A NUMERICAL ANALYSIS FOR THE DYNAMIC PERFORMANCE OF A MULTI-PURPOSE SOLAR THERMAL SYSTEM FOR RESIDENTIAL APPLICATIONS

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ABSTRACT

This paper presents results of a multi-purpose solar thermal system that provides hot service water, space heating and space cooling for residential use during all seasons in Pretoria, South Africa. A pressurized system using evacuated tube solar collectors with internal heat pipes to provide the required hot water and a micro single-effect LiBr -H_2O absorption chiller to provide the required space heating and cooling was considered. For the solar field, collectors consisting of 25-tubes (3.266 m^2 each) are connected in parallel in a 3, 4 and 5 array field. In this study, the focus was on determining the hourly performance trends of the considered system using typical meteorological year weather conditions. The performance of the system was obtained by developing the system's mathematical model whose solution was obtained using Engineering Equation Solver (EES). The model was validated using available experimental data and good agreement was obtained. The absorption chiller model was validated using data from ASHRAE and was shown to be valid within ±2.2%. From the analysis, results show that the 25-tube collector with a 5 array field could operate without interruption throughout the seasons, yielding tank temperatures at 111.75°C and 95.18 °C for the summer and winter seasons, respectively. However, the recommended system, with the lowest scalding hazard risk, is that of the 25-tube collectors with a 4 array field. This system produced a cooling capacity of 5.94-7.3 kW at a cooling coefficient of performance (COP) of 0.78-0.8 in winter and 4.24-5.94 kW at a cooling COP of 0.77-0.73 in summer.

INTRODUCTION

The South African solar thermal market is growing due to increased interest in utilizing this technology for domestic solar water heating (SWH) in residential areas to negate high energy consumption from the common electrical geysers [1]. Several studies have shown that these systems can reduce the monthly consumption by 30%-50% [2]. However, most South African households are not yet fully persuaded to invest in these technologies. This may be due to factors such as low hourly performance trends and lower maximum system capability, or due to incorrect construction parameters that lead to overdesigned systems. The lack of acceptance may also be due to limited functionality, as with the single purpose SWH currently offered in the residential market [1].

Aside from domestic water heating, air conditioning creates another high energy consuming application in buildings. Currently, vapour compression systems, using significantly large amounts of energy and using refrigerants that are harmful to the environment, are commonly used for air condition in buildings all over South Africa. According to the country's energy providers, heating, ventilation and air-conditioning (HVAC), account for approximately 4000 GWh of electricity consumption on an annual basis [3]. A space cooling and heating system utilizing solar energy would assist in reducing environmental impact and reduce electricity need.

NOMENCLATURE

$\dot{Q}, Q^{\cdot}$	[W]	Heat transfer rate
$\dot{m}$	[kg/s]	Mass flow rate
Δx	[mm]	Thickness
COP	[-]	Coefficient of performance
Cp	[kJ/kg.K]	Specific heat
D	[m]	Diameter
h	[J/g]	Enthalpy
L	[m]	Length
M	[kg]	Mass
P	[kPa]	Pressure
R	[K/W]	Thermal resistance
S	[W]	Absorbed irradiance
T	[°C]	Temperature
UA	[W/K]	Overall heat transfer coefficient
V	[m^3]	Volume
X	[-]	Refrigeration-absorbent solution
x	[-]	Refrigerant solution
DEM	[W]	Demand
Special characters		
ε	[-]	Emissivity
η	[%]	Efficiency
ρ	[kg/m^3]	Density
Subscripts		
a		Absorber
am		Ambient
c		Condenser
d		Desorber
DW		Dewar tube
e		Evaporator
f		Fluid
hp		Heat pipe
i		Inlet/inner
INS		Insolation
IT		Inner tube
l		Application loss
man		Manifold

mu	Public main
n	New
o	Outlet/Outer
OT	Outer tube
s	Storage
sat	Saturation
SHX	Solution heat exchanger
st	Storage tank
tl	Tank loss
u	Useful

Few solar thermal heating and cooling systems have been installed or studied so far in South Africa. The first solar powered space cooling unit was installed for cooling a network building for MTN [4]. Bvumbe and Inambao [5] studied and later implemented a cooling unit for a hospital in Pretoria. Although this particular unit was decommissioned due to overdesigning there are some initiatives regarding the use of such systems. The adoption in the country is not widespread and these systems have not been adopted for domestic use.

