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Solar Domestic Hot Water Design and Optimisation in the United Kingdom

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Abstract

Solar energy is being increasingly implemented for domestic uses, such as hot water systems. However, the poor economic gains, particularly in cold climates and winter months, make this environmentally beneficial technology a less attractive investment. So far, research into optimising these designs have focussed on warmer climates and optimising the gains made during the summer months. Furthermore, most high budget research is geared towards industrial implementation which has limited use at the domestic, home-owner scale. To investigate an economically and energetically optimal domestic hot water system design for cold climates, mathematical methods and TRNSYS simulations were used to experiment with the parameters influencing efficiency. This was achieved by creating a parametric model to reveal losses experienced by the system. Next, a fully validated TRNSYS model was built to allow for experimentation from which the system could be adapted for cold climates. The key impact of this research will be to improve the affordability, such as rate of return, and appeal of solar domestic hot water systems for the public. Ultimately, developments such as this will reduce dependence upon fossil fuels in the United Kingdom, and other environmentally conscious nations, and allow large-scale implementation of sustainable technologies.

1. Introduction

Across the United Kingdom the government has pushed for reduced household energy consumption to align with energy reduction targets, leading to an increase in demand for solar energy (Gill *et al*., 2015). One main use of solar energy is domestic hot water heating systems which produce hot water using the energy provided by the sun. The premise of hot water solar technology involves a solar collector, which are typically either flat plate collectors or evacuated tube collectors, alongside a control unit, pump station and multiple storage tanks (Ayompe *et al*., 2011). All solar hot water systems have four main causes for reduced performance. These include the efficiency of the solar collector to absorb available solar energy, heat loses throughout the collector, heat loss throughout the pipe network and heat loss from the storage tanks. All thermal loses are a result of convection, conduction and radiation. These systems tend have a high initial cost and a slow return of investment which is mainly due to their inefficiencies at colder times of the year. This is a specific issue in the United Kingdom where the solar heating system is rendered almost entirely useless for half the year due to low solar intensity and cold ambient air temperatures (Stackhouse, 2016).

There is a lack of whole-system analysis within the study of domestic hot water technology. Furthermore, knowledge of these systems optimality in cold climates has been seldom investigated, with the main focus being upon optimal design in optimal conditions. Recently, Ayompe *et al*. (2011) conducted analysis of the most common system in domestic hot water technology, henceforth described as the base system, in Dublin. This is one of the first and few studies conducted in cold climates. This study builds upon these current findings, in which a combination of different systems will be simulated.

This research is key in an increasingly environmentally-conscious, but money-conscious, society. This study aims to present a more cost-effective, yet equally environmentally beneficial, system. Therefore, the developments and improvements perused will ultimately improve the efficiency of the common domestic hot water system, and consequently save the individual money on this necessity, without costing the environment.

The aim of this project it to effectively model a solar domestic thermal hot water system with the intention to propose an energetically and economically optimal design for annual performance for cold climates. To achieve this aim three main objectives are proposed. This includes 1) produce a parametric model to analyse the individual losses within a previously studied common system, 2) produce and validate a TRNSYS (transient systems simulation program) model to predict the performance of the base system and adapted systems and 3) to propose a more efficient and cost-effective system. This research will be presented using these three main objectives, with the methods and results being separated to explicitly meet each objective and the overall aim of the study.

2. Literature Review

The purpose of this literature review is to highlight, through performance comparisons, different available domestic hot water technologies. The review first briefly outlines the two different collectors considered – solar flat plate collectors and evacuated tube (Duffie & Beckman, 2013).

Solar Collectors

A solar collector is a heat exchanger that uses solar radiant energy and converts it to heat which can then be transported to other parts in the system.

Hot water energy gain is defined as $Q' = \dot{m}C_p(\Delta T)$ where ΔT is the change in temperature of the heat transfer fluid from the entrance and exit of the solar collector and Q'= heat energy transfer rate (W). The primary concept influencing design and performance of a solar collector is to achieve the highest value for Q' possible which is primarily dependent on increasing ΔT. ΔT is dependent on two main factors when keeping control variables such as ṁ constant, heat energy gain from solar irradiance and heat losses across the solar collector; $Q'_{Total} = Q'_{Solar\ irradiance} - Q'_{reflectance} - Q'_{radiation} - Q'_{convection}$ (Duffie & Beckman, 2013).

Flat Plate Collector

Flat plate collectors are typically used for low cost, low demand hot water systems; this is why they have been widely adopted in homes across the UK. A flat plate solar collector consists of a black body with a glazed glass cover which is used to change the wavelength of incoming radiation and trap the thermal radiation. Almost all the solar energy is absorbed into a highly conductive black body and tubing (made from a highly conductive material such as copper) where the thermal energy is absorbed by a moving fluid. The fluid is then transported to a storage tank used to heat the water (IEA, 2011). Improvements to flat plate collector designs aim to increase the number of glass layers to reduce the thermal losses across the panel. As shown by Duffie & Beckman (2013), increasing the number of glazing layers reduces convection and radiation heat losses. However, this will increase manufacturing cost which often make the collector too expensive for the consumer.

Evacuated Tube Collectors

An evacuated tube collector consists of many tubes running in parallel. These tubes are cylindrical to allow for maximum solar absorption throughout the day regardless of the angle of the sun. The solar tubes contain copper heat pipes, and sometimes other highly conductive materials, running through them. They have an absorbing reflector plate situated behind them within a sealed vacuum tube. This vastly improves the solar collector's efficiency because of minimising, and in some areas eliminating, the convective and conductive heat loses (Alternative Energy Tutorials, n.d).

Flat Plate and Evacuated Tube Collector Performance Comparison

Evacuated tube collectors are, on average, 30% more expensive to purchase and produce when compared to flat plate collectors (Sokhansefat *et al.,* 2017). A paper written by Ayompe *et al*. (2011), directly compared the performance of a flat plate collector and an evacuated tube collector with an equal total collector area of 4m^2 in the summer months in Dublin. The TRNSYS simulation model developed was then adopted by Sokhansefat (2017) where the difference in performance between flat plate and evacuated tube collectors during the summer months is negligible. This showed a performance gain of only 2% which leads to the conclusion that in summer months the performance gain is not justified by the increased cost. The method used was confirmed by Ayompe *et al*. (2011) using data with experimental results and TRNYSYS simulations. It was shown by Sokhansefat (2017) that in winter months the evacuated tube collectors greatly outperform flat plate collectors by at least 20%

for heat energy gain per hour. This value was discovered using a validated TRNYSYS model, which was confirmed using external experimental data.

Thermal Energy Storage Tanks

Thermal energy storage tanks are separated into two main categories; sensible heat storage and latent heat energy storage.

Sensible Heat Storage

Sensible heat storage is the most common hot water storage method used in domestic settings due to its low cost when compared to other storage methods (Tian & Zhao, 2013). In sensible heat storage, thermal energy is stored by increasing the temperature of the heat storage material which can be solid or liquid. The amount of heat energy stored depends on the specific heat, density, quantity and temperature of the stored material (Tian & Zhao, 2013).

Latent Heat Energy Storage

Latent heat storage relies on materials that undergo a phase change from a solid to liquid and vice versa, and liquid to gas and vice versa. During this phase change a large amount of thermal energy is stored or released. This is more expensive than sensible heat storage but does offer improved storage densities, where typical values range 5-10 times denser (Tian & Zhao, 2013). However, the drawbacks include poor thermal conductivity, therefore a poor heat transfer rate. Consequently, in low thermal energy conditions, this may result in a very slow and ineffective storage system (Zhao *et al*., 2010*)* However, this experiment was conducted in conditions of higher energy demand than that of typical domestic households. Therefore, the results may not be representative of performance in domestic hot water storage.

Heat Exchangers

Heat exchangers are used to transfer the thermal energy attained from the solar collector to the thermal energy storage tank. The ideal heat exchanger would transfer 100% of the thermal energy to the energy storage material. However, this cannot realistically be achieved (Cengel *et al.,* 2016). To keep the heat transfer rate high, while minimising temperature difference, many designs and sizes of heat exchangers have been adopted to maximise the total heat transfer of the fluid while in the heat exchanger. This is important for cold climates with low solar intensity. The colder the fluid entering the solar collector, the greater the energy absorption will be. Therefore, when there is a small amount of solar irradiance available, maximising this absorption is key to an effective system in cold climate conditions.

