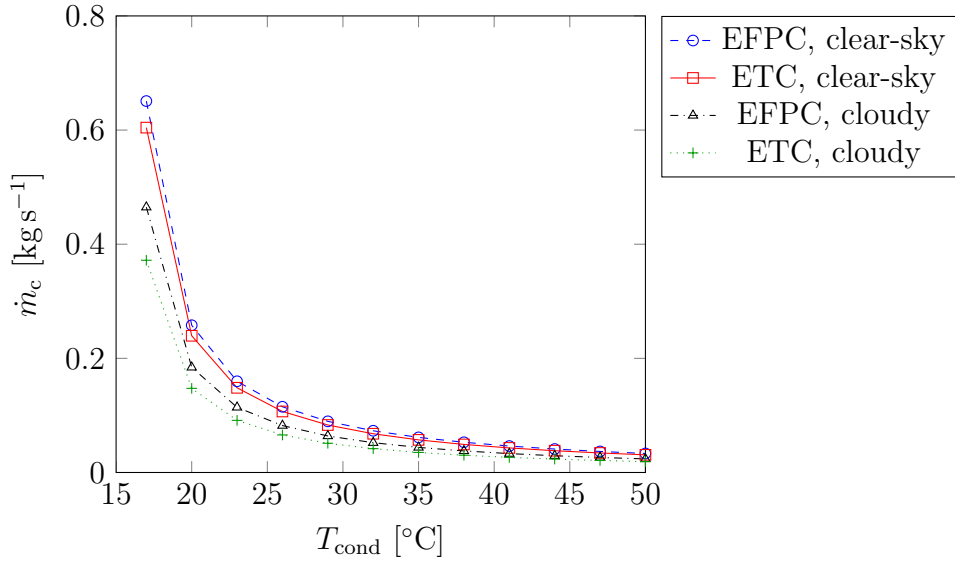


Figure 4.4: Cooling water flow-rate in the condenser plotted against condensation temperature. All other system settings including $\dot{m}_r$, T_{evap} , ΔT_{sh} set to the corresponding maximum power settings in Table 4.1.

mains and that the increased water flow-rate incurs no increase in pumping power that impacts the net work output. For the EFPC system the water demand decreases dramatically from 13 m³/day at $T_{\text{cond}} = 17$ °C, to 0.75 m³/day at $T_{\text{cond}} = 50$ °C; while the daily electrical work output also decreases by 32% (the relative impact on electrical output is more significant for the ETC system where evaporation temperature is lower). At present UK water rates, the associated costs decrease from £17 to 96p per day [184]. These results indicate that the exergetic benefit of mains water as the ORC heat rejection medium compared to air at ambient temperatures may be economically realistic only if the water quantity is available at little or no extra cost (possibly by being shared with another domestic or industrial process). The performance of the system with air as the heat rejection medium is discussed in Section 4.3.

Evaporation temperature and superheating temperature

The evaporation temperature and minimum degree of superheating at the evaporator outlet are strongly linked to the solar collector array temperature and hot-stream mass flow-rate because of the approach temperature difference and pinch point temperature difference, as described in Eqs 3.6 and 3.7. It should be noted that the *minimum* degree of superheating corresponds to

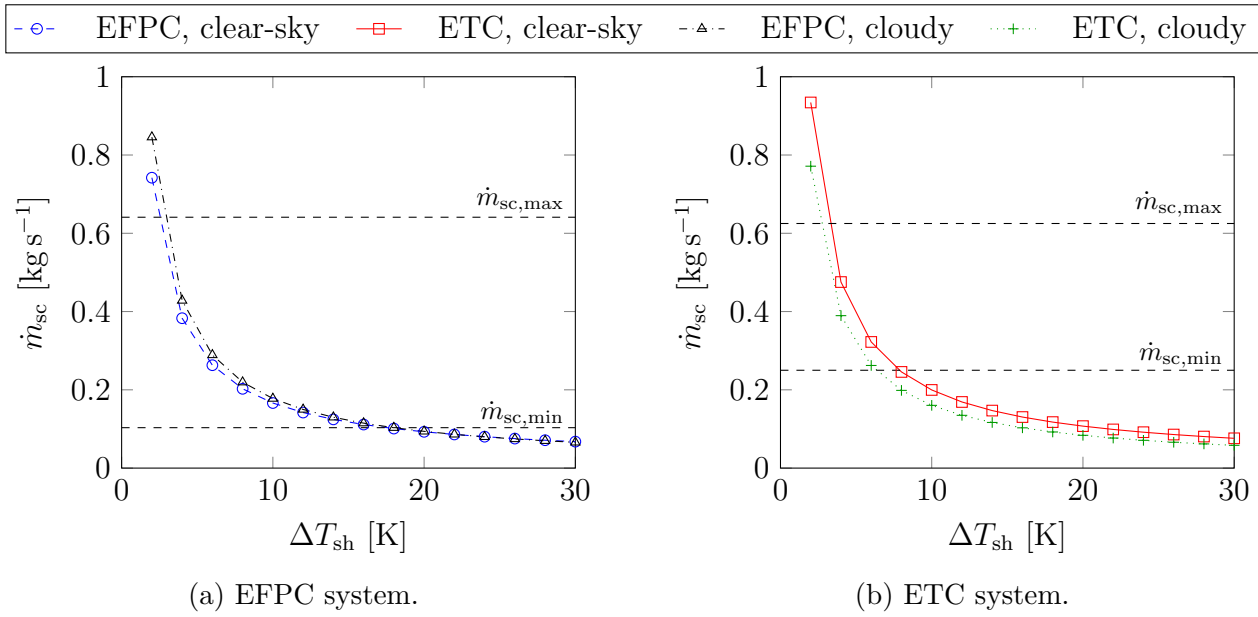


Figure 4.5: Solar collector array mass flow-rate plotted against minimum degree of superheating in the ORC, with all other parameters set to corresponding maximum power settings from Table 4.1. The dashed lines indicate the collector flow-rate range recommended by the manufacturer, scaled for a 15 m² collector array.

the minimum solar heat input required to operate the cycle, and that for higher solar radiation intensities, the degree of superheating will increase if all other system parameters are held constant. Increasing the evaporation temperature leads to an increase in ORC pressure ratio and specific work output, but also to an increase in solar collector temperature which negatively impacts the collector efficiency and the total heat energy input to the cycle. The minimum degree of superheating directly affects the solar collector/hot-stream temperature glide, such that a higher minimum superheating value sets a steeper temperature glide and therefore a lower mass flow-rate in the collector array according to Eq. 3.8. As a consequence the pumping power requirement for circulating the solar collector fluid is highest for a very low minimum superheating value and this leads to a reduction in net power output; while for a very high minimum superheating value the net power output is also limited by the increased solar collector array temperature. Thus the evaporation temperature and the minimum degree of superheating (and hence solar collector fluid mass flow-rate) both have optimal values that are neither at the highest nor the lowest ends of the range.

The relationship between the minimum degree of superheating and the mass flow-rate of the heat carrier fluid in the collector array are shown in Fig. 4.5. Also indicated for each

collector is the minimum and maximum flow-rates recommended by the manufacturer scaled to an equivalent array size of 15 m^2 . From Fig. 4.3d it was found that the optimal minimum superheat for maximum net work output was 28 K for the EFPC and 18 K for the ETC (under “clear-sky” day conditions). In both cases this corresponds to a mass flow-rate in the collector array that is lower than the manufacturer’s recommended range shown in Fig. 4.5. However, this can be addressed by considering the connection of the solar collectors in either series or parallel configurations. The default configuration for the simulation is a parallel configuration with the flow-rate divided equally between the appropriate number of collector modules that comprise the total array. By considering a series configuration instead the flow-rate through the individual modules can be increased while maintaining the same total mass flow-rate in the array. This will be given further consideration in Section 4.4.

Expander power output and volume ratio

The nominal instantaneous power output from the expander when optimised for the “clear-sky” day is between 1 – 1.3 kW, and the nominal volume ratio for the expansion process is between 13 and 24 (depending on the choice of solar collector). For the system optimised for the “cloudy” day, the nominal expander power output is between 0.3 – 0.6 kW and the volume ratio is between 5 and 8. By comparison, the smallest expander available on the market designed specifically for ORC systems (the Air-Squared 1 kW scroll expander [43]), has a nominal power output of 1 kW and a built-in volume ratio of 3.5. This indicates that the system design parameters do not match any readily available expander products in terms of power output, flow-rate and expansion ratio and may require a bespoke design that can deliver a large expansion ratio with a small working fluid flow-rate.

Solar hot water versus electricity

Figure 4.6 illustrates the impact on electricity and hot water provision that results from increasing the fraction of the solar collector fluid that passes through the upper coil in the domestic hot water tank after leaving the ORC evaporator (see also Eq. 3.26 and Figs 3.1 and 3.4). When

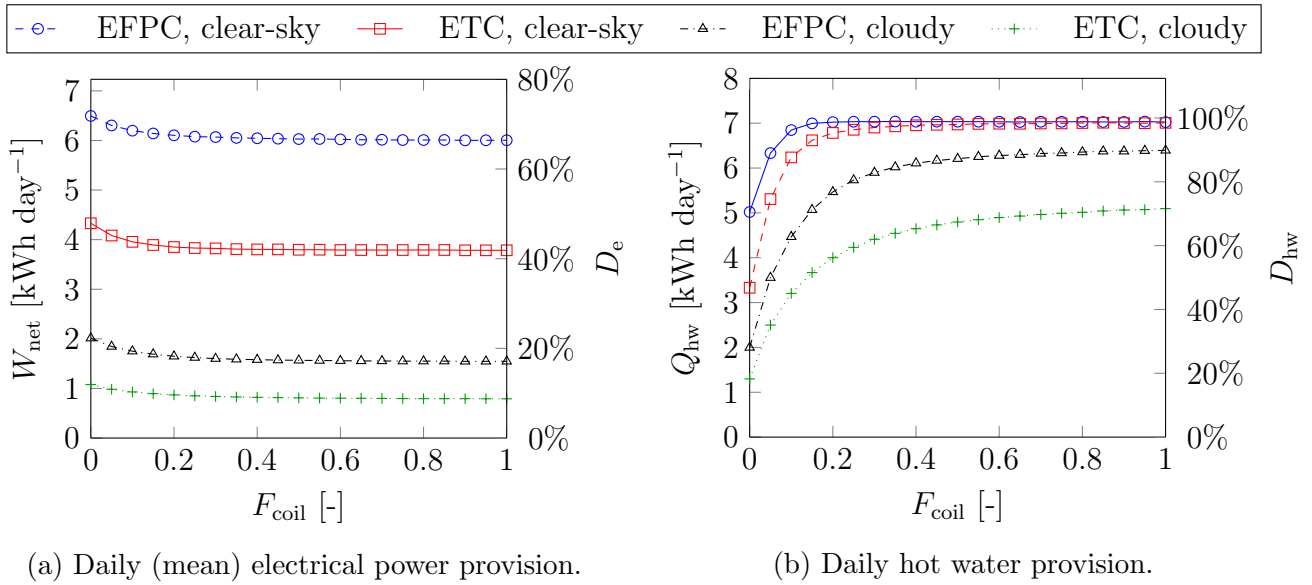
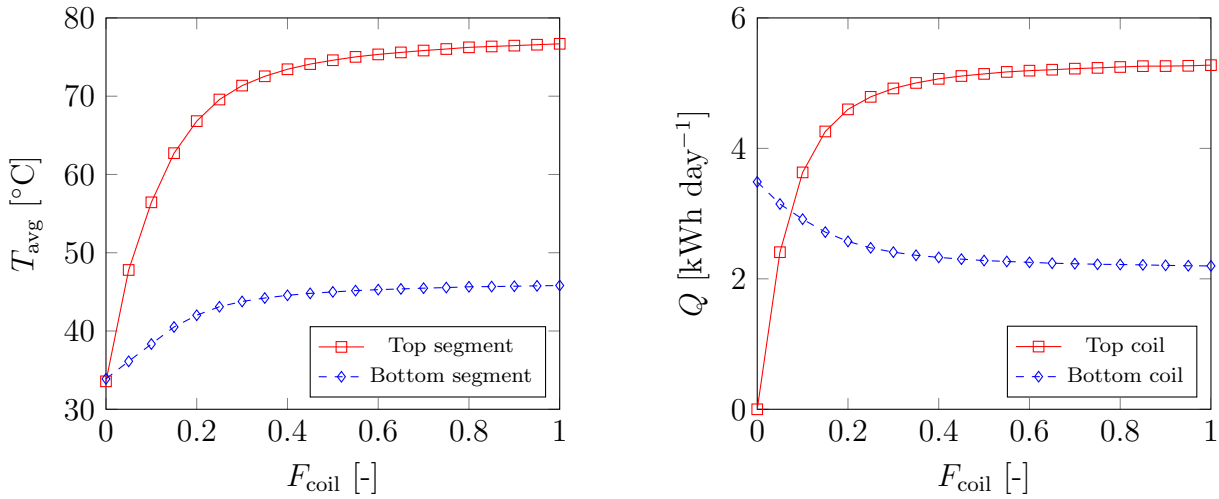


Figure 4.6: System net electrical power output and daily hot water provision plotted as a function of the solar coil flow factor F_{coil} , with all other parameters set to corresponding maximum power settings from Table 4.1.

the coil is entirely bypassed ($F_{\text{coil}} = 0$), the electrical performance of the system is maximised and the thermal input to the hot water tank is solely from the bottom coil in which the ORC working fluid is precooled before entering the condenser (see Fig 3.4). For this setting the EFPC system is able to supply a daily water heating load of $5.0 \text{ kWh}_{\text{th}} \text{ day}^{-1}$ under the “clear-sky day” simulation conditions while the ETC system is able to supply $3.3 \text{ kWh}_{\text{th}} \text{ day}^{-1}$. This is equivalent to a daily hot water demand provision of 72% and 48% respectively (based on a typical household demand of 122 L day^{-1} [125]). This can be considered a “worst-case” prediction for the water heating demand because it is based on an assumption that the required hot water delivery temperature is 60°C . However, as was found in a study by The Energy Saving Trust, in practice hot water supply temperatures from domestic boiler installations in the UK are often much lower, ranging from $42\text{--}62^\circ\text{C}$ with a mean value of 51.9°C (taken from a sample of 112 cases). Although safe delivery temperatures between $41\text{--}44^\circ\text{C}$ are recommended for wash-basin, shower and bath applications, storage temperatures of 60°C or higher are recommended for the prevention of Legionnaires disease [185]. For solar hot water systems, extended periods below 60°C are considered acceptable, but daily to weekly pasteurisation cycles at temperatures above 60°C for periods of two minutes or longer are recommended [186].

If 100% of the solar fluid flow is passed through the top coil ($F_{\text{coil}} = 1$) the daily electrical



(a) Daily mean water temperature in the top and bottom segments of the hot water tank.

(b) Thermal energy supplied to the tank from top and bottom coils.

Figure 4.7: Daily mean tank temperature and coil heating loads plotted against F_{coil} for the Day type 1 (“clear-sky”) simulation with the ETC collector array; where the top coil transfers heat to the top segment of the DHW tank from a fraction F_{coil} of the solar collector fluid exiting the evaporator, while the bottom coil transfers heat to the bottom segment of the tank from the pre-cooling of ORC working fluid exiting the expander.

output drops by 5–10% relative to the $F_{\text{coil}} = 0$ case, but the daily hot water provision increases to 98% for the EFPC system and 91% for the ETC system. Under “clear-sky” conditions, $F_{\text{coil}} = 1$ is found to result in the tank reaching its upper temperature limit of 80 $^{\circ}\text{C}$. It should be recalled that the collector array is far over-sized compared to a conventional solar hot water system with a similar sized tank (collector array sizes of 5 m^2 or less are typical). Therefore, the fraction of solar fluid sent to the top coil can be reduced, and values of $F_{\text{coil}} = 0.1$ are found to provide hot water at temperatures $> 43^{\circ}\text{C}$ for the whole day and $> 60^{\circ}\text{C}$ for sufficiently long periods to meet requirements to kill *Legionella* bacteria. Under “cloudy-day” conditions, the electrical output from the system is significantly lower, but water heating equivalent to 91% and 73% for the EFPC and ETC systems respectively can still be achieved when $F_{\text{coil}} = 1$. This is a significant advantage of the co-generation functionality of the system compared with electricity-only operation. The daily electrical and thermal performance of the system under electricity prioritisation ($F_{\text{coil}} = 0$) and hot water prioritisation ($F_{\text{coil}} = 1$) settings is summarised in Table 4.1. The daily temperature profiles of the hot water tank are discussed in Section 4.2.

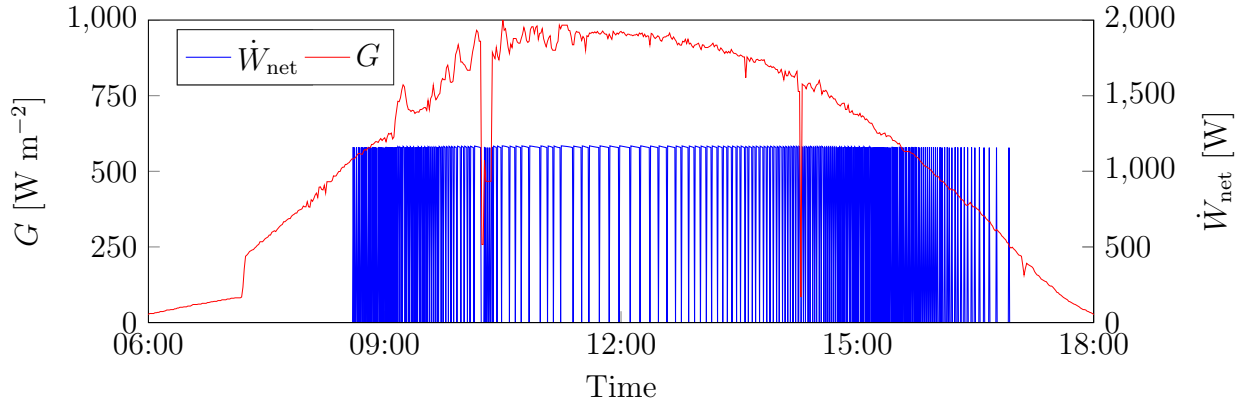
The energy fraction saved by reclaiming preheated cooling water from the condenser

is found to be very small under the maximum power settings. The total daily volume of hot-water drawn from the tank for domestic use is 122 L day^{-1} , which is significantly lower than the volume that is consumed for heat rejection purposes. At the lowest condensation temperature corresponding to maximum power, the cooling water enters the condenser at 10°C and exits at $12\text{--}13^\circ\text{C}$. Some of this preheated water is used to top-up the hot water tank, but this is only possible when there is a simultaneous draw-off of water from the tank. Over the course of the day the total volume of the preheated water that is delivered to the tank is $20\text{--}30 \text{ L}$ which is $\sim 25\%$ of the total demand volume and $< 1\%$ of the volume rejected from the condenser (the remainder of which is not used). It also represents a contribution of $< 1\%$ to the daily energy demand for water heating. When the condensation temperature is increased to 50°C , the temperature of the cooling water exiting the condenser is $50\text{--}60^\circ\text{C}$ and directly reclaiming this water to the tank represents a more significant contribution of up to 30% of the daily water heating demand, although the reduction in electrical work output is also significant (see Fig. 4.3b).

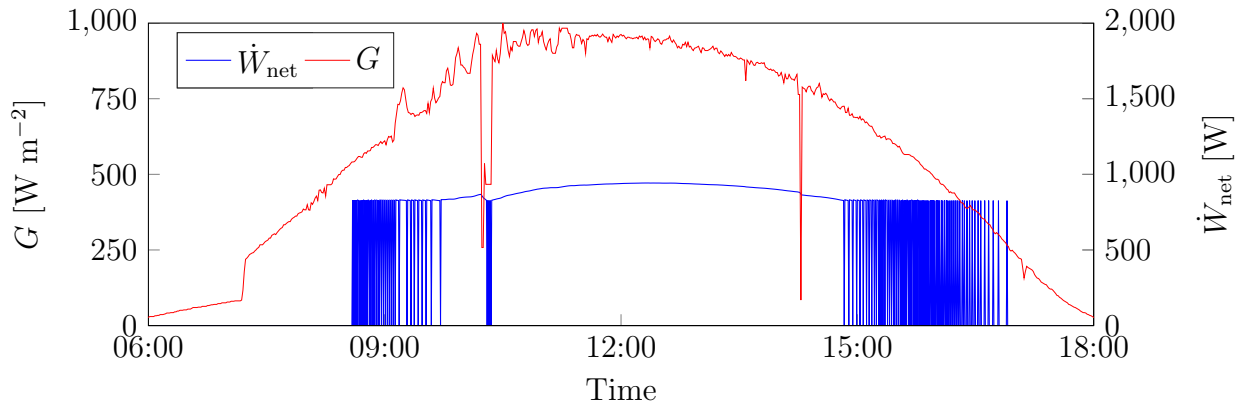
4.2 Diurnal Operation Analysis

In this section, the electricity and hot water performance of the S-CHP system over a 24-hour period will be examined and some practical considerations regarding the system operation will be discussed. Figure 4.8a shows a plot of the instantaneous net power output from the S-CHP on the clear-sky day and with the relevant optimal settings determined in Section 4.1 and listed in Table 4.1. While these settings are found to maximise the electrical work output over the diurnal period, Fig. 4.8a shows that they result in a highly intermittent system performance because the solar thermal input is not sufficient to sustain operation for periods longer than a few minutes at a time. This could be problematic as the dynamic response time of the ORC system components is not accounted for in the ORC sub-system model.

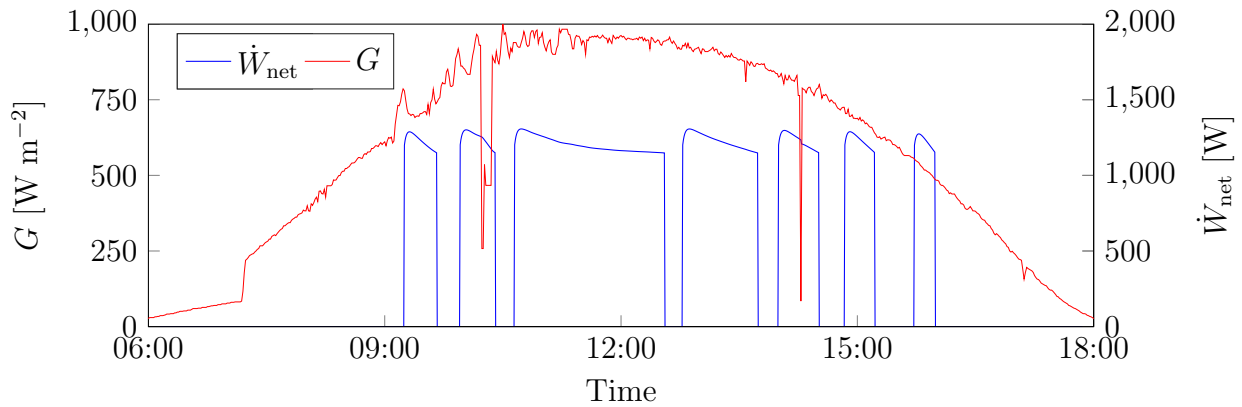
Figure 4.8b shows the effect of sizing the system for a smaller thermal input by reducing the nominal mass flow-rate of working fluid circulating in the ORC. Reducing the working fluid



(a) Non-buffered S-CHP system with EFPC array and optimal settings for the “clear-sky” day from Table 4.1, exhibiting high-frequency switching between ORC “on” and “off” states.



(b) Non-buffered system with ORC working fluid mass flow-rate reduced from 0.028 kg s^{-1} to 0.02 kg s^{-1} .



(c) “Buffered” system configuration (see also, Fig. 4.9), with a 25 L liquid buffering vessel at the collector array outlet in conjunction with dual-setpoint control of the ORC with a 10 K deadband.

Figure 4.8: Electrical performance of the EFPC S-CHP system operating under clear sky conditions with and without thermal buffering for longer sustained periods of continuous electricity output.

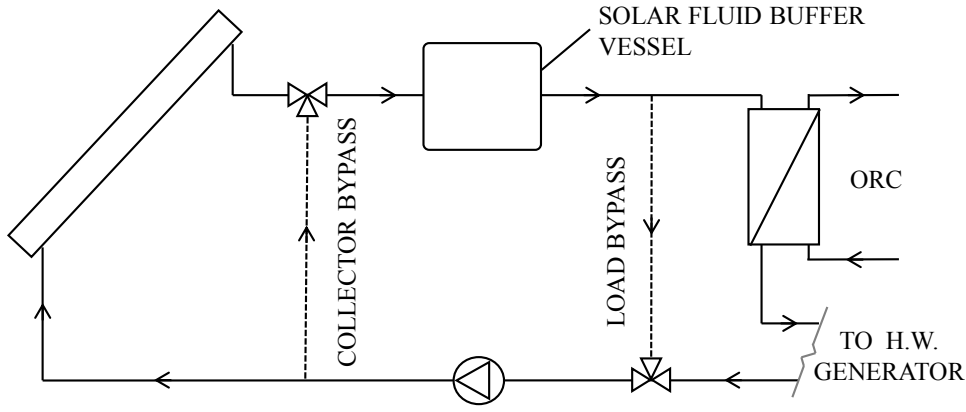
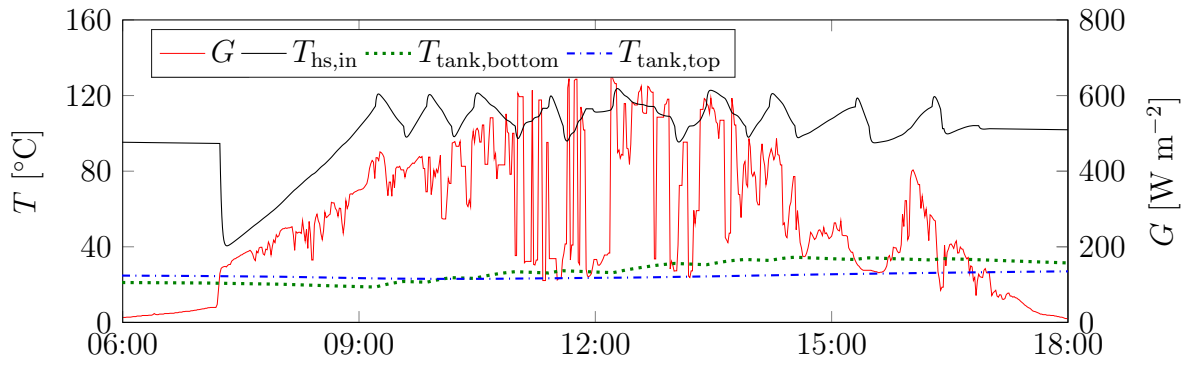


Figure 4.9: Configuration of a the solar fluid loop with a buffering vessel to increase sustained periods of electrical output under intermittent solar irradiance.

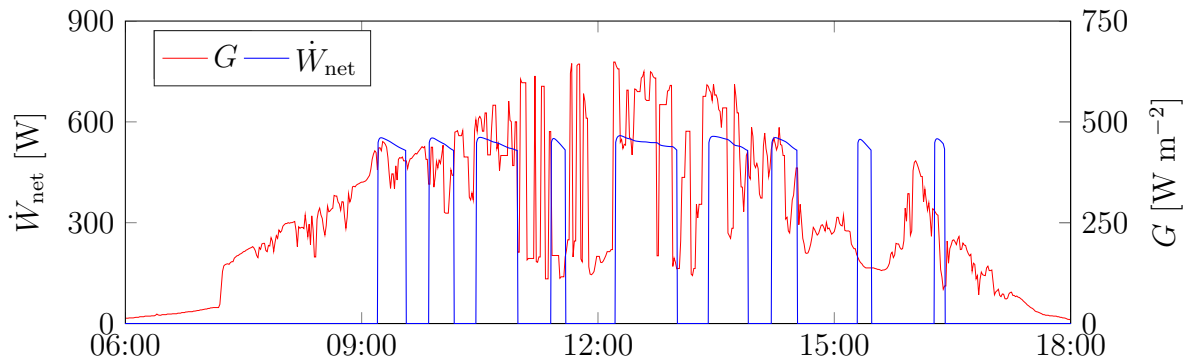
flow-rate from 0.028 kg s^{-1} to 0.02 kg s^{-1} (for the EFPC system) increases the longest period of uninterrupted operation from 9 minutes to 4 hours in the middle of the clear-sky day, at the expense of an overall drop in daily work output of 6%. However, the intermittency problem is still encountered in the morning and afternoon periods when solar irradiance is lower.

Another solution is to incorporate thermal energy storage (TES) to buffer the variations in solar thermal input. There are several options for TES, that will be discussed further in Section 4.6. It should be noted that some degree of thermal buffering is already provided by the collector array itself², which is accounted for in the model by the effective thermal capacity of the collectors in Eq. 3.1. In this Section, additional thermal storage will be provided by means of a liquid storage vessel incorporated into the solar collector fluid loop at the outlet of the solar collector array, prior to the ORC evaporator inlet as shown in Fig. 4.9. The operation of the buffer vessel is such that it can be charged by the solar collector array independently of the ORC operation (by enabling the ORC bypass) and can be discharged by the ORC independently of solar collector operation (by enabling the collector bypass). The buffer vessel is then operated in conjunction with the ORC using a dual set-point control, whereby the heat source set-point temperature (at the buffer vessel) required to initiate the operation of the ORC is higher than the set-point temperature at which the ORC switches off. The temperature difference between the high and low set-points is called the deadband. It is found that a 25 L vessel with a 10 K deadband enables the system to operate for sustained periods of up to 2 hours and no shorter

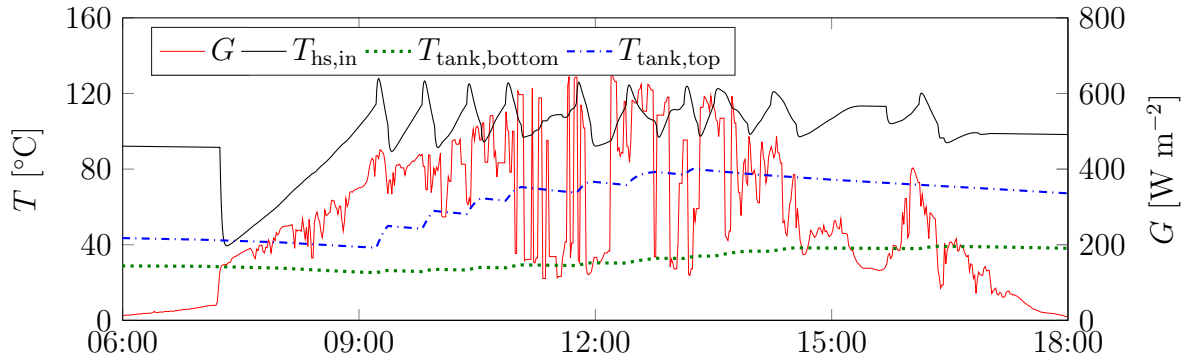
²Based on the collector fluid capacities from the manufacturers data, a 15 m^2 array of EFPC collectors will hold 12 L of fluid while a 15 m^2 array of ETC collectors will hold 20 L of fluid.



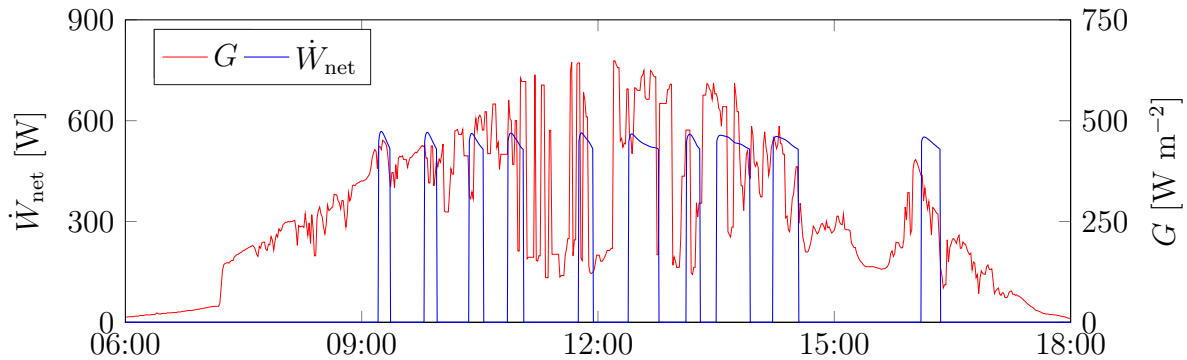
(a) Temperature profiles for the S-CHP system operating under electricity generation priority conditions ($F_{\text{coil}} = 0$). Temperatures shown are that of the solar collector fluid (ORC heat source) and the top and bottom segments of the hot water tank. Solar irradiance is also shown.



(b) Electrical power generation profile, corresponding to Fig. 4.10a for $F_{\text{coil}} = 0$.



(c) Temperature profiles for the S-CHP system operating under conditions for hot water prioritisation ($F_{\text{coil}} = 1$).



(d) Electrical power generation profile, corresponding to Fig. 4.10c for $F_{\text{coil}} = 1$.

Figure 4.10: Daily profile plots of system temperatures, electrical power generation and solar irradiance for the “cloudy” day simulations of the “buffered” variant of the EFPC system, under electricity and hot water prioritisation settings.

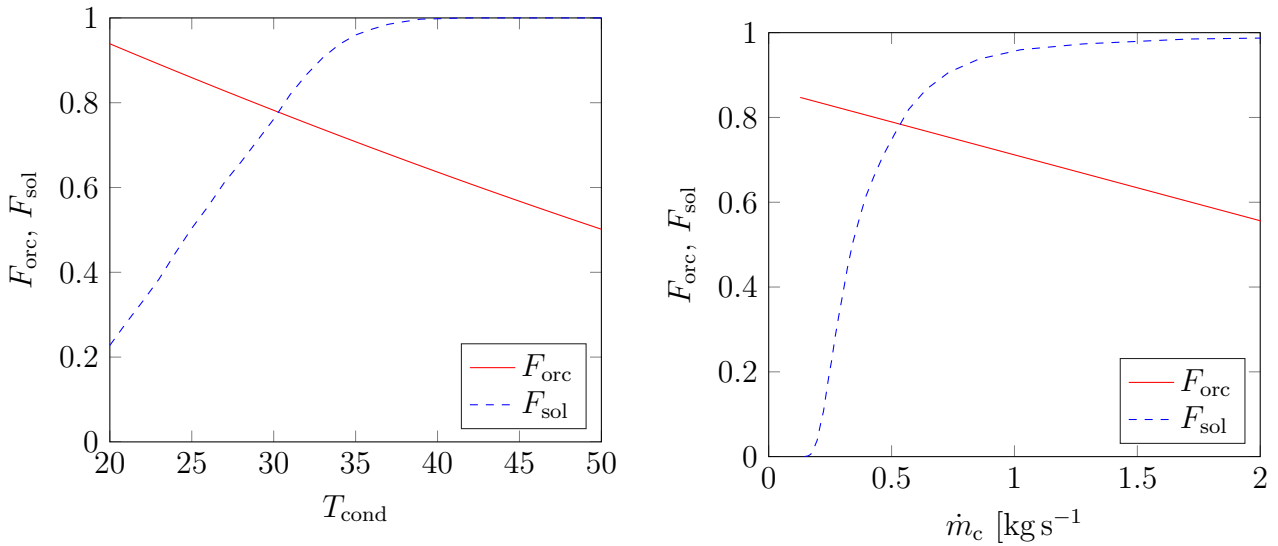
than 15 minutes, at the expense of a reduction in daily electrical output from the system of 8%. This is illustrated in Fig. 4.8c. This variant of the system will be referred to as the “buffered” configuration.

Figure 4.10 shows a comparison of the system operation in “electricity priority” and “hot-water priority” modes, during the intermittently cloudy day. For hot-water prioritisation, supplemental heating is provided to the hot water tank by diverting a portion of the solar collector fluid (represented by the variable F_{coil}) through the second heat exchanger coil located in the top section of the hot water tank. In Fig. 4.10a, F_{coil} is set to zero and as a result the temperature of the water in the top of the tank does not exceed 26 °C. Meanwhile, some water heating is still provided via the heat exchanger in the bottom section of the tank, through which the exhaust vapour from the ORC expander is circulated and precooled. Thus the bottom of the tank undergoes a larger temperature swing than the top of the tank over the diurnal period. The corresponding ORC electrical output is shown in Fig. 4.10b (note that the results shown here are for the “buffered” system described above). The ORC switches on when the heat source temperature reaches the upper set-point value and switches off when the temperature drops below the lower set-point value.

In Fig. 4.10c, F_{coil} is set to 100%, with the full flow of the solar collector fluid being diverted through the upper coil in the hot water tank. As a result, the temperature in the upper part of the tank reaches the maximum of 80 °C in the early afternoon, after which no further heating is provided according to the control strategy in Eq. 3.26. Meanwhile the corresponding total duration of electrical output from the ORC over the daily period shown in Fig. 4.10d is reduced by 27% compared to Fig. 4.10b.

4.3 Air as Heat Rejection Medium

Where a supply of cooling water is not available at low cost, heat rejection to air may be preferable. However, as was shown in Chapter 2, this results in a reduced overall potential for work production due to the larger annual variation in air temperatures compared to mains



(a) F_{orc} and F_{sol} plotted against condensation temperature, for a fixed cold-stream temperature glide of 5 K.

(b) F_{orc} and F_{sol} plotted against cold-stream mass flow-rate for a fixed T_{cond} of 35 °C.

Figure 4.11: The influence of T_{cond} and $\dot{m}_c$ on F_{orc} and F_{sol} for the EFPC collector, air-cooled variant of the S-CHP system.

water temperatures. Maintaining a high instantaneous power output from the system requires setting the condensation temperature as low as possible, placing a constraint on the allowable temperature rise of the cooling medium due to the condenser pinch point. Thus for an air-cooled system a lower condensation temperature requires a higher flow-rate of air and a higher parasitic energy consumption for the fan, and may result in periods where the ambient air temperature is too high for the ORC to operate. On the other hand a higher condensation temperature leads to a reduced specific work output and a larger proportion of the heat supplied to the cycle that is rejected to the cold sink (hence lower gross thermal efficiency of the cycle). In order to calculate the optimal condensation temperature and air flow-rate to maximise the net annual work output from an air-cooled system, two performance parameters are considered. The first parameter is the ORC performance limitation factor:

$$F_{\text{orc}} = \frac{\dot{W}_{\text{net}}}{\dot{W}_*}, \quad (4.1)$$

which is the ratio of the nominal net power output with air as the cooling medium to the nominal net power output under optimal settings with mains water as the cooling medium,

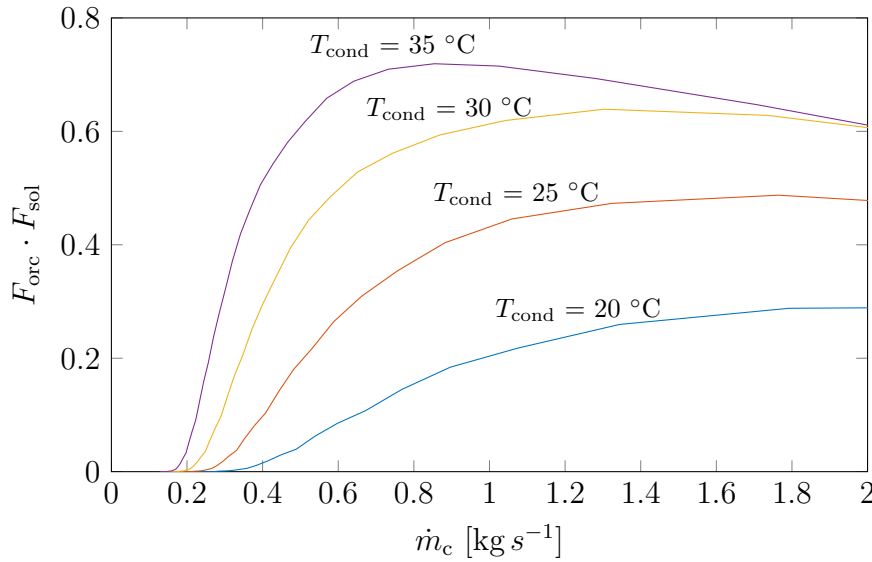


Figure 4.12: The product of $F_{\text{orc}} \cdot F_{\text{sol}}$ plotted against $\dot{m}_c$ for various values of T_{cond} for the EFPC, air-cooled variant of the S-CHP system. Maximising $F_{\text{orc}} \cdot F_{\text{sol}}$ is equivalent to maximising the annual net electrical work output.

$\dot{W}^*$. The second parameter is the solar irradiance limitation factor:

$$F_{\text{sol}} = \frac{\Sigma G (T_a \leq T_{\text{c,in,max}})}{\Sigma G}, \quad (4.2)$$

which is the ratio of the total irradiation during hours when air temperature is low enough for the ORC to operate to total irradiation over the entire annual period. The influence of T_{cond} and $\dot{m}_c$ on F_{orc} and F_{sol} are illustrated in Figs. 4.11 and 4.12. Setting the condensation temperature as low as possible to achieve a high instantaneous power output from the ORC (thereby increasing F_{orc}) places a restriction on the system's ability to make use of solar irradiance when T_a is high, hence F_{sol} decreases sharply for low values of T_{cond} . Meanwhile increasing the mass flow-rate of air in order to achieve a shallow temperature glide raises the maximum air temperature under which the system is able to operate (thereby increasing F_{sol}), while also increasing the fan power consumption and decreasing F_{orc} .