There are several studies on the design and construction of micro capacity chillers (<10 kW) for residential use. Nevertheless, the micro chillers remain commercially unavailable. Franchini, et al. [6] modelled, designed and constructed a 5 kW LiBr-H_2O absorption chiller. They created a computer code to analyze the system to demonstrate the feasibility of such a chiller by changing operational parameters of the cycle. Florides, et al. [7] presented a method to evaluate the characteristics and performances of a single-stage LiBr-H_2O absorption machine. Theoretical equations for mass and energy balances and the heat exchanger design for the machine were solved with a computer program (EES). In a second study, Florides, et al. [8] used TRNSYS to model an 11 kW LiBr-H_2O single-stage absorption chiller for a typical home in Cyprus. This investigation was directed towards the long-term performance, economic analysis, and global warming impact. The investigators concluded that such a system is feasible, especially with the environmental impact in mind. In Turkey, Atmaca and Yigit [9] also studied such a system. A modular computer program was developed to simulate various cycle configurations and solar energy parameters. The results emphasize the effect of hot water inlet temperatures on the coefficient of performance and the surface areas of absorption cooling exchangers. Various types of collectors were studied and it was concluded that the evacuated tube collector (ETC) is the system of choice. Arsalis and Alexandrou [10] designed and modelled a solar heating and cooling (SHC) system to satisfy a single domestic home's thermal loads. The main purpose of the study was to model the overall system and conduct a parametric study that determines the optimum economic system performance in terms of design parameters. Fong and Lee [11] studied the technical effectiveness of SHC units for a typical village house in subtropical Hong Kong. They investigated two systems that consist of driving a single effect LiBr-H_2O chiller by ETC's. The two systems are connected by either separated or integrated collector arrays, in which a year-round dynamic simulation, utilizing TRNSYS, was performed. It was concluded that such a system is technically feasible for such an environment and that the integrated system performed better.

As revealed in the previous studies, the design and performance of a multi-purpose solar powered space heating and cooling system depend on numerous factors such as the solar array size, operational parameters, environmental conditions and the type of refrigeration cycles used. It is also clear that few South African studies have been undertaken to characterize the performance of multi-purpose solar thermal systems. This investigation focus on the prediction of seasonal hourly performance trends. Moreover, the investigation of micro-sized absorption systems specific for South African weather conditions would improve the feasibility and understanding of this kind of technology.

SYSTEM CONFIGURATION

Figure 1 shows the multi-purpose solar thermal system that is considered in this study. An ETHPC absorbs available solar radiation, heats up the heat pipes working fluid and transports the heated fluid to the manifold. A pump circulates a secondary fluid through the manifold, this fluid absorbs the heat and transports it to storage Tank 1. Multiple storage tanks were considered to provide the required demand and a relief valve is used to relieve high pressures in the solar loop. An auxiliary heater is used to assist in increasing the working fluid temperature when a certain temperature tolerance is reached. Tank 1 receives make-up water from the city supply, provides water for tank 2 and is used to maintain the water temperature in Tank 2. A secondary pump transports the hot water to Tank 2, in which the occupant's consumption is supplied, and the intended application. The application can be switched between space cooling (SSC) and heating (SSH) by means of the application switch. The fluid that heats the thermally activated refrigeration system returns to Tank 1 in which it is reheated and the cycle repeated. Experimentally, the system parameters can be measured by the equipment marked FM (flow meter), PM (pressure transducer) and TM (thermocouple). Temperatures can also be measured at state points throughout Tank 1 by the DTC (differential temperature controller).