3. Methodology

The study is separated into two main components:

- A parametric model which will be used to analyse the base system, and the energy losses and gains, to suggest improvements to the base system, areas of further research and understanding of the base system and its technologies.
- A TRNSYS model will be produced to optimise and, if necessary, re-design the base system to produce the most cost-effective solution.

Base System Properties

The base system used in this study was that of Ayompe *et al*. (2011) in which a typical solar hot water system, consisting of a hot water storage unit, a pump, a control unit and a hot water solar plate collector, was designed and investigated. In this case two K420-EM2L flat plate collectors are arranged in parallel. This provides optimal performance and an area totalling 4m^2. In addition, a coil heat exchanger is located inside the hot water storage tank, as well as a heating element. This configuration is suggested to be the most common used in the United Kingdom and as a result forms the base of this study (Ayompe *et al*., 2011). A basic diagram is shown in Figure 1 and main component parameters given in Table 1.

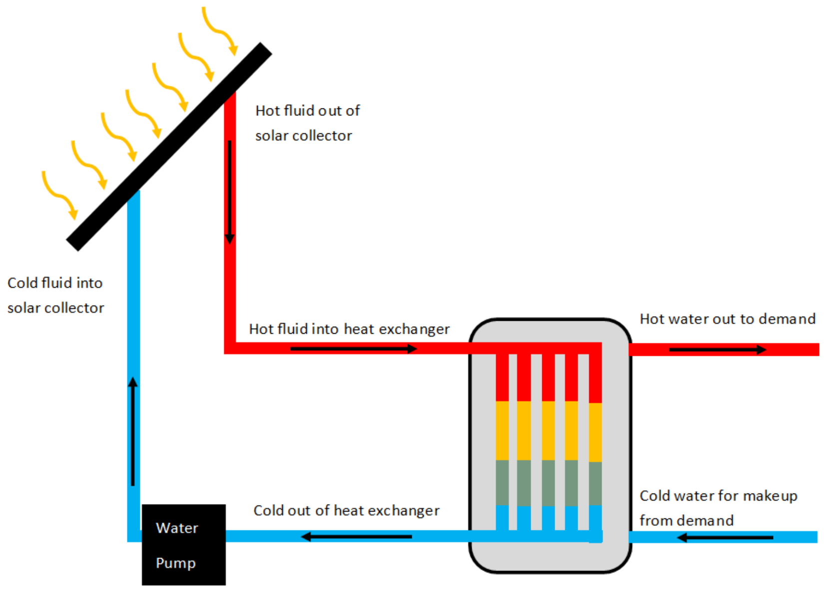

Figure 1: Base system diagram to show different components of the system

Table 1: Basic parameters of hot water system provided by (Ayompe *et al*., 2011).

Parameter	Value
Solar Collector gross area	4.36m^2
Solar Collector absorption area	4m^2
Maximum operating temperature	120^{o}C
Tank Volume	0.3m^3
Heat exchanger inside diameter	0.016m
Heat exchanger outside diameter	0.02m
Heat exchanger surface area	1.4m^2

See supporting documents for full list of parameters used.

The hot water demand used in the experiment performed by Ayompe *et al*. (2011) was used as an input for the current study. This was based upon a typical family of four's hot water usage and respective time of day. The solar data used in the current study will also be identical to that used by Ayompe *et al*. (2011) which represents an average summer day in Dublin.

The mass flowrate for the fluid running through the thermal collector and heat exchanger will be set using the previously discovered optimal equation:

$$\dot{m} = aG_t + b$$

where a = 0.0623 and b = -2.1394 (Ayompe *et al.,* 2011).

Validating the Studies Performed

It is essential that for all results to be refutable a validation process will be required to ensure that there has been little to no error during calculations and simulations.

As suggested by Urbina *et al*. (2005), the best method for validating mathematical models and simulations is to replicate a previous authors experimental process and successfully validated simulation. This allows a direct comparison of these confirmed results with the current study's findings. The statistical methodology used to validate this method were originally used by Ayompe *et al*. (2011) in which the experimental model was used to validate the TRNSYS model. This was subsequently used to calculate Percentage Mean Absolute Error (PMAE) and Percentage Mean Error (PME) for comparison in the study's methodology:

$$PMAE = \frac{100}{N}\sum_{i=1}^{N}\frac{|C_i - M_i|}{M_i}$$

$$PME = \frac{100}{N}\sum_{i=1}^{N}\frac{C_i - M_i}{M_i}$$

Where C_i= the calculated results
M_i= the measured result of base system study

Next, validation of the parametric model and TRNSYS simulation was replicated with the same set up conditions and environmental conditions to the experiment performed by Ayompe *et al*. (2011). This will allow the direct comparison of the results which are expected to be identical, or as close as possible. The results will be given graphically and therefore comparison will be performed to visually evaluate the similarities between the result. Then a numerical test using PMAE and PME will be performed to justify that the parametric model and TRNSYS model are valid to be used for independent studies.

Ayompe *et al*. (2011) suggests that for a simulation to give an accurate result, a maximum PMAE or PME of 18% or less is appropriate to:

- Predict long-term performance of the solar hot water system.
- Simulate the system performances in different weather conditions and locations.
- Be used to optimise the heating system.

Input error from the graphical readings from the environmental data were estimated and the errors from the measurements have been carried though the entire parametric model. This could be achieved using error propagation using differentiation of the equation to estimate the error of the result (Ku, 1966). However, this method is mathematically and time intensive, so for this parametric model the errors were inputted into the results to provide the minimum and maximum values.

Finding the Optimal System

Test Method

As a result of the findings from the parametric model, it was hypothesised that the tank size may limit performance, especially during high solar intensity. Therefore, varying the tank sizes at 0.15m^3, 0.3m^3 and 0.6m^3 will test the hypothesis that tank size will limit performance. The results of the parametric model and literature review indicate that implementing an evacuated tube collector (ETC) would improve the performance of the solar system, especially in colder climates. Therefore, multiple tests on ETCs will be undertaken in TRNSYS using collector areas of 2m^2, 4m^2, 8m^2 and 12m^2. In addition, different flat plate collector (FPC) sizes will be tested to see if the sizing may better suit the demand, using sizes 2m^2, 4m^2, 8m^2 and 12m^2.

The aim of this test, as mentioned, is to optimise the base solar collector system to provide a more cost-effective design and ideally provide faster payback periods and improved return on investments. The biggest issues faced with providing accurate results is the cost of the solar collector systems. Most systems work on a quote basis where precise costs of systems can be tailored to the requirements and provided. Therefore, in this test an estimation will be provided using multiple sources to gain a generic estimation.

Calculating System Performance

To calculate the system performance a basic system was created where annual hot water heating costs can be found under the same demand and climate conditions as shown in the base system.

Each system, including the base system, will be equipped with an external heating unit to ensure all hot water reaches the required hot water temperature of 60 degrees. The external heater is not a necessity but is an advantageous tool to measure useful energy absorbed into the system rather than that absorbed, but not contributing to, the final hot water delivery.

To adequately compare the systems the following equations will be used to aid the analysis of the financial performance each individual system;

$$Savings\ per\ year\ (£) = Anual\ costs\ without\ Solar\ System - Anual\ Costs\ with\ Solar\ System \quad (4)$$

$$Payback\ Time\ (years) = \frac{Cost\ of\ Solar\ System}{Savings\ per\ year} \quad (5)$$

$$Percentage\ Anual\ Return\ on\ Investment = \frac{Savings\ per\ year}{Cost\ of\ Solar\ System} * 100 \quad (6)$$

$$Hot\ Water\ Demand\ Provided\ by\ Solar\ System = \frac{Cost\ of\ Hot\ Water\ Without\ Solar\ system}{Cost\ of\ Hot\ Water\ With\ Solar\ System} * 100 \quad (7)$$

4. Parametric Model Calculations

A parametric model will be produced using the calculations in Microsoft Excel. This will be used once validated to also validate the TRNSYS models, as well as the base and optimised systems, to reveal where inefficiencies occur in greater detail than other techniques allow.