In order to attain the maximum electrical work output from the air-cooled system over the annual period, the product of $F_{\text{orc}} \cdot F_{\text{sol}}$ is maximised by optimising T_{cond} and $\dot{m}_c$ (illustrated in Fig. 4.12). Table 4.2 shows the optimal values for T_{cond} and $\dot{m}_c$, for the system configured with clear-day settings and cloudy-day settings. Because of the dependence of F_{sol} on irradiance

Table 4.2: Optimum settings for the air-cooled configuration of the S-CHP system, for Day profile 1 (clear-sky) and Day profile 2 (cloudy). All other system parameters are set to the maximum power settings from Table 4.1.

	EFPC system		ETC system	
	Clear-day settings	Cloudy-day settings	Clear-day settings	Cloudy-day settings
<i>London UK climate</i>				
$T_{\text{cond}}, ^\circ\text{C}$	37	36	37	33
$\dot{m}_c, \text{kg s}^{-1}$	0.73	0.52	0.68	0.42
$T_{\text{c,in,max}}, ^\circ\text{C}$	25	24	25	21
Hours $T_a \geq T_{\text{c,in,max}}$	72	118	72	395
$\dot{W}_{\text{fan}}, \text{W}$	105	75	98	60
$F_{\text{orc}} \cdot F_{\text{sol}}$	0.73	0.59	0.67	0.45
<i>Larnaca Cyprus climate</i>				
$T_{\text{cond}}, ^\circ\text{C}$	45	44	45	42
$\dot{m}_c, \text{kg s}^{-1}$	0.71	0.44	0.57	0.36
$T_{\text{c,in,max}}, ^\circ\text{C}$	33	31	32	29
Hours $T_a \geq T_{\text{c,in,max}}$	74	244	141	651
$\dot{W}_{\text{fan}}, \text{W}$	105	65	85	52
$F_{\text{orc}} \cdot F_{\text{sol}}$	0.64	0.45	0.56	0.25

and air temperature, Eq. 4.2 requires the use annual climate data for the given location. In Table 4.2, results are reported for the climates of London (UK) and Larnaca (Cyprus).

Taking the example of the EFPC system optimised for the “clear-day” conditions, the optimal value for T_{cond} for the air-cooled configuration is found to be 37°C for the UK; while the optimal cold-stream (air) mass flow-rate is 0.73 kg s^{-1} corresponding to a maximum cold stream inlet temperature of 25°C . Under these settings the instantaneous fan power consumption is equal to 20% of the ORC gross power output, while the temperature glide in the cold stream limits the system operation to periods when ambient air temperatures are $\leq 25^\circ\text{C}$. From the analysis of the UK climate data used for the simulations, this corresponds to 72 hours of the year during which air temperatures are too high for the system to operate. For Cyprus meanwhile, the optimal values of T_{cond} and $\dot{m}_c$ are 45°C and 0.71 kg s^{-1} respectively; and system operation is limited to periods when air temperatures are $\leq 33^\circ\text{C}$ (corresponding to 74 hours of the year for which air temperatures are too high for the system to operate).

4.4 Collector Array Configurations

As noted in Section 4.1, steeper temperature glides for the heat source fluid in the solar collector array require smaller flow-rates, which can be below the minimum flow-rate per collector module recommended by the manufacturer. At flow-rates outside of this range it more likely that the collectors will not perform as their efficiency curve predicts. It was assumed in the parametric analysis that the collectors are arranged in parallel configuration with the total flow-rate split equally between the modules (see Fig. 4.13a). Based on dimensional data for the collectors used in the analysis, a 15 m^2 array corresponds approximately to ~ 8 modules for the EFPC array and ~ 5 modules for the ETC array. For the optimum heat source flow-rates determined in Table 4.1 this results in flow-rates through the individual collector modules that are between 35–85% lower than the minimum stated by the manufacturer.

In order to increase the flow-rate through the individual modules, the array can instead be modelled with the modules arranged in a series configuration, as shown in Fig 4.13b where the total flow passes through each collector one after the other. It is also possible to have an intermediate configuration with two or more parallel branches of collectors arranged in series. In a parallel array, each collector sees the same fluid inlet temperature and is assumed to operate

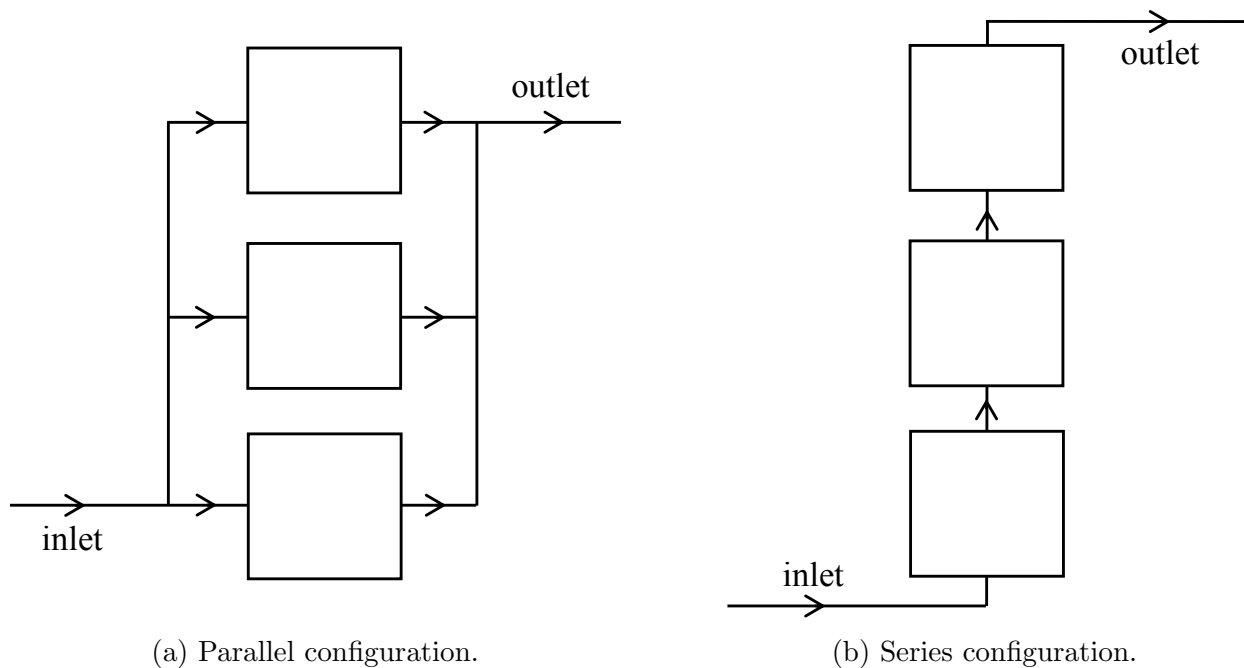


Figure 4.13: Schematic diagrams showing series and parallel array configurations.

Table 4.3: Performance analysis of EFPC and ETC collector arrays in parallel and series configurations, under steady-state conditions of $G = 1000 \text{ W m}^{-2}$ and $T_a = 20 \text{ }^\circ\text{C}$. Flow-rate and inlet temperature set according to optimal settings for clear-sky day in Section 4.1.

	N_{modules} -	A_{array} m^2	$\dot{m}_{\text{array}}$ kg s^{-1}	$\dot{m}_{\text{module}}$ kg s^{-1}	T_{in} $^\circ\text{C}$	T_{out} $^\circ\text{C}$	$T_{\text{out}} - T_{\text{in}}$ K	$\dot{Q}_{\text{array}}$ W	η_{array} %	Δp Pa
<i>EFPC</i>										
Series	8	15.6	0.071	0.071	102	167.7	65.7	8863	56.8	693
Parallel	8	15.6	0.071	0.009	102	168.3	66.3	8946	57.3	87
<i>ETC</i>										
Series	5	14.2	0.118	0.118	95	124.6	29.6	6411	45.3	2091
Parallel	5	14.2	0.118	0.024	95	124.7	29.7	6425	45.4	200

with the same mean temperature and efficiency. In a series array however, each collector will operate at a progressively higher temperature as the fluid is heated by the preceding collector. As a result the final collector in the series will operate with the lowest efficiency and the temperature profile of the fluid will be non-linear with distance between the array inlet and outlet. This will result in different predictions for the mean array temperature (and thus also the total useful energy gain) for a parallel-arranged and series-arranged collector array with the same total flow-rate. An increased pressure drop in the series array is also expected relative to the parallel array as a larger flow-rate of fluid will be passing through each collector, and the flow path is also longer. To assess whether the differences in efficiency and pressure drop between parallel and series configurations are significant in the case of the S-CHP system, a simple test is performed.

A model of a parallel and a series array configuration, each consisting of 8 EFPC collectors, will be compared under steady-state conditions of $G = 1000 \text{ W m}^{-2}$ and $T_a = 20 \text{ }^\circ\text{C}$. For both configurations the array will be subjected to the same total flow-rate of heat transfer fluid, taken as 0.071 kg s^{-1} for the optimal “clear-sky” day conditions, and also the same corresponding inlet temperature for the nominal design condition which is $102 \text{ }^\circ\text{C}$. For the parallel array the total flow-rate will be divided equally between the collector modules in the array, whereas for the series array the total flow-rate will pass through each collector consecutively. The resulting efficiencies, outlet temperatures, and useful heat gain from the collector arrays will be compared in Table 4.3 and Fig. 4.14 for the two configurations. The test will also be performed for an equivalent-sized ETC collector array consisting of 5 modules, a total flow-rate of 0.118 kg s^{-1} and a nominal inlet temperature of $95 \text{ }^\circ\text{C}$ (again corresponding to optimal

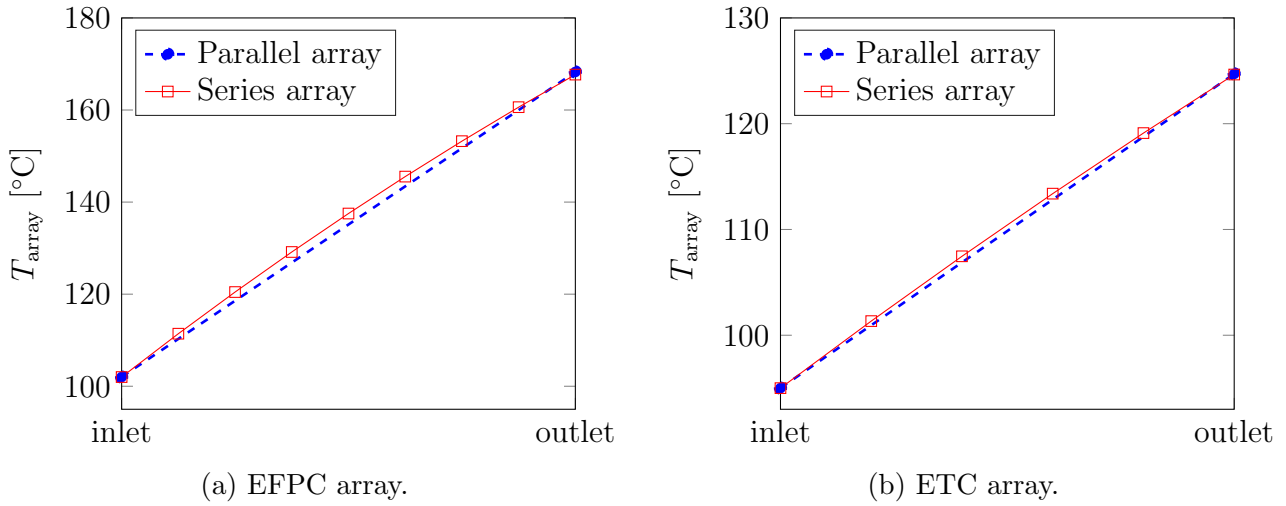


Figure 4.14: Comparison of temperature with length along the flow direction as modelled for the EFPC and ETC arrays in series and parallel configurations.

clear-sky day conditions).

Figure 4.14 shows a very similar temperature profile for the parallel and series arrays. The difference in useful energy output for the series and parallel array configurations is found to be $< 1\%$ for both the EFPC and ETC collector arrays, while the difference in predicted outlet temperatures is < 1 K. This gives reasonable confidence in the simplified assumption of a parallel array for the parametric analysis, despite the minimum flow-rate recommendation of the manufacturer indicating that a series array might be more suitable. It is anticipated that for array sizes considerably larger than 15 m^2 and consisting of more collector modules, the difference in results would become more significant. Meanwhile, the pressure drop through the series array is found to be larger than that through the parallel array by a factor of 8 for the EFPC and by a factor of 10 for the ETC. However due to the design of these collectors for low pressure losses within their intended flow-rate range, the pressure drop in the series array configuration is still low compared to that assumed for the rest of the system (in particular the ORC evaporator heat exchanger, for which the pressure drop is modelled as 50 kPa). Therefore the increase in pumping power for the S-CHP system's solar/heat source pump with a series array configuration instead of a parallel configuration is expected to be $\sim 0.5\text{--}2\%$.

4.5 Installed Cost of System

This section will consider how the S-CHP system performance parameters determined in the parametric analysis influence the overall capital cost of the system. The results of the cost analysis are presented in Table 4.4 and shown as a percentage breakdown in Fig. 4.15.

The total system costs in Table 4.4 have been divided into “electrical sub-system costs” and “hot water sub-system costs” following the approach outlined in Section 3.4.4. The hot-water sub-system cost is most affected by the choice of solar collector, while the electrical sub-system cost is dependent on both the choice of collector and the climatic conditions under which the system is designed to operate. The resulting hot water sub-system cost is between €3,200–3,800 while the ORC sub-system cost is between €5,200–7,000. Among costs associated with the electrical sub-system, 70–80% are attributed to the ORC system components and installation while the remaining 20–30% are additional area-related costs associated with the larger collector array requirement relative to a solar hot water-only system.

The total capital cost is highest when the S-CHP system is designed for maximum instantaneous power output under clear-sky conditions using high-efficiency EFPC collectors. However, based on the diurnal simulations in the parametric analysis the total net work output is significantly higher and hence the specific investment cost is lower. The most significant cost item overall is the solar collector array, which accounts for between 24–31% of the total capital expenditure, depending on the type of collector chosen. The expander is also a significant cost item, accounting for between 13–19% of the total system capital costs and between 35–50% of the purchased equipment cost of the ORC sub-system. The proportional cost breakdown is shown in Fig. 4.15. Apart from the collector and expander, this proportional breakdown is very similar for each of the cases in Table 4.4. Designing the system for nominal operation at a higher irradiance level influences both the expander and ORC pump costs significantly, as higher refrigerant flow-rates lead to larger swept volumes. The cost of the condenser is also influenced due to the larger cooling duty requirement. However, the cost of the evaporator is less affected by the larger heating duty because the increased amount of superheating at the design condition also leads to an increased log-mean temperature difference between the heat

Table 4.4: Results of the system cost analysis

	EFPC system		ETC system	
	Day profile 1 (clear sky)	Day profile 2 (cloudy)	Day profile 1 (clear sky)	Day type 2 (cloudy)
<i>Main parameters affecting cost</i>				
$\dot{W}_{e,net}$ (nominal), W	1149	515	846	297
$\dot{W}_{exp}$, W	1376	615	1006	361
$\dot{W}_{pp,orc}$, W	78.5	19.7	42.5	8.5
$\dot{W}_{pp,h}$, W	10.3	18.4	16.9	18.9
$\dot{V}_{exp,out}$, $\times 10^{-3} \text{ m}^3 \text{ s}^{-1}$	5.60	3.61	4.88	2.79
$A_{evap,hx}$, m^2	0.73	0.71	0.73	0.73
$A_{cond,hx}$, m^2	1.02	0.70	0.92	0.56
M_{wf} , kg	9.7	9.1	9.5	8.9
<i>ORC main component costs, €</i>				
Expander	2098	1385	1840	1093
ORC Pump	382	343	358	336
Evaporator	417	410	417	417
Condenser	507	408	475	364
Generator	520	520	520	520
Liquid receiver tank	110	110	110	110
Working fluid	290	273	285	267
<i>Solar-thermal sub-system costs, €</i>				
Evacuated flat plate collector	3000	3000	2250	2250
Circulation pump	90	95	94	95
Supporting framework	180	180	180	180
Pipework and valves	90	90	90	90
Heat transfer fluid	113	113	113	113
<i>Ancillary costs, €</i>				
Hot water tank	825	825	825	825
Expansion vessel	60	60	60	825
Instrumentation and controls	300	300	300	300
Installation cost	1797	1623	1584	1404
Total investment cost, €	10779	9735	9501	8425
Electrical sub-system cost, €	7008	6046	6208	5220
Hot water sub-system cost, €	3771	3689	3293	3205
SIC (total system), €/W	39.8	116.0	52.7	187.5
SIC (electrical sub-system), €/W	25.9	72.0	34.4	116.1

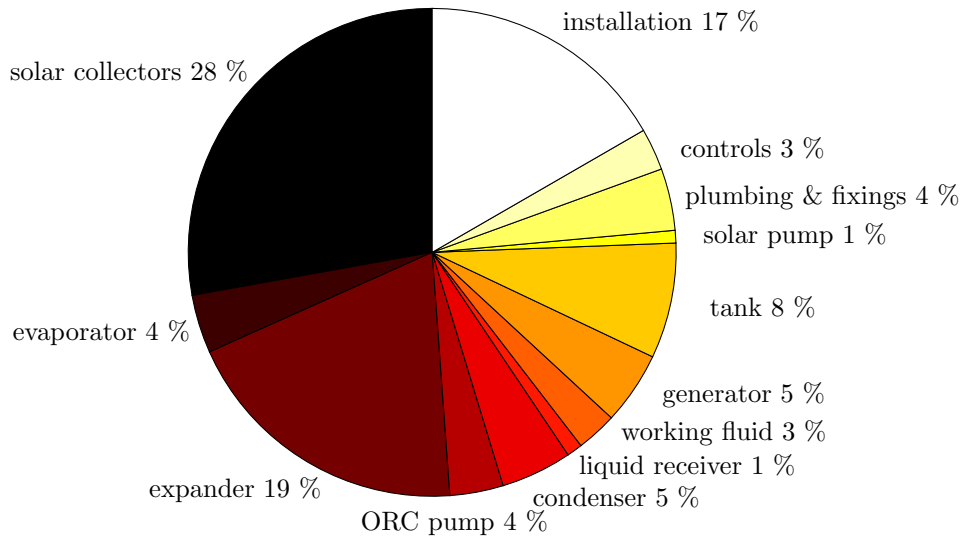


Figure 4.15: S-CHP system capital cost breakdown, based on EFPC system sized for optimal performance under “clear-sky” day conditions.

source and refrigerant fluid streams. Increasing the superheating also leads to an increased temperature gradient and decreased flow-rate in the heat source fluid, and hence a slight decrease in the cost of the solar fluid circulation pump. Apart from the collector and pump, the rest of the solar system costs are area-related and therefore do not change with nominal design condition of the system providing the array area is fixed.

Finally it should be noted that the specific investment cost listed in Table 4.4 is misleading because it only considers the average power output on the “clear-sky” or “cloudy” day for which the system parameters were optimised and thus does not offer a good comparison in terms of economic performance of the system over its operational lifetime in a given climate. In Section 4.6 the specific investment cost will be calculated and compared based on the annual performance of the system, and the levelised cost of energy will also be considered.

4.6 Annual Simulation Results

The annual simulation was performed using the IWECC hourly climate dataset for London published by ASHRAE [120]. For comparison, the system was also simulated using hourly climate data for Cyprus available from the same source. The hourly data was re-sampled in

Table 4.5: Maximum work settings for the annual simulation with UK and Cyprus climate data.

	London, UK climate		Larnaca, Cyprus climate	
	EFPC system	ETC system	EFPC system	ETC system
$\dot{m}_r$, kg s ⁻¹	0.024	0.021	0.028	0.026
$\dot{m}_h$, kg s ⁻¹	0.093	0.128	0.071	0.118
$\dot{m}_w$, kg s ⁻¹	0.461	0.404	0.538	0.450
$T_{\text{evap}}/p_{\text{evap}}$, °C/bar	108/14.9	85/8.9	130	65
$T_{\text{cond}}/p_{\text{cond}}$, °C/bar	20/1.23	20/1.23	20/1.23	20/1.23
ΔT_{sh} , K	21	15	28	18
$r_{v,\text{exp}}$, -	12.8	7.4	23.5	12.1
$\dot{W}_{\text{e,net}}$ (nominal), W	773	513	1103	806

order to run the simulations at the required calculation time-interval of 30 seconds. The annual simulation was performed separately for the S-CHP system configured with the EFPC and ETC collector arrays; with the maximum work settings for the “clear-sky” day and the “cloudy” day and also for a series of intermediate settings. As well as solar irradiance and air temperature, mains cooling water temperature is also modelled as varying over the annual period; applying the same sinusoidal oscillation of ± 2.6 K about a 10 °C mean used for the annual simulations in Chapter 2. This requires the condensation temperature to be set slightly higher than for the maximum work settings in Section 4.1, for which a fixed cooling water temperature of 10 °C was assumed. The condensation temperature is therefore increased from 17 °C to 20 °C.

The annual maximum work settings for the two locations are listed in Table 4.5. The maximum annual electrical work output in the Cyprus simulation are similar to the optimal “clear-sky” settings from Section 4.1; while for the UK the maximum annual electrical work output was achieved with an intermediate setting in-between the “clear-sky” and “cloudy” day settings. With the annual maximum power settings established, the simulation was performed for the electricity prioritisation ($F_{\text{coil}} = 0$) and hot water prioritisation ($F_{\text{coil}} = 1$) cases. The results of the simulations are presented in Table 4.6 and compared for London and Cyprus. Under electricity prioritisation settings, the maximum net electrical work output obtained for London is 628 kWh_e yr⁻¹ with the EFPC array and 360 kWh_e yr⁻¹ with the ETC array; which when averaged over the year is equivalent to 72 W_{e,avg} and 41 W_{e,avg} respectively. This is 10–15% lower than the average power obtained for the “cloudy” day simulation in Section 4.1. The reason for this is partly due to the higher condensation temperature used for the annual

Table 4.6: Results from the annual simulation of the S-CHP system.

	London, UK climate		Larnaca, Cyprus climate	
	EFPC system	ETC system	EFPC system	ETC system
<i>Electricity prioritisation</i>				
Annual net electrical work output, kWh _e yr ⁻¹	628	360	1628	1043
Annual hot water provision, kWh _{th} yr ⁻¹	697	433	1565	998
Average net electrical power output, W _{e,avg}	71.7	41.1	185.8	119.1
ORC operation time, hrs yr ⁻¹	793	690	1460	1290
Annual electricity demand coverage ⁽ⁱ⁾ , %	19.0	10.9	32.7	21.0
Annual hot water demand coverage ⁽ⁱⁱ⁾ , %	27.9	16.8	58.4	30.0
Average solar collector efficiency ⁽ⁱⁱⁱ⁾ , %	24.3	15.5	32.4	23.1
Average ORC efficiency ^(iv) , %	11.5	9.5	13.0	11.3
Annual solar to electrical efficiency, %	3.9	2.2	5.5	3.5
Annual cooling water consumption, m ³ /yr	1318	1003	2829	2320
<i>Hot water prioritisation</i>				
Annual net electrical work output, kWh _e yr ⁻¹	551	295	1527	920
Annual hot water provision, kWh _{th} yr ⁻¹	1618	1317	2504	2221
Average net electrical power output, W _{e,avg}	62.9	33.6	174.4	105
ORC operation time, hrs/yr	694	566	1368	1137
Annual electricity demand coverage ⁽ⁱ⁾ , %	16.7	8.9	30.7	18.5
Annual hot water demand coverage ⁽ⁱⁱ⁾ , %	60.6	49.9	92.3	82.7
Average solar collector efficiency ⁽ⁱⁱⁱ⁾ , %	27.7	17.7	33.4	24.6
Average ORC efficiency ^(iv) , %	11.5	9.5	13.0	11.3
Annual solar to electrical efficiency, %	3.4	1.8	5.2	3.1
Annual cooling water consumption, m ³ /yr	1151	823	2650	2044

⁽ⁱ⁾ Assumes that all electricity generated displaces local demand.

⁽ⁱⁱ⁾ Also considers additional auxiliary heating required to balance heat losses from tank.

⁽ⁱⁱⁱ⁾ Reported value is the mean solar collector efficiency during daylight hours only.

^(iv) Reported value is the mean efficiency during ORC operational hours only.

simulation, and partly due to the characteristics of the annual climate dataset, such that the “cloudy” day irradiance and air temperature profile does not well represent the “annual-average” day. With the EFPC array the electrical output is equivalent to 19% of the annual consumption for a typical UK home, with an additional 28% hot water demand provision through recovery of rejected heat from the ORC. When simulated with Cyprus climate data, the maximum net work output is 1628 kWh_e yr⁻¹ (186 W_{e,avg}) for the EFPC array, which is 2–3 times higher than that achieved in the UK climate simulation, while the hot water demand coverage is 59%.

Under hot-water prioritisation settings, the hot water provision from the S-CHP system with the EFPC array increases by 130% relative to the electricity prioritisation case, while with the ETC array the relative increase is 200%. However annual electricity provision decreases by 12% and 18% respectively. For the EFPC system, the annual heat losses from the hot

water tank are 62 kWh yr^{-1} , which is considerably lower than the $230\text{--}2,900 \text{ kWh yr}^{-1}$ range found in the field trials of real solar hot water systems by the Energy Saving Trust [20], and is partially due to the high level of tank insulation specified for the simulation. However, another partial explanation is that auxiliary water heating is supplied down-stream of the tank in the simulation, which results in an annual mean hot water storage temperature (considered as the temperature at the top of the tank) of 42°C compared to typically expected storage temperatures of up to and above 60°C where auxiliary heating is supplied within the tank [20, 125].

The total number of hours in the year during which conditions are sufficient for the ORC to operate are 793 hours for the EFPC system in London and 1460 hours in Cyprus. This is equivalent to 16% and 29% of the annual daylight hours in the UK and Cyprus respectively. For some of this time the system is operating with higher solar irradiance intensities than the design condition, and this results in an increased collector temperature, a higher degree of superheating at the inlet to the ORC expander and an increased power output. If the system were instead designed with a means of storing the excess thermal energy, the system operational hours could conceivably be extended, with conditions maintained at closer to the design point with lower thermal losses from the collector array, and the further benefit of load shifting capability to better match local electricity demand. Solid-liquid phase change materials (PCM) allow thermal energy to be stored and released at a near uniform temperature that can be matched to the system operational requirements, while also offering high energy storage densities. Hydrated salt and organic PCMs with melting temperatures in the range $80\text{--}120^\circ\text{C}$ typically have volumetric latent heat capacities of $\sim 200 \text{ MJ m}^{-3}$. On the peak solar day in the UK simulation, the EFPC system is found to receive an excess 13.5 MJ of collected solar energy when operating above its design point. The required PCM volume to store this energy would be $\sim 67 \text{ L}$. For Cyprus a smaller store of $\sim 14 \text{ L}$ would be sufficient as the system is sized for a higher nominal output. However these figures are indicative only, and the full optimisation of a system incorporating thermal storage should be considered in detail in order for the solar collector, ORC and thermal store parameters to be optimally matched. The operating temperatures and component sizes will be quite different for system that is designed to provide continuous power

over a longer operating period compared to one that is designed to operate in short bursts when sufficient solar energy is available. Some further considerations for system sizing with PCM thermal storage are presented in Appendix E.

The annual electrical output of the simulated S-CHP system can be compared to the annual endoreversible maximum work determined from the collector analysis in Chapter 2 (Table 2.4). For the UK simulation with the EFPC collector array it is found that the annual electrical work output from the S-CHP system is 38% lower than the endoreversible work prediction; while for the Cyprus simulation it is 25% lower. Although conversion efficiencies predicted by endoreversible analyses are often considered to be realistic targets in terms of what can be achieved by optimally-designed systems, there are some important reasons why the S-CHP achieves a lower than predicted annual work output. Firstly, the endoreversible analysis assumption is that of a Carnot cycle operating between the internal hot and cold temperatures, and therefore the irreversibilities that are internal to the cycle are not considered; namely those associated with the working fluid, pump and expander. For the Rankine cycle, the non-isentropic expansion process and the shape of the fluid saturation envelope result in a superheated vapour exiting the expander and a loss in potential to produce work. In fact, for the S-CHP system some use is made of this remaining exergetic potential by heating water, although this is not considered in the above comparison with the endoreversible analysis. Finally for the S-CHP system, the solar fluid pump, ORC pump and expander all have fixed operating points (flow-rates and compression/expansion ratios). Although this has some advantages in avoiding pump and expander losses due to part-load operation, it also limits the ability of the ORC engine to extract the maximum work from the variable heat source outside of the design conditions; whereas the endoreversible analysis assumed a Novikov efficiency corresponding to maximum power conditions (i.e. optimal collector temperature) at each time-instant.

When the annual simulation is performed with air as the heat rejection medium and using the settings determined from the analysis in Section 4.3, the annual work output is 35–50% lower than that obtained with water as the heat rejection medium. This is approximately in line with what was predicted in the aforementioned analysis, and is a result of a $\sim 25\text{--}35\%$ reduction in gross annual work output due to the increased condensation temperature, and a

Table 4.7: Results from the economic analysis of the S-CHP system.

	London, UK climate		Larnaca, Cyprus climate	
	EFPC system	ETC system	EFPC system	ETC system
<i>Electricity prioritisation</i>				
Total system investment cost, €	9,711	8,467	10,519	9,274
Electrical subs-system investment cost, €	6,027	5,259	6,770	5,999
Specific investment cost, €/W _{e,avg} ⁽ⁱⁱ⁾	84.1	128.0	36.4	50.4
Annual incurred costs (O&M), €	97.1	84.7	105.2	92.7
Annual electricity savings, €	129.0	73.9	334.4	214.2
Annual gas savings, €	44.3	27.5	99.5	63.5
Total annual emissions savings ⁽ⁱ⁾ , kgCO ₂ (e)	425	251	1052	673
Levelised cost of electricity ^(v) , €/kWh	1.00	1.52	0.43	0.60
Levelised cost of water heating, €/kWh	0.59	0.82	0.27	0.37
Payback period ^(vi) , years	49.7	82.3	21.9	29.6
<i>Hot water prioritisation</i>				
Total system investment cost, €	9,711	8,467	10,519	9,274
Electrical subs-system investment cost, €	6,027	5,259	6,770	5,999
Specific investment cost, €/W _{e,avg} ⁽ⁱⁱ⁾	95.8	156.5	38.8	57.1
Annual incurred costs (O&M), €	97.1	84.7	105.2	92.7
Annual electricity savings, €	113.2	60.6	313.6	189.0
Annual gas savings, €	102.9	83.8	159.3	141.3
Total annual emissions savings ⁽ⁱ⁾ , kgCO ₂ (e)	584	407	1,204	874
Levelised cost of electricity ^(v) , €/kWh	1.14	1.86	0.46	0.68
Levelised cost of water heating, €/kWh	0.25	0.27	0.17	0.17
Payback period ^(vi) , years	38.9	50.6	20.1	24.8

⁽ⁱ⁾ Based on a UK-specific carbon intensity for electricity provision from the grid of 0.44548 kgCO₂(e)/kWh given by the Carbon Trust [187].

⁽ⁱⁱ⁾ Costs associated with the electricity-generating related components of the system only (refer to Section 3.4.4).

⁽ⁱⁱⁱ⁾ Reported value is the mean solar collector efficiency during daylight hours only.

^(iv) Reported value is the mean efficiency during ORC operational hours only.

^(v) Assumes that all electricity generated displaces local demand. A discount rate of 8% is chosen for the S-CHP system levelised cost calculations.

^(vi) Fuel savings are not discounted for the payback analysis.

further $\sim 10-20\%$ reduction in the net work output due to the energy consumed by the fan.

In Table 4.7, an economic analysis of the system is presented based on the annual simulation results. A levelised cost analysis is also performed by projecting the annual performance results over an estimated system lifetime of 25 years. Considering the electrical sub-system alone, the best-case specific investment cost is 36 €/W (~ 40 \$/W), and the best-case levelised cost of electricity is 0.42 €/kWh (0.46 \$/kWh), both of which are obtained in the Cyprus climate. This is found to be within the range of 0.25–0.65 \$/kWh for residential c-Si PV systems reported by IRENA [188] (depending on cell type and location of installation). For the UK meanwhile the levelised cost is found to be significantly higher (> 1 euro/kWh). The shortest payback time is obtained for Cyprus, with the system configured for hot water priori-

tisation and by assuming that fuel savings are not discounted, which results in a payback time of 20 years. For the UK analysis the shortest payback time is calculated as ~ 40 years (longer than the assumed system lifetime of 25 years). It is assumed in the analysis that water heating is provided by burning gas in a boiler. However if the comparison is instead made with electric water heating, the best-case payback time decreases significantly to 17 years for the UK and 11 years for Cyprus.

4.7 Comparison with Alternative Technologies

A brief performance and cost comparison of the S-CHP system with three alternative systems for heat and/or power provision from solar energy is presented in Table 4.8. These are (i) a solar CHP system based on hybrid solar photovoltaic-thermal (PVT) collectors, (ii) a solar CHP system based on a side-by-side PV and solar hot water system; and (iii) an electricity-only system based on PV collectors. The common grounds for comparison will be an available roof-space of 15 m^2 that determines the total array size. It is worth noting for this comparison that the capital costs and projected life cycle savings associated with PV in the UK has seen a great deal of change in recent years. In 2011 the typical capital cost of PV (across cell types) according to the Energy Saving Trust was estimated at $3,500 \text{ £/kW}_p$ ($\sim 4,200 \text{ €/kW}_p$). In 2016, this estimated cost has more than halved to around $1,600 \text{ £/kW}_p$ ($\sim 1,900 \text{ €/kW}_p$). Meanwhile, the PV generation Feed-in Tariff (FIT) payment has also decreased, from $43.3 \text{ p kWh}_e^{-1}$ in 2011 to $4.39 \text{ p kWh}_e^{-1}$ in 2016. The net result is that payback times for solar PV systems are estimated to be < 10 years in the UK at present.

Battery storage for PV was not considered as part of this comparison, although a brief market analysis of battery units for the domestic market was undertaken [189–191]. At the time of writing in the UK, the capital cost of small battery units with storage capacities in the range $2\text{--}4 \text{ kWh}_e$ is $\text{€}1,900\text{--}2,600$ for lead-acid technology and $\text{€}2,800\text{--}4,000$ for lithium-ion technology. Thus, at present the addition of battery storage represents a significant capital cost increase of up to 100% for a small domestic PV system.

Table 4.8: UK electrical performance, hot water performance and installed cost of the S-CHP system compared (based on EFPC array) to PVT, PV-only and side-by-side PV and solar hot water systems, each occupying the same 15 m² total array area.

	S-CHP system ⁽ⁱ⁾	PV-only system ⁽ⁱⁱ⁾	Side-by-side system ⁽ⁱⁱⁱ⁾	Hybrid PVT system ^(iv)
Total installed cost, €	9,700	3,500–5,500	8,000–9,000	9,000–11,500
Electrical provision, kWh _e yr ⁻¹	551–628	1,500–2,400	1,100–1,700	1,300–2,400
Hot water provision, kWh _e yr ⁻¹	433–1,618	0	900–1,100	700–1,100
Annual average power, W _{e,avg}	63–72	170–240	130–170	150–270
Total system SIC ^(v) , €/W _{e,avg}	135–155	-	50–70	30–60
Electrical SIC ^(vi) , €/W _{e,avg}	84–96	15–30	15–30	-

⁽ⁱ⁾ Based on the range of annual electricity outputs, hot water outputs and installed system costs listed in Tables 4.6 and 4.7 for the S-CHP system when simulated in the UK climate and configured for electricity and hot water prioritisation with EFPC and ETC collectors.

⁽ⁱⁱ⁾ Based on a PV system cost of €1,600–2,600 (£1,250–2,000 kW_p⁻¹ [16]), and assumes 6–8 m² kW_p⁻¹ for mono-Si PV and 800–950 kWh kW_p⁻¹ yr⁻¹ for the UK [19, 22].

⁽ⁱⁱⁱ⁾ Based on a reduced PV array size of 11 m² to accommodate a solar thermal hot water system array-size of 4 m². Solar hot water purchase and installation cost (including hot water cylinder) and UK annual performance based on typical figures from the Energy Saving Trust [20, 159].

^(iv) Based on a total PVT system capital cost (including hot water cylinder) of €9,000–11,500 (£6,700–8,800) for a 15 m² system, and a range of outputs depending on configurations for maximum electricity or maximum hot water provision (higher electrical outputs corresponding to lower hot-water outputs and vice-versa) [22, 23].

^(v) Defined as the total solar CHP system installed cost divided by the annual average electrical power output.

^(vi) Defined as the component and installation costs of the electricity generating sub-system divided by the annual average electrical power output.

With this in mind, a 15 m² mono-crystalline silicon (mono-Si) PV system (approx. 2–2.5 kW_p) costing around ~€4,500 (without battery storage) can be expected to have an average electrical output of ~200 W_e in the UK climate, giving a specific investment cost (SIC, per unit average power) of €15–30/W_e³. This normalised cost range is around one quarter of the €84–157/W_e range found for the various S-CHP system configurations in Section 4.6 (considering only the installation costs associated with the electricity generating components). Although the PV system has a higher electrical output compared to the S-CHP system for approximately half of the installed cost, the option to provide solar hot-water does not exist because the entirety of the roof space is taken-up by the PV array that does not have a thermal output. The electrical output from the PV-array could be used to provide water heating via an air source heat pump (ASHP). Domestic scale air-to-water heat pumps typically have installed costs ~€9000 (£7000, [192, 193]) which, added to the cost of the PV system brings the total

³By comparison if this analysis were conducted using 2011 price estimates, the SIC for the PV system would be closer to €50/W_e.

installed cost of a PV-ASHP CHP system to $\sim \text{€}13,000$ and the SIC to $\sim \text{€}65/\text{W}_e$.

A like-for-like comparison should be made with various side-by-side configurations for a PV and solar hot-water system sharing the same roof-space (where the electrical and thermal outputs are dependent on how the roof-area is split), or a hybrid PV-Thermal (PVT) system in which the PV and solar-thermal collector components are combined in a single module. In the former case, the electrical output is compromised (due to reduced PV array-size to accommodate the solar thermal collectors) and in the latter the potential for hot-water provision is compromised (due to the collector thermal efficiency as a result of combining with the PV module), while the electrical output will be similar to a PV-only array of equivalent area. The purchase and installation cost of the side-by-side system based on 4 m^2 of solar thermal collectors and the rest of the 15 m^2 roof-space filled with PV would be in the range $\text{€}8,000 - 9,000$, and have an annual average electrical output of $\sim 150 \text{ W}_e$ in the UK climate, giving a SIC of $\sim \text{€}50 - 70/\text{W}_e^4$. This figure should instead be compared to the total (electrical and hot water sub-system) SIC of the S-CHP system which is $\text{€}135 - 252/\text{W}_e$.

For the PVT system the purchase and installation cost is in the region $\text{€}9,000 - 11,500$ ($\text{£}6,700 - 8,800$), with an SIC of $\sim \text{€}30 - 60/\text{W}_e$ based on an output of $150 - 270 \text{ W}_{e,\text{avg}}$ depending on whether the system is configured to provide maximum electricity or hot water output [22, 23]. Although this SIC range is lower than that of the S-CHP system in the UK, it should be noted that the PVT alternative has a significantly reduced capacity for hot water provision, which amounts to 40% of annual demand at best, whereas the S-CHP system can be configured to provide up to 61% of annual hot water demand. In both cases, the upfront cost of the purchase and installation may prove a prohibitive amount for many households. The S-CHP system will arguably have the greater potential for further cost reductions as dedicated small-scale ORC components such as positive displacement pumps and expanders achieve greater market maturity. The S-CHP system and the PVT system also both have the advantage of conceivably being designed as add-ons to be retrofitted to existing solar hot water systems. Finally, the S-CHP system has the unique feature compared to the PVT system and side-by-

⁴By comparison, using 2011 estimates for PV prices the total system SIC for the side-by-side PV and hot water system would be $\text{€}75 - 90/\text{W}_e$.

side PV and solar hot water system that the output ratio of electricity and water heating can be adjusted over time to suit demand, while electricity can also potentially be generated during non-solar hours through the incorporation of thermal energy storage at lower equivalent costs than battery storage for PV⁵.

4.8 Exergy Analysis

The results from the exergy analysis of the S-CHP system are shown in Table 4.9 for the EFPC variant, simulated over the full annual period in the UK in power generation priority mode ($F_{\text{coil}} = 0$). The thermodynamic parameters in the table are calculated according to the exergy topological method outlined in Section 3.3 and used by Mago *et al.* [67]. The proportional breakdown of exergy destruction in each component is shown in Fig. 4.16. As expected, the majority of the exergy is lost in the solar collector array. This is an inevitable consequence of the conversion of solar radiation to enthalpy at a temperature that must be, for practical reasons, far lower than the apparent temperature of the sun as an exergy source.