A single-effect LiBr-H_2O absorption refrigeration system, as identified in Figure 1 as the absorption cycle, is divided into two cycles, namely a power cycle and a refrigeration cycle, as represented on a Dühring chart in Figure 2. The power cycle develops pressure for the condenser and the evaporator, similar to a vapour-compressor system. This is accomplished in an absorption refrigeration system by absorbing low-pressure vapour into an absorbing liquid (absorber), which is then transported, by means of a low powered pump, into a desorber. The desorber introduces heat to the solution and releases a high-pressure vapour. The vapour refrigerant then moves to the refrigerant cycle. Meanwhile, the absorbent returns to the absorber by passing through an expansion device to repeat the cycle. The refrigeration cycle accepts the vapour refrigerant and passes it through a condenser that releases the heat into the surrounding environment/component (SSH). When the refrigerant is cooled to a liquid state it is passed through the refrigeration expansion device to satisfy the pressure difference and enters the evaporator. The refrigerant absorbs the heat from the surrounding environment/component and returns to the absorber to repeat the cycle again (SSC)[12-14].

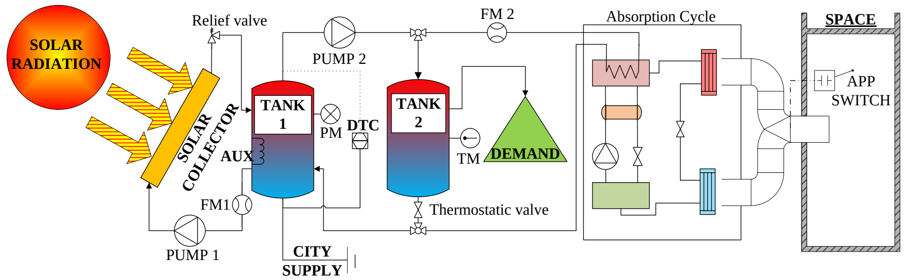

Figure 1 The multi-purpose solar thermal system schematic

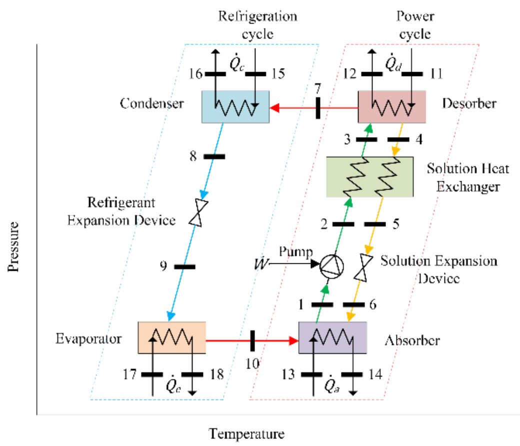

Figure 2 The Dühring plot of a simple absorption system illustrating the power & refrigeration cycle (modified from Herold, et al. [14])

NUMERICAL METHOD

ETHPC system

The thermal electrical analogy of an evacuated tube with internal heat pipe configuration collector (ETHPC) is illustrated in Figure 3.

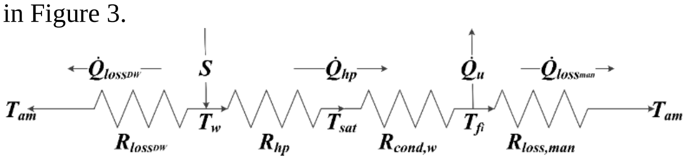

Figure 3 The electrical analogy for an ETHPC

Allowing for steady state performance of the collector, an energy balance equation can be determined as

$$S = \dot{Q}_{loss_{Dw}} + \dot{Q}_{hp} = \dot{Q}_{loss_{Dw}} + \dot{Q}_u + \dot{Q}_{loss_{man}} \quad (1)$$