Flat plate collector

Throughout the flat plate collector parametric model (equation 1 to 49), the formula used, unless otherwise stated, has been adapted from Duffie & Beckman (2013).

Figure 2 shows a simplified diagram of the assumed energy inputs and losses across the plate collector. The assumptions made regarding the flat plate collector, as stated by, include a) performance is steady state, b) the flat plate collector is a sheet of parallel tubes c) the headers cover a very small area and as a result the heat losses are negligible, d) all flow in tubes are uniform, consistent and do not experience natural flow and convection, only forced flow, e) the cover does not absorb solar energy so that it effects heat transfer losses from the collector, f) heat flow through the cover is one dimensional, g) there is negligible temperature drop through the cover, h) the sky is a black body and I) there are no temperature gradients.

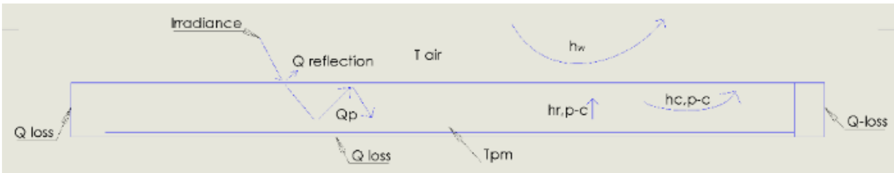

Figure 2: Assumed losses (Chekerovska & Filkoski, 2015)

The simple equation for heat energy gained is:

$$Q_{gain} = Q_{in} - Q_{loss} \tag{1}$$

Across the flat plate collector there are three primary loss factors to consider; 1) convection between the plate and the cover, 2) radiation between the plate and the cover and 3) radiation between the cover and the air. In addition, heat will be taken away from the plate as useful energy to the collector fluid and delivered to the hot water tank.

Before starting the calculations, an estimate for the fluid and plate temperatures will be made, this estimate is essential for calculating loss coefficients and will be corrected to improve the accuracy of the presumption as the calculations continue and develop.

Calculating convection losses

$$h_{c,p-c} = Nu\frac{k}{L} \tag{2}$$

Where $R_a = \frac{g\beta'\Delta T L^3}{\nu\alpha}$ and $Nu = 1 + 1.44\left[1 - \frac{1708(\sin(1.8\beta)^{1.6}}{Ra*Cos(\beta)}\right]\left[1 - \frac{1708}{Ra*Cos(\beta)}\right] + \left[\left(\frac{(Ra*Cos(\beta)}{5830}\right)^{\frac{1}{3}} - 1\right]$ (10)

Calculating radiation losses

$$h_{r,p-c} = \frac{\sigma({T_p}^2+{T_c}^2)(T_p-T_c)}{\frac{1}{\varepsilon_p}+\frac{1}{\varepsilon_c}-1},\ h_{r,c-a} = \varepsilon_c\sigma({T_c}^2 + {T_s}^2)(T_c + T_s) \tag{3}$$

$$T_c = T_p - \frac{U_t(T_p-T_a)}{h_{c,p-c}+h_{r,p-c}} \rightarrow\ U_t = \left(\frac{1}{h_{c,p-c}+h_{r,p-c}} + \frac{1}{h_w+h_{r,c-a}}\right)^{-1} \text{ (Cengel, et al., 2016)} \tag{4}$$

Calculating losses from the top, bottom and edge of the flat plate collector

$$q = h_w(T_c - T_a) + \sigma\varepsilon_p({T_c}^4 - {T_a}^4) \tag{5}$$

$$f = \left(1 + 0.089h_w - 0.1166h_w\varepsilon_p\right)(1 + 0.07866N), \tag{6}$$

$$c = 520(1 - 0.000051\beta^2), \tag{7}$$

$$E = 0.43(1 - \frac{100}{T_{pm}}) \tag{8}$$

$$U_t = \left(\frac{N}{\frac{C}{T_{pm}} \left[\left| \frac{T_{pm} - T_a}{N+f} \right| \right]^E} + \frac{1}{h_w} \right)^{-1} + \frac{\sigma({T_{pm}}^2 + {T_a}^2)(T_{pm} + T_a)}{(\frac{1}{\varepsilon_p + (0.00591 N * h_w)} + \frac{2N + f - 1 + 0.133\varepsilon_p}{\varepsilon_g} - N)}, \tag{9}$$

$$U_b = \frac{k}{L_{back}}, \tag{10}$$

$$U_e = \frac{\frac{k}{Edge\ insulation\ thickness} * Perimeter * Collector\ thickness}{Collector\ area}, \ U_L = U_e + U_b + U_t \tag{11}$$

Finding heat distribution factors

As shown by Figure 3 the flat plate acts as a fin to the tubes which increase its surface area and therefore absorbance. However, in addition to this, this increased size results in additional heat losses which result in the need for a fin efficiency factor to be derived to accurately predict the overall performance of the flat plate collector.

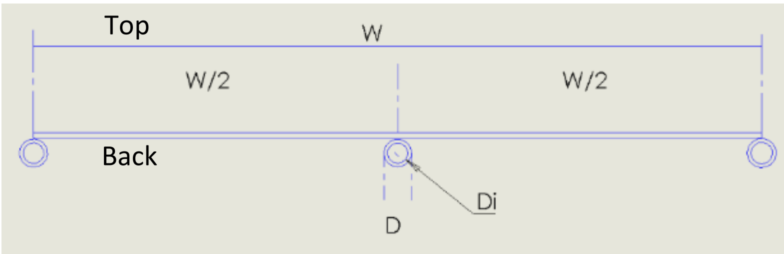

Figure 3: Flat plate collector fin dimensions.

To find the efficiency factor, an energy balance must be completed in a region of length in flow direction Δx for a fin of length (W-D)/2 shown from Figure 11 using Figure 10 the energy balance can be found using:

$$S\Delta x - U_L \Delta x(T - T_a) + \left(-k\delta \frac{dT}{dx}\right)\Big|_x - \left(-k\delta \frac{dT}{dx}\right)\Big|_{x+\Delta x} = 0 \tag{12}$$

When Δx is taken to be very close to zero:

$$\frac{d^2T}{dx^2} = \frac{U_L}{k\delta}(T - T_a - \frac{S}{U_L}) \tag{13}$$

Temperature can be assumed to be a local base temperature (T_b) because the material used in all appropriate solar collector fins are good conductors. Therefore, the temperature gradients should be very small and, in this case, can be assumed negligible:

$$\text{Hence } T\big|_{x=\frac{W-D}{2}} = T_b \text{ and } \frac{dT}{dx}\Big|_{x=0} = 0 \tag{14}$$

To reduce complexity of formulas let:

$$A = T_b - T_a - \frac{S}{U_L} \tag{15}$$

And:

$$B^2 = \frac{U_L}{k\delta} \tag{16}$$

Therefore:

$$\frac{d^2A}{dx^2} = B^2A \tag{17}$$

The general solution is:

$$A = C_1 \sinh(Bx) + C_2 \cosh(Bx) \tag{18}$$

By substituting in boundary conditions this becomes:

$$\frac{T - T_a - \frac{S}{U_L}}{T_b - T_a - \frac{S}{U_L}} = \frac{\cosh(Bx)}{\cosh(\frac{B(W-D)}{2})} \tag{19}$$

Using Fourier's law:

$$q_{fin-1side} = -k\delta \frac{dT}{dx}\bigg|_{x=\frac{W-D}{2}} = \left(\frac{k\delta B}{U_L}\right) * \left(S - U_L(T_b - T_a)\right) * (\tanh\left(\frac{B\frac{(W-D)}{2}}{2}\right)) \tag{20}$$

$$q_{fin} = (W - D)\left(S - U_L(T_b - T_a)\right) * \frac{\tanh\left(B\frac{(W-D)}{2}\right)}{B\frac{(W-D)}{2}} \rightarrow (W - D)F\left(S - U_L(T_b - T_a)\right) \tag{21}$$

Where:

$$F = \frac{\tanh\left(B\frac{(W-D)}{2}\right)}{B\frac{(W-D)}{2}} = \frac{tan\left[\frac{\sqrt{\frac{U_L}{k\delta}}(w-D)}{2\sqrt{\frac{U_L}{k\delta}}}\right]}{\left[\frac{\sqrt{\frac{U_L}{k\delta}}(w-D)}{2}\right]} \tag{22}$$

The fin efficiency shown above only includes loses and gains from the fins. To obtain a more accurate measurement of collector performance the entire collector efficiency factor needs to be accounted for.