When the exergy analysis of the EFPC system is compared to that for the ETC system, it is found that the while the total destroyed exergy is only 2% lower for the EFPC system, the total used exergy is 73% higher. The annual exergy efficiency of the collector array in the EFPC system is 10.9% compared to 6.3% for the ETC system. The EFPC array is able to use 73% more exergy than the ETC array, and as a result the available exergy provided to the ORC components is larger. This in turn also results in approximately a 73% increase in the amount of exergy used by the ORC system, even though the exergy efficiencies of the individual ORC components are not significantly different between the EFPC and ETC systems. The analysis confirms the findings of the previous chapter that the more efficient EFPC collector provides a greater potential for work production from the system compared to the lower-cost ETC collector.

The analysis also shows that the evaporator heat exchanger is responsible for more than

⁵Currently best-case cost estimations for PV (lead-acid) battery storage are $\sim 170 \text{ €/kWh}_e$ [24–26], while organic and inorganic PCM storage options show a wide range of in the range $10\text{--}100 \text{ €/kWh}_{th}$ [79, 194].

Table 4.9: Exergy characteristics of the S-CHP system and its components, based on the UK annual simulation of the EFPC system presented in Table 4.6. Here, X_{loss} ($= X_{\text{in}} - X_{\text{out}}$), X_{use} and X_{ava} are as defined in Section 3.3.

Component	X_{in} (MJ)	X_{out} (MJ)	X_{loss} (MJ)	X_{use} (MJ)	X_{ava} (MJ)	ν (%)	β (%)	η_{ex} (%)
Solar collector array	73,077	24,920	48,157	5,906	54,063	34.1	83.9	10.9
Solar/heat source pump	19,039	19,014	25	13	39	99.9	0.1	35.0
ORC pump	133	85	48	70	118	64.0	0.2	59.5
Evaporator HX	11,615	9,918	1,697	4,220	5,917	85.4	9.2	71.3
Expander	4,305	3,564	741	2,688	3,429	82.8	5.3	78.4
Precooler/hot water tank	876	543	334	96	348	61.9	0.5	27.7
Condenser HX	536	128	408	105	513	23.4	0.9	20.5
Total system	109,580	58,172	51,409	13,100	64,427	53.1	-	20.3
Total ORC ⁽ⁱ⁾	36,504	33,252	3,252	7,194	10,364	91.1	16.1	69.4

⁽ⁱ⁾ Not including solar collector array, but including solar/heat source pump.

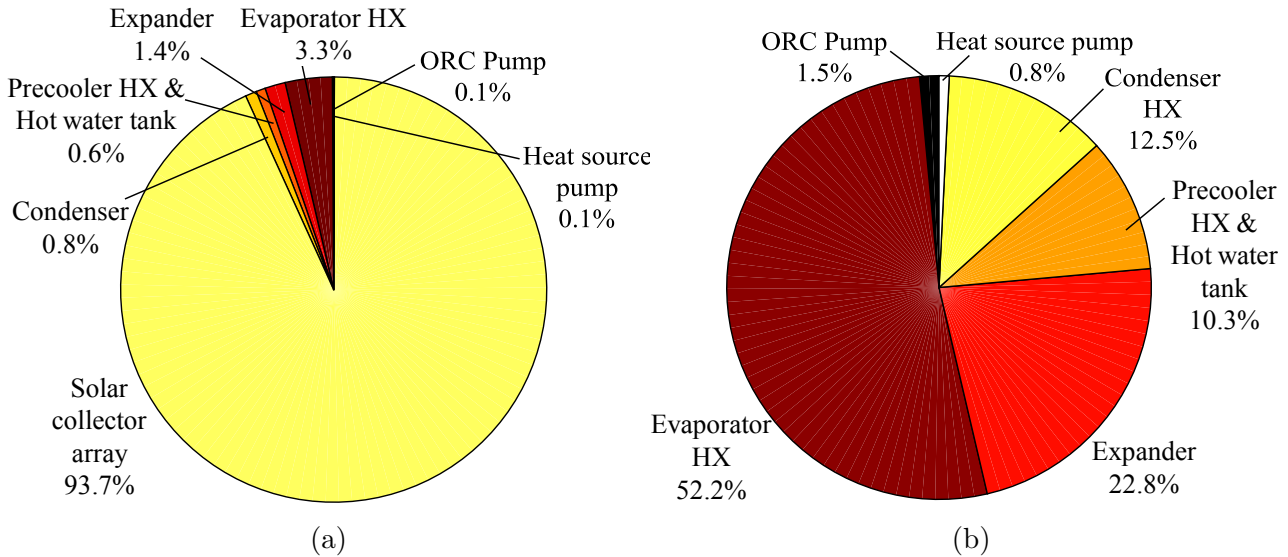


Figure 4.16: Percentage of exergy loss per component in: (a) the entire system including solar collector array, and (b) the ORC system only excluding the solar collector array. Based on the UK annual simulation of the EFPC system presented in Table 4.6 and Table 4.9.

50% of the exergy destruction downstream of the collector array, which represents a significant area of attention for possible performance improvement. This component is less likely to be improved significantly through design modifications but rather through further consideration of the system configuration. A radical design modification that might lead to improvement in this area would be to omit the heat exchanger and heat-transfer fluid loop and to evaporate the ORC working fluid directly in the solar collector array. While this would directly address the exergy losses due to heat transfer between the two fluid streams across a finite temperature difference

(and might also significantly lower the total system cost) it also introduces further challenges for the modelling of the system; specifically regarding the performance of the solar collector with a refrigerant working fluid undergoing flow boiling. Such a deviation from the normal intended use of the collector may significantly affect its efficiency to the point that it is no longer predicted accurately by the standard efficiency curve. While the performance of concentrating PTC solar collectors during direct steam generation is a mature area of research, there are only a few studies that have provided a detailed examination for direct vapour generation using organic fluids in non-concentrating collectors (typically with smaller tube diameters than PTCs) with solar-ORC applications in mind [44, 195–197]. This represents a possible important area for further experimental and modelling research. The expander meanwhile is also shown to contribute more than 20% of exergy losses down-stream of the collector and this is an area already identified as showing strong potential for efficiency improvements in small-scale ORC systems, with a great deal of research already being dedicated to the design and performance analysis of suitable positive-displacement expanders as noted in Chapter 1.

4.9 Summary

In this chapter we have presented the results from a thermo-economic modelling analysis to investigate the potential performance and cost of the S-CHP system, compared for London (UK) and Larnaca (Cyprus). The results of the parametric study (based on simple component efficiency data, load profiles and operational control regimes), have shown that the electrical output from the system is sensitive to the flow-rates, temperatures and pressures in the ORC sub-system and to the design and operation of the solar collector array. The working fluid R245fa was chosen for the present study, although R1233zd(e) was also found to provide a similar level of performance in the range of operating temperatures and pressures, and may prove more suitable in the longer term given the recent amendment to the Montreal Protocol for a global phase out of high global warming potential HFC refrigerants [198].

Annual simulations have shown that for a fixed flow-rate system operation with a 15 m²

solar collector array, the system produces an average power in the region of 41–72 W_e (360–628 kWh_e/yr) in the UK, or 119–186 W_e (1,043–1,628 kWh_e/yr) in Cyprus, for an approximate total capital cost in the region of €8,500–10,500, (£6,500–8,000). The wide range in performance and cost are related to the choice of solar collector between cheaper evacuated tube collectors (designed primarily for domestic hot water applications) or more expensive high-efficiency evacuated flat plate collectors (designed for higher temperature process-heat applications). With approximately $\sim 35\%$ of this capital cost accounting for the basic solar hot water system components, the capital cost attributed to the electricity generating sub-system comprising the ORC and the additional collectors and components to extend the array size is in the region of €5,000–6,800 (£4,000–5,200). For the UK annual results this is equivalent to a total specific investment cost (SIC) per unit average power generation of €135–205/ W_e , or €84–128/ W_e if one considers the costs associated with electricity generation alone, and represents ~ 160 –280 $kgCO_2(e) yr^{-1}$ in emissions reductions due to local power generation. For Cyprus, the SIC is a more promising €56–79/ W_e (total system cost) or €36–50/ W_e (electrical sub-system cost only) due to the larger annual solar resource that results in a higher predicted electrical work output from the system. In addition to the electrical output noted above, the S-CHP system model has demonstrated a potential for producing up to 27% of the required hot water for household consumption under maximum electrical output settings in the UK, corresponding to an additional 130 $kgCO_2(e) yr^{-1}$ in emissions reductions giving a total emissions savings of up to 425 $kgCO_2(e) yr^{-1}$. Under hot water prioritisation settings the hot water demand coverage is up to 61% in the UK, and an increased total annual emissions savings of 584 $kgCO_2(e) yr^{-1}$ is predicted, at the expense of a 13% reduction in electrical work output. For Cyprus the potential for hot water generation in addition to electricity is found to be $\sim 30\%$ higher than for the UK.

When compared to PV-based systems the ORC-based S-CHP system is limited in terms of electrical output, providing less than half of the annual electricity output for an equivalent-sized PV array in the UK. A possible area for further investigation highlighted in the analysis was to use the solar collector as the evaporator, removing the need for a separate heat transfer fluid loop and potentially improving both the cost and efficiency of the system. Further potential

advantages in terms of flexibility have only partly been explored here. As well as the previously noted potential to vary the proportion of electricity and hot water provision to suit demand, the further interesting potential exists to use cheap thermal energy storage to offset electricity generation to coincide with times of peak demand for increased savings, or to operate the system in areas that are not served by an electricity grid. The rejected heat from the ORC may also be useful for a wider range of thermal applications such as low-temperature absorption refrigeration and water desalination. Finally, when compared with the endoreversible analysis from Chapter 2, the S-CHP system is found to achieve around 60% of its annual endoreversible work potential in the UK and 75% in Cyprus. This highlights the difference between the electrical performance potential of the solar collectors and the current capability of the S-CHP system, and should provide motivation for further work to consider improvements through alternative system configurations, working fluids, etc.

In the next chapter a series of experiments will be introduced for the purpose of evaluating the sub-system models of the solar collector array and the ORC engine.

Chapter 5

Steady-state and Dynamic Testing of Solar Collectors

In this chapter, a series of experiments are described to measure the efficiency of a solar collector under steady-state and dynamic conditions, for the purpose of evaluating the solar collector sub-model component of the S-CHP whole-system model. As part of the work for this chapter, a dedicated outdoor testing facility was designed and constructed. The experiments were designed to address two questions, discussed in the preceding chapters, that are specifically related to this work: (1) how well does the solar collector steady-state efficiency curve predict the performance of the solar collector at higher heat collection temperatures that are outside the typical testing range for domestic hot-water applications, and (2) how well does the solar collector model predict performance under time-varying climatic conditions, and can this be improved by modifying the dynamic term that accounts for the thermal inertia of the collector.

5.1 Introduction

The steady-state efficiency curve allows solar collector products to be compared across a range of operating temperatures and solar irradiance conditions. However, the conditions under which the collector is tested are often limited to the narrow range specified in the testing standards and

this can result in performance predictions that are based on extrapolation of the efficiency curve. EN 12975-2 [106] specifies that solar collectors should be tested at four inlet temperatures spaced evenly across the normal range of operation. Most non-concentrating collectors are targeted for water heating applications and therefore are tested at maximum operating temperatures around 80 °C. Furthermore, collectors tend to be tested at a single irradiance level representing “clear-sky” conditions (specified in the standard as $> 700 \text{ W m}^{-2}$). An exception is the EFPC collector investigated in the previous chapter, which is targeted for process heating applications and therefore is tested at temperatures of up to 200 K above ambient and at three irradiance levels between 400–1000 W m^{-2} [117].

The dynamic behaviour of solar thermal collectors has an important bearing on their annual performance that varies across different geographical settings. As was found in Chapter 2, the year-round intermittency of direct solar irradiance due to cloud interruption is a dominant feature of northern European climates such as the UK. Southern European locations such as Cyprus on the other hand experience clear-sky conditions for the majority of the year, and solar hot water systems are widely installed on most homes [182]. Steady-state testing and characterisation of solar collectors according to the standardised method in EN 12975-2 [106] is well established in Europe. As a result, steady-state performance data is published as standard by manufacturers and testing institutions and is freely available for comparison via various online resources such as the Institute for Solar Energy database [110]. The European Standard also includes an experimental and a calculation-based “weighted-component” method for characterising the dynamic performance of solar collectors. The calculation-based procedure is typically preferred in practice due to its greater simplicity although its accuracy has been a matter of critical discussion in subsequent reviews [199]. The experimental procedure on the other hand requires suitable conditions when performed outdoors and can therefore be costly and time-consuming [200]. Detailed 3-dimensional dynamic thermal models of solar collectors such as the one developed by Guarracino *et al.* [201] can be a useful alternative for predicting their performance, providing sufficient information about the geometry and materials are available.

In this chapter, an experimental apparatus for solar collector testing under outdoor con-

ditions will be described. Following this, results will be presented from the steady-state testing of a standard ETC collector at temperatures above its hot water system operational range, for which certified test data was already available. Knowledge of how actual performance differs from predicted performance outside of this range is vital in order to validate the results of the S-CHP system model with the lower-cost ETC collectors. This can also provide a useful indication as to whether such water heating collectors are suitable for higher temperature operation in unconventional applications such as the S-CHP system.

A further series of experiments are described to characterise the thermal response of the ETC collector and obtain a value for the effective thermal capacity. This value will be compared to the value published in the Keymark test report for the same collector, which was obtained by the approximate weighted-component method [128]. The latter value was also used as the collector thermal capacity in the S-CHP model (reported in Table 3.1 as coefficient c_5 for the ETC collector). The experiment is also performed for two other collectors: a flat plate collector (FPC) and a photovoltaic-thermal hybrid collector (PVT). For each collector, a comparison is also made with the thermal-response time constant predicted by the 3-D numerical model of Guarracino *et al.* [201]. Finally, lumped collector models incorporating the experimental and weighted-component effective thermal capacity values are implemented into a simple solar hot water system model and compared to the 3-D collector model by way of a 24-hour simulation using real climate data.

5.2 Apparatus

A schematic diagram of the experimental set-up is shown in Fig. 5.1. The apparatus was located on the roof of the Department of Chemical Engineering, Imperial College London (latitude 51.5° , longitude -0.2°). The arrangement is based on the configuration proposed for liquid-heating solar thermal systems by ASHRAE, (reported in Kredier and Kreith [202]) and EN 12975-2 [106]. The apparatus allows the solar collector efficiency, defined as the useful energy (enthalpy) gained by the fluid divided by the total solar radiation incident on the

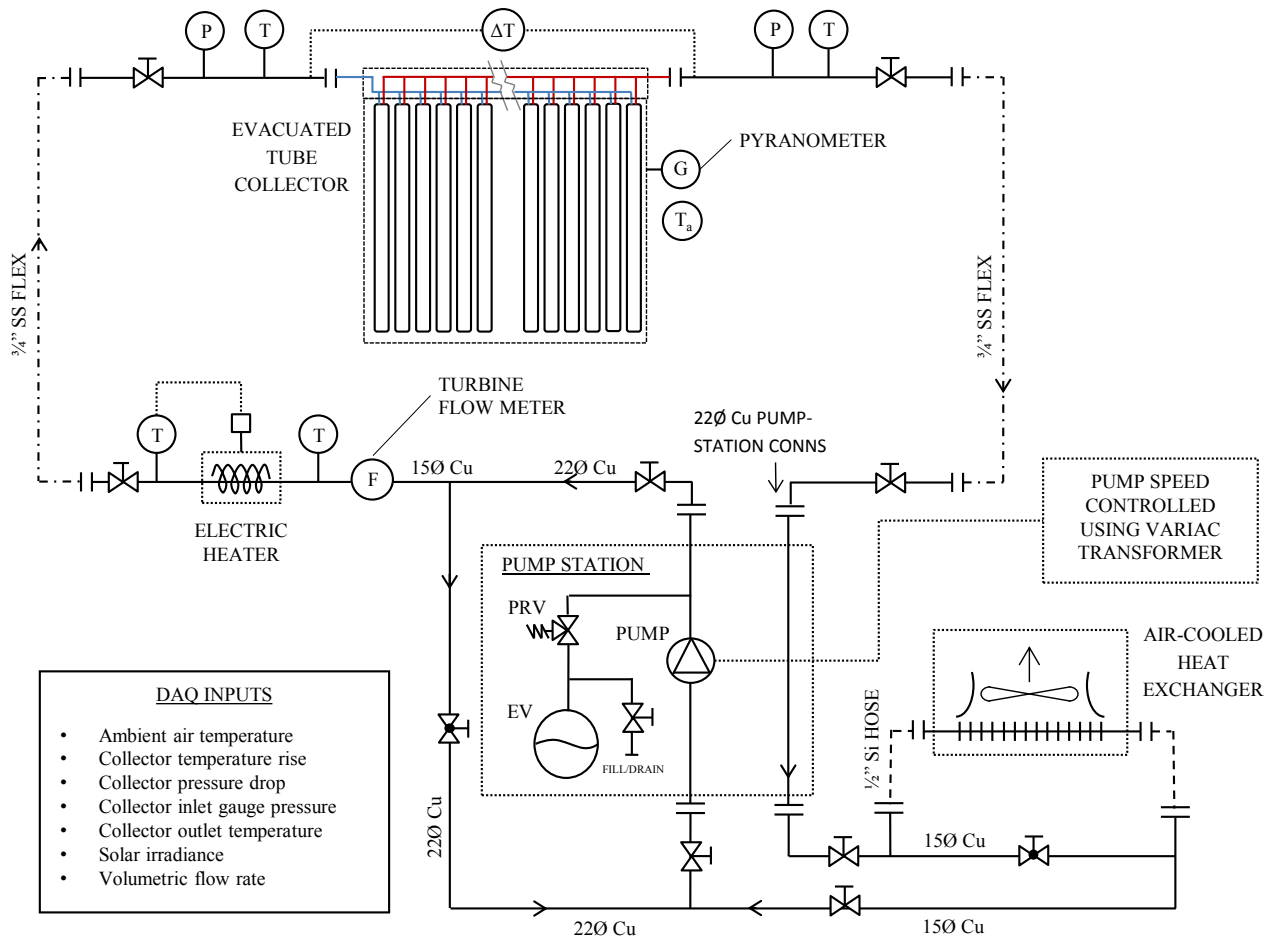
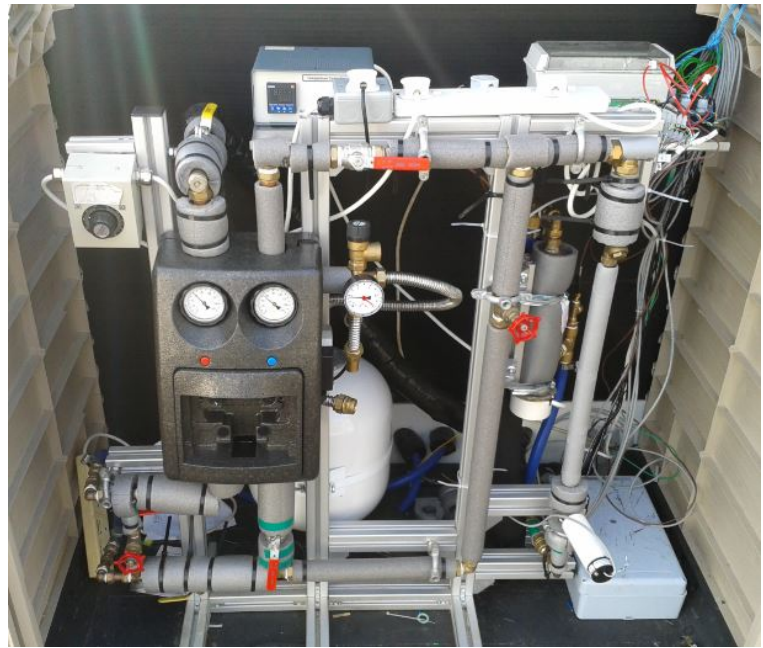


Figure 5.1: Schematic representation of the solar-collector experiment apparatus.

collector surface, to be determined under real sun conditions. In the test procedure, a liquid heat transfer fluid is brought to a desired inlet temperature before passing through the solar collector. The inlet temperature is controlled using an in-line electric heater, controlled using a proportional-integral-derivative (PID) controller. Feedback to the PID controller is provided by a T-type thermocouple measuring the temperature at the outlet of the heater. A turbine flow meter is situated before the inlet of the heater to measure the volumetric flow-rate of the heat transfer fluid. Thermocouples are positioned at the inlet and outlet of the collector to measure fluid inlet and outlet temperatures. Additionally, and for improved accuracy, the fluid temperature rise across the collector is measured using a differential temperature thermocouple arrangement. Upon leaving the collector, the fluid can be cooled to a varying degree using the air-cooled heat exchanger. The heat exchanger acts as the thermal load in the circuit, returning the fluid to the desired steady-state inlet condition. The amount of cooling is controllable by



(a)



(b)

Figure 5.2: (a) Solar collector array with pyranometer and inlet and outlet temperature measurement sections, and (b) enclosure containing pump and instrumentation and for control and measurement of fluid temperature and flow-rate.

adjusting a gate valve positioned on the heat exchanger bypass loop, allowing a portion of the total liquid flow to be diverted through the heat exchanger. The amount of cooling can also be boosted using the fan.

Solar radiation is measured using a pyranometer mounted perpendicular to the inclined plane of the solar collector. Ambient air temperature is measured close to the collector in a location shielded from direct solar radiation. The apparatus is intended primarily for the investigation of evacuated solar collectors for which convective losses to the ambient air are relatively small, and therefore for the initial tests the surrounding air speed was not measured.

Solar thermal collector

The primary component of the test apparatus is the evacuated-tube collector (ETC), shown in Fig. 5.2a. The ETC used for the tests was a 20-tube Thermomax DF-100 model, designed primarily for use in domestic water-heating applications but also investigated for use with the S-CHP system in Chapter 4. The collector is of the direct-flow, coaxial-tube variety. The

standard circulating fluid is a mixture of water and polypropylene-glycol (60% water), which is used to prevent freezing in winter. The fluid is pressurised to approximately 3 bar(a), which raises its boiling point to approximately 130 °C. The collector was installed facing due south at inclination angle of 45°.

Pump

The fluid circulation pump is a Wilo Star ST-25/6 centrifugal pump. The maximum rated operating temperature is 120 °C. The pump speed has three manually switchable speed settings. Finer control of flow-rate is also achieved by adjusting the input power to the pump using an A.C. variable autotransformer (variac).

Air-cooled heat exchanger

The air cooled heat exchanger is shown in Fig. 5.3 and consists of a finned-coil heat exchanger and an axial fan mounted in an enclosure. The fan is controlled manually (by switching on or off). The heat exchanger is also installed with a fluid bypass loop and a gate valve that is adjusted manually in order to vary the cooling delivered to the fluid stream.

Solar radiation sensor (pyranometer)

The solar radiation sensor is a silicon photodiode transducer, which provides an electrical current output proportional to the incident radiation intensity. An integral amplifier converts the transducer current to a 0 to +2.5 V D.C. output signal that is received by the data acquisition device. The spectral response of the transducer is reported by the manufacturer in the range 400–1100 nm. The sensor was calibrated using a UNISIM Solar Simulator and a reference measurement from a series of isotype triple-junction photovoltaic cells with known responses in the relevant parts of the solar spectrum. Further details of the calibration procedure are provided in Appendix F.



Figure 5.3: Experimental apparatus including air-cooled heat exchanger (left), solar collector array (right), and pump station (centre).

Data aquisition unit

A Novus Fieldlogger multi-function data acquisition (DAQ) unit with internal data storage was used to log the measurements. For thermocouple measurements the DAQ's internal cold junction was used to convert the readings into absolute temperature values.

Thermocouples and differential temperature sensor

T-type thermocouples were used to measure fluid temperature at the inlet and outlet of the solar collector. The stated accuracy of the thermocouples is ± 0.5 K. An accurate measurement of fluid temperature rise through the collector is required in order to calculate the useful energy output and the collector efficiency. The systematic (bias) errors determined from the stated accuracies of the inlet and outlet thermocouples and that the data acquisition unit can lead to total uncertainties of 20–30% for the obtained value of the collector efficiency. However the stated accuracies for thermocouples are for absolute temperature measurement, while the bias error associated with measurements of ΔT can be eliminated through simple calibration of the

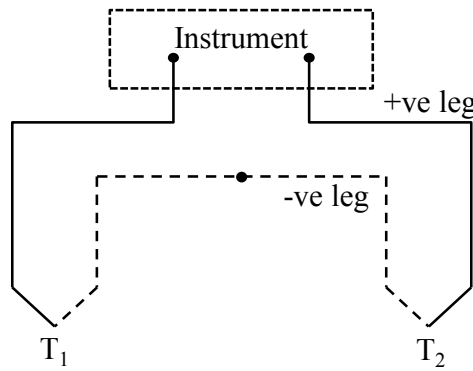


Figure 5.4: Schematic representation of thermocouples configured for differential temperature measurement.

thermocouples. A common method is to determine a correction factor by calibrating against one or more known reference temperatures [203]. Thus a two-point calibration at 0 °C and 100 °C was performed by comparing repeated readings from the thermocouples when immersed in an ice-bath and a flask of boiling deionised water. The absolute temperature readings were not of primary interest, but rather the ΔT between the two thermocouples which was expected to be zero if the fixed errors for both thermocouples were the same. A correction factor was then applied to the readings as a linear function of the mean fluid temperature measured by the two thermocouples. The fixed errors associated with the DAQ and extension cables could also be corrected through this method so that the total uncertainty value associated with the ΔT measurement was dominated by random error rather than systematic error. As a result the total uncertainty associated with the obtained value of the collector efficiency was approximately halved.

An alternative ΔT measurement across the collector was provided by a second pair of thermocouples connected in series to form a “diferential thermocouple”, as illustrated in Fig. 5.4. This simple arrangement is similar to a thermopile, which normally consists of several thermocouples connected in series to provide a more sensitive circuit for differential temperature measurement. The arrangement eliminates the requirement for a reference junction whose temperature must also be measured (usually an internal feature and process performed by the DAQ system). In the case of the differential thermocouple, a connection is made between the two positive (or two negative) legs while the two negative (or two positive) legs are connected to the measuring instrument. A voltage difference is measured that corresponds to a difference

in temperature between each of the positive-negative junctions. The relationship between temperature difference and measured voltage difference changes with absolute temperature, due to the non-linear thermoelectric sensitivity of the thermocouple materials over their temperature measurement range. Thus the reference measurement of absolute temperature in the inlet and outlet fluid streams was used to provide a reference voltage with which to calibrate the reading. A precision amplifier circuit was also designed in order to boost the signal from microvolt to millivolt range which presented benefits for overcoming extraneous signal noise in the outdoor setting (approximately 5 m of extension cable was required between the collector and the waterproof enclosure containing the DAQ).

Turbine flow meter

An Omega FTB-101 turbine flow meter was used to measure volumetric flow-rate in the circulating fluid. A flow meter was required that could operate at temperatures $> 120\text{ }^{\circ}\text{C}$ when the system is operated in full recirculation mode with the cooling load bypassed. The FTB-101 is rated to $232\text{ }^{\circ}\text{C}$. The K-factor provided from the manufacturer's calibration was for water, and therefore a calibration was performed for the glycol-water mixture used in the experiments. The calibration curve is provided in Appendix F.

5.2.1 Second apparatus for Cyprus testing

In addition to the testing of the ETC using the apparatus described above, a second apparatus was also used to perform dynamic performance tests on a flat plate collector (FPC) and a PV-thermal hybrid collector (PVT). The second apparatus, located at the Cyprus University of Technology in Limassol (latitude 34.7° , longitude 33.0°), was set up during winter 2015 with results obtained over a two-week period in early December 2015. Limassol is located on the south coast of Cyprus, approximately 60 km west of Larnaca. The two locations have similar weather conditions over the entire annual period [204], such that the analysis performed using Larnaca climate data in Chapters 2–4 of this thesis can also be considered broadly applicable to Limassol.

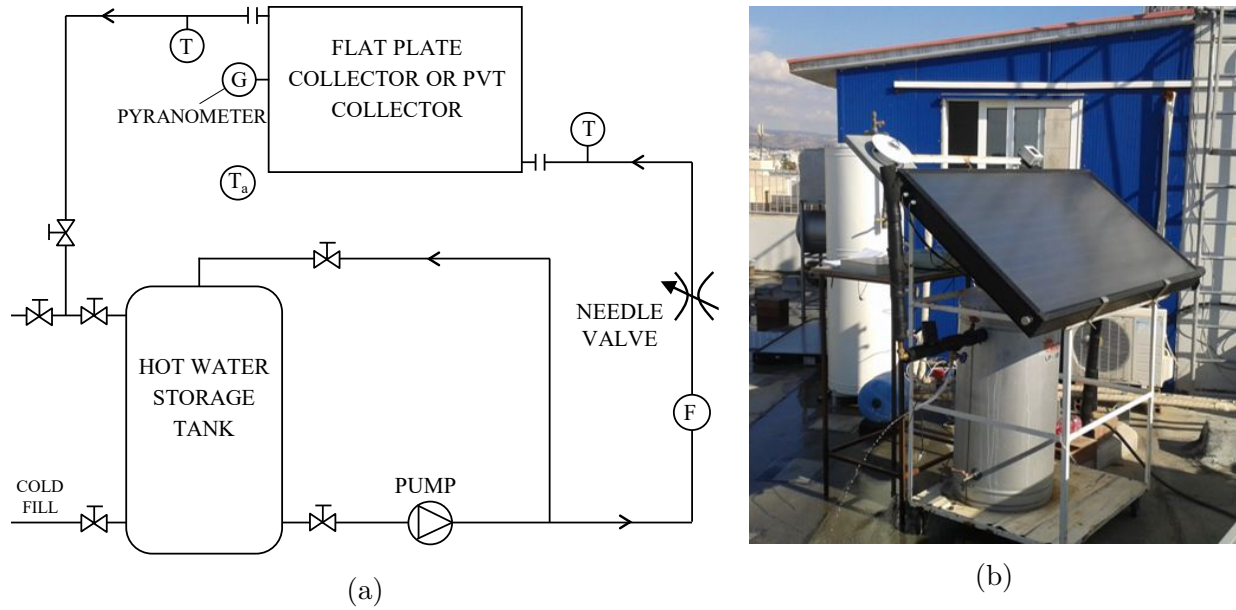


Figure 5.5: Experimental apparatus in Limassol Cyprus: (a) shown schematically, and (b) photographed during PVT collector testing, with hot water storage tank below collector and pyranometer top-left.

The Limassol apparatus is similar to the London apparatus described previously, but with some important differences. The Limassol apparatus is shown in Fig. 5.2.1 and consists of collector, pump and storage tank in a closed-loop configuration. In this configuration, water from the tank is directly heated in the collector and returned to the tank. Alternatively, if constant inlet temperature to the collector is required as in the transient performance test, the system can be configured in “open-loop” to take cold water supplied continuously from the utility mains which is discharged to the drain after passing through the solar collector. Inlet and outlet temperatures to the collector are measured using K-type thermocouples. Fluid flow-rate is measured using a turbine flow meter and can be adjusted by means of a needle valve. Solar irradiance is measured using an Eppley PSP pyranometer, installed in the plane of the collector. The calibration of the various instruments was performed using the same methods described for the UK apparatus. The technical specifications for the FPC, ETC, and PVT collectors (including the key geometrical parameters used in the 3-D numerical model and the steady state efficiency curve coefficients stated in the manufacturer’s data sheet) are listed in Table 5.1.

Table 5.1: Solar collector parameters, taken from manufacturer's data [128, 205, 206].

	FPC	PVT	ETC
Manufacturer/model	Elcora 1.5	Solimpeks Powertherm	Thermomax DF-100
Aperture area, m ²	1.44	1.42	2.15
Mass (dry), kg	30.2	34.4	54.8
Fluid capacity, L	3.1	1.2	3.8
Number of risers/tubes	7	14	20
Riser/tube diameter, mm	15	8	8 (inner), 12 (outer)
η_0 , -	0.67	0.486	0.773
c_1 , m ² K W ⁻¹	4.859	4.028	1.43
c_2 , m ² K ² W ⁻¹	0.012	0.067	0.0059

5.3 Methodology

5.3.1 Steady-state tests

For the steady-state test, data points were obtained over as large a range of collector temperatures as possible. Requirements regarding allowable variations in the steady-state measured parameters were followed according the specifications in EN 12975-2 [106]. The steady-state solar collector efficiency is calculated from the experimental data as follows:

$$\eta_{sc} = \frac{\dot{m}c_{p,f} (T_{sc,out} - T_{sc,in})}{GA_{sc}}. \quad (5.1)$$

The data will be compared to the collector efficiency curve plotted using the coefficients provided by the manufacturer. The convention is to plot the efficiency curve against the reduced temperature difference T_m^* , where:

$$T_m^* = \frac{\bar{T}_{sc} - T_a}{G}. \quad (5.2)$$

Following the procedure in EN 12975-2 [106], a non-linear least squares fit will be applied to the data so that:

$$S = \sum \left[\eta_{sc} - \left(\eta_0 - c_1 \frac{\bar{T}_{sc} - T_a}{G} - c_2 \frac{(\bar{T}_{sc} - T_a)^2}{G} \right) \right]^2. \quad (5.3)$$

where η_0 , c_1 , c_2 are the coefficients obtained from the fitting procedure.

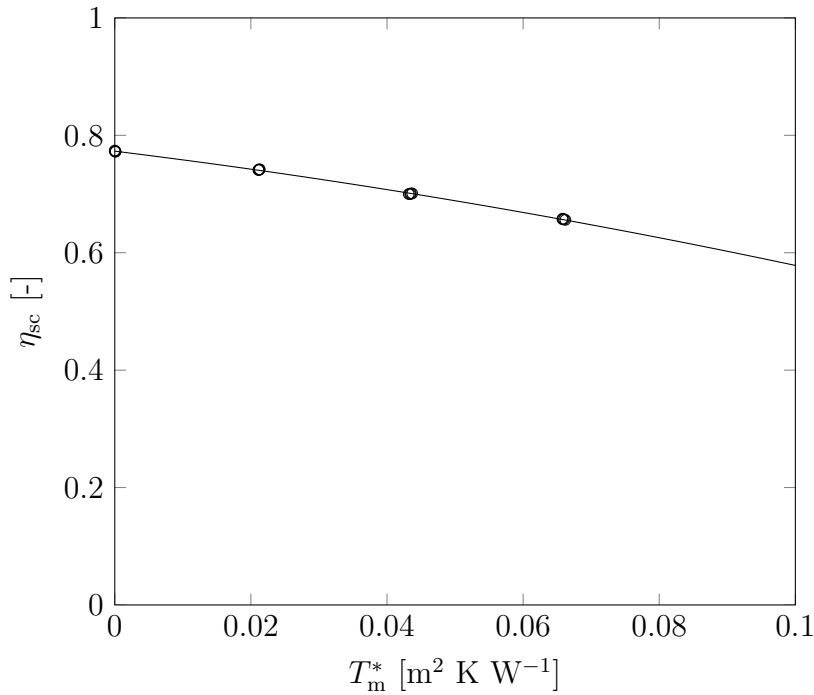


Figure 5.6: Test data from the original test report for the ETC collector, from which the following efficiency curve coefficients were obtained: $\eta_0 = 0.773$, $c_1 = 1.43$, $c_2 = 0.0059$. The efficiency curve corresponding to the mean test irradiance condition $G = 874 \text{ W m}^2$ is also shown passing through the data.

Figure 5.6 shows the original test data from the original ISFH test report for the Thermomax DF-100 collector [128], with the manufacturer's efficiency curve passing through the data. The tests described in the report were conducted indoors using a solar simulator. The conditions were limited to the typical range encountered in water heating applications as specified in the standard, with inlet temperatures in the range $21 - 80 \text{ }^\circ\text{C}$. The other environmental and operational conditions were maintained in a relatively narrow range: $G = 874 \pm 2 \text{ W m}^2$; $\dot{m} = 150 \pm 1 \text{ kg hr}^{-1}$ and $T_a = 26 \pm 1 \text{ }^\circ\text{C}$. In fact, evacuated tube collectors are capable of operating with reasonable efficiency at higher inlet temperatures than this providing irradiance is sufficient, and it was found in Chapter 2 (Table 2.1) that for coaxial direct-flow collectors such as this, mean collector temperatures of $\sim 160 \text{ }^\circ\text{C}$ are optimal for maximum exergy efficiency under peak irradiance conditions of 1000 W m^2 . Therefore in the present study data points were obtained at the highest achievable inlet temperatures, in excess of $100 \text{ }^\circ\text{C}$.

Error analysis

The uncertainty in the solar collector efficiency values obtained from the experiments is determined by taking into account the random errors and systematic errors involved in each of the measurements. Random errors are those observed over repeat measurements, and tend to follow a statistical distribution. Systematic errors are those due to measurement errors such as sensor inaccuracies, and tend to be reproduced consistently over repeat measurements. The uncertainty associated with each directly measured parameter during steady-state operation of the collector over a series of N measurements can be estimated as the standard deviation of the mean:

$$u_{\bar{x}} = \sqrt{\frac{\sum_{i=1}^N (x_i - \bar{x})^2}{N(N-1)}}. \quad (5.4)$$

For the steady-state tests in this chapter, measurements are logged at ten second intervals over a period of at least ten minutes from the moment when the collector reaches steady state. The uncertainty associated with systematic errors for each measurement are calculated from the individual uncertainties for each of the instruments used to obtain the measurement. Typically this will include the sensor and the data acquisition unit, and may include additional sources of error such as signal amplifiers and extension cables. Stated accuracy values for the various sensors (reported in Table 5.2) are converted to standard uncertainty values using the following equation [207]:

$$u = \frac{\alpha}{\sqrt{3}}, \quad (5.5)$$

and the total uncertainty for each measurement resulting from multiple independent sources, u_i is:

$$u = \left(\sum_i u_i^2 \right)^{\frac{1}{2}}. \quad (5.6)$$

The combined standard uncertainty in a calculated result R (for parameters such as the efficiency and the reduced temperature difference) are obtained from the uncertainties of the individual variables (assumed independent) using the following equation [203]:

$$u_R = \left[\sum_{i=1}^N \left(\frac{\partial R}{\partial x_i} u_i \right)^2 \right]^{\frac{1}{2}}. \quad (5.7)$$

Table 5.2: Reported instrument and DAQ accuracies.

Sensor/instrument	Manufacturer/ model	Range	Accuracy	Additional comments
K-type thermocouple	T.C.	-40 – 1000 °C	± 1.0 K	
T-type thermocouple	T.C.	-200 – 350 °C	± 0.5 K	
Pyranometer (UK apparatus)	Davis Instruments 6450	0 – 1800 W m ²	± 5% of full scale	± 1.5 W m ² m ⁻¹ extension cable
Pyranometer (Cyprus apparatus)	Eppley PSP	0 – 2800 W m ²	± 0.5% of full scale	
Data acquisition unit (UK apparatus)	Novus Fieldlogger	0 – 5 V or 0 – 60 mV	± 0.15% full scale	
Data acquisition unit (Cyprus apparatus)	Omega DAQPRO 5300	0 – 10 V or 0 – 50 mV	± 0.5% full scale	

5.3.2 Transient performance characterisation

EN 12975-2 [106] contains two alternative methodologies for obtaining parameters to characterise the transient performance of solar collectors. These parameters are the time constant τ and the effective thermal capacity C . An experimental method is described for the time constant, while the effective thermal capacity may be obtained either by an experimental method or a calculation procedure. Both approaches will be compared in this study.

The time constant is obtained by exposing the collector to a step-change in solar irradiance and recording the change in collector temperature with time between the initial state (State 1) and final state (State 2). The final state is identified when the outlet temperature of the collector is no longer observed to change significantly with time ($< 0.05 \text{ K min}^{-1}$, as required by EN 12975-2), at which point the collector is approximated to be in steady state. The step-change in irradiance is achieved by initially shielding the collector from solar irradiance using a reflective cover. A steady fluid inlet temperature is required for the duration of the tests, approximately equal to the ambient temperature. The time constant τ is the time taken for the transition from the initial state to the instant at which the temperature rise through the collector is as follows:

$$(T_{\text{out}} - T_{\text{a}})_{\tau} = (1 - e^{-1}) [(T_{\text{out}} - T_{\text{a}})_2 - (T_{\text{out}} - T_{\text{a}})_1] . \quad (5.8)$$

The effective thermal capacity parameter meanwhile provides a simplified approach to transient

modelling of a solar collector by a modification of the steady-state efficiency equation to include an additional time-derivative term. In this approach, the useful thermal power extracted by the solar collector at a particular instant in time is approximated by:

$$\dot{m}c_{p,f}(T_{\text{out}} - T_{\text{in}}) = \eta_0 A (K_{\theta,b}G_b + K_{\theta,d}G_d) - c_1 A (\bar{T}_{\text{sc}} - T_a) - c_2 A (\bar{T}_{\text{sc}} - T_a)^2 - C \frac{d\bar{T}_{\text{sc}}}{dt}. \quad (5.9)$$

Note that C is essentially the same as the coefficient c_5 in Eqs. 2.1 and 5.9, apart from that c_5 is the effective thermal capacity per unit aperture area (in $\text{J K}^{-1} \text{m}^{-2}$), while C is for the entire collector module (in J K^{-1}).