In which, $\boldsymbol{S}$ is the product of the incident solar radiation and the transmittance-absorptance product with regard to a chosen sky model; $\dot{\boldsymbol{Q}}_{\boldsymbol{loss_{Dw}}}$ the product of the overall heat loss of the Dewar tube components; $\dot{\boldsymbol{Q}}_{\boldsymbol{hp}}$ the heat pipe rejected energy to the circulating fluid flowing inside the manifold; $\dot{\boldsymbol{Q}}_{\boldsymbol{loss_{man}}}$ the heat loss from the manifold itself; and $\dot{\boldsymbol{Q}}_{\boldsymbol{u}}$ the useful heat transferred out of the collector into the required system.

A single simplified expression is developed for the useful energy obtained from the ETHPC in terms of all collector components, in which the absorbed solar radiation, manifold fluid temperature, and ambient temperature are known:

$$\dot{Q}_u = \frac{S}{1+R_{hr}+R_{cr}} - \frac{T_f - T_{am}}{1+R_{hr}+R_{cr}}\left(\frac{1}{R_{loss_{DW}}} + \frac{1}{R_{loss_{man}}}(1 + R_{hr} + R_{cr})\right) \quad (2)$$

In which, the following ratios are used in the numerical manipulation

$$R_{hr} = \frac{R_{hp}}{R_{loss_{DW}}} \quad \& \quad R_{cr} = \frac{R_{cond,w}}{R_{loss_{DW}}} \quad (3)$$

The temperature leaving the manifold can be acquired by means of the formula [15]

$$\dot{Q}_u = \dot{m} * C_p * (T_{fo} - T_{fi}) \quad (4)$$

Lastly, the collector efficiency can be obtained as the ratio of actual useful heat to the absorbed solar radiation.

$$\eta = \frac{\dot{Q}_u}{S} \quad (5)$$

Storage system

Assuming a fully mixed (un-stratified) storage scenario, an energy balance for a storage tank can be acquired as

$$\dot{Q}_{st} = \dot{Q}_u + \sum \dot{Q}_l + \dot{Q}_{tl} \quad (6)$$

Where the storage capacitance ($\dot{\boldsymbol{Q}}_{\boldsymbol{st}}$), at uniform temperature and operating over a finite temperature difference ($\Delta \boldsymbol{T}_s$), is expressed as

$$\dot{Q}_{st} = (M * Cp)_s * (\Delta T_s) \quad (7)$$

In which, $\boldsymbol{M}$ is the mass of storage capacity.

Then, the rate of energy loss from the tank itself can be expressed in terms of $(\boldsymbol{UA})_s$ being the storage tank's overall heat transfer coefficient-area product and $\boldsymbol{T}_{\boldsymbol{env}}$ is the storage tank's surroundings environment temperature [15].

$$\dot{Q}_{tl} = (UA)_s * (T_s - T_{env}) \quad (8)$$

Considering the application load ($\dot{\boldsymbol{Q}}_{\boldsymbol{l}}$), the crucial parameter of water consumption /demand ($\boldsymbol{DEM}$) over a certain time period can be determined [16] as

$$DEM = V * \rho * Cp * \frac{(T_s - T_{mu})}{t} \quad (9)$$

$$\sum \dot{Q}_l = DEM + \dot{Q}_d \quad (10)$$

In which, $\dot{\boldsymbol{m}}$ is the mass consumption rate for a set time period, $\boldsymbol{V}$ is the volumetric consumption capacity, $\boldsymbol{T}_s$ the water distribution temperature, $\boldsymbol{T}_{mu}$ the public main water temperature and $\dot{\boldsymbol{Q}}_{\boldsymbol{d}}$ the heat rate required for the absorption cycle. In simulation, hourly repetitive water demand profiles should be used [17, 18].