The useful heat gain above the tube:

$$q_{tube} = D(S - U_L(T_b - T_a)) \tag{23}$$

Therefore, the total heat gain can be defined as:

$$q_u = q_{tube} + q_{fin} = \left(S - U_L(T_b - T_a)\right) + (W - D)F\left(S - U_L(T_b - T_a)\right) \tag{24}$$

To transfer this useful heat gain to the fluid it must overcome the resistance of the heat transfer to fluid from tube and the bond. This adapts the equation to include:

$$q_u = \frac{T_b - T_f}{\frac{1}{hf_i \pi D_i} + \frac{1}{C_b}} \tag{25}$$

where hf_i = heat transfer between fluid and tube wall, Di = inside diameter and Cb = bond conductance, which can be expressed as:

$$C_b = \frac{k_b b}{\Upsilon} \tag{26}$$

where k_b = bond thermal conductivity, b = bond width and ϒ = bond thickness.

To get a useful expression for q_u, T_b must be removed. This was achieved by substituting:

$$q_u = \left(S - U_L(T_b - T_a)\right) + (W - D)F\left(S - U_L(T_b - T_a)\right) \quad (27)$$

Into:

$$q_u = \frac{T_b - T_f}{\frac{1}{hf_i \pi D_i} + \frac{1}{C_b}}. \quad (28)$$

$$q_u = w\frac{\frac{1}{U_L}}{\frac{1}{U_L(D(W-D)F)} + \frac{1}{C_b} + \frac{1}{hf_i \pi D_i}}\left(S - U_L\left(T_f - T_a\right)\right) = wF'\left(S - U_L\left(T_f - T_a\right)\right) \quad (29)$$

Where:

$$F' = \frac{\frac{1}{U_L}}{\frac{1}{U_L(D(W-D)F)} + \frac{1}{C_b} + \frac{1}{hf_i \pi D_i}} \quad (30)$$

Duffie & Beckman (2013) explained that, the collector heat removal factor relates the useful heat energy gain to the heat energy gain, if the entire collector was the same temperature as the fluid inlet. This is shown as:

$$F'' = \frac{F_R}{F\prime} \text{ where } F_R = \frac{\dot{m}C_p(T_{fo} - T_{fi})}{A_c(S - U_L(T_{fi} - T_a))} = \frac{\dot{m}C_p}{A_c U_L}\frac{\left(\frac{S}{U_L} - (T_{fi} - T_a)\right) - (\frac{S}{U_L} - (T_{fo} - T_a))}{\frac{S}{U_L} - (T_{fi} - T_a)} \quad (31)$$

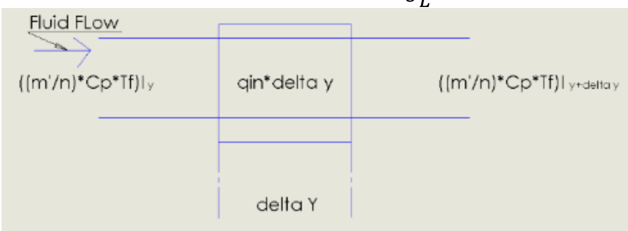

Figure 4: Energy balance diagram.

From Figure 4 an energy balance can be produced:

$$\frac{\dot{m}}{n}C_p T_f\Big|_y - \frac{\dot{m}}{n}C_p T_f\Big|_{y+\Delta y} + \Delta y q'_u = 0 \quad (32)$$

The symbol Δy shows that its length perpendicular to length of fin used previously.

When Δy is very close to zero the energy balance can be re-written by substituting:

$$q_u = WF'\left(S - U_L\left(T_f - T_a\right)\right) \quad (33)$$

Into:

$$\dot{m}C_p\frac{dT_f}{dy} - nWF'(S - U_L\left(T_f - T_a\right) = 0 \quad (34)$$

where n is the number of parallel tubes.

Duffie & Beckman (2013) suggests that assuming F' and UL as independent from position is appropriate and therefore allowing the energy balance to be solved.

$$\frac{T_f - T_a - \frac{S}{U_L}}{T_{fi} - T_a - \frac{S}{U_L}} = -\frac{U_L nWF\prime y}{\dot{m}C_p} \quad (35)$$

As suggested by Dunkel & Cooper (1975), U_L can be expressed as a linear function of Tf-Ta and if L is in the flow direction. Then T_{f0} can be found by substituting L for y allowing for this transformation;

$$\frac{T_f - T_a - \frac{S}{U_L}}{T_{fi} - T_a - \frac{S}{U_L}} = e^{-\frac{U_L nWF'y}{\dot{m}C_p}} \tag{36}$$

As $F_R = F''F' = \frac{\dot{m}C_p}{A_c U_L}\left(1 - e^{-\frac{U_L nWF'y}{\dot{m}C_p}}\right) \rightarrow F'' = \frac{\dot{m}C_p}{A_c U_L F'}\left(1 - e^{-\frac{U_L nWF'y}{\dot{m}C_p}}\right)$ (37)

Calculating Collector performance values

$$Q_u = A_c F_R (S - U_L(T_i - T_a)) \tag{38}$$

where

$$S = G_T(\alpha\tau)_{av} \tag{39}$$

$$Collector\ efficiency = \frac{Q_u}{A_c I_t} \tag{40}$$

Finding actual fluid and plate temperatures

$T_{fluid\ mean=} T_{fluid\ inlet} + \frac{\frac{Q_u}{A_c}}{F_R U_l}(1 - F'')$ and $T_{pm=} T_{fluid\ inlet} + \frac{\frac{Q_u}{A_c}}{F_R U_l}(1 - F_R)$ (41)

As a result, the process can be repeated replacing the initial estimated temperatures with the calculated ones. This process can be repeated until the fluid temperatures calculated, from the fluid temperatures estimated, vary a negligible amount. It was discovered that three iterations were optimal for the parametric model.

Heat exchanger and hot water storage tank

Dimension values

$$A_{crossectional,inner\ tube} = \frac{\pi {d_i}^2}{4} \tag{42}$$

$$A_{inner\ tube} = \pi d_i L \tag{43}$$

$$U_{tube\ fluid} = \frac{\dot{m}}{A * \rho} \tag{44}$$

$$A_{crossectional,tank} = \frac{\pi {d_{tank}}^2}{4} \tag{45}$$

$$A_{tube\ to\ tank} = \pi d_0 L \tag{46}$$

$$A_i = A_{inner\ tube} * N \tag{47}$$

where N = number of tubes,

$$A_o = A_{tube\ to\ tank} * N \tag{48}$$

NTU method for calculating final temperatures

Due to the exit temperatures in the heat exchanger and storage tank, the NTU method is required to accurately estimate the heat transfer rate of the heat exchanger, and therefore calculate the tank temperature and heat exchanger exit temperature.

Therefore, Reynolds number and Prantle number was calculated:

$$Re_{length} = \frac{\rho UL}{\mu} \tag{49}$$

And:

$$Pr = \frac{\mu * C_p}{k_{fluid}}, \tag{50}$$

(Cengel *et al*., 2016)

Nusselt number is a key component to calculating the heat transfer rate, however the formula used to determine the result and its accuracy is given by the accuracy of Reynolds number present. In pipe flow Reynolds number is regularly greater than 10000. Therefore, under the guidance of Hewitt *et al*. (1994), the Dittus-Boelter Equation for Reynolds number is most suitable.

$$Nu = 0.023 Re^{0.8} Pr^{n} \tag{51}$$

Where n=0.4 during heating and n=0.3 during cooling.

$$h_i = \frac{Nu * k_{fluid}}{d_i} \tag{52}$$

$$h_o = \frac{Nu * k_{fluid}}{d_o} \tag{53}$$

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + \frac{L}{k} \tag{54}$$

$$NTU = \frac{(U * A_o)}{(\dot{m} * C_p)} \tag{55}$$

Where $\dot{m} * C_p$ for the minimum value from either side of the heat exchanger.

$$\varepsilon = 1 - e^{-NTU} \tag{56}$$

Adapted from Navarro & Cabezas-Gomez (2007)

$$Q_{actual} = \varepsilon * Q_{max} \ where \ Q_{max} = \dot{m} C_p (T_{in\ hot} - T_{in\ cold}), \tag{57}$$

(Cengel, et al., 2016).