An approximation method is presented in EN 12975-2 for estimating the effective thermal capacity that requires only the mass of the individual material components of the collector to be measured. The effective thermal capacity of each component is calculated as the product of its mass, specific heat capacity, and a weighting factor p that describes the degree of thermal interaction it has with the circulating fluid. Thus the metal absorber is designated a high weighting factor while the insulation and external glazing layer are designated low weighting factors. The total effective thermal capacity of the collector is found by summing the values for each element:

$$C_{\text{wt}} = \sum p_n m_n c_n \quad (5.10)$$

This simplified model is known to deviate from 3-D simulations and practical results [199], nevertheless it is a simple expression that is useful in providing approximate estimates of dynamic performance. In the standard, recommended weighting factors are provided only for a generalised example of a glazed FPC which includes the absorber, insulation, glazing and heat transfer fluid, but not the frame or fixing brackets which are assumed to be thermally isolated from the heat transfer fluid. Component weightings applying to other collector designs (for example the manifold tube and casing for an ETC) are not provided in the standard. This weighted-component calculation method is widely used for determining the effective thermal capacity and is the approach used in the official test reports for all three of the collectors compared in this study. However, an alternative experimental method is also described in EN12975-2 that uses the same step-response procedure as for the time constant test. By inte-

grating and rearranging Eq. 5.9:

$$C_{\text{exp}} = \frac{A\eta_0 \sum_{t_1}^{t_2} G dt - \dot{m}c_{p,f} \int_{t_1}^{t_2} \Delta T_f dt - AU \left[\int_{t_1}^{t_2} (T_{\text{in}} - T_a) dt + \frac{1}{2} \int_{t_1}^{t_2} \Delta T_f dt \right]}{\bar{T}_{\text{sc},2} - \bar{T}_{\text{sc},1}}. \quad (5.11)$$

Results from both methods are discussed and compared in Section 5.4.2. The time constant of the thermal response obtained from the experimental method will also be compared to that predicted by the 3-D dynamic thermal model developed by Guarraçino *et al.* [201]. The model is not described in detail in the present work; instead a short summary is given in Appendix G with some of the key equations.

5.3.3 Hot water system modelling

Lumped dynamic models based on Eq. 5.9 incorporating the effective thermal capacities obtained from the experimental and weighted-component methods will be compared to a quasi-steady collector model based on the steady-state efficiency equation (see Eq. 2.2 in Chapter 2), and the 3-D dynamic thermal collector model. For the comparison, the various collector models are implemented into a simple hot water system model, consisting of a pump, collector and storage tank (see Fig. 5.7) based on the closed-loop arrangement of the Limassol testing apparatus. This allows the models to be compared in terms of their predictions of the total thermal energy transferred to the tank over a diurnal period. Two climate days will be compared: a clear day

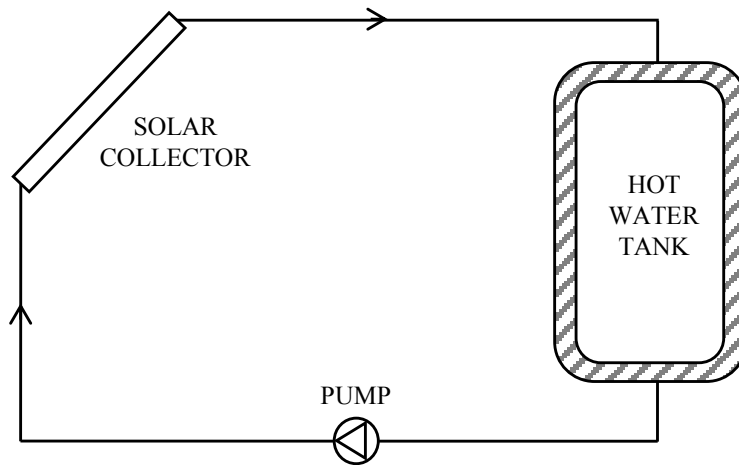


Figure 5.7: Schematic of the solar-thermal hot water system model with hot water storage tank, representing the closed-loop variant of the Limassol testing apparatus.

with an approximately parabolic solar irradiance profile and a cloudy day with short periods of intermittent cloud cover causing a high degree of variability in irradiance intensity. Climate data used in the simulation was collected from the Limassol testing apparatus at a 1-second time resolution. The default system model uses a fully-mixed (lumped) model for the storage tank. This was also compared to an alternative tank model in which thermal stratification in the tank is modelled using a 1-D finite difference approach based on the equations in Section 3.2.4. The tank parameters chosen for the model are based on those for the hot water tank used in the Limassol closed-loop testing apparatus. The UA value of the tank is 1.9 W K^{-1} , the tank volume is 100 L and the fixed flow-rate of the pump is set to 1.68 L min^{-1} . The total energy transferred to the hot water tank over the daily operational period is calculated as:

$$\Delta Q_{\text{tank,net}} = m_{\text{tank}} c_{p,w} (\bar{T}_{\text{tank,final}} - \bar{T}_{\text{tank,initial}}) . \quad (5.12)$$

5.4 Results and Discussion

5.4.1 Steady-state testing

The inlet temperature range over which repeatable steady-state readings were obtained in the present work was $18\text{--}112 \text{ }^{\circ}\text{C}$. Although fluid temperatures exceeding $130 \text{ }^{\circ}\text{C}$ were obtained for short periods at peak irradiance conditions $\sim 1000 \text{ W m}^2$ with the system in full-recirculation mode (heat exchanger bypassed), at such temperatures it was not possible to maintain a steady state for periods long enough to extract a meaningful measurement. Furthermore boiling of the heat transfer fluid was encountered at these temperatures which led to a sharp rise in system pressure and venting of steam through joints (due to failure of the vulcanised-rubber sealing washers at the pipework joints). Thus for improved capability to test the system at higher temperatures approaching the stagnation temperature, improvements are needed to the present set-up to avoid boiling of the heat transfer fluid.

The experimental data shows reasonable agreement with the manufacturers efficiency curve across the full range of test conditions, but deviates further from the manufacturer's curve

Table 5.3: Measured data from the steady-state efficiency tests, with uncertainties reported for each parameter.

G W m^{-2}	$\dot{V}$ L min^{-1}	T_{in} $^{\circ}\text{C}$	T_{out} $^{\circ}\text{C}$	T_{a} $^{\circ}\text{C}$	ΔT_{f} K
850 ± 10	2.6 ± 0.2	18.7 ± 0.8	28.4 ± 0.8	20.8 ± 0.8	9.5 ± 0.1
850 ± 10	2.6 ± 0.2	18.7 ± 0.8	28.2 ± 0.8	20.9 ± 0.8	9.3 ± 0.1
860 ± 10	2.5 ± 0.2	18.7 ± 0.8	28.2 ± 0.8	20.7 ± 0.8	9.3 ± 0.1
860 ± 10	2.6 ± 0.2	18.8 ± 0.8	28.0 ± 0.8	21.0 ± 0.8	9.0 ± 0.1
860 ± 10	2.6 ± 0.2	19.8 ± 0.8	28.8 ± 0.8	21.0 ± 0.8	8.7 ± 0.1
860 ± 10	3.4 ± 0.2	51.4 ± 0.8	57.9 ± 0.8	22.7 ± 0.8	6.4 ± 0.2
860 ± 10	3.6 ± 0.2	55.6 ± 0.8	61.9 ± 0.8	23.1 ± 0.8	6.3 ± 0.2
870 ± 10	3.6 ± 0.2	56.1 ± 0.8	62.3 ± 0.8	23.3 ± 0.8	6.2 ± 0.2
870 ± 10	3.6 ± 0.2	56.2 ± 0.8	62.4 ± 0.8	23.4 ± 0.8	6.1 ± 0.2
880 ± 10	3.6 ± 0.2	65.5 ± 0.8	71.4 ± 0.8	21.8 ± 0.8	6.0 ± 0.2
880 ± 10	3.1 ± 0.2	68.3 ± 0.8	75.2 ± 0.8	22.5 ± 0.8	7.0 ± 0.2
880 ± 10	3.2 ± 0.2	68.3 ± 0.8	75.2 ± 0.8	22.5 ± 0.8	7.0 ± 0.1
880 ± 10	3.1 ± 0.2	68.3 ± 0.8	75.2 ± 0.8	22.5 ± 0.8	6.9 ± 0.2
880 ± 10	3.8 ± 0.2	74.4 ± 0.8	79.6 ± 0.8	22.8 ± 0.8	5.3 ± 0.1
880 ± 10	3.8 ± 0.2	74.4 ± 0.8	79.8 ± 0.8	23.1 ± 0.8	5.5 ± 0.1
880 ± 10	3.8 ± 0.2	74.5 ± 0.8	79.5 ± 0.8	22.6 ± 0.8	5.1 ± 0.1
880 ± 10	3.8 ± 0.2	74.6 ± 0.8	80.0 ± 0.8	23.4 ± 0.8	5.4 ± 0.1
890 ± 10	3.1 ± 0.2	84.7 ± 0.8	90.9 ± 0.8	24.8 ± 0.8	6.3 ± 0.2
890 ± 10	3.4 ± 0.2	106.0 ± 0.8	111.1 ± 0.8	18.1 ± 0.8	5.4 ± 0.2
890 ± 10	3.9 ± 0.2	111.0 ± 0.8	115.2 ± 0.8	24.2 ± 0.8	4.5 ± 0.2
890 ± 10	3.8 ± 0.2	111.5 ± 0.8	115.8 ± 0.8	24.4 ± 0.8	4.7 ± 0.3
890 ± 10	3.7 ± 0.2	112.2 ± 0.8	116.9 ± 0.8	24.7 ± 0.8	5.1 ± 0.2
890 ± 10	3.7 ± 0.2	112.4 ± 0.8	117.0 ± 0.8	24.8 ± 0.8	5.0 ± 0.2
890 ± 10	3.8 ± 0.2	112.5 ± 0.8	117.2 ± 0.8	24.9 ± 0.8	5.0 ± 0.2

at higher values of T_{m}^* . Comparing Fig. 5.8 to Fig. 5.6, the results from this study are extended to values of $T_{\text{m}}^* = 1.04$ which is considerably higher than the ISFH test report and corresponds to collector inlet temperatures up to 112°C . At values of $T_{\text{in}} < 85^{\circ}\text{C}$ the obtained efficiency values are on average within 4% of the values predicted by the manufacturer's curve, whereas at values of $T_{\text{in}} > 110^{\circ}\text{C}$ the obtained values are on average 8.5% higher than predicted; however the relative uncertainty in the calculated efficiency value is $\pm 6\%$. The fitted parameters for the efficiency curve from this study are $\eta_0 = 0.786$; $c_1 = 1.27$ and $c_2 = 0.0037$; compared to the manufacturers values $\eta_0 = 0.773$, $c_1 = 1.43$, $c_2 = 0.0059$. The most notable difference is the second-order loss coefficient which is found to be smaller in this study than the manufacturer's value and reflects a less significant degradation in efficiency at higher collector temperatures.

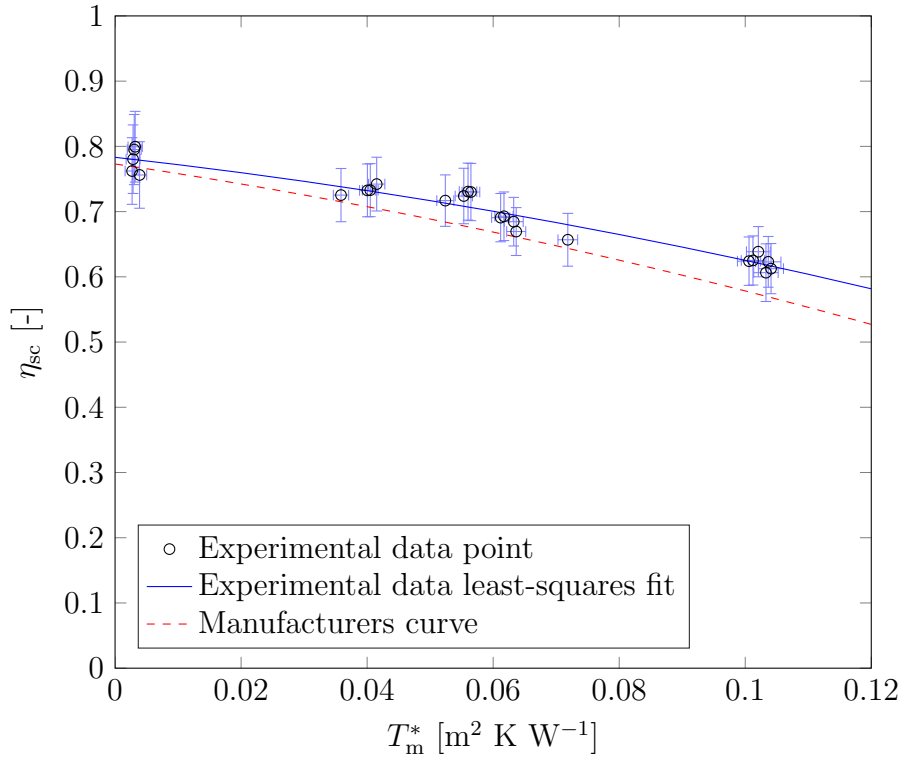


Figure 5.8: Data from the steady-state collector efficiency tests (reported in Table 5.3) compared to the manufacturer's efficiency curve.

5.4.2 Time constant and effective thermal capacity

The experimental conditions for the step response tests are presented in Table 5.4. The same values are used as inputs to the numerical model, and the experimental and numerically obtained values of τ for each collector are compared in Table 5.5. Experimental values of C are also presented in the table, and compared to those provided in the official test reports cited in the manufacturer's documentation, which in all cases are obtained using the non-experimental weighting-factor method discussed in Section 5.3.2. Figures 5.9 and 5.10 show the thermal response plotted against time for the ETC and PVT collectors, respectively. Visual comparisons of the experimental data and the results of the 3-D model are provided in Fig. 5.9b (for the ETC) and Fig. 5.10b (for the PVT collector). Also shown are the predictions of the single-node models using C_{wt} and C_{exp} as the lumped thermal capacity for the collector.

For the FPC, the effective heat capacity value calculated from the experimental method was found to agree closely with the manufacturers value obtained using the weighted-component method. However for the ETC and PVT collectors there was a large difference in the values

Table 5.4: Experimental conditions for the transient response tests; the deviation from the mean measured value of each parameter over the period of the test is given to a 95% confidence interval

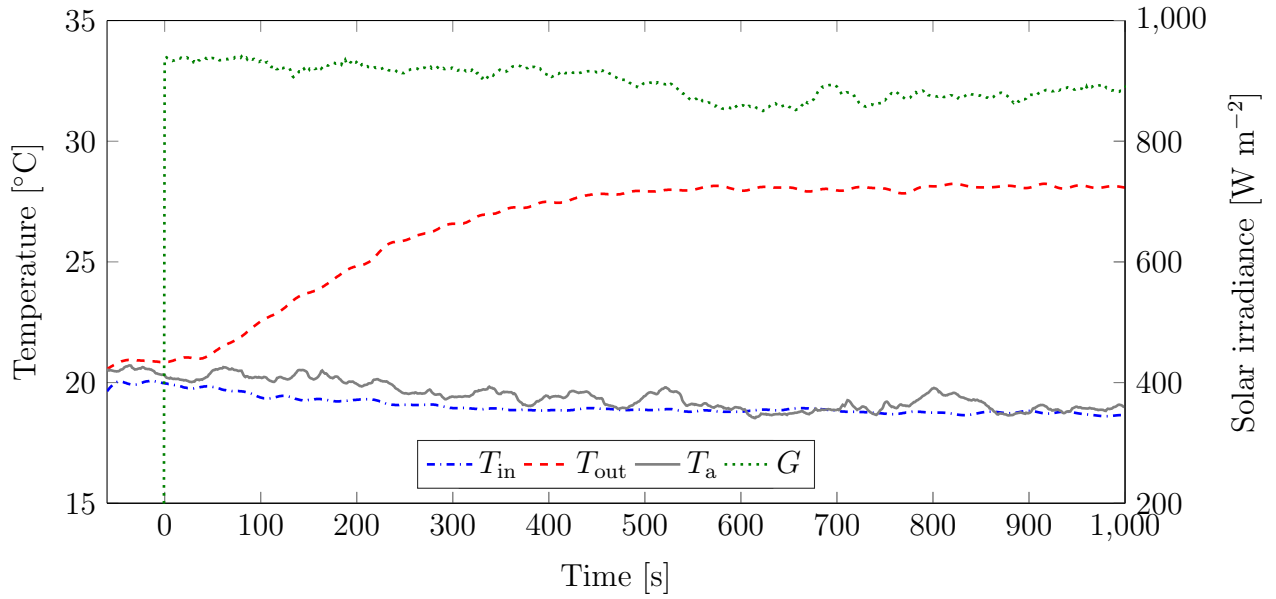
	Collector	Location	$\dot{V}$, L min ⁻¹	G , W m ⁻²	T_a , °C	T_{in} , °C
Test #1	FPC	Limassol	1.7 ± 0.1	950 ± 20	20.5 ± 0.6	21.9 ± 0.2
Test #2	FPC	Limassol	1.7 ± 0.1	950 ± 30	20.2 ± 0.6	20.9 ± 0.2
Test #3	PVT	Limassol	1.7 ± 0.1	930 ± 20	22.6 ± 0.6	25.0 ± 0.2
Test #4	PVT	Limassol	1.4 ± 0.1	910 ± 20	22.1 ± 0.4	26.2 ± 0.4
Test #5	PVT	Limassol	1.7 ± 0.1	930 ± 40	18.3 ± 0.4	20.0 ± 0.4
Test #6	ETC	London	2.2 ± 0.1	840 ± 10	16.3 ± 0.8	15.1 ± 0.6
Test #7	ETC	London	2.3 ± 0.1	900 ± 20	17.6 ± 0.8	16.6 ± 1.0
Test #8	ETC	London	2.5 ± 0.1	950 ± 90	20.0 ± 0.7	19.8 ± 1.2
Test #9	ETC	London	2.5 ± 0.1	910 ± 50	19.6 ± 1.0	19.1 ± 0.6

Table 5.5: Transient performance tests results; the time constant is compared to the value obtained using the 3-D dynamic numerical model, while the effective thermal capacity is compared to the manufacturers value obtained using the weighted-component method

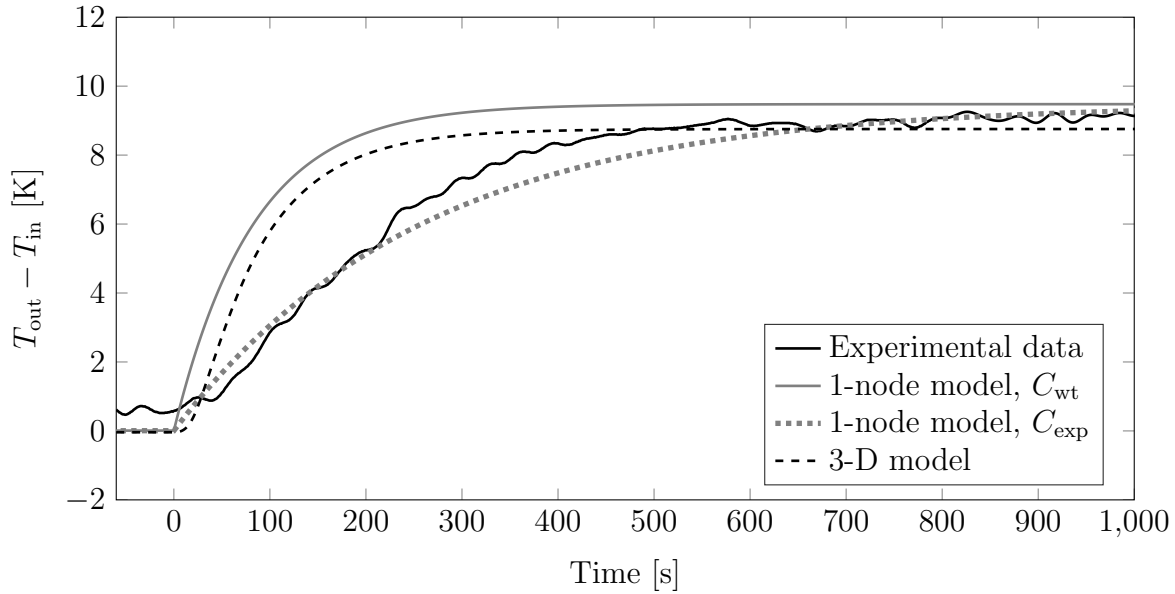
	Collector	Manufacturer value	Experimental results ⁽ⁱ⁾		Numerical model
		C_{wt} , kJ K ⁻¹	τ , s	C_{exp} , kJ K ⁻¹	τ , s
Test #1	FPC	14.9	115	13.9	75
Test #2	FPC	14.9	98	15.8	79
Test #3	PVT	20.3	545	292.9	140 (272)
Test #4	PVT	20.3	605	195.6	191 (332)
Test #5	PVT	20.3	516	165.7	179 (269)
Test #6	ETC	20.1	283	77.8	117
Test #7	ETC	20.1	292	65.7	107
Test #8	ETC	20.1	189	68.7	96
Test #9	ETC	20.1	204	82.2	82

⁽ⁱ⁾ Value reported in brackets is the PVT model modified to include the 2 mm air gap between PV module and absorber.

obtained from the two methods. For the ETC collector the experimental value was 3–4 times larger than the weighted-component value, and for the PVT the experimental value was 8–15 times larger. Meanwhile for all cases the 3-D numerical model was found to under-predict the time-constant of the thermal response. Again, this difference was most significant for the ETC (50–65% under-predicted) and the PVT (65–75% under-predicted, in the first iteration of the model). Thus the real behaviour of the collectors appears to deviate from the predicted behaviour most noticeably for the collectors with complex geometries and a larger number of components in thermal interaction; whereas for the relatively simple FPC the behaviour was reasonably well predicted.

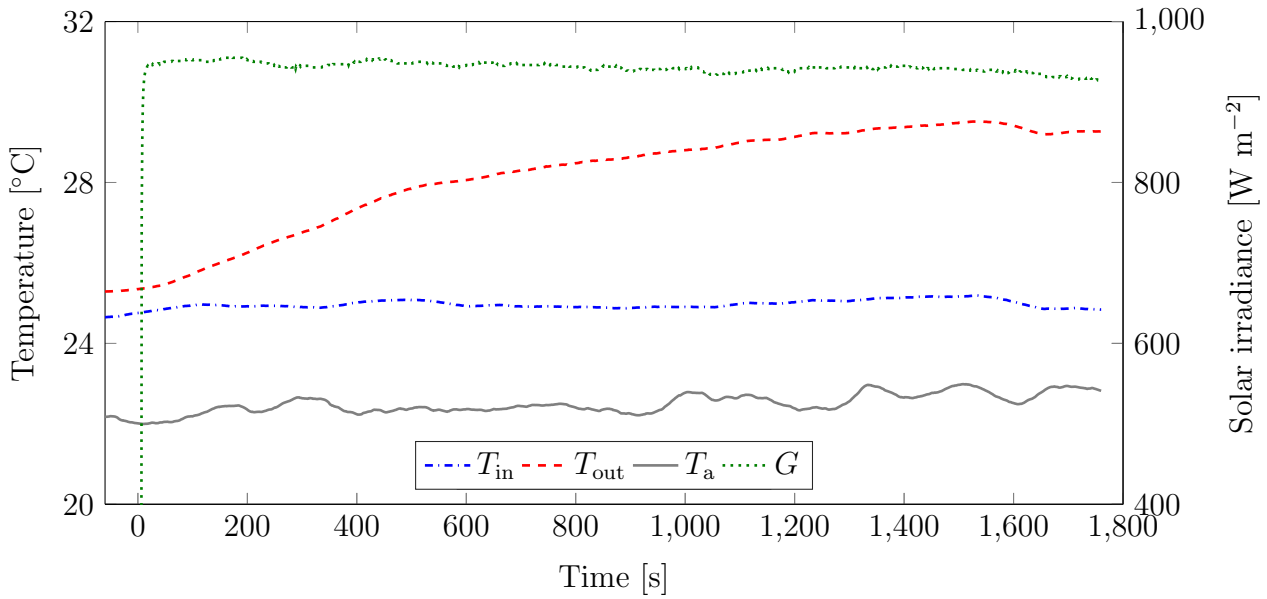


(a) Collector inlet and outlet temperatures, ambient temperature and solar irradiance (see Table 2 for test conditions).

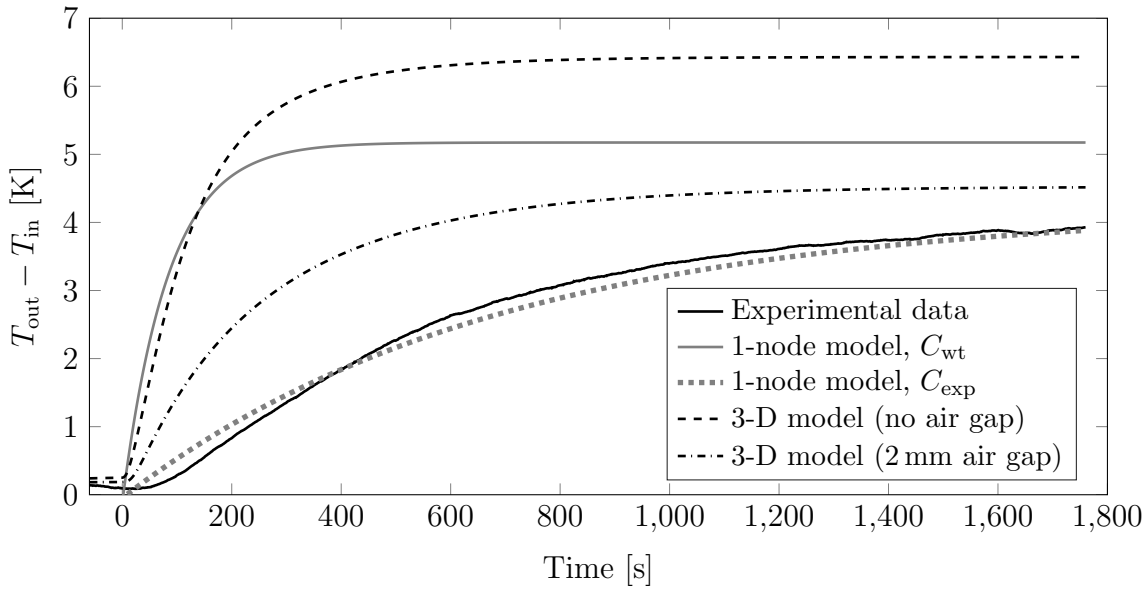


(b) Comparison of the experimental results with the predictions of the various collector models, showing collector temperature rise ($T_{\text{out}} - T_{\text{in}}$).

Figure 5.9: Step response of the ETC collector, Test #9.



(a) Collector inlet and outlet temperatures, ambient temperature and solar irradiance (see Table 2 for test conditions).



(b) Comparison of the various collector model predictions with experimental results, showing collector temperature rise ($T_{out} - T_{in}$). Numerical results are presented for the 3-D dynamic model with and without a 2 mm air gap between the PV layer and copper absorber.

Figure 5.10: Step response of the PVT collector, Test #3.

Table 5.6: ETC component masses (determined by weighing), specific heat capacities and weighting factors (according to EN 12975-2)

Component	Material	M , kg	c_p , kJ kg ⁻¹ K ⁻¹	p , - [106]
Absorber plate/tubing	copper	12.7	0.4	1
External glazing	borosilicate glass	26.5	0.8	0.014
Insulation	foamed melamine resin	0.06	1.4	0.5
Heat transfer fluid	water/glycol mixture	4.3	3.6	1
Manifold pipe	copper	2.1	0.4	-
Manifold casing	aluminium	6.0	0.9	-
A-frame/brackets	aluminium	18.6	0.9	-
Instrumentation sections	brass/copper	5.2	0.4	-

In the case of the PVT collector it was also found that the 3-D numerical model under-predicted the steady-state efficiency of the collector across the range of operating temperatures. It was found upon physical inspection of the collector that there was poor thermal contact between the PV module and the copper sheets that form the thermal absorber plate, due to insufficient application of thermal grease between the layers. This situation was emulated in the 3-D model by including a small air gap resistance (equivalent to 2 mm thickness) between the PV module and the absorber plate. This improved the prediction of the time-constant to within 45–50% of the experimental value for the second iteration of the 3-D model, and an improved prediction of the final steady-state outlet temperature delivered by the collector (see Fig. 5.10b).

5.4.3 Factors influencing the effective thermal capacity

It was shown in Section 5.4.2 that the effective thermal capacity values determined from the experimental data for the ETC and PVT collectors are larger than those obtained from the weighted component calculation method in the official test reports for each collector. In the latter method, only the thermal capacity of certain components are included in the calculation, based on their estimated degree of thermal interaction with the heat transfer fluid. In Table 5.6, the full list of material components making up the ETC and test assembly are shown. The mass of each component was measured and the fluid capacity of the collector tubes and manifold piping were also measured. The recommended weighting factors for the components included

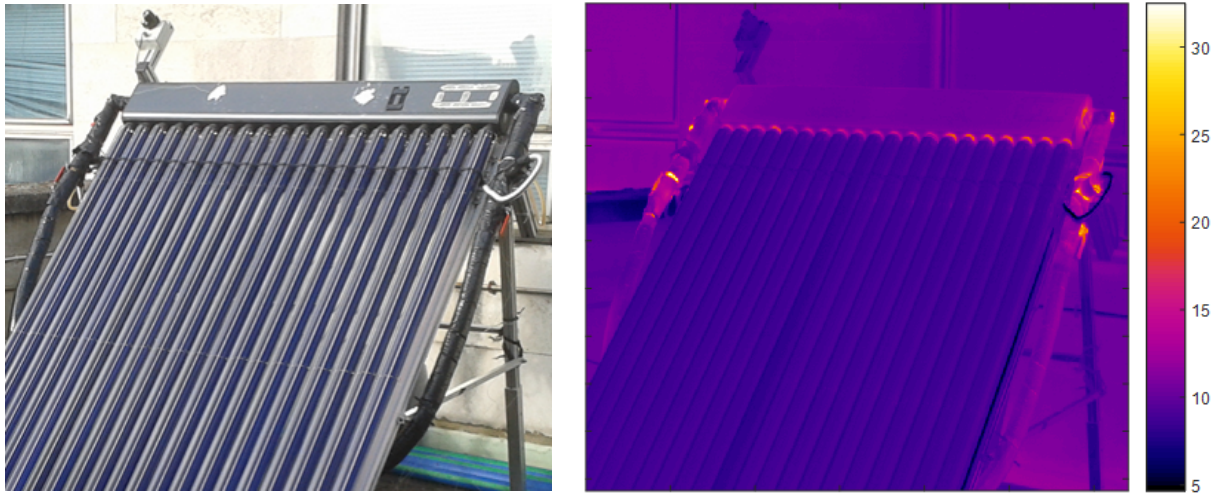


Figure 5.11: Infrared image of the ETC (right) in steady-state operation with $T_{\text{in}} = 50\text{ }^{\circ}\text{C}$, and $G < 100\text{ W m}^{-2}$.

in the effective heat capacity calculation according to EN 12975-2 are included where available.

Components such as the frame and the manifold casing are assumed to be thermally isolated from the collector fluid, but thermal bridging due to poor installation practices can render this assumption invalid. The glazing is also assumed to have minimal thermal interaction; however a loss of vacuum in one or more tubes will result in convective heat transfer between the copper absorber and the glass. In order to investigate the validity of the assumption that thermal interaction between the fluid and the external/structural elements of the collector (in particular the frame, glazing and manifold casing) is negligible, a series of infra-red images were taken as the circulating fluid is raised to an elevated temperature and then maintained at steady state. Meanwhile, temperatures were measured with thermocouples located at various points on the frame, glazing and manifold casing. The image in Fig. 5.11 shows that the glazing and manifold casing are reasonably well thermally-isolated from the fluid, and that the vacuum in the tubes are not compromised. Some hotter regions are identified at the points of connection between the tubes and the manifold, and in the instrumentation sections connected at the inlet and outlet (comprising of brass fittings including bleeding and isolation valves, and ports for temperature and pressure sensors). However thermal bridging to more significant structural elements was not found to occur, and in particular the frame was also found to have little or no thermal interaction with the fluid.

Table 5.7: Solar hot water system simulation results for cloudy day and clear-sky day showing the net energy gain to the tank over the simulation period; experimental results are shown for the PVT system obtained under the same climate conditions as used in the simulations. Also shown for the PVT is the percentage deviation between the predictions of the various models and the experimental result.

	Cloudy day				Clear-sky day			
	$\Delta Q_{\text{tank,net}}$ MJ				$\Delta Q_{\text{tank,net}}$ MJ			
	FPC	ETC	PVT	%	FPC	ETC	PVT	%
1-node dynamic model ($C = C_{\text{exp}}$)	6.14	12.76	3.12	-4.3	10.52	22.11	6.16	28.1
1-node dynamic model ($C = C_{\text{wt}}$)	6.14	13.34	3.47	6.4	10.52	23.16	6.84	42.2
Quasi-steady model ($C = 0$)	6.20	13.59	3.52	8.0	10.85	23.62	6.94	44.3
3-D dynamic model	6.72	15.54	3.20	-1.8	11.92	26.80	5.95	23.7
Experimental result	-	-	3.26	-	-	-	4.81	-

5.4.4 Hot-water system simulation

The results of the hot water system simulation using the lumped (single-node) dynamic model, lumped quasi-steady state model and 3-D dynamic model of the FPC, ETC and PVT collectors are reported in Table 5.7. The results are compared in terms of the net energy gain to the tank over the simulation period in order to evaluate the impact of the dynamic collector behaviour on the energy yield prediction for the system¹. Results are presented for a clear-sky day and an intermittently cloudy day. Results for the PVT system model are compared and evaluated using experimental data for the real PVT system². For the single-node dynamic collector model, two comparison values of effective thermal capacity are used. The first is the manufacturers value determined using the weighted-component thermal capacities C_{wt} . The second is the average of the experimentally obtained values C_{exp} presented in Table 5.5. The quasi-steady state model uses the steady-state collector efficiency equation (equivalent to setting $C = 0$) to predict the rise in enthalpy of the fluid flowing through the collector at each time instant.

The results in Table 5.7 show that of the lumped single-node models, the lowest value of C corresponds to the highest prediction of net energy gained by the tank. In the case of the PVT collector for which comparison experimental data is available, this is shown to be an over-

¹To evaluate the extent to which thermal stratification in the hot-water tank influences the daily system performance, simulations were also performed for a 2-node, stratified tank model and the results compared to those for the fully-mixed (lumped) tank model. The difference in daily net energy accumulation between the two tank models was found to be $\sim 1\%$.

²At the time of writing, comparison test results for the FPC and ETC collectors in the hot water tank configuration have not been obtained.

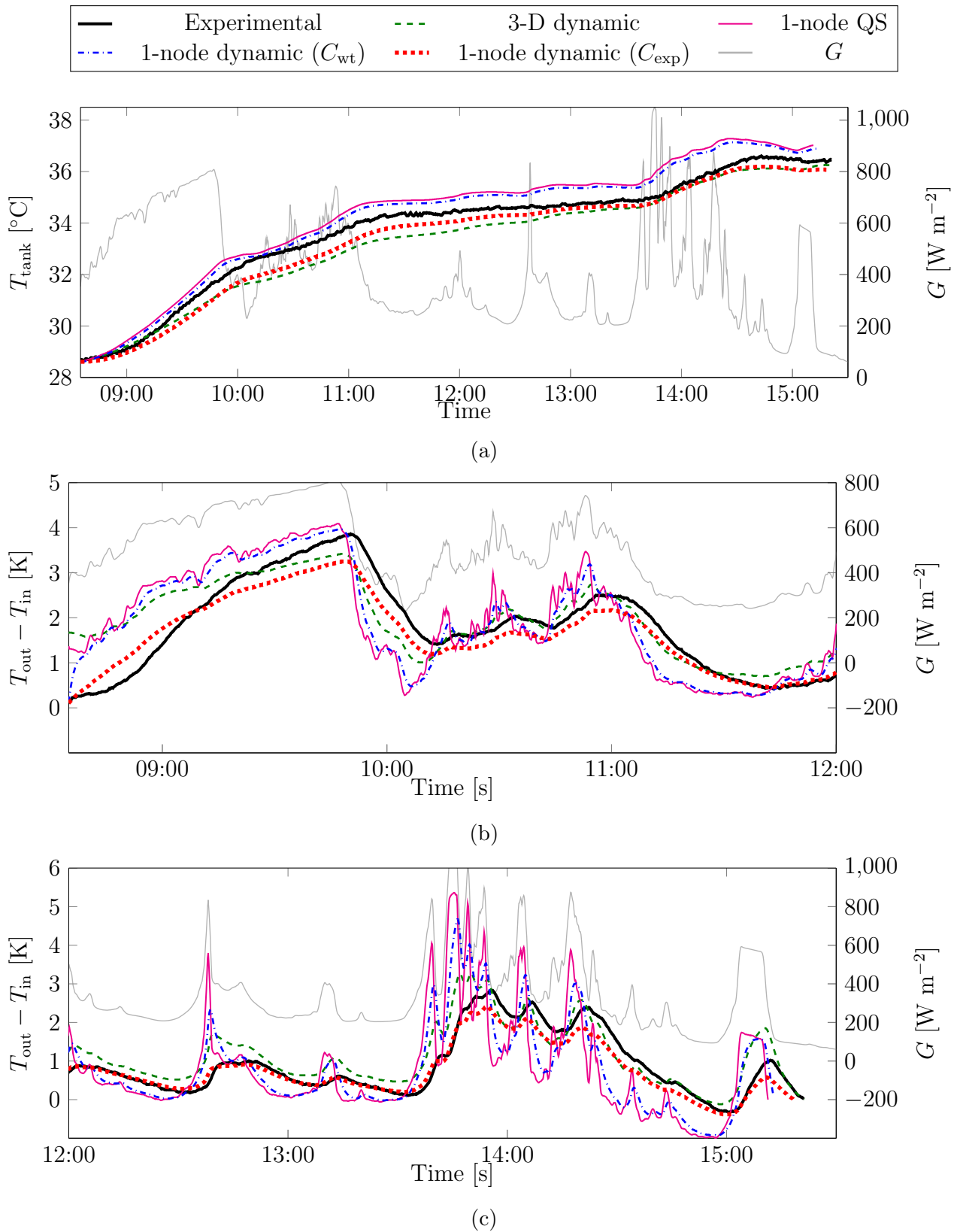


Figure 5.12: Diurnal simulation plots for the 3-D dynamic model of the PVT collector, compared to experimental data for the same irradiance profile: a) tank temperature; b) collector temperature rise (morning), c) collector temperature rise (afternoon); also shown are simulation results for the 1-node quasi-steady (QS) collector model and the 1-node dynamic collector model using the lumped collector thermal capacities C_{wt} and C_{exp} .

prediction. The single-node collector model simulated with the higher, experimentally obtained effective thermal capacity value C_{exp} is shown to under-predict the net thermal energy transfer to the tank by 4% over the cloudy-day simulation period, however the instantaneous peaks in absorbed thermal energy predicted by this model coincide relatively well with the experimental data, as shown in Figures 5.12b and 5.12c. The single-node dynamic model using the effective thermal capacity value C_{wt} obtained via the weighted-component method is found to result in a 6% overestimate in net energy gain compared to the cloudy-day experimental results. This model also shows an appreciable response to the very fast spikes in irradiance (such as that occurring in Fig. 5.12c at time 12:38), which are entirely attenuated by the collector in the experimental results, and by the single-node model with the experimental value of C_{exp} . The quasi-steady model responds instantaneously to variations in irradiance as also shown in Fig. 5.12c and, for the case of the PVT collector, results in the largest over-prediction (8% on the cloudy day) in net thermal energy gain to the tank of all the models. The 3-D dynamic model predicts a lower energy yield compared to the lumped models for the PVT collector but a higher energy yield for the FPC and ETC collectors, which suggests that the reliability of the results from the 3-D model is dependent on the quality of the input data describing the materials and geometry and that further refinement of this input data may be needed.

5.5 Summary

The primary aim of the steady-state tests in this section was to evaluate the performance of a standard ETC water heating solar collector at temperatures in excess of the typical operating range encountered in hot water systems, and closer to the temperatures found to be optimal for maximum power predicted by the endoreversible analysis in Chapter 2. The existing data for the collector published in the official ISFH test document reported an efficiency of 66% at a maximum testing temperature of 80 °C (measured at the inlet) and a solar irradiance of 874 W m⁻². Extrapolating the efficiency curve beyond this point to inlet temperatures of 110 °C results in a predicted efficiency of 57%.

By comparison, experimental data obtained under similar conditions in the present study have provided encouraging results with efficiencies of 61–64% obtained at collector inlet temperatures between 111–112.5 °C. While the results are found to deviate further from the manufacturer’s efficiency curve prediction but 7–12% at higher temperatures, this deviation results in a higher than predicted efficiency and a modified efficiency curve in which the second-order heat loss coefficient is smaller. This provides some degree of validation for the results in Chapters 2 and 4, in which most of the analysis was performed at temperatures that required extrapolation of the manufacturer’s efficiency curve outside of the tested range.

Further work in this area should extend to testing the collector at low irradiance levels, in order to validate the “cloudy-day” simulation results in Chapter 4. For outdoor testing this requires cloud cover to be uniform such that irradiance is maintained at an approximate steady state within the acceptable margin of the testing standard. For the outdoor tests performed so far, the measured irradiance during cloudy conditions has been found to exhibit too much short time-scale variation to be usable for the steady-state performance analysis, however it has been useful for evaluating the dynamic performance.