Validation of Parametric Model

During the set-up of the base model it was evident that some details were absent from the experimental result provided by Ayompe *et al*. (2011) which could result in some differences between the parametric model results and the experimental results. This included;

- Pipe length and insulation could lead to increased losses in the experimental results compared to the neglected pipe losses in the parametric model.
- The thermostats opening and closing temperatures between the two-hot water storage tanks could double the storage tanks capacity if the thermostat is open as this would decrease the average tank fluid temperature and increase the heat transfer rate in the heat exchanger.

The results of Ayompe *et al*. (2011) were presented on graphs, rather than tables of exact results, giving an estimated error of ± 2 Kelvin.

As shown in Figure 5 the difference between the experimental data and the parametric results were very small, with all results being within the error allowance. The only exceptions were at 9 hours which shows a difference of 12.2%, at 15 hours with a difference of 9.8% and 19 hours with a difference of 23.8%. This could be an error within the parametric model but is more likely a result of imprecise readings from the original graph. Consequently, during the assembly of the parametric calculations each method used was checked against worked examples to ensure the exact results were obtained.

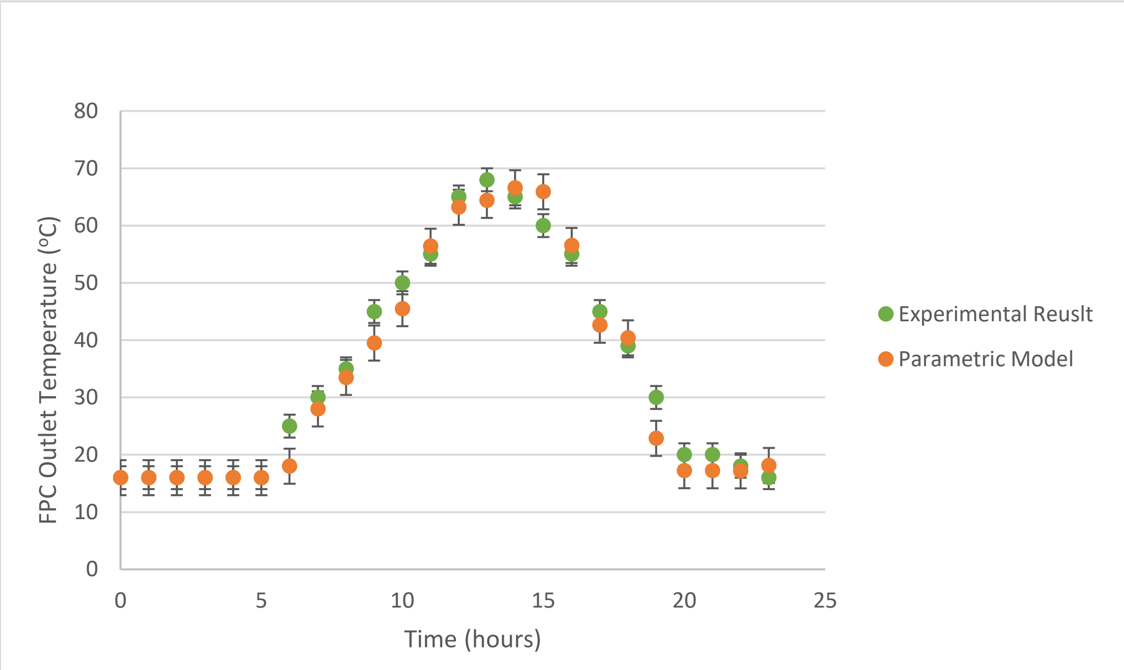

Figure 5: Comparing experimental results (Ayompe et al., 2011) with parametric model results for flat plate collector (FPC) outlet temperature (°C).

Further analysis of the results provided gave a PMAE of 13.85%, and a PME of -13.85%. The negative value indicates that these results are an under estimation of the experimental results. As suggested previously, a maximum PMAE of 18% or less would indicate that the simulation model would be a good, accurate representation of the experimental system. Therefore, the system would be appropriate for use in evaluating the existing system, as well as initiate changes to the system. This may include changes to climate, hot water demand or temperature. As this system achieved a PMAE of 13.85% it is within the allowance stated and therefore, for this specific use, it is a successfully validated system.

5. TRNSYS Simulation

Building the TRNSYS Model

Initially, the TRNSYS model was set up to represent the base system, which would be used for validation. The TRNSYS model would then be modified to meet the overall aim of the study to propose an energetically and economically optimal design for annual performance for cold climates. Figure 6 shows a simplified model of the base TRNSYS system.

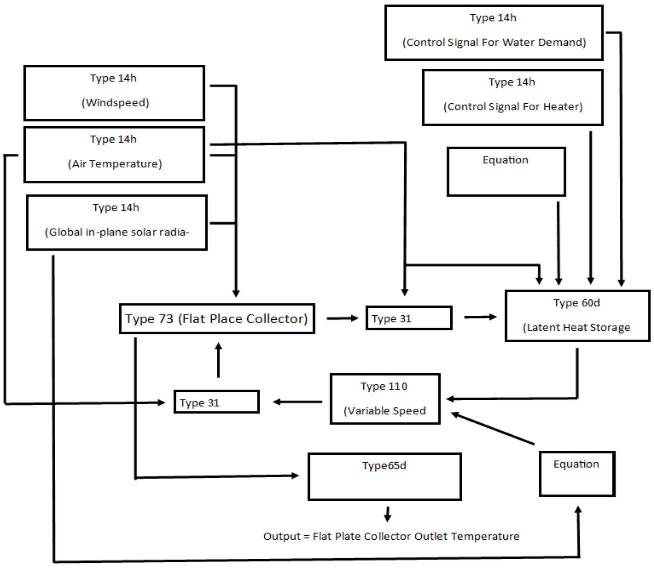

Figure 6: Simplified TRNSYS set up for base system for validation

Following the same method for the set-up of the parametric model, the flat plate collector and hot water tank were set up using parameters provided by Ayompe *et al*. (2011). The weather data was set up using Type 14h (Time Dependent Forcing Functions) opposing the more commonly used method of Type 15 Weather Data (Klein *et al*., 2011). This choice was made with the intention to accurately replicate the base system used by Ayompe *et al*. (2011). Therefore, identical weather data was required. This method was the simplest way of accurately replicating the environmental conditions. The pipe length and material were unspecified, therefore reasonable estimations of 10m copper pipes with thermal losses of 0.83W/m^{2}K, were used. The error that would be carried through is ±3.06; the same as the error in the parametric model. The calculations performed in TRNSYS are very similar to the parametric model and thus would inherit similar traits (Klein *et al*., 2012).

TRNSYS Model Validation

Following the same method for the set-up of the parametric model, some details were absent from the experimental result provided by Ayompe *et al*. (2011) which could result in some differences between the TRNSYS simulation's results and the experimental results.