The second part of the experimental effort was conducted primarily to consider various models for predicting the dynamic performance of solar collectors with particular emphasis on a lumped dynamic model that uses an effective thermal capacity. More importantly the wider impact of this dynamic behaviour on the performance of solar thermal energy systems (in this case, a simple solar hot water system) is of interest in order to understand the importance of capturing this behaviour accurately. This is relevant for the evaluation of the S-CHP system model, which uses an effective thermal capacity value obtained by the weighted-component method as an input parameter in the solar collector array subs-system model. In many cases this estimated value is the only information available for modelling the dynamic behaviour of solar collectors. Where more detailed information on geometry and materials are available, detailed multi-dimensional models may be preferred for a higher level of accuracy. In this study a further comparison was also performed with a bespoke 3-D dynamic thermal model developed by Guarracino *et al*, configured with the geometrical and material parameters of the three collectors for which the experiments were conducted.

The results have shown a significant difference between the values obtained from the experimental and the weighted-component estimation methods for the effective thermal capacity, with the experimental method resulting in typically a ~ 4 times higher value for the ETC and a ~ 10 times higher value for the PVT; while for the FPC the two methods produce a similar value. Further work is required to investigate the reason for the observed differences. For the ETC the possibility of a higher than expected degree of thermal interaction between the absorber and the glass envelope, due to a loss of vacuum in the tubes, was investigated and ruled out. The 3-D numerical model meanwhile was also found to under-predict the effect of the thermal inertia for all of the collectors, underestimating the transient response time by 20–35% for the FPC and by 50–65% for the ETC. For the PVT, the response time was initially under-predicted by 65–75% by the 3-D model; but by modifying the model to include a 2 mm air gap representing the poor thermal contact between the PV and copper absorber plate layers, the prediction improved to within 45–50% of the experimental value. The ETC and PVT collectors are considerably more complex in design than the FPC, with features that involve greater modelling complexity such as the optical efficiency and thermal loss mechanisms in the manifold for the ETC, and the PV-thermal absorber interface for the PVT collector.

Implementing the collector models into a simple solar hot water system has shown that a smaller thermal capacity in the lumped collector model results in a higher prediction of the daily solar energy gain, and generally an over-prediction compared to that observed for the real collector under the same conditions. The predicted daily energy gain is 5% higher from the lumped model with $C = C_{\text{wt}}$ compared to $C = C_{\text{exp}}$ for the ETC, whereas for the PVT the difference is 12%. The model in which $C = C_{\text{exp}}$ also shows a much improved prediction of transient response prediction under intermittently cloudy day conditions when compared to actual performance data for the PVT collector. The predictions of the 3-D model meanwhile result in both under-predictions and over-predictions of the daily energy gain when compared to the lumped models, depending on the collector type. This suggests that while complex multi-dimensional models are highly useful as design tools, their extra value in predicting the performance of a wide range of existing collector designs within a broader energy system is dependent on the available information with which to model the collector.

Chapter 6

Organic Rankine Cycle Engine Testing

In this section we describe a small-scale (~ 1 kW) ORC test bench, designed for the testing of various system components, and the models used for their performance predictions. Some preliminary results are presented from the test bench in a water-cooled configuration with a scroll expander and the working fluid R245fa, operating with heat source temperatures in the range 100–120 °C. The components of the system are discussed and compared to the design requirements determined from the S-CHP system modelling. The preliminary results indicate that the ORC model over-predicts the gross power output from the system by between 30–90% when configured with the experimental system operating parameters as inputs. This is attributed largely to thermal losses and pressure losses in the first version of the experimental system. Cycle analysis is performed using data from the ORC rig and based on the findings, design improvements are suggested to improve efficiency. Finally some analysis of the dynamic behaviour of the system is performed and the implications on the S-CHP model predictions are discussed.

6.1 Test Bench Description

A schematic diagram of the test bench apparatus is shown in Fig. 6.1. The apparatus consists of a non-recuperative ORC engine, an electrically heated thermal oil circuit as the heat source,

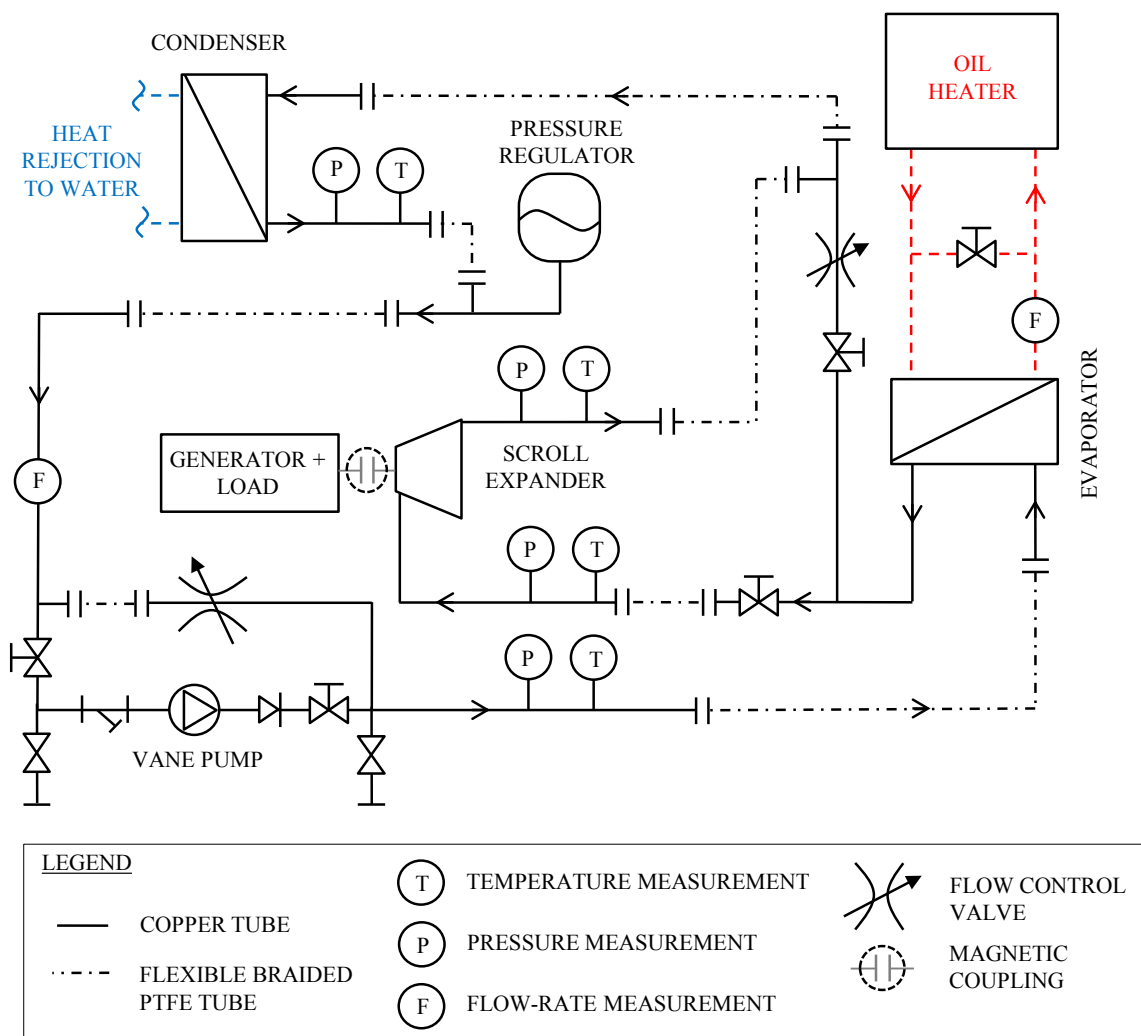


Figure 6.1: Schematic representation of the ORC test bench.

a mains cooling water circuit and various instruments for measurement and control. The individual components will now be discussed in detail.

Electric oil heater

The thermal input to the ORC test rig is an 18 kW electrical oil heater (Frigosystem TRO-200-18/15), shown in Fig. 6.2 The oil heater has an internal reservoir and circulation pump. The thermal oil used in the oil heater circuit is MARLO THERM SH. Temperature is controllable via a PID controller to a maximum of 200 °C. For fine control of temperature, the oil heater also uses an active water cooling circuit. The oil pump fixed flow-rate is 4800 L hr⁻¹, however the flow-rate of oil through the ORC evaporator heat exchanger can be adjusted by means of

a bypass valve. The flow-rate of oil through the evaporator is monitored using a turbine flow meter, also shown in Fig. 6.2.



Figure 6.2: 18 kW electric oil heater used as heat source for the ORC test bench, delivering heat source temperatures in the range 100 – 120 °C. Also shown (right) is the turbine flow meter measuring thermal oil flow-rate through the evaporator.



(a) Vane pump.



(b) Scroll expander coupled to A.C. generator.

Figure 6.3: Pump and expander/generator used in the ORC test bench

Table 6.1: Rotary vane pump specifications, taken from Ref. [208].

Manufacturer/model	Fluid-o-Tech TMFR 051A
Maximum flow-rate	250 L hr ⁻¹
Maximum head (continuous duty)	16 bar
Flow-rate at maximum head	180 L min ⁻¹
Speed range	1,100–3,500 RPM

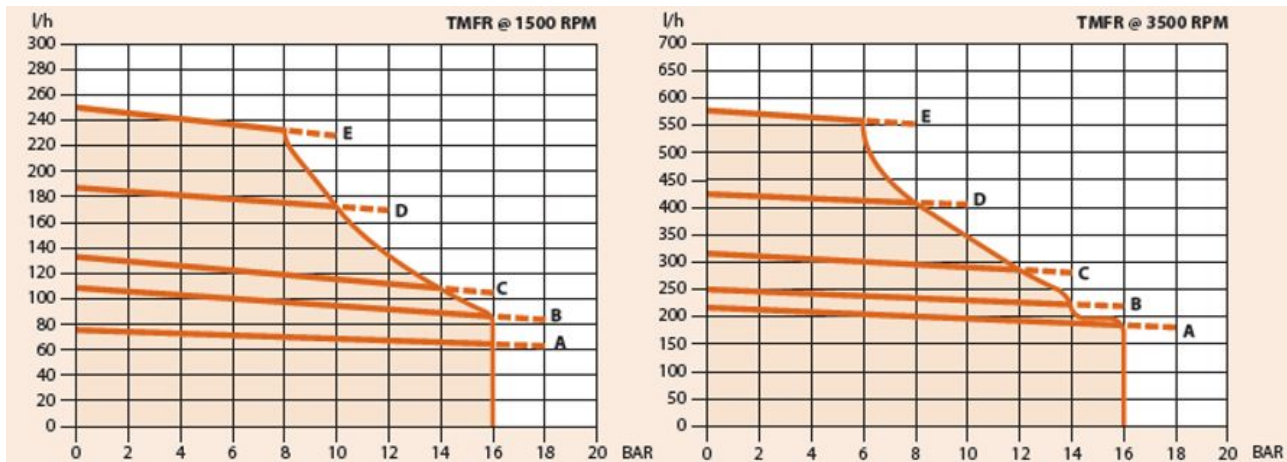


Figure 6.4: Manufacturer's hydraulic performance curves for the Fluid-o-Tech TMFR 051A vane pump (line A) at 1500 RPM (left) and 3500 RPM (right). Performance curves obtained with water at 20 °C. Taken from Ref. [208]

ORC pump

The pump selected for the the ORC system was a Fluid-o-Tech TMFR 051A series magnet-driven vane pump, shown in Fig. 6.3a. The pump has a maximum rated flow-rate (with water) of 180 L hr⁻¹ at a rotational speed of 3,500 RPM and a maximum delivery pressure of 16 bar. The pump speed is adjustable in finite intervals between 1,100–3,500 RPM using a dedicated speed controller unit also supplied by Fluid-o-Tech. The pump specifications are reported in Table 6.1 and the characteristic performance curves from the manufacturer's documentation (for operation at 1,500 RPM and 3,500 RPM) are shown Fig. 6.4.

Expander

One of the intended applications for the experimental facility is to test multiple types of small-scale positive-displacement expanders. For the preliminary tests, the test bench was configured with an Air Squared E15H22 scroll expander, shown in Fig. 6.3b. This is one of the few “off-

Table 6.2: Scroll expander specifications, taken from Ref. [43].

Manufacturer/model	Air Squared E15H22
Nominal output	1 kW _e
Nominal inlet pressure ⁽ⁱ⁾	13.8 bar
Nominal exhaust pressure ⁽ⁱ⁾	1.25 bar
Nominal expander efficiency ⁽ⁱ⁾	70%
Volumetric efficiency	80%
Volume ratio	3.5
Displacement (suction volume)	14.5 cc/rev
Maximum rotational speed	3,600 RPM
Maximum rated inlet pressure	13.8 bar

⁽ⁱ⁾ Corresponding to 1 kW_e output and nominal speed of 3000 RPM with R245fa as working fluid.

the-shelf” products specifically for use in small-scale ORC systems. The specifications of the expander are listed in Table 6.2. The expander is intended for use with a range of refrigerant working fluids, including R245fa and R134a.

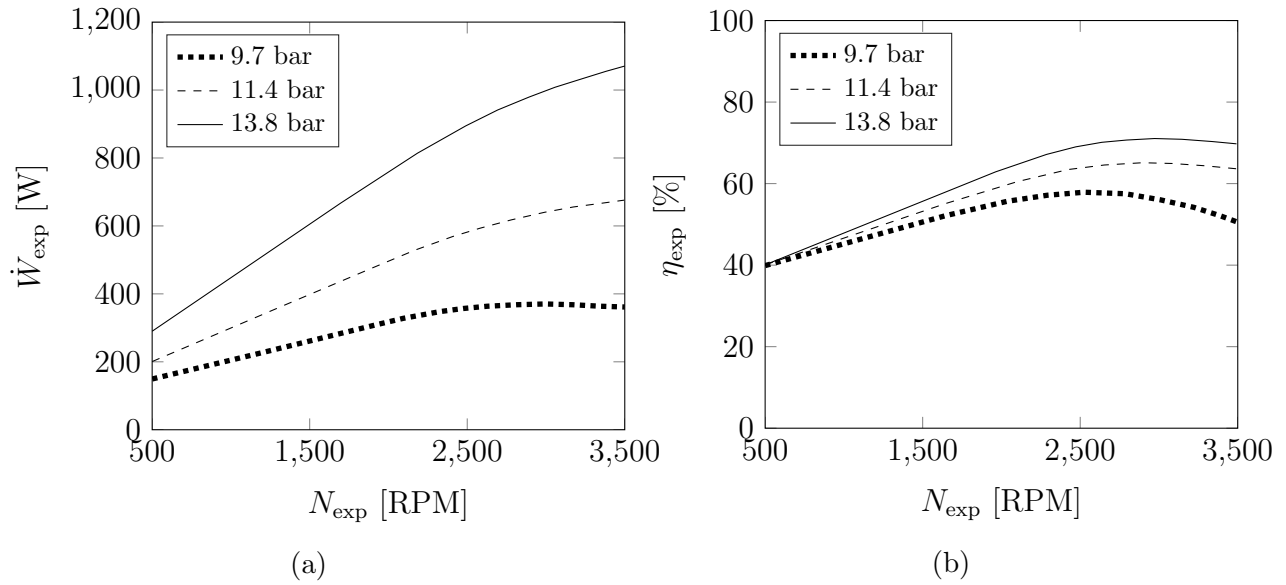


Figure 6.5: Manufacturer’s performance curves for the Air Squared E15H22 scroll expander, showing (a) shaft power output and (b) expander efficiency, at various rotational speeds. Performance curves obtained with R245fa with a condensation pressure of 1.25 bar(a). Reproduced from Ref. [43]

Generator

The expander is magnetically coupled to a Voltmaster AB-30 series A.C. electric generator. The generator is connected to an electrical load made up of light bulbs and heat-dissipating

Table 6.3: Evaporator and condenser plate heat exchanger specifications and nominal sizing conditions.

	Evaporator	Condenser
Heat duty	18.0 kW	22.6 kW
Length	287 mm	527 mm
Width	117	113 mm
Number of plates	30	30
Total surface area	0.672 m ²	1.624 m ²
Overall HTC	400 W m ⁻² K ⁻¹	2810 W m ⁻² K ⁻¹
Nominal refrigerant flow-rate	0.07 kg s ⁻¹	0.07 kg s ⁻¹
Nominal oil/water flow-rate	80 L hr ⁻¹	1.075 kg s ⁻¹
Refrigerant-side pressure drop	2 kPa	15 kPa
Oil/water-side pressure drop	50 kPa	45 kPa

resistors connected in parallel circuits. The individual circuits can be switched in or out in order to regulate the electrical load and modify the rotational speed and electrical power delivered by the expander/generator under different operating conditions.

Heat exchangers

Brazed plate heat exchangers are used for the evaporator and condenser. For the heat addition process, the preheater, evaporator and superheater stages are combined into a single heat exchanger. Likewise for the heat rejection process, the precooler, condenser and subcooler are also combined into a single heat exchanger. The evaporator and condenser specifications are listed in Table 6.3. As will be shown in Section 6.2, the mass flow-rate of R245fa required for the nominal 1 kW_e output under the nominal design conditions is 0.043 kg s⁻¹. However, the heat exchangers are over-sized for a refrigerant flow-rate of 0.07 kg s⁻¹, in order to provide greater flexibility over a wider range of experimental conditions.

Instrumentation

Temperatures in the refrigerant, thermal oil and cooling water circuits were measured using T-type thermocouples. Pressures were measured using Omega PX and PXM-series pressure transducers. The refrigerant flow-rate was measured unobtrusively using an Omega FDT-21 ultrasonic flowmeter, which was also used to calibrate the reading of the turbine flow-

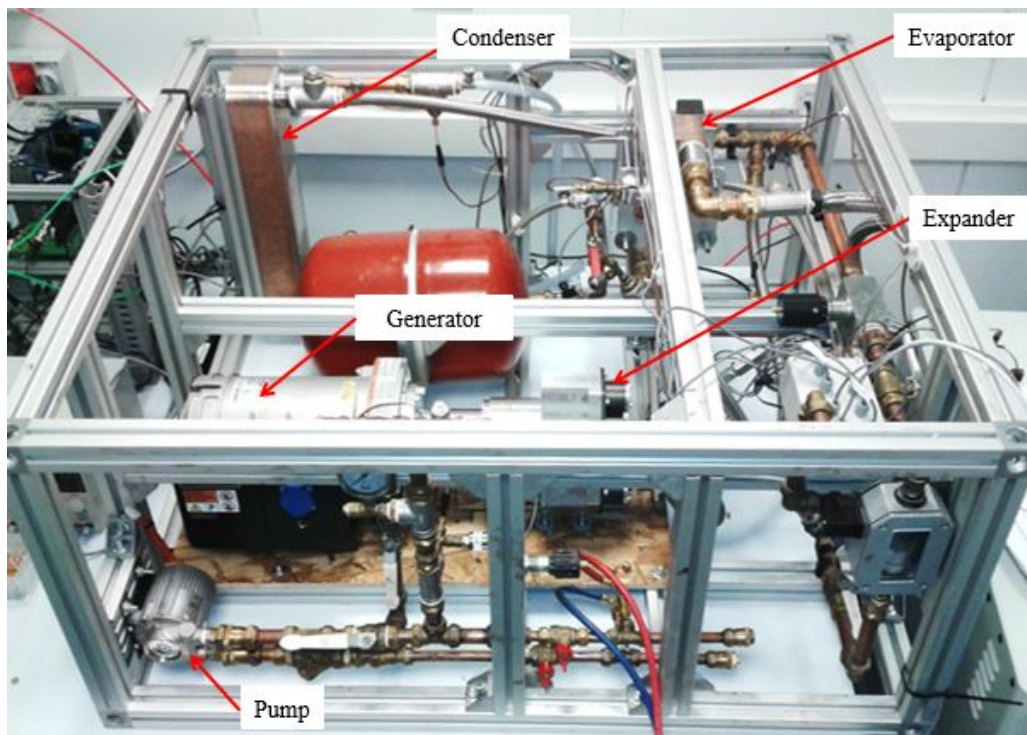


Figure 6.6: Photograph of the ORC test bench during construction, with major components labelled.

meter in the thermal oil circuit. Voltage, current and power output from the generator were measured using a Rohde & Schwarz HM8115 power meter, while the rotational speed of the expander/generator was determined by measuring the frequency of the A.C. voltage at the generator terminals using a Fluke multi-meter. The electrical power consumed by the pump was measured using a Prodigit 2000-MU power meter.

Control of system operating parameters

The system was designed to allow as much control as possible of the many operating parameters. Flow-rates of refrigerant, thermal oil and cooling water are all adjustable through the use of flow-control valves, bypass branches and by adjusting the pump speed in the case of the refrigerant. Adjustment of the pump speed in conjunction with the electrical load on the expander/generator also allows for control of the pressure ratio across the expander. Maximum pressure ratio is achieved when the pump is set to its maximum power point, with the full electrical load engaged across the generator. Increasing the load also causes a reduction in the expander/generator rotational speed, a drop in output voltage and an increase in current. An

expansion vessel similar to those used in central heating systems is installed in the refrigerant liquid line between the condenser and pump. The vessel is partially filled with air, separated from the refrigerant by a flexible diaphragm. By adding or removing air from the vessel the ORC condensation pressure can be adjusted.

6.2 Design Conditions and System Sizing

Based on the scroll expander nominal power of 1 kW, a sizing calculation was performed to determine the nominal temperatures, pressures and flow-rates in the system. Preliminary observations were made regarding the nominal conditions in the manufacturer's datasheet. The evaporation and condensation pressures corresponding to 1 kW nominal power with R245fa are 13.8 bar and 1.25 bar respectively, equivalent to evaporation and condensation temperatures of 104 °C and 20 °C and a pressure ratio of 11. This corresponds to a volume ratio of ~ 12 , which is between 3 and 4 times larger than the built-in volume ratio of the expander. Therefore the conditions for nominal power output may not necessarily correspond to the conditions for maximum expander efficiency. Using the manufacturers information on displacement volume and volumetric efficiency, an estimated volumetric flow-rate at the expander inlet for a nominal rotational speed of 3000 RPM (50 Hz) can be calculated:

$$\dot{V}_{\text{sw,exp}} = V_{\text{exp,in}} \cdot N_{\text{exp}} \cdot \eta_{\text{vol}} = 14.5 \times 10^{-6} \cdot 50 \cdot 0.8 = 0.58 \times 10^{-3} \text{ m}^3 \text{ s}^{-1}. \quad (6.1)$$

For the purposes of the test bench sizing, the evaporation temperature will be assumed to be 105 °C and the condensation temperature will be assumed to be 35 °C to allow for the possibility of cooling water temperatures of up to 20 °C (particularly considering the long mains water pipework runs in the building). Assuming superheating of 10 K at the expander inlet, the refrigerant density at the expander inlet is 76.2 kg m^{-3} . This gives a working fluid mass flow-rate of 0.044 kg s^{-1} based on $\dot{V}_{\text{exp,in}}$ calculated in Eq. 6.1. The remaining cycle parameters can now be calculated and are presented in Table 6.4.

Table 6.4: Design conditions for the ORC test rig.

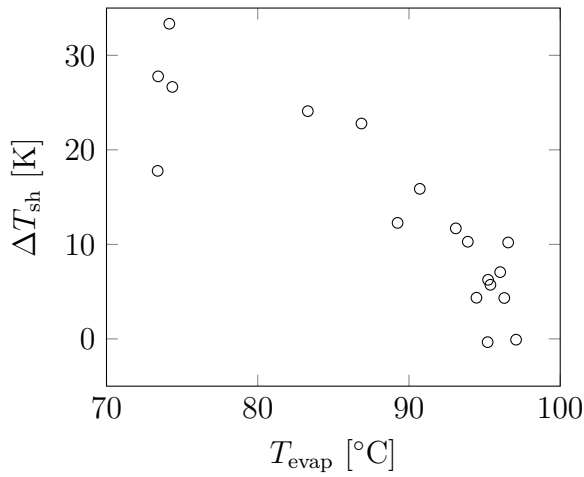
Parameter	Value
$\dot{m}_r, \dot{m}_h, \dot{m}_c$	$0.044 \text{ kg s}^{-1}, 0.34 \text{ kg s}^{-1}, 0.21 \text{ kg s}^{-1}$
$T_{\text{evap}}, p_{\text{evap}}$	$105 \text{ }^{\circ}\text{C}, 14.1 \text{ bar}$
$T_{\text{evap}}, p_{\text{evap}}$	$35 \text{ }^{\circ}\text{C}, 2.1 \text{ bar}$
T_1, T_2, T_3, T_4	$25 \text{ }^{\circ}\text{C}, 25.8 \text{ }^{\circ}\text{C}, 115 \text{ }^{\circ}\text{C}, 70.1 \text{ }^{\circ}\text{C}$
h_1, h_2, h_3, h_4	$232 \text{ kJ kg}^{-1}, 234 \text{ kJ kg}^{-1}, 491 \text{ kJ kg}^{-1}, 465 \text{ kJ kg}^{-1}$
$\eta_{\text{exp}}, \eta_{\text{pump}}, \eta_{\text{gen}}, \eta_{\text{m}}$	$70\%, 65\%, 90\%, 90\%$
$\Delta T_{\text{sub}}, \Delta T_{\text{sh}}$	$10 \text{ K}, 10 \text{ K}$
$\Delta T_{\text{app}}, \Delta T_{\text{pin}}$	$5 \text{ K}, 5 \text{ K}$
$\dot{W}_{\text{exp}}, \dot{W}_{\text{pp}}$	$1144 \text{ W}, 68 \text{ W}$
$\dot{W}_{\text{e,gross}}, \dot{W}_{\text{e,net}}$	$1029 \text{ W}, 962 \text{ W}$
$\dot{Q}_h, \dot{Q}_c$	$11.4 \text{ kW}, 10.3 \text{ kW}$

6.3 Discussion of Preliminary Results

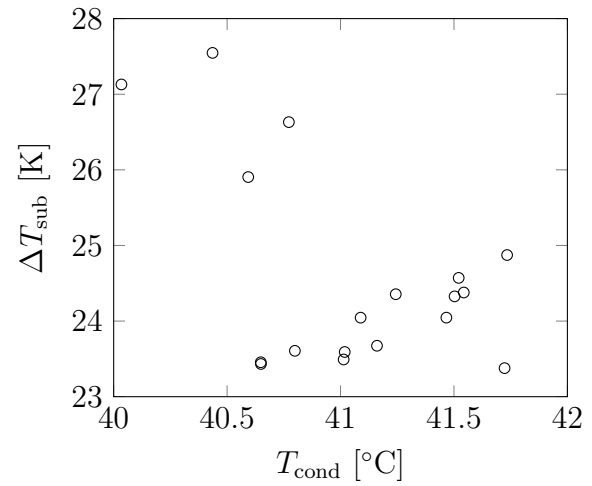
Preliminary results from the ORC test bench were obtained over a range of pump settings, heat source temperatures and expander loadings. The key experimental parameters measured over the range of steady state tests are plotted in Figs 6.7 and 6.8.

Figure 6.8 shows that increased power output was generally obtained at higher evaporation temperatures and pressure ratios, however maximum power did not correspond to maximum thermal efficiency, which was also influenced by the characteristics of the heat source. The maximum gross electrical power output obtained from the preliminary runs was 560 W at an evaporation temperature of $97 \text{ }^{\circ}\text{C}$. The maximum expander inlet pressure achieved during the tests was 11.8 bar, however Fig. 6.5a indicates that a value closer to the maximum rated pressure of 13.8 bar is required in order to achieve the nominal 1 kW electrical output.

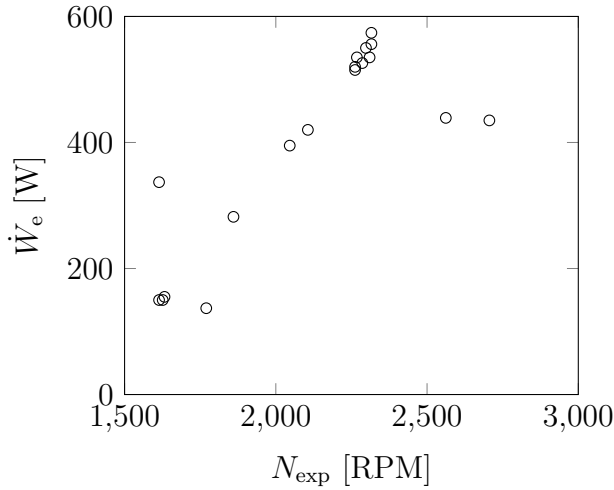
The range of gross thermal efficiencies obtained was 3.1–5.3%, with the higher efficiencies generally obtained at higher power outputs. The range of net thermal efficiencies were found to be considerably lower due to the low efficiency of the pump, which was between 11–27%. The pump power consumption that corresponded to the maximum obtained expander power output was 160 W. The highest gross efficiency was obtained for a power output of 526 W at a heat source temperature of $114 \text{ }^{\circ}\text{C}$. Increasing the heat source temperature to $120 \text{ }^{\circ}\text{C}$ (also increasing the heat energy input) allowed a marginally higher power output to be obtained but



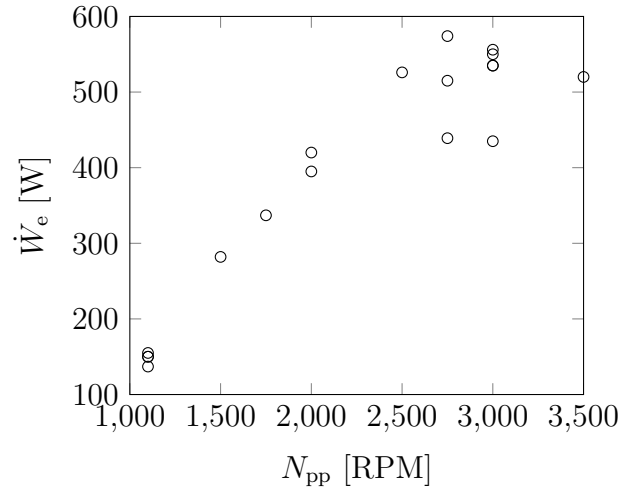
(a) Superheating at expander inlet vs evaporation temperature.



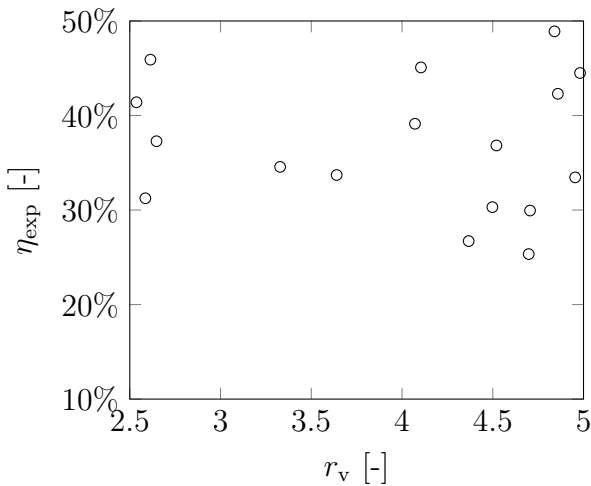
(b) Subcooling at condenser outlet vs condensation temperature.



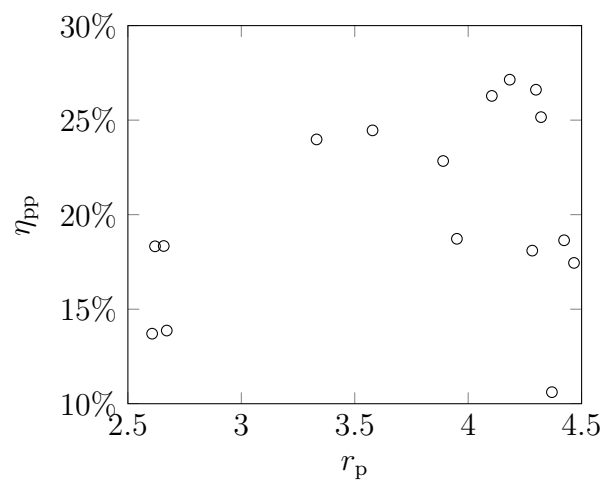
(c) Electrical work output vs expander rotational speed.



(d) Electrical work output vs pump rotational speed.



(e) Expander efficiency vs volume ratio.



(f) Pump efficiency vs pressure ratio.

Figure 6.7: ORC performance parameter plots from the preliminary experiments.

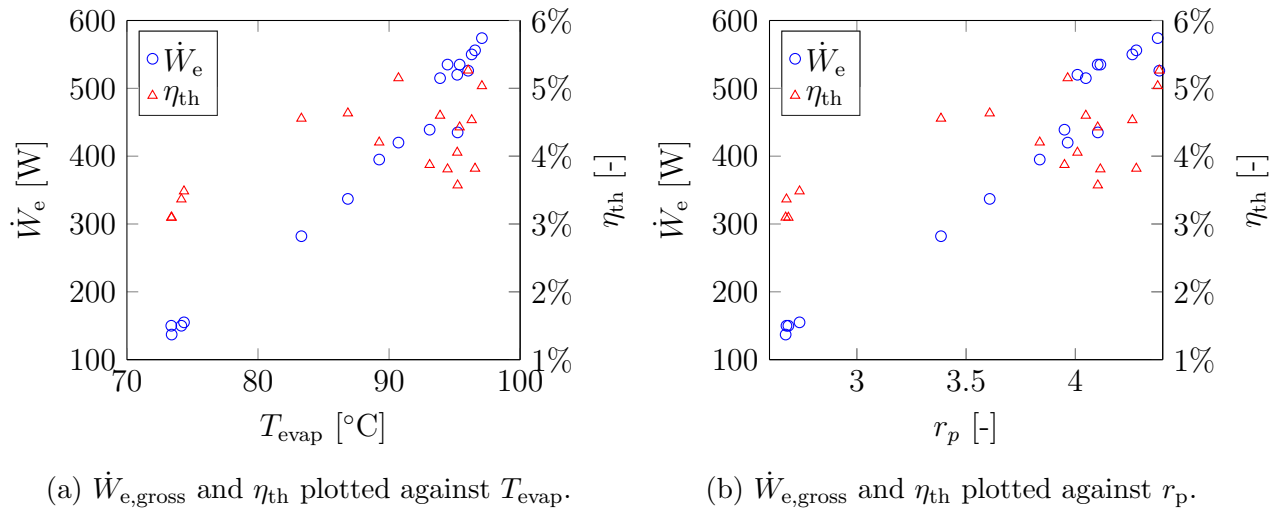


Figure 6.8: Gross electrical power output and ORC gross thermal efficiency plotted against evaporation temperature and pressure ratio.

with a significant drop in efficiency to $< 4\%$.

When evaluating the expander performance, Eq. 3.17 from Chapter 3 is found to give a potentially misleading indication of the expander efficiency, because of the assumption that the expansion process is adiabatic and that enthalpy is entirely converted into shaft work. For the experimental system, heat losses to the environment during the expansion process result in a lower temperature measured in the exhaust vapour than would be expected for an equivalent adiabatic expansion process. The result when plotted on a T - s diagram is an expansion process line with a positive gradient indicating an apparent isentropic efficiency of $> 100\%$ for some of the results¹. Therefore an alternative expression for the overall isentropic efficiency of the expander is instead applied to the experimental results:

$$\eta_{exp} = \frac{\dot{W}_e}{\dot{m}_r(h_3 - h_{4s})}, \quad (6.2)$$

which includes also the mechanical and electrical losses in the generator. This expression is found to result in expander efficiencies between 25–50% for the preliminary test results, but is subject to some degree of uncertainty regarding the measurement of the refrigerant flow-rate (see Section 6.4).

It was found that a substantial amount of subcooling was required prior to the pump inlet

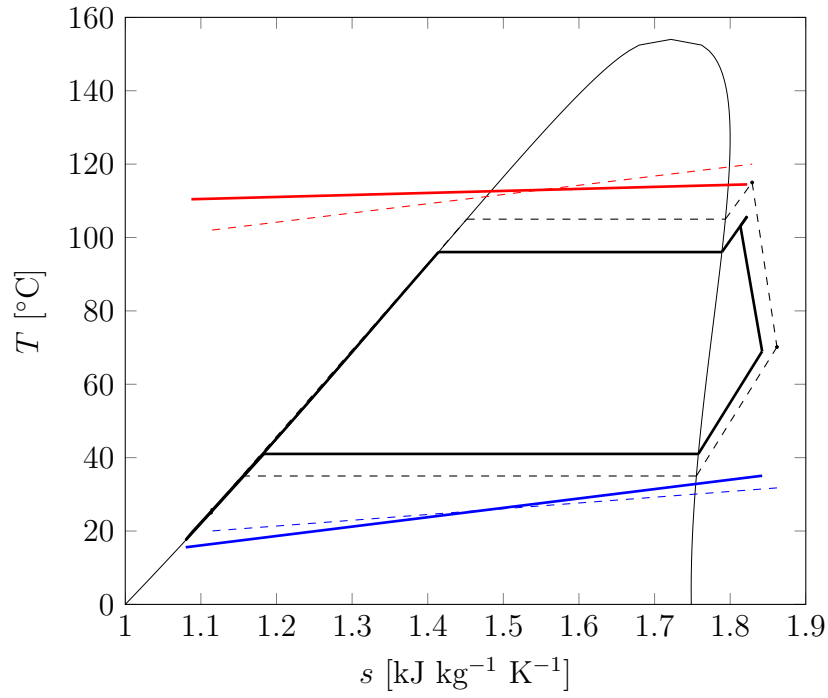
¹This was mostly found to occur for the lower power output cases at lower pump speeds

Table 6.5: Comparison of experimental values (for maximum power case) and design values for the ORC test bench, corresponding to the T - s plots in Fig. 6.9b.

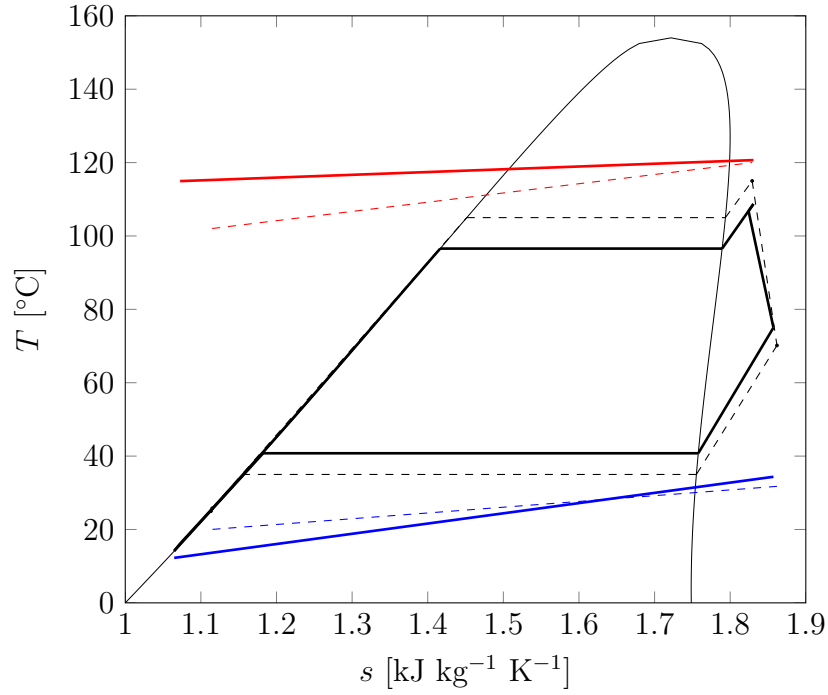
	Experimental values		Design value
	Maximum efficiency case	Maximum power case	
$\dot{W}_{e,gross}$	526 W	556 W	1029 W
$\dot{W}_{pp}$	144 W	160 W	68 W
$\dot{W}_{e,net}$	382 W	396 W	962 W
$\eta_{th(gross)}$	5.3%	3.8%	9.1%
T_{evap}, p_{evap}	96.0 °C, 11.6 bar	96.6 °C, 11.7 bar	105 °C, 14.1 bar
T_{cond}, p_{cond}	41.0 °C, 2.6 bar	40.8 °C, 2.6 bar	35 °C, 2.1 bar
$\dot{Q}_h$	10.0 kW	14.6 kW	11.3 kW
$\dot{m}_r$	0.038 kg s ⁻¹	0.059 kg s ⁻¹	0.044 kg s ⁻¹
$\dot{m}_h$	1.3 kg s ⁻¹	1.3 kg s ⁻¹	0.36 kg s ⁻¹
$\dot{m}_c$	0.2 kg s ⁻¹	0.2 kg s ⁻¹	0.21 kg s ⁻¹
N_{exp}	2286 RPM	2316 RPM	3000 RPM
$T_{h,in}$	114.5 °C	120.7 °C	120 °C
$T_{c,in}$	15.6 °C	12.2 °C	20 °C
ΔT_h	4.0 K	5.7 K	5 K
ΔT_c	19.5 K	22.1 K	5 K
r_p	4.5	4.6	6.7
$r_{v,exp}$	4.8	5.0	7.3
$\eta_{exp}^{(i)}$	49%	33%	63%
$\eta_{pump}^{(ii)}$	17%	25%	58.5%
ΔT_{sub}	23.5 K	26.6 K	10 K
ΔT_{sh}	7.1 K	10.2 K	10 K
$\Delta T_{app,evap}$	8.7 K	12.0 K	5 K
$\Delta T_{app,cond}$	2.0 K	1.8 K	5 K
$\Delta T_{pin,evap}$	16.1 K	20.8 K	5 K
$\Delta T_{pin,cond}$	8.2 K	9.4 K	5 K

⁽ⁱ⁾ Combined expander and generator efficiency. ⁽ⁱⁱ⁾ Combined pump and motor efficiency.

in order for the pump to operate effectively (noisy operation with lower subcooling indicated the possible occurrence of vapour bubbles at the suction side). To avoid this phenomenon, between 23–28 K of subcooling was required which caused the condensation temperature to be raised to over 40 °C (compared to 35 °C for the design case). This limits the potential for work conversion from the cycle, but can be addressed through improvement to the layout of the experimental apparatus to ensure a larger liquid head at the pump inlet (installing the pump below the condenser with an intermediate liquid receiver tank). Mains water temperatures encountered at the inlet to the condenser were between 10.8–15.6 °C during the preliminary tests.



(a) Maximum efficiency case: $\eta_{th} = 5.3\%$, $\dot{W}_e = 526$ W.



(b) Maximum power case: $\eta_{th} = 3.8\%$, $\dot{W}_e = 556$ W

Figure 6.9: T - s plots of the maximum efficiency and maximum power cases from the preliminary experiments compared to the design conditions for the ORC test bench (shown in dashed lines) and with heat source and cold sink streams shown in red and blue respectively. See Table 6.5 for comparison of the performance parameters.

Maximum power output was achieved at expander rotational speeds between 2250–2350 RPM with the generator fully loaded. This is below the nominal rotational speed of 3000 RPM and indicates that the expander was being under-powered. The pump speed at maximum power was between 2500–3500 RPM, with most of the runs performed at 3000 RPM. The poor expander efficiency and lower than expected fluid temperature measured at the expander outlet suggest significant thermal losses to the environment from the expander casing. Insulating the expander may be one way to increase the ORC efficiency and power output, without further increasing the thermal input from the oil heater which in the present tests was delivering up to $15 \text{ kW}_{\text{th}}$ (85% rated output) for the maximum power output case.

The results from the experimental system under the maximum obtained efficiency and maximum delivered power are compared to the design values in Table 6.5 and by way of a T - s plot in Fig. 6.9. It can be seen in Fig. 6.9 that the lower evaporation temperature and higher condensation temperature result in a smaller cycle envelope compared to the design case. The plots also show a drop in temperature between the evaporator outlet and expander inlet, indicating the occurrence of thermal losses in the connecting pipework. The main difference between the maximum power case and the maximum efficiency case is the increased heat source inlet temperature and thermal power provided by the oil heater. The cooling water inlet temperature is lower than the design value, while the cooling water flow-rate is also lower than the design value resulting in a steeper temperature glide. The heat source flow-rate is significantly higher than the design value, resulting in a shallower temperature glide and a larger than expected temperature difference between the temperature profiles of the heat source and working fluid. It was found during the experiment that a high flow-rate of thermal oil was necessary to achieve a dry vapour at the inlet to the expander. At lower heat source flow rates and higher refrigerant flow-rates, it was not possible to achieve a the small amount of superheating, and noisy operation of the expander indicated the likely presence of liquid droplets in the vapour entering the expander.

The above observation, together with the lower than expected power output (and rotational speed) of the expander, and the lower than expected evaporation pressure, indicate that insufficient heat input is being delivered to the refrigerant working fluid during the evaporation

process. The temperature profiles of heat source and working fluid streams in Fig. 6.9b may further indicate that insufficient heat is being extracted from the heat source due to limited heat transfer. Despite the heat exchanger being over-sized for the required design duty (see Tables 6.3 and 6.4), it is possible that the compact volume and the single-stage configuration of the heat transfer process limits the effectiveness that is achieved at lower heat source temperatures. It is common in heat exchanger design to separate the evaporation and superheating stages between two heat exchanger stages, however this may also result in an increased system cost. As a first measure, improvements to the insulation of the heat source pipe loop and the evaporator heat exchanger, and modifications to shorten the pipework run between the evaporator and expander to reduce thermal (and pressure) losses, may provide an improvement in the heating delivered to the cycle and a closer temperature match between the heat source and working fluid to be achieved. Further, this improve the vapour quality at the expander inlet, enabling operation at higher refrigerant flow-rates and higher evaporation pressures.

The gross thermal efficiency of 5.3% in the maximum efficiency case is just over half of the design value (9.1%). The thermal energy input supplied by the hot stream for the maximum efficiency case is also slightly lower than the design case (10 kW compared to 11.3 kW). For the maximum power case the thermal input is higher (14.6 kW) which is 46% higher than the maximum efficiency case, although the relative improvement in power output is only 6%. Thus it can be concluded that increasing the heating input from the oil heater results in disproportionately larger thermal losses from the system which should be reduced through improved insulation.

6.4 Discussion of Experimental Uncertainties

The preliminary results are subject to some significant uncertainties associated with the measurements. A full uncertainty analysis is not presented here, however the main uncertainties associated with the calculation of the expander efficiency and the ORC thermal efficiency are indicated in Fig. 6.10. The expander efficiency is calculated as the electrical work divided by

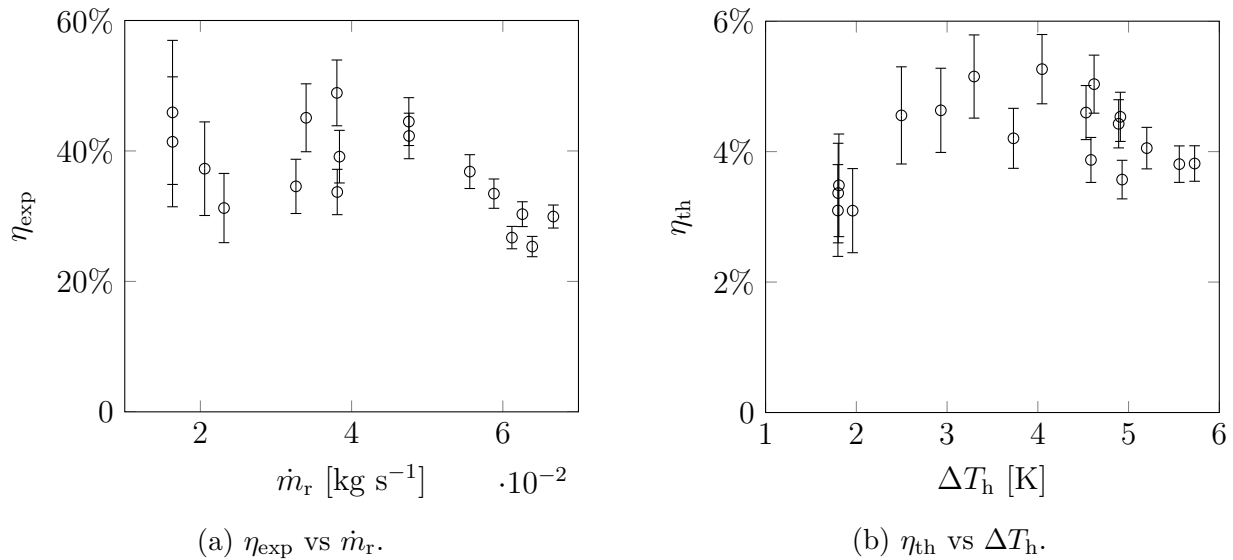


Figure 6.10: Experimental uncertainties indicated for the expander efficiency (associated with flow-rate measurement of the refrigerant working fluid) and ORC thermal efficiency (associated with differential temperature measurement of the thermal oil).

the isentropic enthalpy drop across the expander. At present the most significant source of uncertainty associated with the latter is the measurement of the mass flow-rate using the ultrasonic flow meter. In the preliminary experiments flow-rate readings are read periodically from the flow meter's digital display during each run and an average is taken; however the readings are subject to variation over time and several readings are required to give a confident estimate of the mean value for each steady-state run. The uncertainty in these results can be improved through automatic logging of the readings at a higher frequency sampling rate. Further improvement is also possible by direct measurement of mass flow-rate using a Coriolis-type mass flow meter, as is the practice in other ORC experimental studies [143, 147, 158, 209].

Flow-rate measurement in the thermal oil circuit also affects the uncertainty in the ORC thermal efficiency, however the main source of uncertainty at present (shown in Fig. 6.10b) is associated with the temperature measurement of the thermal oil at the inlet and outlet of the evaporator. The calculated heat input to the ORC is directly proportional to the temperature drop of the heat source for a given flow-rate, and therefore the relative uncertainty in the overall efficiency is more significant for smaller temperature differences. This uncertainty can be reduced significantly through improved calibration of the thermocouples for differential temperature measurement across the evaporator.

6.5 Dynamic Behaviour of the ORC System

Observations of the ORC system's dynamic behaviour can be used to further evaluate the S-CHP system model. Figure 6.11 shows the response of the system temperatures and evaporation pressure to a change in the system operating condition. At time 16:22 shown in the figure, the pump speed was increased from 2000 RPM to 3000 RPM (it should be noted that at the same time the oil heater set-point was also increased from 115 °C to 120 °C, although this is observed to have a less significant impact on the dynamic behaviour of the system). The change in pump speed causes a sudden increase in pressure measured at the evaporator, which is quickly absorbed by the system before a more gradual evolution to the new steady state, under which the increased pressure ratio and refrigerant flow-rate causes the ORC to extract more thermal power from the thermal oil loop. This change in the steady state takes approximately 35 minutes to fully occur, with the time constant of the thermal response calculated as 11 minutes (obtained using a similar expression to Eq. 5.8 in Section 5.3.2). Instantaneous power output was not logged during this period, however a gradual increase in power that followed the gradual increase in evaporator pressure was observed, from an initial steady state of 400 W to a final steady state of 550 W.

The implications of such findings are relevant for the S-CHP system model results from Chapter 4, Section 4.2, for which fast on-off switching behaviour of the ORC engine was observed over short time periods (of the order of once per minute in the worst case) under insufficient solar thermal input conditions. The model did not account for the dynamic response time due to the thermal mass of the ORC components (such as the expander, heat exchangers and working fluid) and the slow evolution of the pressures and temperatures with time as a result of the system inertia that are observed for the experimental system. The experimental results indicate that this dynamic behaviour may significantly affect the predictions of instantaneous power output for such systems. In order for the quasi-steady state sub-model of the ORC to be a reliable predictor of instantaneous power output under intermittent thermal input conditions, either the ORC system must be designed to have very low thermal inertia, or sufficient buffering of the thermal energy input must be applied up-stream so that the ORC is not subjected to

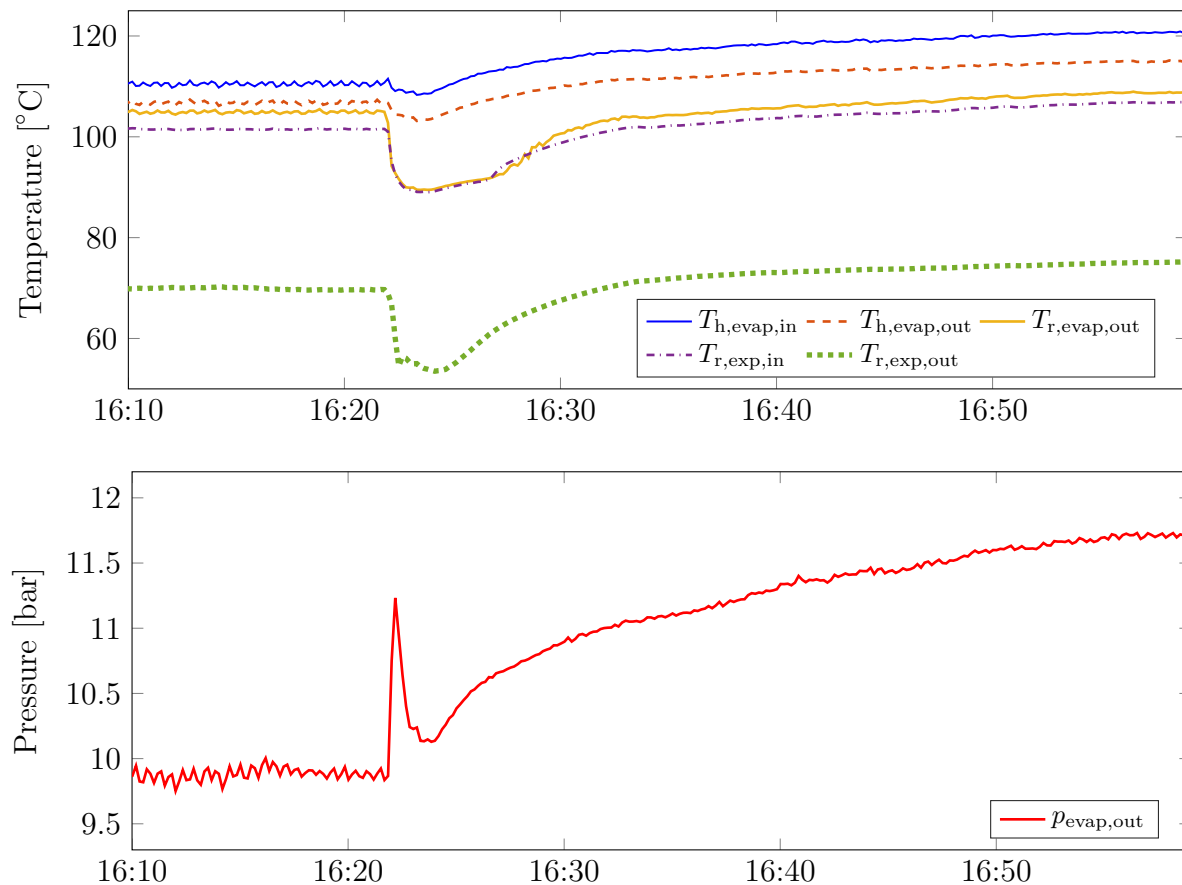


Figure 6.11: Response of the ORC system temperatures (top) and evaporator pressure (bottom) to a change in operating condition. At time 16:22, the pump speed is increased from 2000 RPM to 3000 RPM and the heat source set-point is also increased from 115 °C to 120 °C.

fast changes in the thermal input condition. As was shown in Chapter 4, when the S-CHP system model was modified to include a 25 L buffering vessel at the outlet of the solar collector array (similar to the volume of fluid in the thermal oil circuit of the experimental system), the period of the on-off switching was increased to between 15 minutes and 2 hours, which is more in line with the time-scale of the dynamic response witnessed for the experimental system.

6.6 Summary

The preliminary results from the ORC test bench have proved useful in understanding the operating parameters that influence the cycle efficiency and also in identifying various design improvements through which the efficiency can be improved. At present the ORC system is achieving just under 60% of its design efficiency with the 1 kW scroll expander, which is

operating below its nominal rotational speed; suggesting that a higher output power should also be achievable as a result of improvements. The scroll expander used for the preliminary testing obtained overall isentropic efficiencies of up to 49%. This is lower than the 68% (combined expander and generator efficiency) assumed for the S-CHP system, however the results suggest that this may be in some part due to off-design operational conditions (in particular a lower than expected evaporation pressure) and thermal losses from the expander. The range of expander efficiencies obtained so far do not show a clear relationship with the degree of under or over-expansion (with reference to the built-in volume ratio of the expander), suggesting that other loss mechanisms are dominating the overall expander losses in the preliminary tests. The maximum overall pump efficiency obtained is 27%, which is less than half of the 59% (combined pump and motor efficiency) assumed for the S-CHP system model. This results in a more significant impact on the net power output at present, with the pump consuming equivalent to $\sim 30\%$ of the gross electrical power output, compared to $< 7\%$ for the modelled system. If the rest of the system can be improved to achieve the nominal 1 kW from the expander, the pump power consumption will be a far lower percentage of the gross electrical output.

When the ORC sub-system model described in Chapter 3 is configured to represent the experimental system (using the measured temperatures, pressures and flow-rates as inputs), the gross electrical power output is over-predicted by 30 – 90%. Thermal losses and pressure losses in the evaporator, expander and intermediate pipework is thought to have a significant influence at present, although uncertainty in the refrigerant mass flow-rate measurement, and assumptions regarding mechanical and electrical losses in the generator may also make a further contribution. Further efforts to reduce thermal losses, pressure losses and experimental uncertainties should be addressed in future work.

The observations of the system's dynamic behaviour have revealed that the system takes several minutes to respond to changes in operational settings such as the pump speed. The thermal inertia of the system components is significant, and while this may result in longer times to achieve full power on start-up, it may also have some advantages for the buffering of short-duration dips in thermal energy input while reducing the requirement for additional

thermal storage for such a system operating with an intermittent heat source. These results have proven to be useful for the evaluation of the S-CHP system model, which at present does not include dynamic models of the ORC components.

Chapter 7

Conclusions

The aim of this research has been to investigate a small-scale system for the combined provision of electricity and hot water utilising solar thermal energy. In considering the technology for domestic applications, a system based on low-cost components and simple control strategies was proposed. The potential of the system for heat and power provision in northern European locations was of interest, in addition to that in more southerly locations where the viability of large-scale solar thermal power provision is already established. Non-concentrating (or low-concentration) stationary solar thermal collectors were of interest due to their suitability for small-scale applications and their ability to use diffuse irradiation. The organic Rankine cycle (ORC) was also chosen as the power-generation technology of interest for this study because of its demonstrated efficiency at low temperatures ($< 300\text{ }^{\circ}\text{C}$) compared to rival technologies such as the steam-Rankine and Stirling cycles. In order to achieve the main goal of the project, the following objectives were defined:

- to perform a maximum work analysis for a range of non-concentrating and concentrating solar thermal collectors under a range of solar irradiance conditions, and to extend this to an annual analysis in two comparison climates, London (UK) and Larnaca (Cyprus), and with an array size limited to 15 m^2 (typical of a UK domestic roof-space);
- to develop a thermo-economic model of a solar combined heat and power (S-CHP) system

based on sub-component models of the solar collector array, ORC engine and domestic hot water system;

- to use the system model to assess the performance and cost of the system in various configurations in the UK and Cyprus climates;
- to design and construct an experimental apparatus for the steady-state and dynamic testing of solar collectors under realistic outdoor conditions, in order to evaluate the various solar collector sub-models;
- to design and construct a small-scale ORC test bench for the testing and mapping of various ORC components over a range of conditions in order to evaluate the ORC sub-model.

7.1 Summary of Thesis Findings

The maximum work analysis of a range of solar collectors confirmed that for the UK, non-concentrating collectors offer a higher potential for work production over the annual period compared to concentrating collectors with single-axis tracking that can only make use of the direct-beam irradiance component. From an analysis of the climate data it was found that for London the annual global irradiance received by an optimally tilted plane is 46% lower than for Larnaca, and that the total diffuse fraction is significantly higher (50%, compared to 30%). In Larnaca, a parabolic trough collector tracking the sun on a single axis was found to offer a 15–20% improvement compared to the best non-concentrating collector. The maximum work determined from an endoreversible analysis of a non-concentrating evacuated flat plate collector (EFPC) operating in London was $67.2 \text{ kWh m}^{-2} \text{ yr}^{-1}$. For a 15 m^2 array this is equivalent to 1008 kWh yr^{-1} , or 115 W averaged over the annual period, and is 54% lower than the maximum annual work found for the same collector in Larnaca. It was assumed in the analysis that the collector is operated at its optimum temperature and flow-rate at each time instant.

The S-CHP system model was based on a configuration in which a heat transfer fluid

circulating in the collector array transfers heat to an ORC engine via a heat exchanger. R245fa was chosen as the ORC working fluid from a preliminary screening of pure refrigerant working fluids, based on thermodynamic performance and environmental considerations. Cogeneration of domestic hot water was provided using heat rejected from the cooling of the ORC expander exhaust vapour via a heat exchanger coil located in the bottom part of a domestic hot water tank. A further option for additional water heating at higher temperature was provided by diverting a variable fraction of the solar heat transfer fluid through a heat exchanger coil in the top of the hot water tank. The system was initially simulated with mains water as the heat rejection medium flowing through the ORC condenser.

When optimised for maximum power from a 15 m² EFPC array as the heat source, an ORC evaporation temperature of 130 °C was found to be optimal under clear-sky conditions using R245fa. This results in a system sized for a nominal expander power output of 1.37 kW. Under nominal design conditions, the temperature of the vapour exiting the expander is 86 °C, with the potential to provide a maximum of 1.8 kW for water heating if cooled down to the nominal condensation temperature of 17 °C. The system was also optimised for cloudy-day conditions, for which a lower evaporation temperature of 85 °C and a reduced nominal expander power output of 0.62 kW was found to provide the maximum daily work output with the EFPC array.

When simulated over an annual period in the London climate, a maximum annual net electrical work output of 628 kWh yr⁻¹ was predicted from the S-CHP system with the EFPC array, corresponding to an average power of 72 W, and a solar to electrical efficiency of 3.9%. This is equivalent to 19% of the typical annual household electrical demand in the UK, with a thermal output equivalent to 28% of the annual water heating demand. When simulated for the Larnaca climate, the maximum net electrical output was 1,628 kWh yr⁻¹, corresponding to 186 W average and a solar to electrical efficiency of 5.5%. This is equivalent to 33% of the typical Cyprus household electrical demand, with a thermal output equivalent to 58% of the annual water heating demand. Under hot water prioritisation conditions, an increased hot water demand coverage of 61% was demonstrated for the London simulation, however the electrical output decreased to 17% of the annual demand. The annual solar to electrical

efficiencies are low compared with PV (12–15% for mono-Si PV), however comparison with the endoreversible analysis showed that the S-CHP system is only achieving around 60% of the annual endoreversible work potential for the EFPC in the UK and around 74% in Cyprus. This provides motivation for further improvements in the design and operation of the system.

For the Cyprus simulation, climate conditions were found to exhibit less variation on a daily basis and therefore the maximum annual electrical performance was achieved with the optimal “clear-sky day” system settings. For the UK, the day-to-day climate conditions are far more variable, and achieving an annual electrical performance closer to the endoreversible target requires a system that can maintain collector operation at the optimal temperature by varying the operating conditions of the load (i.e. the ORC engine). Thus further consideration for the optimal control strategy (bearing in mind system cost and suitability for the domestic application) is required. The use of thermal energy storage in the collector fluid circuit to buffer instantaneous variations in solar irradiance was not modelled in detail in this work. This may offer some advantages in avoiding part-load operation and intermittent performance of the ORC, however this may also reduce the potential for achieving solar to electrical efficiencies approaching the endoreversible values under higher irradiance conditions that favour high temperature operation. An alternative is to hybridise the solar system with a secondary, higher-grade heat source (such as a biogas burner) to provide a steady expander inlet condition and a supplementary thermal input to the ORC during times of low solar irradiance.

Cost models were also developed based on a market analysis of the various system components. For the ORC, small-scale positive-displacement expander costs were determined from correlations for equivalent-sized refrigeration compressor prices. Solar collector system costs were found to vary over a wide range depending on location, and UK market prices were found to be higher than those in countries where markets for solar thermal systems are more established. Total installed cost estimates for the S-CHP system, sized for maximum electrical work output with EFPC collectors, was €9,700 for the UK and €10,500 for Cyprus. The investment costs for the electrical sub-system (separated from those associated with the basic solar hot water system) were found to account for 60–65% of the total investment cost. Compared with a side-by-side PV and hot-water system producing both electricity and hot water (divided

approximately 75% PV / 25% solar thermal) the total investment costs are similar, while compared to a hybrid PV-Thermal (PVT) system the cost is potentially 15–25% lower. In both cases the electrical output is still lower for the S-CHP system, however the hot-water output has the potential to be higher depending on how the S-CHP system is configured. The unique advantage of the S-CHP system is the ability to vary the proportion of hot water or electrical output through the diversion of the heat source fluid through the secondary heat exchanger coil in the hot water tank, and the potential to use thermal energy storage (at a potentially lower cost than battery storage for PV, for which the current market price of small 2 kWh_e domestic units was found to be upward of €2,000) to offset electricity generation from the ORC in order to meet a greater share of the local demand during peak hours.

An exergy analysis of the system revealed that downstream of the collector, the most significant component in terms of exergy destruction in the system was the evaporator. This suggests that further attention should be given to configurations for direct vapour generation in the solar collector, although it is noted for standard evacuated tube collectors (ETCs) this represents a significant departure from their intended use. Omitting the evaporator could also bring about a decrease in system costs, and a reduction in the response time of the ORC sub-system under transient solar irradiance conditions.

The use of water as the heat rejection medium, while exergetically favourable, leads to a large volume of water required over the annual period. This represents possibly an unsustainable use of a limited resource and a significant additional cost if purchased from the mains. Thus this may only be a viable option in rare cases where a large volume of water is easily available at low cost. The comparison with air as the heat rejection medium showed a reduced potential for work production during summer when air temperatures are higher and a further reduction in net work output due to the energy consumption associated with the required fan. The result is a significant (35–50%) reduction in the net work output that can be achieved from the S-CHP system.

An experimental apparatus was designed and constructed for the purpose of evaluating the solar collector in the S-CHP system. The steady-state solar collector experiments per-

formed with the ETC showed that the collector was able to operate reliably at temperatures significantly higher than its intended application range. The experiments were performed at a maximum inlet temperature of 112 °C, compared to 85 °C being the maximum testing temperature for solar hot water applications. The measured efficiency was 7–12% higher than the efficiency predicted by the collector efficiency curve at the highest inlet temperature. The dynamic collector testing meanwhile showed that various methods for determining the thermal capacity of the solar collector lead to a wide range of results. The standard method for estimating the effective thermal capacity involves using a weighted mean calculation based on information about the collector component materials. This method was found to provide an under-prediction of the thermal response time when compared to experimental methods. A diurnal simulation using a lumped collector model with the experimentally obtained effective thermal capacity value was compared to real data over a diurnal period under intermittent irradiance conditions. The model provided a reasonably realistic prediction of the collector's dynamic behaviour that was an improvement on the weighted-component value. The results were significant for the S-CHP system modelling, which by default used the thermal capacity value provided by the manufacturer and determined using the weighted-component value method.

An ORC test bench apparatus was also designed and constructed for the purpose of validating models of small-scale ORC components. Preliminary steady-state tests were performed using a 1 kW_e scroll expander, the refrigerant R245fa as the working fluid, and heat source temperatures in the range of 100–120 °C, which is also broadly representative of the ORC configuration in the S-CHP system model. A maximum gross thermal efficiency of 5.3% was obtained during the preliminary testing. The electrical output predicted by the ORC model using the experimental system parameters as input variables was between 30–90% higher than that obtained in the tests. Off-design operation of the expander, as well as significant thermal losses in the evaporator, expander and intermediate piping were identified as major factors in the deviations between predicted and achieved performance; however improvements to the experimental apparatus should result in closer agreement. The pump efficiency was also found to be significantly lower than the value assumed in the model, although the vane pump used

in the experiment was not specifically designed for the ORC application and organic working fluid. Observations were also made of the system's dynamic behaviour when subjected to a change in operating conditions. The response time was several minutes due to the significant thermal inertia associated with the heat exchangers, expander and working fluid. This suggests that for the S-CHP system simulated in a climate characterised by intermittent solar irradiance and without thermal storage, dynamic models of the ORC components are required for a more realistic prediction of system performance.

7.2 Summary of Thesis Outcomes

The important outcomes from the research are as follows:

- a whole-system computational platform, comprised of detailed sub-component models (including a solar-collector array, an ORC engine and a thermal store);
- two experimental apparatus, for validation of the solar collector array and ORC sub-system component models (to be performed in further work);
- three journal publications (to date) describing results from the model and from the maximum work analysis of the solar collectors;
- work presented at seven international conferences/seminars.

7.3 Future Work

Several areas of further work have been highlighted as a result of this research. These include investigations into alternative system configurations incorporating thermal storage or hybridisation with supplementary heat sources. This work did not consider the benefits of thermal storage in great detail, however an early-stage model of a conventional liquid thermocline store and a 1-D model of a solid-liquid phase change material store were developed, and these should

be implemented into future versions of the S-CHP system model. A comparison of thermal storage and battery storage should also be performed for the present application that takes into account the difference in the economic and exergetic values of electrical and thermal energy.

Polygeneration systems including cooling cycles are also of interest, and during the thesis a model was also developed of a small-scale diffusion absorption refrigeration (DAR) system using an ammonia-water working fluid. An experimental system was also acquired and some preliminary tests performed to assess the coefficient of performance over a range of heat source conditions. The DAR model and experimental apparatus can be used to investigate alternative working fluids and configurations for integration with a solar heat source in a CCP or CCHP configuration.

The experimental ORC apparatus can be used for further testing of small-scale components, including a reciprocating expander that is currently under development. The next steps for the development of the ORC test bench in its present configuration (with the scroll expander) should include improvements to the insulation of the expander and pipework, and further testing to determine the maximum power achievable for the present configuration. Following this, modifications should be made to the apparatus to improve the layout of the components, especially the pump and expander. The length of piping between the evaporator and expander should be reduced in order to reduce the pressure and temperature losses observed between the evaporator outlet and expander inlet, while the pump installation should be modified to give a greater net positive suction head at the inlet in an effort to reduce the necessary subcooling.

The solar collector experimental apparatus can be used to test alternative collector designs and to investigate performance under low irradiance and intermittent irradiance conditions. The steady-state test results presented in this work were at higher irradiance conditions, comparable to those specified in the standard testing procedures. However, test data under low irradiance conditions are rarely presented for conventional water-heating collectors and this is important in evaluating their performance for locations such as the UK. Some data was already gathered under low irradiance conditions, however due to time constraints it was not possible to collect data over a wide enough range of fluid temperatures to map the performance sufficiently.

Finally, the potential to use solar collectors for direct vapour generation with organic fluids has also been highlighted as an area for further investigation. Although this subject is discussed in a number of papers there are no known operational systems in the solar ORC literature, and there would appear to be a need for an experimental effort in this area.

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Appendix A

Solar calculations

A.1 Calculations for solar irradiance of a tilted surface

The calculations are performed using hourly solar irradiance data and using the equations provided in Chapter 2 of Kalogirou [122]. The total irradiance incident on a tilted surface is calculated as the sum of the beam, diffuse and ground-reflected components:

$$G_t = G_{b,t} + G_{d,t} + G_{G,t}, \quad (\text{A.1})$$

$$G_t = (G_{b,h} + G_{d,h}A) R_b + G_{d,h} (1 - A) \left[\frac{1 - \cos(\beta)}{2} \right] \left[1 + \sqrt{\frac{G_{b,h}}{G_{d,h} + G_{d,h}}} \sin^3 \left(\frac{\beta}{2} \right) \right] + (G_{b,h} + G_{d,h}) \rho_g \left[\frac{1 - \cos(\beta)}{2} \right]. \quad (\text{A.2})$$

where $G_{b,h}$ and $G_{d,h}$ are the horizontal beam and diffuse radiation components, β is the surface inclination angle from the horizontal, A is the anisotropy index, and ρ_g is the ground reflectance, and R_b , is the beam tilt radiation factor. A typical value of 0.2 is assumed for ρ_g [122]. The anisotropy index A is calculated as the ratio of the direct normal irradiance to the normal extraterrestrial radiation:

$$A = \frac{G_{b,n}}{G_{o,n}}. \quad (\text{A.3})$$

The beam tilt radiation factor R_b is calculated as follows:

$$R_b = \frac{\sin(L - \beta) \sin(\delta) + \cos(L - \beta) \cos(\delta) \cos(h)}{\sin(L) \sin(\delta) + \cos(L) \cos(\delta) \cos(h)}. \quad (\text{A.4})$$

where L is the latitude angle, δ is the declination angle and h is the hour angle.

A.2 Incident angle modifier calculations

For the incident angle modifier calculations, the incidence angle is calculated using the following expression:

$$\begin{aligned} \cos(\theta) = & \sin(L) \sin(\delta) \cos(\beta) - \cos(L) \sin(\delta) \sin(\beta) \cos(Z_s) \\ & + \cos(L) \cos(\delta) \cos(h) \cos(\beta) \\ & + \sin(L) \cos(\delta) \cos(h) \sin(\beta) \cos(Z_s) \\ & + \cos(\delta) \sin(h) \sin(\beta) \sin(Z_s). \end{aligned} \quad (\text{A.5})$$

where Z_s is the surface azimuth angle, relative to due south.

The solar incidence angle θ is also used to calculate the correction factor known as the incidence angle modifier (IAM). This IAM is used to modify the optical efficiency of the collector η_0 for non-perpendicular incident beam angles, and is usually measured in longitudinal and transversal planes relative to the collector for non-symmetrical collectors. The following equations are used to obtain the longitudinal and transversal projections of the incidence angle:

$$\theta_{lo} = | \text{atan} \{ \tan(\Phi) \cdot \cos(h - Z_s) \} - \beta |, \quad (\text{A.6})$$

$$\theta_{tr} = \left| \text{atan} \left\{ \frac{\sin(\Phi) \cdot \sin(h - Z_s)}{\cos(\theta)} \right\} \right|, \quad (\text{A.7})$$

where Φ is the solar zenith angle. The overall incidence angle modifier for beam radiation is then obtained as follows:

$$K_{\theta b} = K_{\theta b, lo} \cdot K_{\theta b, tr}. \quad (\text{A.8})$$

Appendix B

Numerical model methodology comparison

In an early version of the S-CHP system model, a numerical time-marching (TM) method was used to solve heat and mass balance equations in the solar collector circuit (see Chapter 3) and to obtain the response of the overall S-CHP system. For the purposes of model evaluation, the solution from this method was compared to a quasi-steady (QS) equivalent solution, which neglects the thermal inertia of the collector but not of the hot water tank that continues to obey the model described in Section 3.2.4. For simplicity, the ORC engine is omitted from the system model for this analysis and all of the fluid leaving the collector is passed through the hot water tank heating coil. Under the TM method, the solar collector fluid temperature exiting the hot water tank heating coil at time instant i , $T_{\text{coil,out}}(i)$, becomes the entering temperature into the collector at the next time instant $i + 1$, $T_{\text{sc,in}}(i + 1)$. The QS method involves solving the heat fluxes and temperatures in all loops and components such that the whole system remains in steady-state at each time instant. This method effectively assumes that changes in the external/boundary conditions influencing the system are slow in comparison to the internal dynamics of the system itself. The steady-state solution then varies with time as it responds to the time-varying external/boundary conditions imposed on the system. It is expected that the QS solution should closely approximate the TM solution, providing that the time-step interval

over which the TM calculations are performed is suitably small and the above QS assumption is valid. The TM and QS models were compared for various time-step intervals; in Fig. B.1 the simulation results are plotted for a time-step interval of 60 s showing strong agreement between the two models for the predicted temperatures in the system.

As a further test, the following model parameters were adjusted: (1) the length of the time-step Δt ; and (2) the number of discrete spatial elements n_{sc} into which the collector *array* is split (equivalent to splitting the array into a number of individual collector modules connected in series), while keeping the total collector area constant. In fact, the two parameters are closely linked within the set-up of the model. As the number of elements (i.e. collector modules) n_{sc} that make up the array is increased, the mass associated with each element decreases and therefore the time-step length Δt required to avoid calculation instability also decreases. For this reason, the time-step and number of collector elements are varied proportionally, as can be seen in Fig. B.2 where the results are shown. Based on a total available area of 15 m² and a 1.2 m² area for the Microtherm SK-6 ETC collector, the maximum number of in-series collector modules is 12. When the array resolution is increased beyond 9 elements, it is found that there is very little change in the system temperatures predicted by the model. For the single-element array there is a slight under-prediction of the solar collector temperature compared to the higher

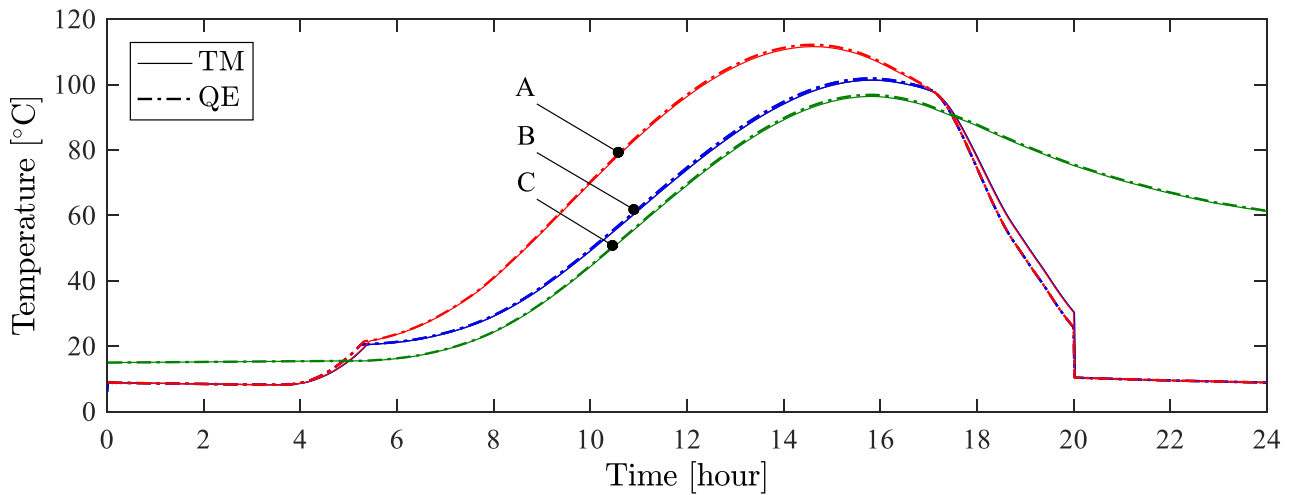


Figure B.1: Collector array inlet temperature $T_{sc,in}$ (“A” in the figure), outlet temperature $T_{sc,out}$ (“B”), and hot water tank temperature T_{tank} (“C”). Compared for simulations using time-marching (TM) and quasi-steady (QS) approaches. ETC collector used with “annual average day” climatic conditions (see Section 4.2). Time-step interval $\Delta t = 60$ s.

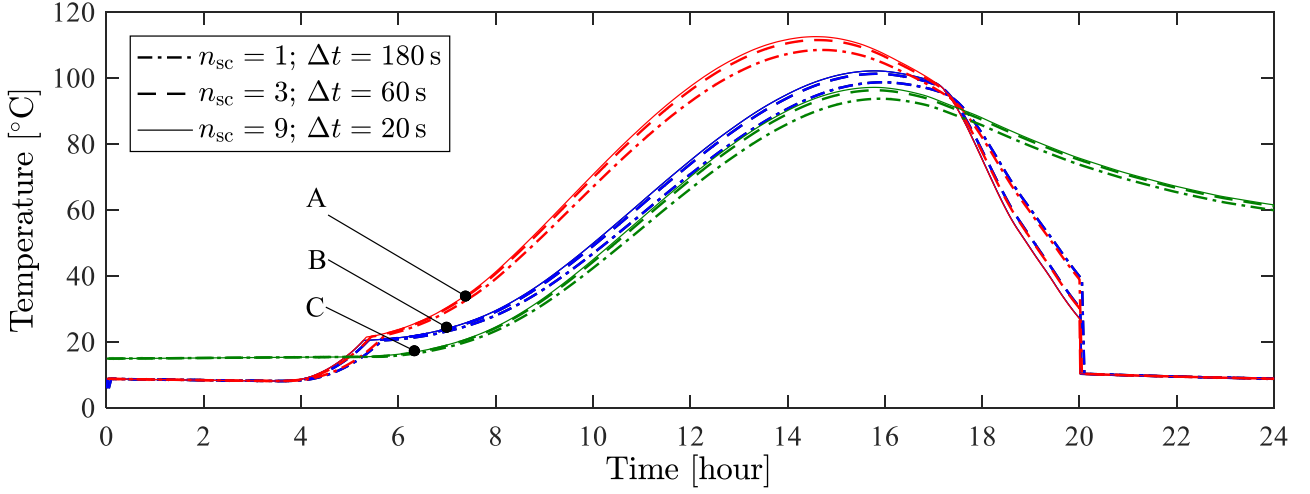


Figure B.2: Collector array inlet (“A”), outlet (“B”) and hot water tank (“C”) temperatures for model variants with increasing resolution of collector array finite elements n_{sc} and decreasing size of time-step interval Δt . Collector mass flow-rate $\dot{m}_{sc} = 0.03$ kg/s.

resolution arrays. The average difference in collector temperature between the single-element and the 9-element model is 2.7°C . This is equal to 2.6% of the amplitude of the daily collector temperature variation, which is deemed to be an acceptable error margin. Therefore, due to the significant reduction in calculation time, the single-element model was considered suitable for annual simulation purposes.

Finally, a further detail that must be considered is the effect of increasing the solar collector flow-rate on the calculation stability. It is generally observed that the calculation becomes unstable when $\dot{m}_{sc}\Delta t \geq M_{sc}$. For the single-element collector array model, a time-step interval of 60 s is found to be effective at providing a stable calculation output for $\dot{m}_{sc} \leq 0.1$ kg/s. However, at higher flow-rates the calculations become unstable and therefore the time-step interval is decreased as necessary to maintain stability.

Appendix C

Working fluid assessment

In this section, suitable pure working fluids will be considered for further investigation as ORC working fluids in the the S-CHP system model.