As can be seen in Figure 7 there is some difference between the simulation results and the experimental results, although only a small amount. All values are within the error allowance, except 5, 12, 19 and 20 hours where values of 62.5%, 12.3%, 20% and 20% are recorded. After calculating the PMAE and PME, providing answers of 9.7% and 9.7% respectively, the positive values indicate an over approximation

Figure 8 shows the difference in results between the TRNSYS simulation results and the parametric model, to further test their validity. As show, they perform near identically where the only significant deviation can be found at hours 5 and 6 which would be at sunrise. The deviation is small and occurs at sunrise, therefore making a negligible difference to the total. The reasons for the error is likely to be a difference in pre-heater efficiency and timing. As for the TRNSYS model, all heater modes were left to default. It can be assumed that this will not invalidate the model because of the small impact on total energy absorption

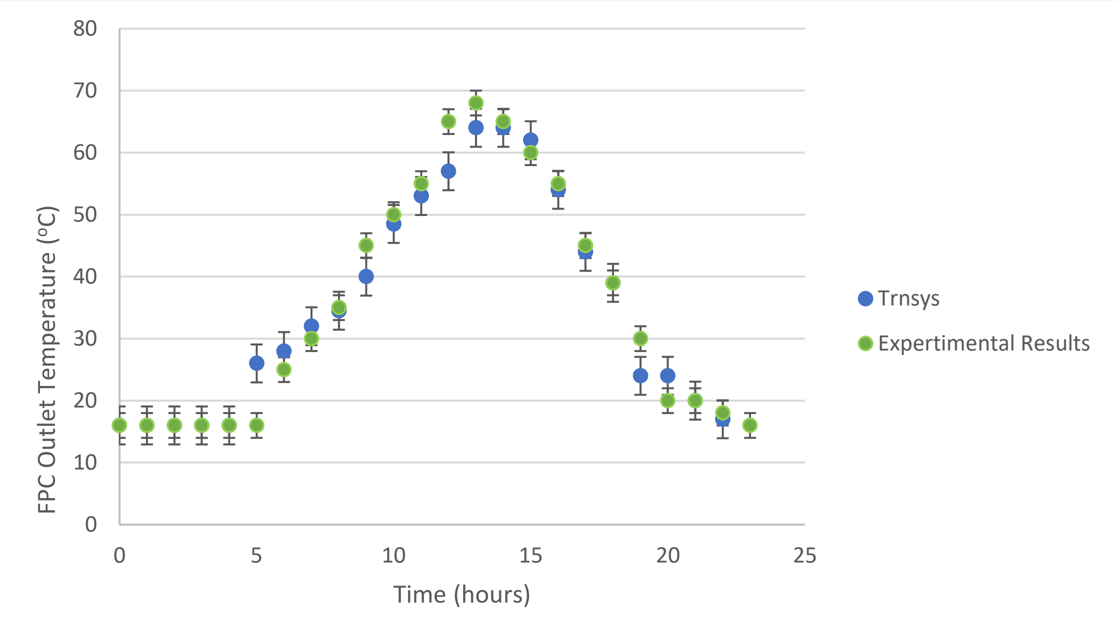

Figure 7: TRNSYS vs Experimental Results of flat plate collector (FPC) outlet temperature.

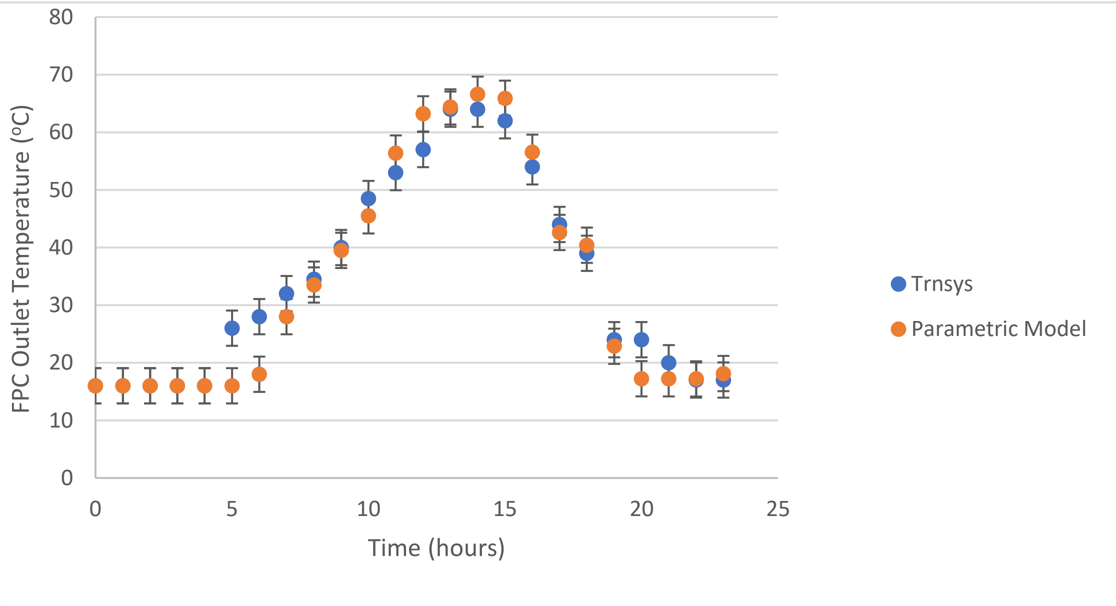

Figure 8: TRNSYS vs Parametric model results of flat plate collector (FPC) outlet temperature.

The PMAE is 4.48% and PME is -4.48%. Using the results displayed from the graphs in Figure 7 and 8 the TRNSYS model has near identical results to the parametric and experimental results. As mentioned previously, a TRNSYS model with a maximum

PMAE and PME when compared to experimental results of 18% and below, would be valid to use to test and compare systems. Therefore, despite the small amount of error, the TRNSYS model is suitable to provide reliable results while testing the base, and different, systems.

The same error of ±3.06, as calculated for the parametric model, was adopted for the TRNSYS model. The reasons for this include that the TRNSYS model adopted the same inputs as the parametric model, from the experimental data and their set-up parameters. Therefore, it is assumed that the models experience near identical error spread. This has been presumed to make a negligible difference to the error between the TRNSYS and parametric model.

The System Representation

The base system has identical properties to that of the validated base system, with the addition of the external heater. From the adapted base system, the flat plate collector sizes will be changed to $2m^2$, $8m^2$ and $12m^2$ and the tank size will also be changed to $0.15m^3$ and $0.6m^3$ as previously explained. The same alterations will be made to the system which has an evacuated tube collector.

Figure 9 shows the addition of a rock bed collector and heat exchanger. This system would use an evacuated tube collector or flat plate collector, based on which collector and size performed most effectively. In addition, a heat exchanger was added to separate the collector fluid from the hot water supply which would be used to supply the rock bed with thermal energy. Two diverting valves, type 649 mixing valve, would supply cold water to the system to make up hot water demand and a type 647 diverting valve would be used to supply hot water to the demand via the external heater. The flow rate, like the solar fluid side, would be the same and controlled by an identical variable speed pump at the same flow rate which is located after the type 647 diverting valve.

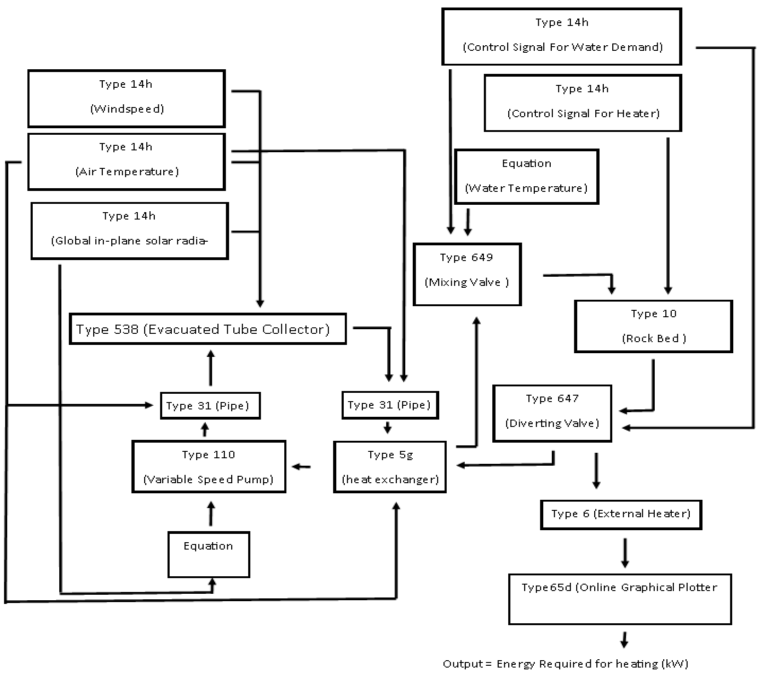

Figure 9: System rock bed storage solar collector system in TRNSYS

6. Results

Parametric Model Results

Hot Water Storage Tank

Surprisingly, one of the main factors that impacted performance of the solar collector system was the poor thermal storage flexibility of the tank. As shown in Figure 10 below, the tank size required in autumn and colder months is much smaller than that of the summer.