C.0.1 Methodology

The assessment will be performed by modelling the performance of the working fluids in a simplified solar-ORC system model under steady state conditions representing a “high irradiance” and “low irradiance” condition. A simple non-recuperative ORC architecture is used for the analysis. Fixed values of 65% and 75% are chosen for the isentropic efficiencies of the pump and expander respectively. The ORC working fluid is heated in a counter-flow heat exchanger by a hot fluid stream leaving the solar collector. The calculations were performed for a steady-state condition at a fixed solar irradiance and air temperature condition. After exchanging heat with the ORC working fluid, the hot fluid stream is returned to the solar collector array for further heating. The collector is modelled using the simplified steady-state collector efficiency equation given in Eq. 2.2, and uses the same efficiency-curve coefficients obtained empirically in the collector testing procedure.

It is assumed that heat is rejected from the cycle to a cold fluid stream with an inlet temperature of 20 °C. The condenser heat exchanger is assumed to have a minimum approach

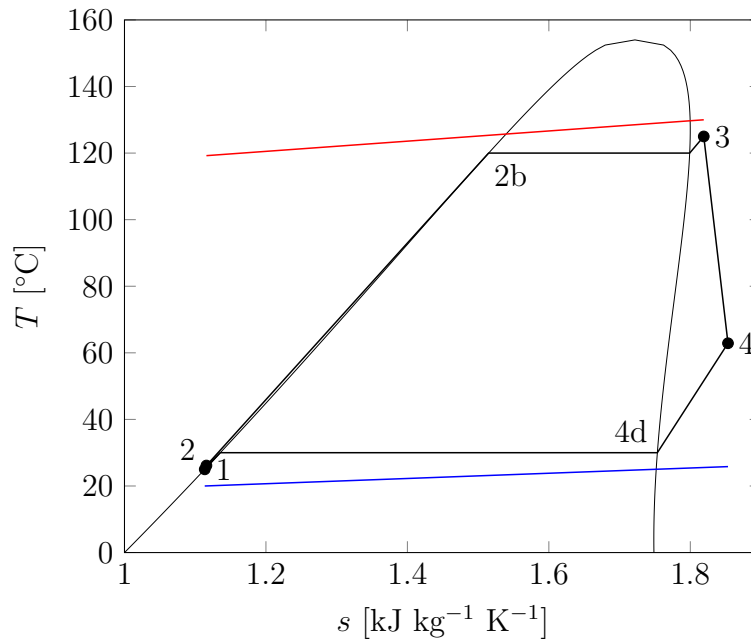


Figure C.1: T - s diagram showing the basic ORC processes in the cycle analysis.

temperature difference at the cold fluid inlet of 5 K, and the ORC working fluid is also assumed to be subcooled by 5 K at the inlet to the pump (State 1 in Fig. C.1); thus the condensation temperature for the cycle is fixed at 30 °C. The minimum condenser pinch-point temperature difference occurring at the working fluid dew-point (State 4d) is chosen as 5 K, which allows the calculation of the cold fluid heat capacity flow-rate. The evaporator heat exchanger is also assumed to have a minimum approach temperature difference of 5 K at the hot fluid inlet and a minimum pinch-point temperature difference occurring at the working fluid bubble-point (State 2b) is also chosen as 5 K.

A list of 121 pure working fluids from the NIST REFPROP library [183] will be screened for their suitability. The fluids in the library can be divided into the following categories:

- hydrocarbons (HC);
- fluorocarbons (FC);
- chlorofluorocarbons (CFC);
- hydrochlorofluorocarbons (HCFC);
- hydrofluorocarbons (HFC);

- refrigerant ethers (RE);
- natural refrigerants, such as ammonia and carbon dioxide;
- siloxanes;
- xylenes;
- fatty acid methyl esters (FAME);
- cryogenics;
- noble elements;
- water;
- miscellaneous substances including ethanol and methanol.

An initial screening of the fluids is performed as follows: for a minimum cycle evaporation temperature of $T_{\text{evap},\text{min}} = 40^\circ\text{C}$, it is required that the corresponding evaporation pressure is both greater than or equal to atmospheric pressure (taken as $p_{\text{atm}} = 101.3 \text{ kPa}$) and less than the critical pressure of the working fluid.

$$p_{\text{atm}} \leq p_{\text{evap},\text{min}} < p_{\text{cr}} . \quad (\text{C.1})$$

A maximum pressure limitation in the cycle is defined as 30 bar, or the critical pressure of the working fluid if this is lower than 30 bar:

$$p_{\text{evap},\text{max}} = \min \{p_{\text{cr}}, 30 \text{ bar}\} . \quad (\text{C.2})$$

For practical reasons it is also desirable that the condensation pressure in the cycle is above atmospheric pressure to prevent ingress of air into the system through leakage routes. Thus, if the default condensation temperature of 30°C is found to result in a sub-atmospheric condensation pressure, then a higher condensation temperature corresponding to atmospheric pressure is chosen:

$$p_{\text{cond}} = \max \{p_{\text{atm}}, p_{\text{cond}}(T_{\text{cond}} = 30^\circ\text{C})\} . \quad (\text{C.3})$$

An optimisation of evaporation pressure/temperature and expander inlet temperature is then performed for each working fluid, with the objective of maximising the overall efficiency of the system within the constraints defined above. The overall efficiency is here defined as the net work output per unit solar energy received, or the product of the solar collector efficiency and the ORC efficiency:

$$\eta_{ov} = \frac{\dot{W}_{out} - \dot{W}_{in}}{GA_{sc}} = \eta_{sc} \cdot \eta_{orc} . \quad (C.4)$$

C.0.2 Results and discussion

The optimisation is performed for the peak irradiance (1000 W m^{-2}) and the low irradiance (250 W m^{-2}) conditions, and using the properties of the EFPC and ETC-DF collectors listed in Table 2.3. The maximum overall efficiency is reported in Tables C.1 and C.2, as well as the corresponding optimal operating conditions determined for each working fluid including evaporation temperature T_{evap} , expander inlet temperature T_3 , expansion ratio r_v and the required collector array area per kW of net power output A_{sc} . Depending on the shape of the fluid saturation curve, the expansion process may result in a so-called “wet” expansion into the two-phase region. Thus the vapour quality exiting the expander x_4 is also reported.

Of the 121 fluids in the REFPROP database, 70 were found to perform with a maximum overall efficiency greater than zero. The top 20 fluids are listed in order of maximum overall efficiency in Tables C.1 and C.2. The relative range in overall efficiency between the fluids is found to be largest for the EFPC collector in the high irradiance case. The fluids that perform with the highest maximum efficiency across the range of cases are the HCFC fluid R21 (dichlorofluoromethane) and the CFC fluid R11 (trichlorofluoromethane). At peak irradiance, R21 is shown to provide maximum efficiency with a larger amount of superheat than R11 and has a lower critical temperature and lower evaporation temperature corresponding to the maximum pressure limitation. Under low irradiance conditions all fluids are found to operate below the maximum pressure limitations and therefore the required superheating for the maximum overall efficiency condition is minimal. Consequently, the optimal evaporation temperature range across the range of working fluids is also found to be much smaller than for the high irradiance

Table C.1: Solar-ORC performance parameters for maximum overall efficiency, calculated for an array of direct-flow ETC (Thermomax DF 100) collectors. Listed for the 20 highest performing pure working fluids from the REFPROP database.

(a) Peak irradiance, $G = 1000 \text{ W m}^{-2}$

	fluid type	η_{ov} %	η_{sc} %	η_{orc} %	T_{crit} °C	T_3 °C	T_{evap} °C	T_{cond} °C	p_{evap} bar	r_v -	A_{sc} m ² /kW	x_4 -
R21	HCFC	5.93	36.5	16.2	178.3	185.6	142.4	30	29.3	12.5	16.9	> 1
R11	CFC	5.90	38.8	15.2	198.0	151.0	135.7	30	16.5	12.9	16.9	> 1
R141b	HCFC	5.76	39.2	14.7	204.3	140.6	134.8	32.0	13.7	13.7	17.4	> 1
cis-butene	HC	5.66	38.2	14.8	162.6	171.8	141.3	30	30	13.1	17.7	> 1
R123	HCFC	5.64	39.2	14.4	183.7	149.6	135.2	30	16.1	15.7	17.7	> 1
R1233zd(E)	HFC	5.58	36.6	15.3	165.6	193.5	155.3	30	30	21.6	17.9	> 1
trans-butene	HC	5.56	39.1	14.2	155.5	167.2	137.9	30	30	12.3	18.0	> 1
sulfur dioxide	misc	5.51	45.6	12.1	157.5	103.8	103.8	30	30	5.9	18.2	0.90
butene	HC	5.43	40.9	13.3	146.1	157.4	128.9	30	30	9.8	18.4	> 1
R245fa	HFC	5.43	38.9	13.9	154.0	174.0	143.2	30	30	19.3	18.4	> 1
R245ca	HFC	5.41	39.6	13.7	174.4	149.2	134.8	30	19.3	17.9	18.5	> 1
isopentane	HC	5.38	40.9	13.2	187.2	127.4	127.4	30	12.5	13.0	18.6	> 1
R142b	HCFC	5.37	42.3	12.7	137.1	153.3	119.9	30	30	8.3	18.6	> 1
isobutene	HC	5.37	41.2	13.0	144.9	154.9	127.4	30	30	9.6	18.6	> 1
butane	HC	5.36	39.6	13.5	152.0	163.4	137.7	30	30	12.5	18.7	> 1
RE245fa2	RE	5.29	39.9	13.3	171.7	146.8	133.9	30	17	19.0	18.9	> 1
diethyl ether	misc	5.28	39.9	13.2	193.5	133.8	132.2	34.4	12.8	14.1	18.9	> 1
R114	CFC	5.27	40.2	13.1	145.7	173.8	140.9	30	30	14.2	19.0	> 1
CF3I	misc	5.24	43.5	12.0	123.3	156.2	107.3	30	30	5.2	19.1	> 1
R40	misc	5.21	43.4	12.0	143.1	155.3	95.8	30	30	3.9	19.2	> 1

(b) Low irradiance, $G = 250 \text{ W m}^{-2}$

	fluid type	η_{ov} %	η_{sc} %	η_{orc} %	T_{crit} °C	T_3 °C	T_{evap} °C	T_{cond} °C	p_{evap} bar	r_v -	A_{sc} m ² /kW	x_4 -
sulfur dioxide	misc	2.29	30.2	7.58	157.5	68.7	68.7	30	13.8	2.7	174.6	0.95
R11	CFC	2.28	30.2	7.54	198.0	68.7	68.7	30	4.0	3.0	175.5	> 1
R21	HCFC	2.28	30.3	7.52	178.3	68.7	68.7	30	6.5	2.9	175.7	0.99
R40	misc	2.23	30.4	7.35	143.1	68.5	68.5	30	17.2	2.5	179.0	0.95
ammonia	nat. refr	2.23	32.1	6.96	132.2	65.7	65.7	30	30.0	2.4	179.0	0.93
R123	HCFC	2.23	30.5	7.31	183.7	68.4	68.4	30	3.6	3.3	179.7	> 1
cis-butene	HC	2.21	30.5	7.25	162.6	68.3	68.3	30	7.1	2.8	180.9	> 1
R1233zd(E)	HFC	2.19	30.6	7.16	165.6	68.2	68.2	30	4.9	3.2	182.5	> 1
trans-butene	HC	2.19	30.6	7.15	155.5	68.2	68.2	30	7.6	2.8	182.8	> 1
propyne	HC	2.18	30.7	7.12	129.2	68.1	68.1	30	17.3	2.6	183.2	0.98
R245ca	HFC	2.18	30.7	7.12	174.4	68.1	68.1	30	4.1	3.4	183.2	> 1
isopentane	HC	2.18	30.7	7.11	187.2	68.0	68.0	30	3.4	3.1	183.3	> 1
R141b	HCFC	2.16	29.6	7.31	204.3	69.8	69.8	32	3.2	3.1	184.9	> 1
R245fa	HFC	2.16	30.8	7.02	154.0	67.9	67.9	30	5.8	3.3	185.2	> 1
RE245fa2	RE	2.16	30.8	7.01	171.7	67.9	67.9	30	3.6	3.5	185.3	> 1
butene	HC	2.16	30.7	7.03	146.1	68.0	68.0	30	9.2	2.7	185.4	> 1
butane	HC	2.15	30.7	6.99	152.0	67.9	67.9	30	7.7	2.8	186.1	> 1
isobutene	HC	2.15	30.7	6.98	144.9	67.9	67.9	30	9.4	2.7	186.4	> 1
R142b	HCFC	2.14	30.8	6.97	137.1	67.9	67.9	30	10.6	2.8	186.6	> 1
cyclopropane	HC	2.13	30.8	6.92	125.1	67.9	67.9	30	19.9	2.4	187.9	0.98

Table C.2: Solar-ORC performance parameters for maximum overall efficiency, calculated for an array of EFPC (TVP Solar MT-Power) collectors. Listed for the 20 highest performing pure working fluids from the REFPROP database.

(a) Peak irradiance, $G = 1000 \text{ W m}^{-2}$

	fluid type	η_{ov} %	η_{sc} %	η_{orc} %	T_{crit} °C	T_3 °C	T_{evap} °C	T_{cond} °C	p_{evap} bar	r_v -	A_{sc} m ² /kW	x_4 -
R21	HCFC	8.76	52.7	16.6	178.3	201.3	143.9	30	30	12.6	11.4	> 1
R11	CFC	8.75	48.4	18.1	198.0	221.4	171.9	30	30	23.2	11.4	> 1
R141b	HCFC	8.45	50.9	16.6	204.3	187.0	159.7	32.0	21.2	21.4	11.8	> 1
R123	HCFC	8.37	50.3	16.6	183.7	213.9	171.3	30	30	29.7	11.9	> 1
R1233zd(E)	HFC	8.20	53.7	15.3	165.6	196.1	155.3	30	30	21.4	12.2	> 1
cis-butene	HC	8.19	55.2	14.8	162.6	175.8	141.3	30	30	13.0	12.2	> 1
trans-butene	HC	7.98	56.1	14.2	155.5	170.5	137.9	30	30	12.2	12.5	> 1
R245ca	HFC	7.95	53.0	15.0	174.4	190.1	158.7	30	30	28.3	12.6	> 1
RE245fa2	RE	7.86	53.3	14.7	171.7	198.5	164.1	30	30	34.8	12.7	> 1
R245fa	HFC	7.81	56.0	13.9	154.0	175.3	143.2	30	30	19.2	12.8	> 1
sulfur dioxide	misc	7.74	56.5	13.7	157.5	196.8	103.8	30	30	5.2	12.9	> 1
isopentane	HC	7.74	53.6	14.4	187.2	164.2	150.7	30	18.9	20.5	12.9	> 1
butene	HC	7.70	57.7	13.3	146.1	161.1	128.9	30	30	9.7	13.0	> 1
diethyl ether	misc	7.69	51.8	14.8	193.5	180.2	159.5	34.4	21.0	23.8	13.0	> 1
butane	HC	7.67	56.6	13.5	152.0	164.7	137.7	30	30	12.4	13.0	> 1
isobutene	HC	7.58	58.1	13.1	144.9	158.1	127.4	30	30	9.5	13.2	> 1
R142b	HCFC	7.53	58.9	12.8	137.1	159.5	119.9	30	30	8.1	13.3	> 1
R114	CFC	7.50	57.2	13.1	145.7	174.7	140.9	30	30	14.1	13.3	> 1
pentane	HC	7.49	53.0	14.1	196.5	161.8	152.1	36.1	16.5	19.1	13.4	> 1
cyclopentane	HC	7.45	50.7	14.7	238.6	161.3	160.3	49.2	14.0	14.8	13.4	> 1

(b) Low irradiance, $G = 250 \text{ W m}^{-2}$

	fluid type	η_{ov} %	η_{sc} %	η_{orc} %	T_{crit} °C	T_3 °C	T_{evap} °C	T_{cond} °C	p_{evap} bar	r_v -	A_{sc} m ² /kW	x_4 -
sulfur dioxide	misc	4.49	44.6	10.06	157.5	86.2	86.2	30	20.7	4.1	89.2	0.92
R11	CFC	4.47	44.4	10.08	198.0	86.6	86.6	30	6.1	4.6	89.5	> 1
R21	HCFC	4.46	44.5	10.02	178.3	86.3	86.3	30	9.9	4.3	89.7	0.99
R40	misc	4.34	45.0	9.65	143.1	85.4	85.4	30	24.5	3.5	92.1	0.92
R123	HCFC	4.34	44.7	9.69	183.7	85.9	85.9	30	5.7	5.2	92.2	> 1
R141b	HCFC	4.31	43.8	9.84	204.3	87.6	87.6	32	5.1	4.8	92.8	> 1
cis-butene	HC	4.30	44.9	9.57	162.6	85.6	85.6	30	10.5	4.3	93.1	> 1
R1233zd(E)	HFC	4.25	45.0	9.44	165.6	85.4	85.4	30	7.5	5.0	94.1	> 1
trans-butene	HC	4.24	45.1	9.41	155.5	85.3	85.3	30	11.2	4.3	94.3	> 1
R245ca	HFC	4.22	45.1	9.37	174.4	85.3	85.3	30	6.5	5.5	94.7	> 1
isopentane	HC	4.22	45.1	9.36	187.2	85.2	85.2	30	5.2	4.9	94.8	> 1
propyne	HC	4.21	45.6	9.22	129.2	84.3	84.3	30	24.5	3.8	95.1	0.97
butene	HC	4.17	45.3	9.19	146.1	84.9	84.9	30	13.2	4.1	96.0	> 1
R245fa	HFC	4.16	45.3	9.19	154.0	84.9	84.9	30	8.9	5.3	96.0	> 1
RE245fa2	RE	4.16	45.3	9.19	171.7	85.0	85.0	30	5.7	5.7	96.1	> 1
butane	HC	4.15	45.3	9.15	152.0	84.9	84.9	30	11.2	4.2	96.5	> 1
isobutene	HC	4.14	45.4	9.11	144.9	84.8	84.8	30	13.5	4.1	96.7	> 1
R142b	HCFC	4.13	45.5	9.06	137.1	84.5	84.5	30	15.2	4.1	96.9	> 1
cyclopropane	HC	4.08	45.8	8.91	125.1	83.9	83.9	30	27.3	3.4	97.9	0.97
dimethylether	misc	4.08	45.9	8.89	127.2	83.9	83.9	30	24.1	3.7	98.0	0.99

case. Sixteen fluids are identified that appear in the the top 20 for both the high and low irradiance cases with ETC and EFPC collectors. This short-list can be shortened by omitting fluids such as sulfur dioxide with negative saturated vapour lines such that significant superheating is required to avoid a wet expanded vapour in the low irradiance case. Fluids not cited in ORC or refrigeration literature are also omitted (the alkenes butene, isobutene, cis-butene and trans-butene). This leaves the hydrocarbons butane (R600) and isopentane (R601a); the HCFC fluids R21, R123, R141b and R142b; the HFC fluids R245fa, R245ca and R1233zd(E); the CFC R11 and the refrigerant ether RE245fa2.

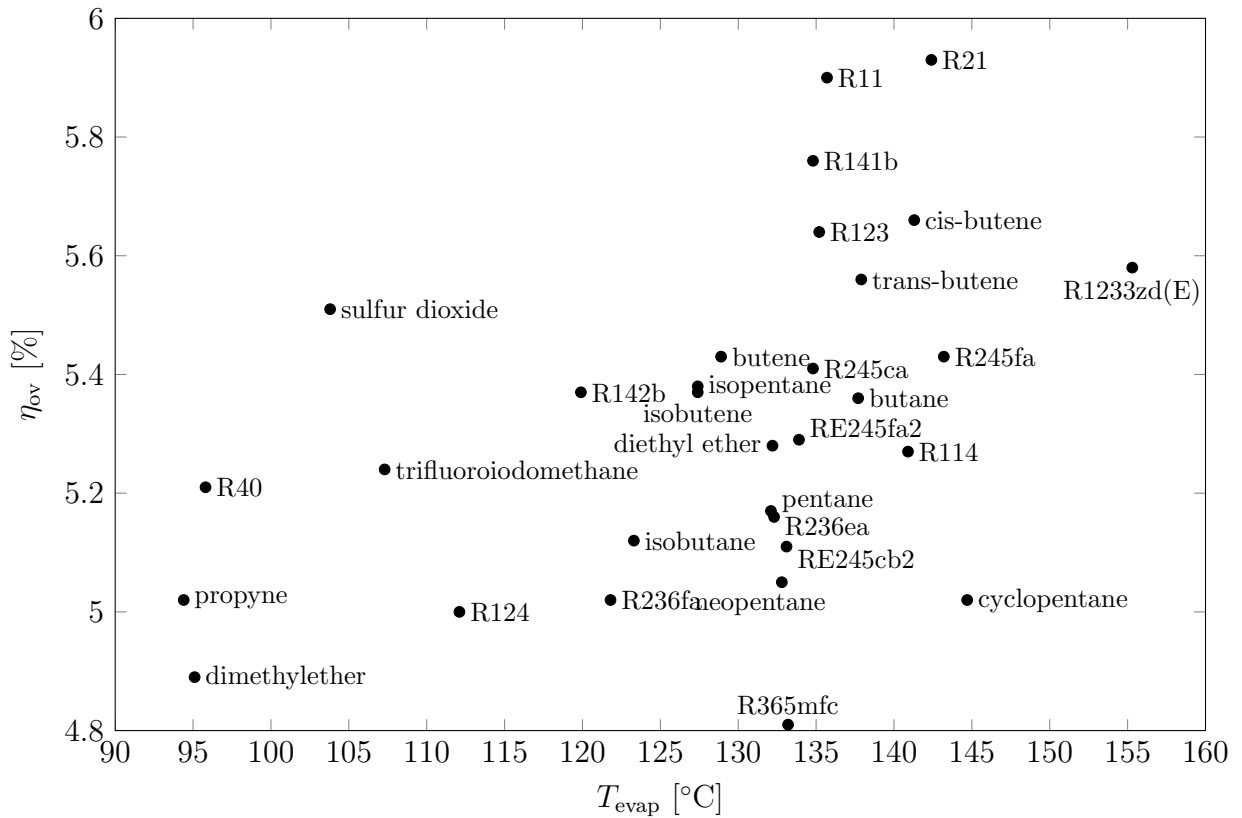
In Table C.3 the short-listed fluids are compared in terms of ozone-depletion potential (ODP), global warming potential (GWP) and atmospheric lifetime (ALT). The CFC and HCFC fluids all have non-zero ODP values, and are therefore phased out or scheduled for phase-out as a result of the Montreal Protocol and subsequent EU Ozone regulation.

C.0.3 Conclusion and recommendation for the present study

The two fluids chosen for further consideration in this study are the HFC fluids R245fa and R1233zd(E). The latter fluid is a an unsaturated HFC (hydrofluoroolefin, or HFO) and as such has a low GWP value compared to the other saturated HFCs in Table C.3. The fluid is relatively new on the market for refrigeration applications and as such is not widely available. By comparison, R245fa is more widely available and well established as a working fluid in low-temperature ORC systems, although it has a far higher GWP. The two fluids are comparable in terms of market price at approximately 30 £/kg in the UK [169]. R1233zd(E) is found to have a better overall performance in the solar ORC-sytem than R245fa at higher temperatures. For the peak irradiance case in Table C.2, R1233zd(E) is within 7% of the benchmark HCFC fluid R21, whereas R245fa is within 11%.

Table C.3: Environmental characteristics of shortlisted ORC working fluids.

	fluid type	ODP -	GWP -	ALT years	Ref.
R11	CFC	1	4660	45	[210, 211]
R21	HCFC	0.04	148	1.7	[211]
R123	HCFC	0.01	79	1.3	[210, 211]
R1233zd(E)	HFC	0	1	0.07	[212]
R141b	HCFC	0.12	782	9.2	[210, 211]
R142b	HCFC	0.06	1980	17.2	[210, 211]
R245ca	HFC	0	716	6.2	[211]
R245fa	HFC	0	858	7.7	[211, 213]
RE245fa2	RE	0	286	2.2	[214]
isopentane	HC	0	< 25	12±3	[213]
butane	HC	0	< 25	12±3	[213]

Figure C.2: Maximum overall efficiency plotted against ORC evaporation temperature for the range of fluids investigated. Shown for the ETC array and $G = 1000 \text{ W m}^{-2}$.

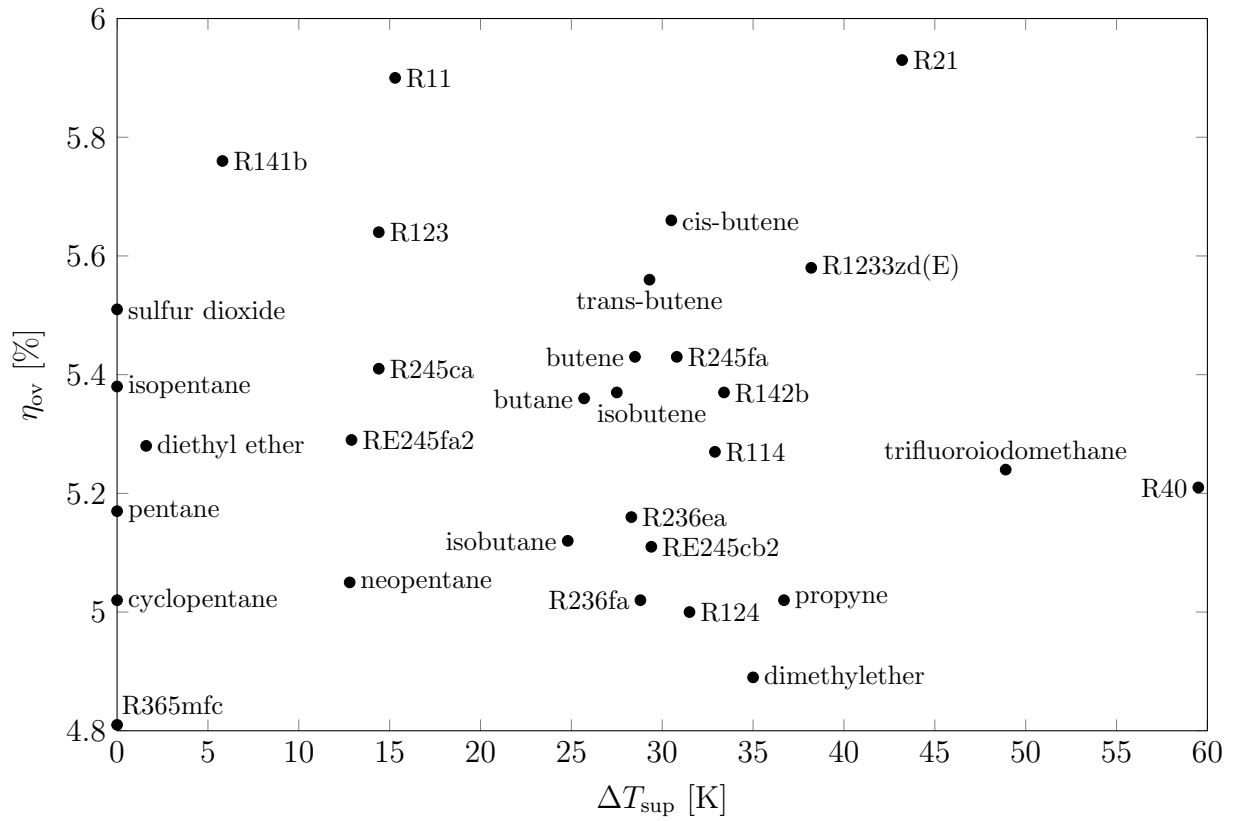


Figure C.3: Maximum overall efficiency plotted against degree of superheating for the range of fluids investigated. Shown for the ETC array and $G = 1000 \text{ W m}^{-2}$.

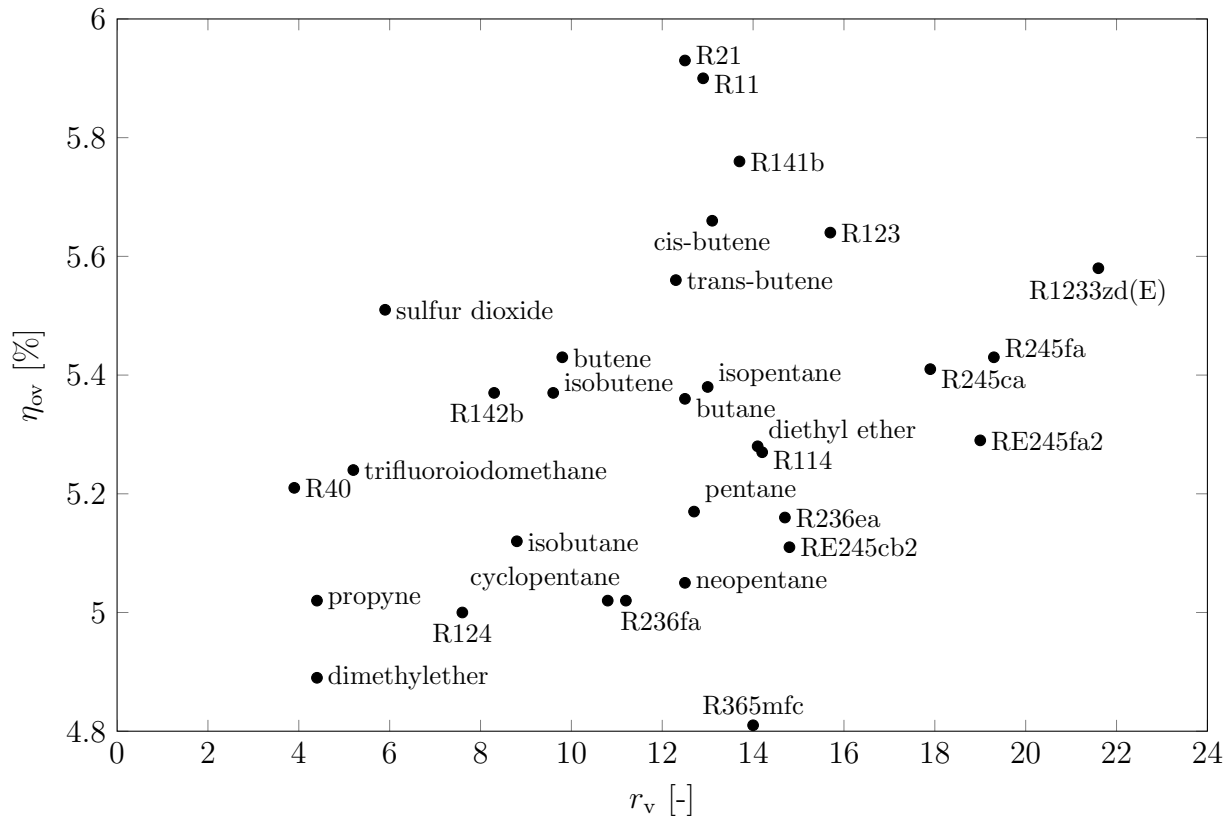


Figure C.4: Maximum overall efficiency plotted against expansion ratio for the range of fluids investigated. Shown for the ETC array and $G = 1000 \text{ W m}^{-2}$.

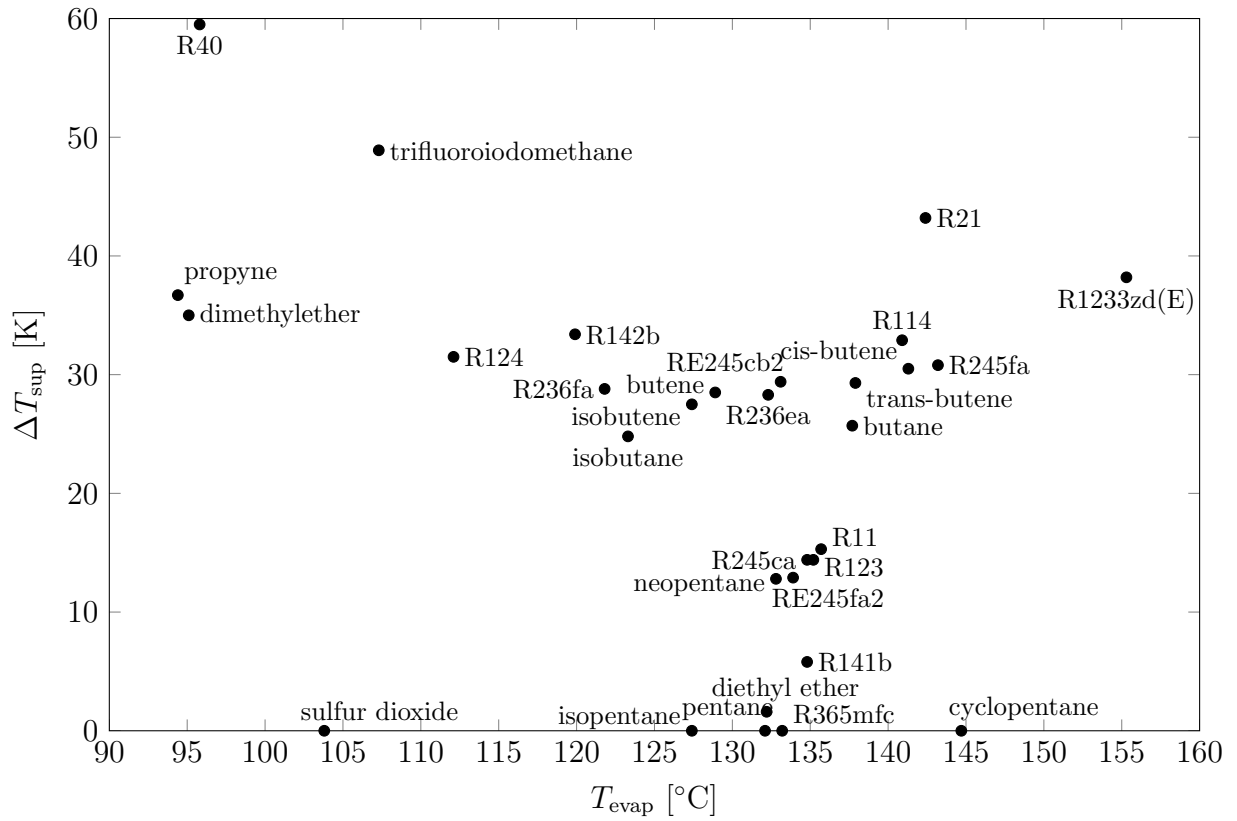


Figure C.5: Degree of superheating plotted against ORC evaporation temperature for the range of fluids investigated. Shown for the ETC array and $G = 1000 \text{ W m}^{-2}$.

Appendix D

Heat transfer calculations

D.1 Concentric tube heat exchanger heat transfer correlations

The following correlations are used for the calculation of the heat transfer coefficients at each node along the length of the tube.

D.1.1 Single-phase flow region

For heat transfer in the region of single-phase liquid or vapour flow, the local Nusselt number is obtained using the Gnielinski correlation:

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}, \quad (\text{D.1})$$

where for the refrigerant flow in the inner tube

$$Re_r = \frac{4\dot{m}_r}{\pi D_i \mu_r N_{\text{tubes}}}, \quad (\text{D.2})$$

and for the hot stream flow in the annulus

$$Re_h = \frac{4\dot{m}_h}{\pi(D_o + D_i)\mu_h N_{\text{tubes}}} . \quad (\text{D.3})$$

The Gnielinski correlation is valid for $0.5 \leq Pr \leq 2000$ and $3000 \leq Re \leq 5 \times 10^6$. The Darcy friction factor is obtained using the Petukhov correlation:

$$f = (0.79 \ln(Re) - 1.64)^{-2} . \quad (\text{D.4})$$

D.1.2 Two-phase flow region (boiling)

For heat transfer between the pipe wall and the refrigerant where the refrigerant undergoes saturated flow boiling, the following calculation method (described in Turaga and Guy [215] and Wang and Toubert [142]) is used to calculate a local heat transfer coefficient that is dependent on the local vapour quality, x . For the so-called wetted wall region where $0 < x \leq x_d$:

$$h_e = h_{\text{tp}}(x) . \quad (\text{D.5})$$

and for the dry-out region where $x > x_d$:

$$h_e = h_{\text{tp}}(x_d) - \frac{(x - x_d)^2}{1 - x_d} (h_{\text{tp}}(x_d) - h_s) . \quad (\text{D.6})$$

For two-phase flows in horizontal pipes, the correlation of Guerrieri and Talty [216] is found to be applicable for commonly used refrigerants:

$$h_{\text{tp}}(x) = 3.4 \left(\frac{1}{X_{\text{tt}}} \right)^{0.45} h_l , \quad (\text{D.7})$$

where X_{tt} is the Lockhart Martinelli parameter:

$$X_{\text{tt}} = \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{1 - x}{x} \right)^{0.9} . \quad (\text{D.8})$$

The dry-out vapour quality is calculated as follows:

$$x_d = 7.943 \left[Re_v \left(2.03 \times 10^4 Re_v^{-0.8} \Delta T - 1 \right) \right]^{-0.161}, \quad (D.9)$$

where ΔT is the temperature difference between the bulk fluid and the pipe wall. h_l and h_s are calculated using the Dittus-Boelter equation, for the liquid and vapour phases of the two-phase fluid:

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4} (\lambda_l / D_i), \quad (D.10)$$

$$h_s = 0.023 Re_v^{0.8} Pr_v^{0.4} (\lambda_v / D_i). \quad (D.11)$$

D.2 Plate heat exchanger calculations

The following calculations are used for the sizing of a plate heat exchanger (as an alternative to the concentric tube configuration), and follow the methodology described in Quoilin *et al.* [143] and LeCompte *et al.* [138].

D.2.1 Single-phase flow region

For single-phase heat transfer in the plate heat exchanger channels, the Nusselt number correlation used is as follows:

$$Nu = 0.299 Re^{0.645} Pr^{1/3}, \quad (D.12)$$

while for friction factor:

$$f = 0.685 Re^{-0.172}. \quad (D.13)$$

The above correlations are valid for $200 \leq Re \leq 15000$ and for the chevron angle $\beta = 45^\circ$.

D.2.2 Two-phase flow region (boiling)

An average heat transfer coefficient is calculated for the two-phase region using the boiling number correlation obtained from the experiments of Hsieh and Lin [217] for saturated flow boiling of the HFC refrigerant R-410A in a vertical plate heat exchanger:

$$h_e = 88h_l Bo^{0.5}, \quad (D.14)$$

where

$$Bo = \frac{Q_{\text{evap}}}{h_l \dot{m}_r}. \quad (D.15)$$

D.3 Condenser heat exchanger calculations

The following calculations are used for the sizing of the water-cooled and air-cooled condensers. The methodology followed is similar to that by LeCompte *et al.* [138] and Stewart [218].

D.3.1 Single-phase flow region

In the single phase liquid and vapour zones on the refrigerant side, the Dittus-Boelter correlation (for surface cooler than fluid) is used:

$$Nu = 0.023 Re^{0.8} Pr^{0.4}. \quad (D.16)$$

D.3.2 Two-phase flow region (condensation)

In the two-phase region the Shah correlation for condensation inside horizontal tubes is used. The correlation assumes an approximately linear variation in vapour quality with length along the tube.

$$h_c = h_l \left[0.55 + \frac{209}{(p/p_{\text{crit}})^{0.38}} \right]. \quad (D.17)$$

D.3.3 Air-side heat transfer and pressure drop (air cooled heat exchanger only)

For the air-side heat transfer across the bank of finned tubes, the Briggs and Young correlation is used:

$$Nu = 0.134 Re^{0.68} Pr^{1/3} \left(\frac{H_{\text{fin}}}{S_{\text{fin}}} \right)^{-0.2} \left(\frac{d_{\text{fin}}}{S_{\text{fin}}} \right)^{-0.12}, \quad (\text{D.18})$$

where d_{fin} is the fin thickness, H_{fin} is the fin height and S_{fin} is the fin spacing. For the calculation of the Reynolds number, the hydraulic diameter according to Mon [219] can be expressed as:

$$D_h = 4 (D_o + 2H_{\text{fin}}) \frac{A'_{\text{free}}}{A'_{\text{total}}}, \quad (\text{D.19})$$

where

$$A'_{\text{free}} = P_{\text{tube}} - D_{\text{fin}} + (D_{\text{fin}} - D_o) \left(1 - \frac{d_{\text{fin}}}{S_{\text{fin}} + d_{\text{fin}}} \right) \quad (\text{D.20})$$

is the minimum free flow area of finned tube per unit length, P_{tube} is the tube pitch, and

$$A'_{\text{total}} = A'_{\text{fin}} + A'_{\text{root}} \quad (\text{D.21})$$

is the overall heat transfer surface area per unit length. The pressure drop on the air side is calculated by way of a Fanning friction factor obtained using the Robinson and Briggs correlation:

$$\Delta p = 2 f_F N_{\text{rows}} \rho u_{\text{max}}^2, \quad (\text{D.22})$$

where

$$f_F = 9.456 Re_a^{-0.316} \frac{p_{\text{tube}}}{D_o}^{-0.927}, \quad (\text{D.23})$$

and

$$u_{\text{max}} = u_{\text{face}} \frac{p_{\text{tube}}}{A'_{\text{free}}}. \quad (\text{D.24})$$

D.3.4 Overall heat transfer coefficient for air-cooled heat exchanger

The overall heat transfer coefficient with reference to the internal heat exchanger surface area is calculated as follows:

$$\frac{1}{U} = \frac{d_{\text{tube}}}{\lambda_{\text{tube}}} \frac{A_i \ln \left(\frac{D_o}{D_i} \right)}{2\pi L} + \frac{A_i}{h_a A_o \eta_o} + \frac{1}{h_r}, \quad (\text{D.25})$$

where

$$\eta_o = 1 - \frac{A'_{\text{fin}}}{A'_{\text{tot}}} (1 - \eta_{\text{fin}}) \quad (\text{D.26})$$

is the overall surface efficiency, while fin efficiency according to the Schmidt approximation is:

$$\eta_{\text{fin}} = \frac{\tanh(m\phi D_o)}{m\phi D_o}, \quad (\text{D.27})$$

where

$$m = \sqrt{\frac{2h_a}{\lambda d_{\text{fin}}}} \quad (\text{D.28})$$

and

$$\phi = \left(\frac{D_o + 2H_{\text{fin}}}{D_o} - 1 \right) \left[1 + 0.35 \ln \left(\frac{D_o + 2H_{\text{fin}}}{D_o} \right) \right]. \quad (\text{D.29})$$

Appendix E

Thermal energy storage for the S-CHP system

Calculations are presented below for the approximate sizing of a thermal energy store (TES) for use in the S-CHP system, considered in combination with the nominal sizing parameters of the solar collector array and ORC engine. A range of phase change materials (PCMs) will be considered for use as TES media with specific attention given to the selection of the PCM physical properties (in particular, the melting temperature and latent heat capacity) for maximisation of both energy storage density and ORC power output.