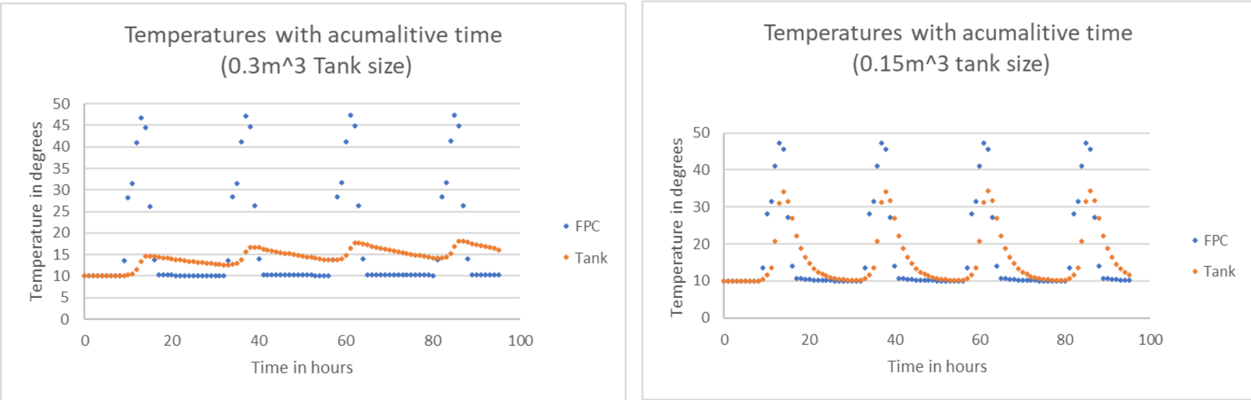

Figure 10: Tank and Flat Plate Collector temperature for tank sizes 0.3m^3 and 0.15m^3 for a typical November Day.

It was found that, to achieve useful desired temperatures outside of the hottest months in June, the tank size needed to be reduced. In the hotter summer months, a tank size less than $0.3m^3$ reduces the solar collector efficiency due to warmer tank temperatures. Rodriguez-Hidalgo *et al*. (2012) shows identical results in an industrial setting. This shows how undersized hot water storage in high solar energy climates can drastically reduce collector efficiency and how oversizing reduces usability of the absorbed energy.

Possible solutions to this problem include a) using the hot water collector as a pre-heater and removing the heating element inside; this would allow all heat energy to be used and allows for a larger hot water tank. The absorbed energy will still be supplied to the heater as demand requires. Consequently, all electrically heated water will be supplied to demand, rather than sitting in the tank below the desired temperature and resulting in larger power usage, b) as suggested by Zhao et al (2011), the use of a phase changing material could be used which has been shown to make gains for performance by increasing water temperatures and therefore energy absorption. However, a recent experiment by Kilickap *et al*. (2018) shows how phase changing materials, despite improving performance in the summer, shows only a 4-degree hot water temperature improvement. As the costs of phase changing materials are so high this improvement is not justified and c) the use of two tanks in parallel with a thermostat to the secondary tank, however this does induce more thermal losses across that tank due to the effect of volume: area ratio and increased losses through increased pipe lengths.

Flat Plate Collector

The parametric model indicated that the most noticeable component resulting in reduced performance is the flat plate collector and the reduction in efficiency as temperatures drop. The collector efficiency dropped as low as 0.31 (31%). Through analysing the parametric model there is very little noticeable change in the total heat loss coefficient, with summer having a slightly higher total heat loss coefficient (see Figure 11 for results).

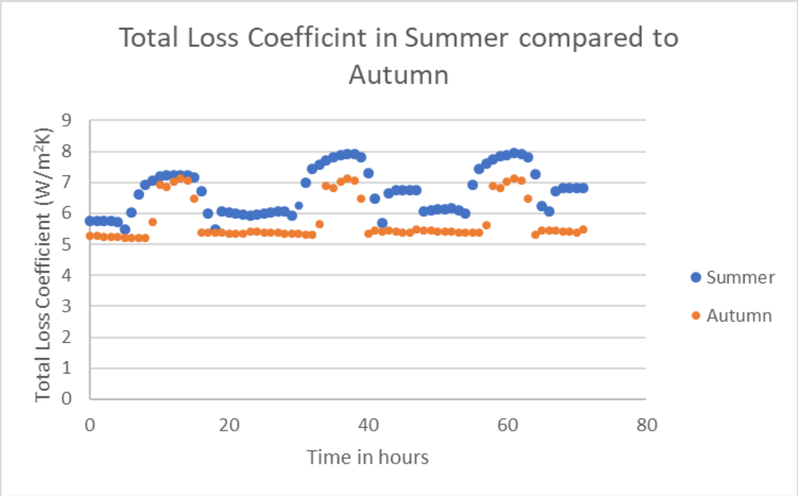

Figure 11: Total Heat Loss Coefficient in Summer Compared to Winter

The total loss coefficient is broken down into two main components; convective losses and radiation losses. Through analysing the results, the radiation losses are consistently higher in summer than winter, as would be expected, as the flat plate collector temperature is higher. However, convective heat losses are relatively unchanged, with a maximum of an estimated 40% decrease in Autumn compared to summer. A decrease would be expected due to reduced flat plate collector temperatures, but in combination with vastly reduced air temperatures, a decrease of 40% is surprising (Cengel *et al*., 2016). This directly conflicts with the results shown from an experimental and TRNSYS study performed by Sokhansefat *et al*. (2017). The study shows that the evacuated tube collector performs much better than the flat plate collectors, especially in winter as a likely result of their significantly reduced convective heat transfer losses in colder climates. However, the results from the parametric model show that the glazing from the flat plate collector is adequate to limit convective heat losses. This might suggest that the potential cold climate benefits suggested by Sokhansefat *et al*. (2017) may be less useful in this chosen set-up and weather conditions. This will be further investigated in the TRNSYS simulation to ensure these conflicting results are thoroughly investigated.

TRNSYS Results

Comparing Flat Plate Collector to Evacuated Tube Collector

The flat plate collector is directly compared to an evacuated tube collector using the base system set-up. As can be seen from Figure 12**,** the evacuated tube collector performance far exceeds that of the flat plate collector and results in a 17.04% improvement in summer and a 79.25% improvement in winter. This fully supports the study provided by Sokhansefat *et al*. (2017) showing that, in warmer higher solar energy conditions, evacuated tube collectors only marginally outperform flat plate collectors. Whereas, in lower solar energy conditions and colder ambient

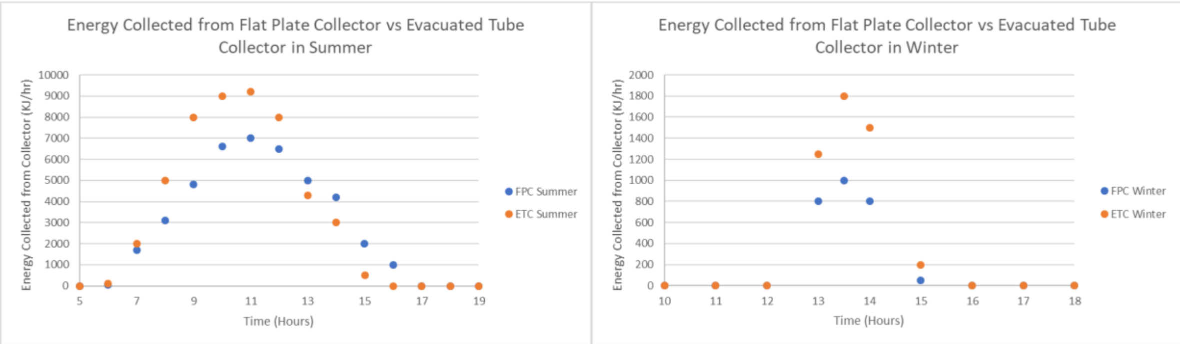

Figure 12: Energy Collected from Flat Plate Collectors and Evacuated Tube Collectors in Summer and Winter Simulated in TRNSYS

temperatures evacuated tube collectors greatly exceed flat plate collectors. This outcome is expected. As explained by Pluta (2011), the manufacturing process causes reduced thermal conductivity due to the heat energy absorbed being conducted through additional metal elements within the tube of the collector rather than directly to the fluid similarly to flat plate collectors. Hence, as solar energy is increased, the performance gain is reduced. Therefore, flat plate collectors are a far more affordable and effective heating solution for high solar energy climates.

However, this conflicts with the results suggested by the parametric model. The convective losses shown by the parametric model would suggest a greater reduced performance from the evacuated tube collector than is observed. This conclusion is supported by Pluta (2011) who found that the performance improvement from evacuated tube collectors was minimal.

Hot Water Storage Tank

As discussed previously, the hot water storage tank sizing can be responsible for greatly reduced performance. It was discovered that the use of a range of tank sizes is key for optimal tank design throughout the year with varying solar energy. All methods of variable tank sizes have their own unique problems, however as discussed by Sokhansefat *et al*. (2017) and Zhao *et al*. (2010) using latent heat energy storage may be the most effective method.

Figure 13 shows the internal energy change within the storage tank and rock bed storage whilst comparing their performance in winter and summer.

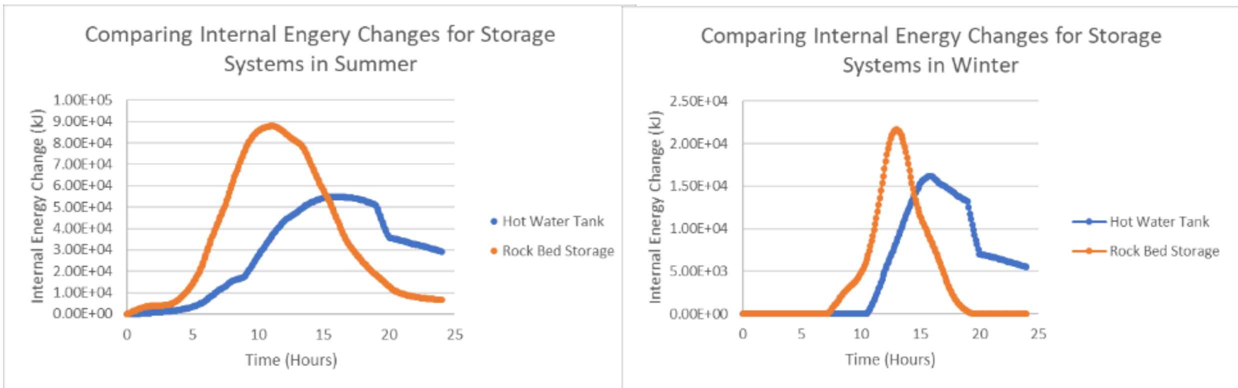

Figure 13: Comparing the Internal Energy Changes for Sensible Hot Water Storage and Rock Bed Storage for both Summer and Winter from a TRNSYS Simulation.

As can be seen from the results the rock bed performs consistently better than the hot water tank for absorbed energy. The initial assumption for this performance gain was that the flat plate collector inlet temperature remains cooler in summer, and thus is more efficient. Whereas, in winter there is lower area and therefore reduced heat losses when compared to a single hot water storage tank. However, the results presented by Buonomano *et al*. (2013) show that the performance difference between multiple variable volume tank sizes compared to just one was minimal. In fact, it is not economically viable to use variable volume tanks, even in winter. This suggests two possible outcomes;

- The improved performance is due to the phase changed materials inducing a more effective heat exchange.
- The improved performance is due to the vastly reduced thermal losses through the walls of the tank caused by the reduced surface area.

The evidence from the literature offers no clear cause that can be concluded by this study and further analysis would need to be conducted. However, there is a clear performance improvement using rock bed storage over traditional hot water storage methods.

New System Proposal

At the time of this study, the maximum UK electricity costs was £0.17 per kWh. Table 2 shows the economic results derived from the methodology described above. These results justify an improvement recommendation to successfully optimise the system investigated. Thus, meeting the main aim of the study to model a solar domestic thermal hot water system and propose an energetically, and economically, optimal design for annual performance in cold climates.

The base system (Ayompe *et al*., 2011) has the most optimal configuration using the flat plate collector and hot water storage tank set-up. However, this is not the most effective choice of technology. This study has shown that the evacuated tube collector is the best choice for winter performance. Nevertheless, its expenses and lack of pricing make it very difficult to compare. The rock bed storage technology has the most potential as it provides the greatest amount of energy saving. However, it’s not easily available for domestic applications.

Overall, the most economically and energetically optimal system is the evacuated tube collector at the 2m^2 system parameters. This system has a 2m^2 single evacuated tube collector connected to a 3m^3 hot water storage tank (HM 300L) which contains a coil heat exchanger (U44332) where the flow of the solar fluid is driven by a variable speed pump. Pipe length is dependent on the house specific user requirement. The insulation provides a conductance of 0.8W/m^{2}K. This system produces the optimal financial results, providing a very short payback period of 7.35 years which far exceeds the base system suggested by Ayompe *et al*. (2011) and provides the best lifetime savings of a minimum of £4473.10. This far exceeds the base systems lifetime savings of £2883.30, giving a 55% improvement over a 20-year period. The energy savings are fully validated and consequently reliable. The main limitation of this study is the difficulty in accurately pricing the system. Nevertheless, the aim to model a solar domestic thermal hot water system with an energetically and economically optimal design for annual performance for cold climates has been met by the results of this study.

Table 2: Economic results given by the study output for an optimally designed solar domestic thermal hot water system.

Electricity Cost per kWh (£)	**Electricity demand per month in Summer (kWh)**	**Electricity demand per month in Spring/ Autumn (kWh)**	**Electricity demand per month in Winter (kWh)**	**Annual Electricity Costs for hot water system (£)**
0.17	11	11	12	711.45

System Type	**Cost of System (£)**	**Annual Energy Savings (£)**	**Payback Period (years)**	**Lifetime savings (£)**
Base	2500	269.16	9.28	2883.30
FPC $2m^2$	2000	232.26	8.61	2645.30
FPC $8m^2$	3500	319.26	10.96	2885.05
FPC $12m^2$	4500	306.53	13.73	2053.41
FPC ($0.15m^3$ Water tank)	3500	307.38	11.39	2647.67
FPC (Base $0.3m^3$ Water tank)	3500	319.26	10.96	2885.05
FPC ($0.6m^3$ Water tank	4000	301.91	13.25	2038.24
ETC $2m^2$	2600	353.66	7.35	4473.1
ETC $4m^2$	3250	371.87	8.74	4187.72
ETC $8m^2$	4550	389.17	11.69	3233.41
ETC $12m^2$	5850	379.45	15.42	1738.96
ETC with Rock Bed Storage	10270	415.01	24.74	-1969.75

Further Research

TRNSYS was a useful tool in this project, however it has not been used to its full potential. The software is capable of far more complex and precise simulations in which entire houses can be modelled, therefore further research could be used to suggest the most optimal collector system. This could even predict performance accurately for each house sizing, and could suggest ideal boiler locations relative to the demand and house orientation.

Conclusions

Using the mathematical methods and experimental data provided by Duffie & Beckman (2013) and Ayompe *et al*. (2011) a validated parametric and TRNSYS model were produced. This model was used to test multiple systems to find the best economic performance for domestic hot water systems in colder climates.

The base system suggested by Ayompe *et al*. (2011) was the best configuration for conventional, older technologies. The flat plate collector sizing was optimal for the average household and was relatively cheap, with a good payback period. The hot water tank sizing was ideal for storing the optimal amount of heat energy and for a sufficiently long period. However, this system is easily outperformed by the evacuated tube collector design, and most of the possible arrangements, proposed in this study.

Ultimately the optimal economics of the domestic hot water system depends heavily on the capital the individuals should spend and how long they intend to gain from the system. For quick paybacks and great short term gains the 2m^2 ETC system is ideal, however for long term (20+ year) investments the 4m^2 ETC system is ideal.

The rock bed storage collector shows promise, giving a huge improvement on the sensible heat storage tank, however costings were difficult to estimate and price estimations were far too expensive to justify the purchase, despite the far more desirable performance, especially in the cold seasons and climates.

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