E.1 Methodology

Figure E.1 shows the layout of the S-CHP system of interest. A thermal store (TES) acts as the interface between the collector array in place of the evaporator heat exchanger. The present study considers two types of non-concentrating collector: (i) a new-generation evacuated flat-plate (EFP) collector developed especially for medium temperature process-heating applications (including ORC systems); and (ii) a standard evacuated-tube heat pipe (ETHP) collector designed for low-temperature domestic water heating applications. The collector array is modelled as south-facing, inclined at 36° , and constrained to within the dimensions of a

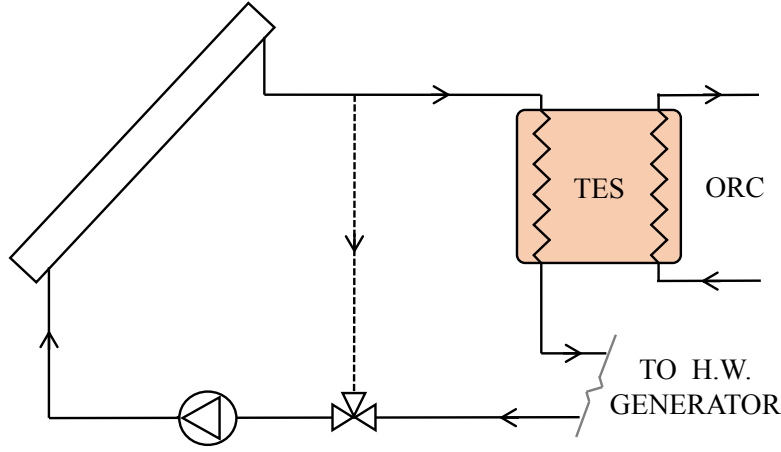


Figure E.1: Schematic of the TES scheme for the S-CHP system.

Table E.1: Efficiency-curve coefficients (stated for gross module area), of the EFPC and ETC-HP collectors modelled in the analysis.

Collector	Manufacturer/Model	A_{sc} , m ²	η_0 , -	c_1 , W m ⁻² K ⁻¹	c_2 , W m ⁻² K ⁻²
EFPC	TVP solar HT-power	1.95	0.774	0.376	0.006
ETC-HP	Thermomax HP-200	2.84	0.556	0.888	0.006

typical domestic roof-space, assumed to be 15 m².

Calculations are performed at hourly time-intervals, subject to the following assumptions:

(i) the system is operating under quasi-steady state conditions, with the collector thermal capacity neglected; (ii) the system is operational only for the hours of the day when the solar incidence angle on the tilted collector plane is $< 70^\circ$; (iii) thermal losses from the store are negligible; and (iv) the thermal store begins each day in a fully discharged (solidified) state.

The performance of the solar collectors is simulated using hourly climate data for London and Larnaca [120]. The collectors are modelled using the steady state efficiency equation E.1, which is solved by assuming that the inlet fluid temperature is equal to the melting temperature of the PCM being considered. Thus the useful solar energy gain is a function of both the choice of collector and the storage medium that governs the collector operating temperature:

$$\frac{\dot{Q}_u}{A_{sc}} = \eta_0 (K_{\theta,b} G_b + K_{\theta,d} G_d) - c_1 (\bar{T}_{sc} - T_a) - c_2 (\bar{T}_{sc} - T_a)^2, \quad (\text{E.1})$$

The parameters used for the thermodynamic cycle analysis of the ORC engine and solar

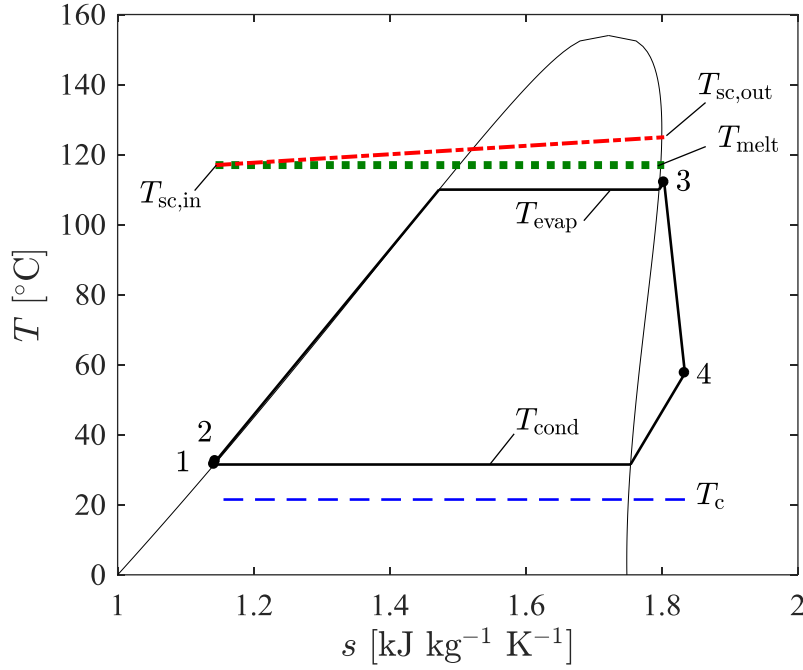


Figure E.2: T - s diagram of solar-TES-ORC processes. For the ORC the following parameters are assumed: $\eta_{pp} = 65\%$, $\eta_{exp} = 75\%$, $\Delta T_{app} = 5$ K, $\Delta T_{sub} = 5$ K, and $\Delta T_{sh} = 5$ K.

collector array are summarised in Figure E.2 and Table E.1. Two widely-available working fluids are compared in the analysis: (i) R245fa, which has been previously investigated for use in low temperature solar-ORC systems and has a critical temperature of 154 °C; and (ii) pentane, which has been investigated for use in higher-temperature solar-ORC systems and has a critical temperature of 197 °C.

The sizing of the ORC engine in relation to the solar input and the store volume requires careful consideration. If the engine is over-sized in relation to the store, its operation will be limited to short periods, whereas if it is under-sized in relation to the solar energy input, a larger storage volume will be required to make full use of any surplus energy collected when solar irradiance is abundant. In the monthly analysis, the ORC working fluid mass flow-rate is set based on Eq. E.2 and the monthly mean solar energy gain (during operational hours), while the TES volume requirement is calculated as the difference between the maximum and mean solar energy yield per day, shown in Eq. E.3:

$$\dot{m}_r = \frac{\dot{Q}_{u,mean}}{h_3 - h_2}, \quad (E.2)$$

Table E.2: Phase change material properties [79].

Name	PCM type	$T_{\text{melt}}, ^\circ\text{C}$	$L, \text{kJ kg}^{-1}$	$\rho, \text{kg m}^3$
S89	Hydrated salt PCM	89	1450	151
S117	Hydrated salt PCM	117	1550	160
A144	Organic PCM	144	880	115
A164	Organic PCM	164	1500	290

$$V_{\text{store}} = \frac{\dot{Q}_{\text{u,max}} - \dot{Q}_{\text{u,mean}}}{(\rho \cdot L)_{\text{PCM}}}, \quad (\text{E.3})$$

where L is the latent heat capacity of the PCM in kJ kg^{-1} .

The required storage volumes and resulting ORC net power outputs for the various PCM options will be compared to that for an equivalent system with water as the energy storage medium. This requires the consideration of a large temperature range for sensible heat uptake rather than a small temperature range for latent heat uptake. It will therefore be assumed that the sensible storage medium operates between a minimum temperature, sufficient for the operation of the ORC engine, and a maximum temperature for which the storage medium (water) remains in its liquid state. This is taken as 130°C , corresponding to a typical unvented hot water cylinder working pressure of 2 bar(g). The allowable daily temperature swing, and hence the required volume of the store, is dependent on the ORC evaporation temperature.

E.2 Results and discussion

Figure E.3 shows the mean and maximum daily solar irradiation for each month of the year in Larnaca and London, respectively. The plots confirm that the Cyprus solar resource is both more abundant and undergoes less variation on a daily, seasonal and annual timescale than the UK. In Larnaca the mean daily irradiation on the tilted plane is $5.3 \text{ kWh m}^{-2} \text{ day}^{-1}$ compared to $2.9 \text{ kWh m}^{-2} \text{ day}^{-1}$ for London. The variation in daily irradiation for Larnaca is far smaller in summer than in winter, with a standard deviation of $0.2 \text{ kWh m}^{-2} \text{ day}^{-1}$ (3%) between the months of June to August; compared to $1.8 \text{ kWh m}^{-2} \text{ day}^{-1}$ (39%) over the same period for London. Thus a TES system designed to buffer daily variations in solar gain in Cyprus can be significantly smaller than one designed for the UK, and furthermore, can be significantly

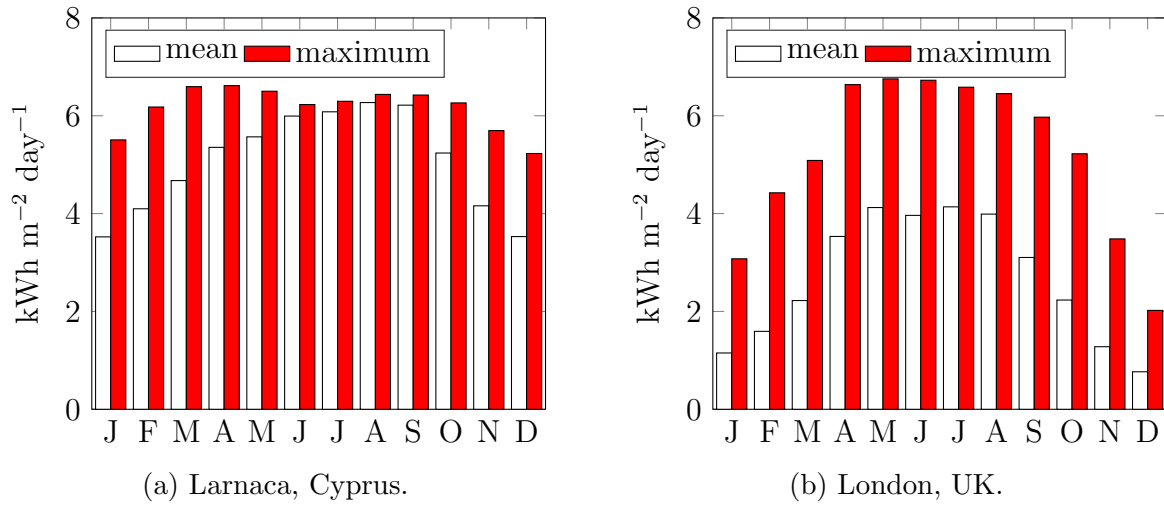
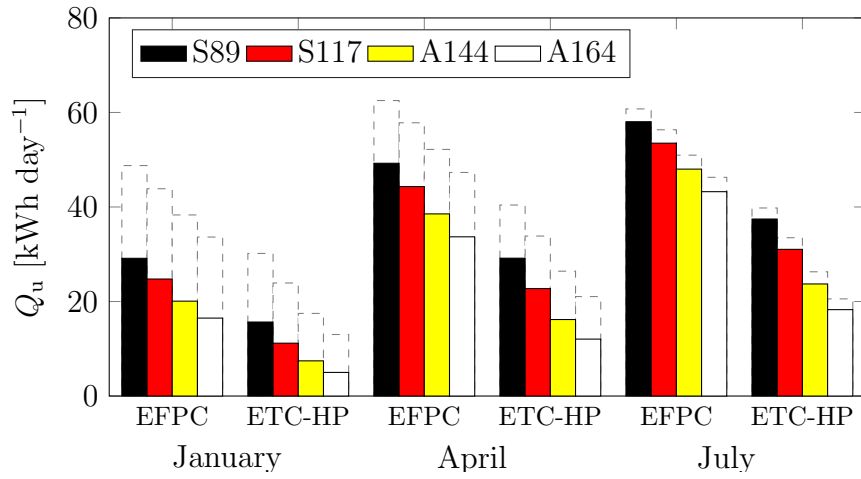


Figure E.3: Mean and maximum solar irradiation per day for: a) Larnaca and b) London.

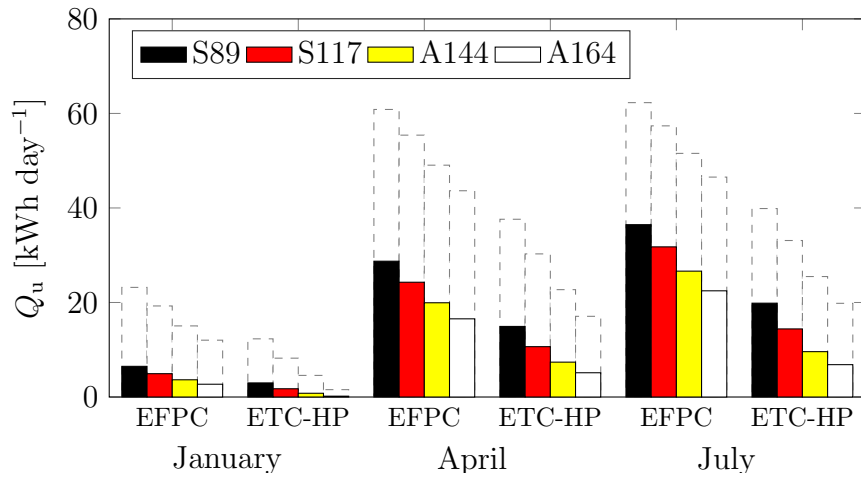
reduced in size if it is designed for summer-only operation compared to year-round operation, which is not true for the UK.

Figure E.4 shows the mean daily solar thermal energy collected by each collector and PCM combination in Larnaca and London, in selected months (January, April and July). The performance of the collector is highly dependent on the PCM melting temperature which is shown by the respective decrease in solar gain. The EFP collector is shown to operate with a significantly higher efficiency than the ETHP collector throughout the year in both the UK and Cyprus climates. The difference between the mean and maximum irradiation (solid and dashed bars) in Fig E.4 informs the sizing of the thermal store shown in Fig. E.5, which is also dependent on the physical properties of the PCM listed in Table E.2. It can be seen from the table that the density and mass-related latent heat capacity of the two hydrated-salt PCMs are quite similar; while for the two organic PCMs there is a considerable difference in both of these properties, resulting in a volumetric latent heat capacity that differs by a factor of four.

Figure E.5 shows that, compared to the latent heat capacity, the PCM melting temperature and its effect on the collector efficiency has a relatively small impact on the required size of the store. For London, a larger storage requirement occurs in the summer and mid-season periods (240–1050 L for the EFP collector array, depending on the choice of PCM), while for Larnaca the largest storage requirement is in the winter (170–740 L), and in summer it is considerably smaller (25–105 L).



(a) Larnaca, Cyprus.



(b) London, UK.

Figure E.4: Useful solar energy gain as a function of collector type, TES medium and month of the year for: a) Larnaca and b) London.

In Fig. E.6 the system power output and required storage volume for summer operation are plotted for each TES medium. The power output is related to the evaporation temperature of the ORC, which is in turn related to the temperature of the thermal store (see Fig. E.2); therefore, for the PCMs this is a fixed value, but for the water storage this is shown as a range of values corresponding to a range of allowable temperature-swings for the daily storage cycle. For Larnaca in the month of July, the highest net power output of 687 W is achieved with the EFP collector and the organic PCM A144, requiring a storage volume of 92 L. PCM A164 meanwhile provides only a slightly lower net power output (678 W) but requires a far smaller storage volume (19 L) due to its higher latent heat capacity. For London, by comparison, the highest net power output is 405 W, and is achieved with the EFP collector and the hydrated

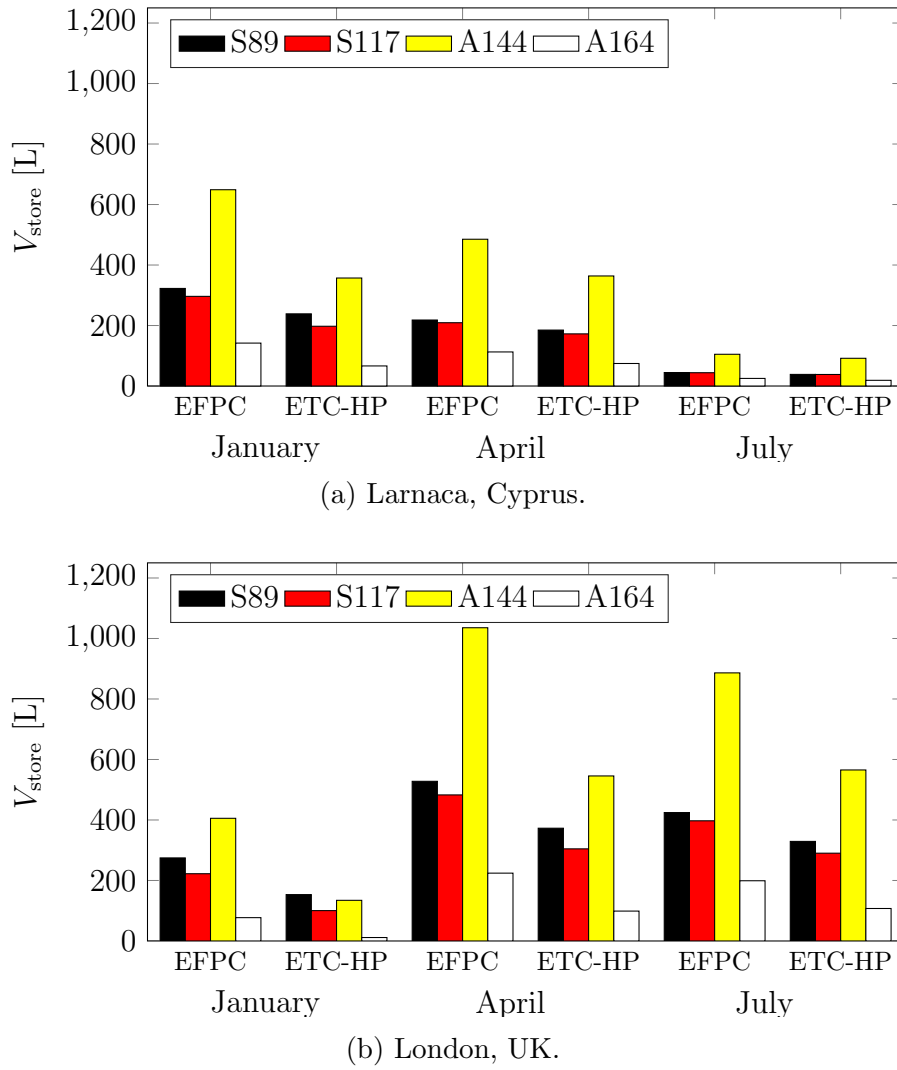
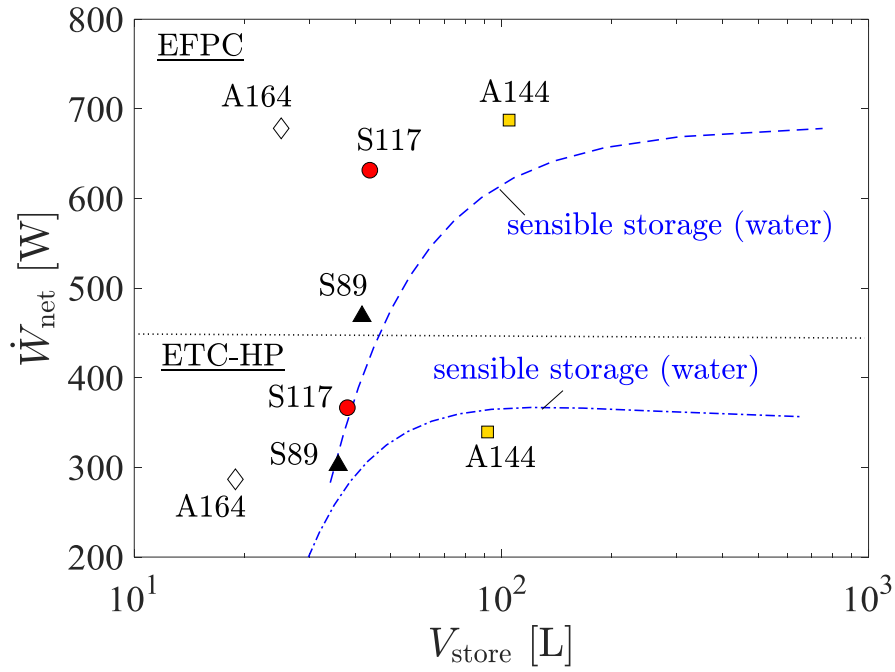


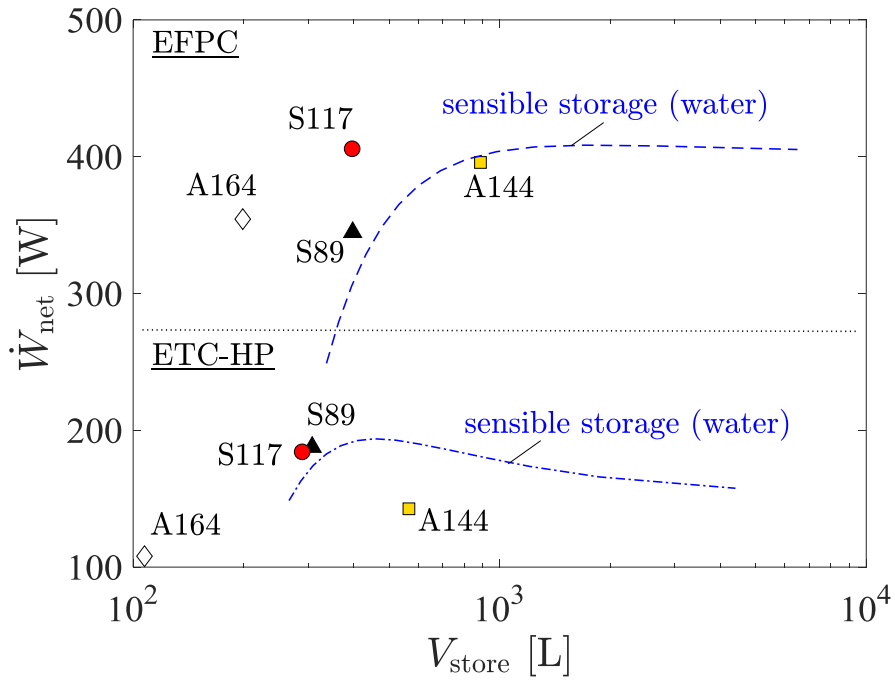
Figure E.5: TES volume requirement as a function of collector type, TES medium and month of the year for: a) Larnaca and b) London.

salt PCM S117, confirming that the optimal collector/ORC operating temperatures are lower than those for Cyprus. The required storage volume is 397 L, which gives a power output per storage volume ratio that is ~ 7 times lower than that for Larnaca. By choosing the PCM with the highest latent heat capacity (A164) instead, the storage volume requirement is reduced to 199 L, but at the expense of a reduction in net power output of 14%.

The choice of working fluid makes a relatively small difference in the above results over the range of conditions considered. For Larnaca, pentane gives a 5% increase in the maximum net power output relative to R245fa, whereas for London R245fa gives a 4% increase relative to pentane. The EFP collector system provides a higher net power output for all of the TES options investigated in both climates. For the sensible heat storage option using water, a small



(a) Larnaca, Cyprus.



(b) London, UK.

Figure E.6: ORC power output for operation in the month of July for: a) Larnaca and b) London, plotted against store volume requirement for each TES options and for EFPC collector array and ETC-HP collector array

temperature variation (and hence a large storage volume) is required to achieve a sufficient evaporation temperature and a power output that is comparable to that achieved with the PCMs. The PCMs therefore generally show a higher power output per required storage volume than the water storage option, with the exception of A144, which has a low latent heat capacity compared to the other PCMs.

When the S-CHP system sizing is performed for winter operation, there are two important implications for the ORC system compared to the summer case. Firstly, a lower average power output is observed (55% lower for Larnaca and 75% lower for London), and secondly, a lower optimal PCM melting temperature for maximum power output is required. The PCM that gives the highest system performance in each geographical location is therefore different in winter compared to summer months, however, the EFP collector still gives the highest power output in both cases.

E.3 Further discussion and conclusions

The work presented in this analysis has been concerned with evaluating a range of latent TES options for use in a solar-powered combined heat and power system based on an ORC engine for operation in the contrasting climates of Cyprus and the UK. The system evaluated here was constrained to an array size that is equivalent to a domestic rooftop area (15 m²). PCMs for latent thermal energy storage have been shown to provide a superior power output from the S-CHP system for a smaller equivalent storage volume than liquid water (used as a benchmark sensible storage medium). However, this is dependent on the selection of a PCM with a melting temperature that is matched to the optimal operating temperatures of the collector and ORC engine.

By designing the system for continuous operation during daylight hours, the resulting summertime nominal net power output in both Cyprus and the UK is below 1 kW, which is typically the lower limit of “small-scale” systems described in the ORC literature. Furthermore, the expander volume ratio corresponding to the best-case system configuration presented here

is found to be 8.7 for London (with R245fa) and 11.6 for Larnaca (with pentane). Commercially available and experimental small-scale positive displacement expanders based on scroll machines tend to have built-in volume ratios < 5 . This suggests that in order to avoid losses in efficiency associated with part-load/off-design operation, the proposed solar ORC system would benefit from a specially-designed expander. Alternatively, the system could be sized for a higher nominal power output over a reduced number of operating hours. One possibility is to operate the solar collector array during the day to fully charge the store, and operate the ORC engine in the evening at full output to off-set grid electricity use during hours of peak demand (when grid electricity is most expensive).

Taking the PCM with the highest volumetric latent heat capacity (A164) and assuming an ORC engine sized for 1 kW output, the system can provide approximately 1.5 hours of continuous operation per 100 L of thermal energy storage. Future work should consider the relative benefits in terms of daily work output, system run-time and potential energy (heat and power) bill savings as a result of using thermal energy storage to off-set the ORC operation to coincide with evening peak electricity demand.

Appendix F

Experimental methods: sensor calibration

F.1 Pyranometer calibration

The radiation sensor was calibrated under laboratory conditions, using a UNISIM Solar Simulator. The simulator was configured to approximately simulate the solar AM 1.5 spectrum using a metal-halide lamp for the shorter-wavelength (blue) part of the spectrum and a quartz-halogen lamp for the longer-wavelength (red) part of the spectrum. Isotype solar cells were used to calibrate the intensity levels of the two lamps. An isotype cell is a triple-junction cell for which only one of the three subcells is activated so that the radiation energy converted to electricity corresponds only to that absorbed in a specific part of the spectrum. Each sub-cell isotype will generate a known short-circuit current when subjected to the standard AM 1.5 spectrum (corresponding to a total irradiance of one Sun or 1000 W m^{-2}). For the calibration exercise, only the top and middle sub-cell isotypes were used, as these correspond to the majority of the spectral response range of the pyranometer. For the top sub-cell (indium gallium phosphide, 300–700 nm spectral response), the expected short-circuit current corresponding to AM 1.5 was 0.72 A, whereas for the middle sub-cell (indium gallium arsenide, 670–900 nm spectral response), the expected short-circuit current was 0.75 A. As the total incident radi-

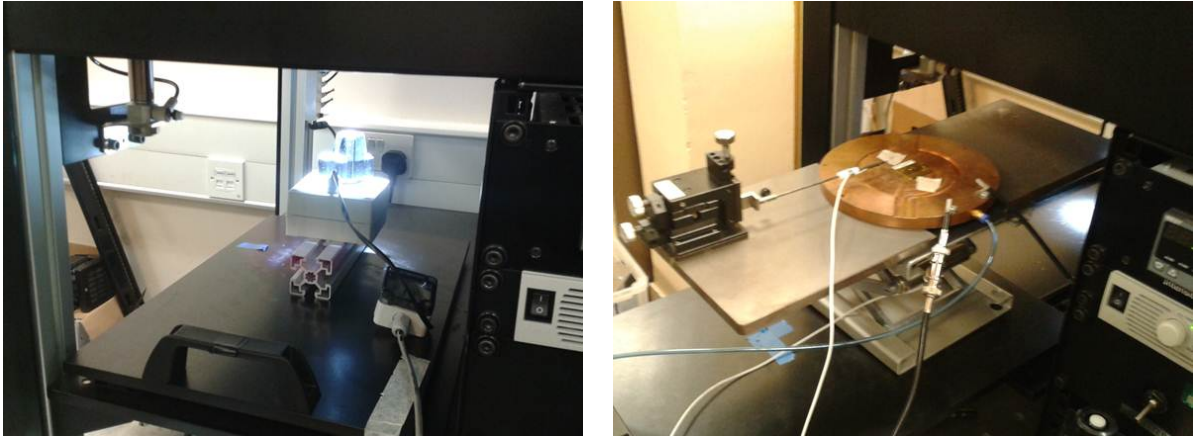


Figure F.1: Testing of the pyranometer in the solar simulator (left), and calibration using an isotype reference cell (right, shown on a conductive copper plate).

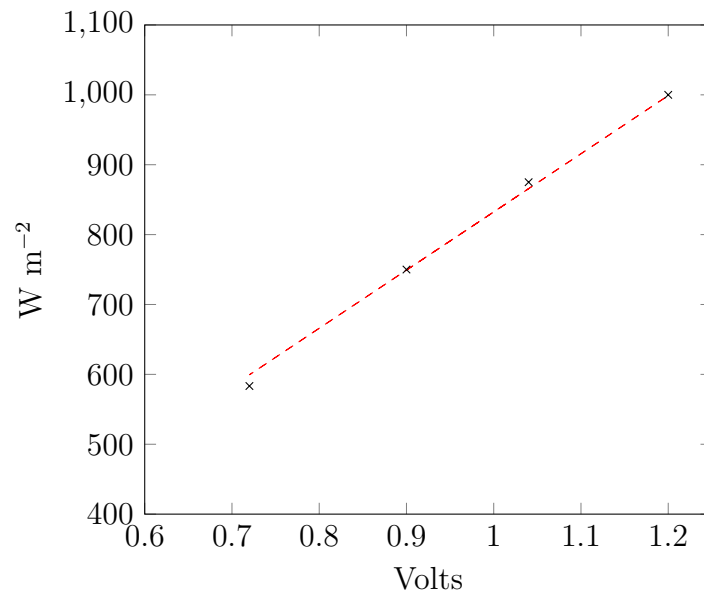


Figure F.2: Calibration data for solar sensor with linear least squares fit passing through the data with the equation $y = 832.47x$ and coefficient of determination $R^2 = 0.9964$.

ation intensity is adjusted, the current output from the sub-cell isotypes can be expected to vary in an approximately linear fashion, providing that the relative spectral distribution was held constant.

Thus the following approach was taken for the calibration of the pyranometer. The output voltage from the pyranometer was measured at four approximately evenly spaced intervals between 1 sun ($1000 W m^{-2}$) and the minimum turn-down setting on the solar simulator ($580 W m^{-2}$). The isotype cells were used to set the irradiance level for each lamp and to ensure that the radiation intensity was varied uniformly across the spectrum. In order to limit the im-

fact of systematic measurement errors specific to the experiment conditions, the pyranometer reading was performed using the same DAQ system and length of extension cable as required for the outdoor setup. The results were plotted and a linear least-squares fit was applied to the data in order to minimise the quantity:

$$S = \sum_{i=1}^n [y_i - (ax_i)]^2 . \quad (\text{F.1})$$

The conversion factor obtained was found to be 832.5 W m^{-2} per Volt and the standard error for the estimate was found to be $\pm 13 \text{ W m}^{-2}$ using the following equation:

$$\text{Standard error} = \sum \left[\frac{(y_i - ax_i)^2}{n - 2} \right]^{1/2} . \quad (\text{F.2})$$

The inverse of the conversion factor is $1.2 \text{ mV per W m}^{-2}$, which was considerably lower than the nominal value of $1.67 \text{ mV per W m}^{-2}$ listed in the manufacturer's datasheet.

F.2 Turbine flow meter calibration

The FTB-101 turbine flow meter was calibrated for use with the water/polypropylene glycol mixture used in the experiments. The calibration was performed in outdoor conditions at ambient temperature. The procedure involved setting the pump at a range of different speeds and measuring the time taken to fill a 2 L measuring cylinder with a stop-watch. This was then converted into a volumetric flow-rate in litres per second for each measurement. The estimated error in the time measurements was $\pm 0.5 \text{ s}$ while the estimated error in the volume measurement was $\pm 0.05 \text{ L}$. Direct measurement was taken of the pulse frequency outputted by the flow-meter at each pump-speed setting and logged on the data acquisition system. This was then converted into a pulse per litre “K-factor” by dividing by the flow-rate. The pump speed was adjusted at a series of regular intervals on the pump-speed controller and the measurements were repeated twice for each setting. It was found that the pump-speed controller settings did not correspond linearly with flow-rate, and thus the regular intervals in pump-speed setting

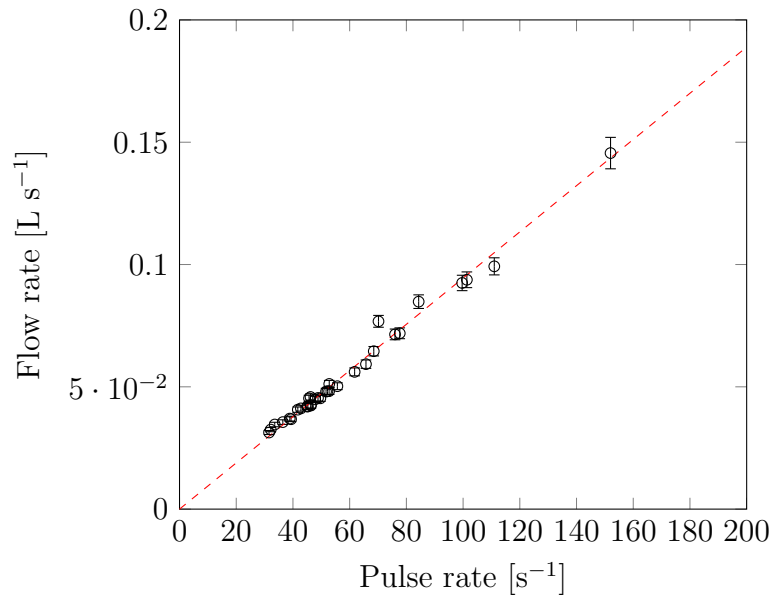


Figure F.3: Calibration data for solar sensor with linear least squares fit passing through the data with the equation $y = 0.945 \cdot 10^{-3}x$ and coefficient of determination $R^2 = 0.9875$.

corresponded to increasingly larger intervals in flow-rate at higher pump speeds. However it was found that the relationship between pulse-rate and volumetric flow-rate was approximately linear across the measured range and that therefore the K-factor was approximately constant across this range.

Appendix G

Summary of the 3-D dynamic thermal collector model

The following is a brief summary of the 3-D dynamic thermal solar collector model developed by Guarracino *et al.* [201] and evaluated in Chapter 5. The model solves the energy balance at each component layer of the collector (glass, thermal absorber, pipe and fluid, discretised as in Fig. G.1). The following assumptions are used in the model: (i) thermophysical properties of solid components do not vary with temperature; (ii) the collector is well insulated (i.e. the thermal losses at the collector edges and manifold are neglected); (iii) for sheet-and-tube configurations, the temperature distribution along the absorber surface is symmetrical between adjacent parallel riser tubes; (iv) the total flow-rate of the fluid flowing in the header/manifold is distributed evenly between the individual riser tubes (or between the individual evacuated tubes for the ETC configuration). The model takes into account the thermal capacity of the collector components and the energy balance is solved numerically at each finite element. The main component energy balance equations are summarised below (for the complete set of detailed equations refer to Guarracino *et al.* [201]).

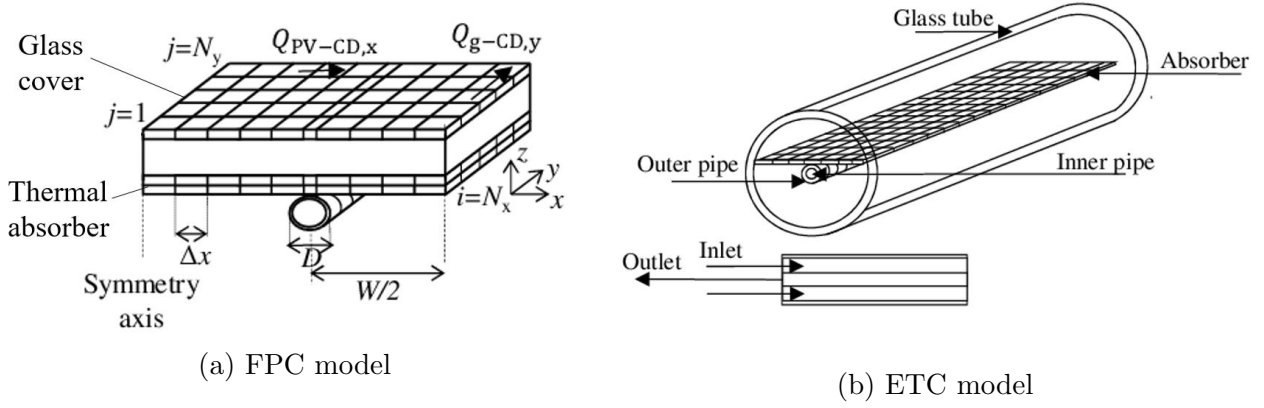


Figure G.1: Schematic diagram summarising the 3-D numerical model of the FPC and ETC collectors, with coaxial flow arrangement for the ETC collector also indicated.

$$M_{gl}c_{gl}\frac{dT}{dt} = \dot{Q}_{gl-con,(x,y)} - A_{xy}\varepsilon_{gl}\sigma(T_{gl}^4 - T_{sky}^4) - A_{xy}h_{top}(T_{gl} - T_a) + A_{xy}\frac{\sigma(T_{ab}^4 - T_{gl}^4)}{1/\varepsilon_{gl} + 1/\varepsilon_A - 1} + A_{xy}h_{gap}(T_{ab} - T_{gl}) + \alpha_{gl}G; \quad (G.1)$$

$$M_{ab}c_{ab}\frac{dT}{dt} = \dot{Q}_{ab-con,(x,y)} - \frac{\sigma(T_{ab}^4 - T_{gl}^4)}{1/\varepsilon_{gl} + 1/\varepsilon_A - 1} - A_{xy}h_{gap}(T_{ab} - T_{gl}) - \frac{A_{xy}(T_{ab} - T_p)}{R_p} - \frac{A_{xy}(T_{ab} - T_a)}{R_{loss}} + \bar{\tau}\alpha G; \quad (G.2)$$

$$M_{pipe}c_{pipe}\frac{dT}{dt} = \dot{Q}_{pipe-con,(y)} + A_{xy}\lambda_{bond}\delta_{bond}^{-1}(T_{ab} - T_{pipe})\pi Dh_f dy(T_{pipe} - T_f); \quad (G.3)$$

$$M_fc_f\frac{dT}{dt} = \pi Dh_f dy(T_{pipe} - T_f) + \dot{m}c_f(T_{in} - T_f); \quad (G.4)$$

$$h_{top} = (h_{wind}^3 + h_{free}^3)^{\frac{1}{3}}, \quad \text{where } h_{wind} = 2v_{wind} + 3.8; \quad (G.5)$$

$$h_{gap} = \frac{\lambda_{air}}{\delta_{gap}} \left\{ 1 + 1.44 \left(1 - \frac{1708}{Ra\cos(\beta)} \right)^* \left(1 - \frac{1708(\sin(1.8\beta))^{1.6}}{Ra\cos(\beta)} \right) + \left[\left(\frac{Ra\cos(\beta)}{5830} \right)^{\frac{1}{3}} - 1 \right]^* \right\}, \quad (G.6)$$

where the asterisk indicates terms that are set to zero when their value is negative;

$$h_f = 4.36 \frac{\lambda_f}{D_H}, \quad \text{for } Re < 2300; \quad (G.7)$$

$$h_f = 0.23Re^{0.8}Pr^{0.4}, \quad \text{for } Re > 2300. \quad (G.8)$$