Generally, with some mathematical manipulation, the fully mixed storage tank temperature to be supplied to the next collector loop ($\boldsymbol{T}_{s\text{-}n}$) can be obtained as by [16]

$$T_{s-n} = T_s + \frac{\Delta t}{(M * Cp)_s} * (\dot{Q}_u - \sum \dot{Q}_l - (UA)_s * (T_s - T_{env})) \tag{11}$$

Absorption System

The cycle plot, shown in Figure 2, is modelled by assigning mass and energy balances to each component of the system to obtain the thermal properties of each state point. The mass and energy balances are formulated for each component in Table 1 and Table 2. Note that $\boldsymbol{x}$ represents the refrigerant quality and $\boldsymbol{X}$ that of the refrigerant-absorbent solution.

Table 1 Mass balances of the absorption cycle states

Component	Mass balance	State
Pump	$\dot{m}_1 = \dot{m}_2$ $X_1 = X_2$	LiBr-H_2O solution
SHX	$\dot{m}_2 = \dot{m}_3$ $\quad \dot{m}_4 = \dot{m}_5$ $X_2 = X_3$ $\quad X_4 = X_5$	LiBr-H_2O solution
SED	$\dot{m}_5 = \dot{m}_6$ $X_5 = X_6$	LiBr-H_2O solution
Absorber	$\dot{m}_{10} + \dot{m}_6 = \dot{m}_1$ $\dot{m}_{10}x_{10} + \dot{m}_6X_6 = \dot{m}_1X_1$	LiBr-H_2O solution
Desorber	$\dot{m}_4 + \dot{m}_7 = \dot{m}_3$ $\dot{m}_4X_4 + \dot{m}_7x_7 = \dot{m}_3X_3$	LiBr-H_2O solution
Condenser	$\dot{m}_7 = \dot{m}_8$ $x_7 = x_8$	Inlet vapour
RED	$\dot{m}_8 = \dot{m}_9$ $x_8 = x_9$	Liquid refrigerant
Evaporator	$\dot{m}_9 = \dot{m}_{10}$ $x_9 = x_{10}$	Inlet liquid

Table 2 Energy balances of the absorption cycle states

Component	Energy balance
Pump	$\dot{W} = \dot{m}_2h_2 - \dot{m}_1h_1$
SHX	$\dot{Q}_{SHX} = \dot{m}_2h_2 - \dot{m}_3h_3 = \dot{m}_4h_4 - \dot{m}_5h_5$
SED	$h_5 = h_6$
Absorber	$\dot{Q}_a = \dot{m}_{10}h_{10} + \dot{m}_6h_6 - \dot{m}_1h_1$
Desorber	$\dot{Q}_d = \dot{m}_4h_4 + \dot{m}_7h_7 - \dot{m}_3h_3$
Condenser	$\dot{Q}_c = \dot{m}_7h_7 - \dot{m}_8h_8$
RED	$h_8 = h_9$
Evaporator	$\dot{Q}_e = \dot{m}_{10}h_{10} - \dot{m}_9h_9$

The performance of the system can be calculated after completion of the component modelling. Using a similar method to Stoecker and Jones [12] the cooling load coefficient of performance (*COP*) can be calculated as:

$$COP_{cooling} = \frac{\dot{Q}_e}{\dot{Q}_d} \tag{12}$$

The COP for the heating load is calculated according to [14]

$$COP_{heating} = \frac{\dot{Q}_a + \dot{Q}_c}{\dot{Q}_d} = COP_{cooling} + 1 \tag{13}$$

ENVIRONMENTAL CONDITIONS

In this study, a typical meteorological year (TMY) was generated, based on data received from the South African Weather Service (SAWS) for Pretoria (25.75°S, 28.21°E) [19, 20]. Figure 3 illustrates the irradiance and ambient temperatures for 21 June (winter), and 21 December (summer). A maximum wind velocity constant was taken as 7.3 m/s and 5.1 m/s for these seasons, respectively [20].

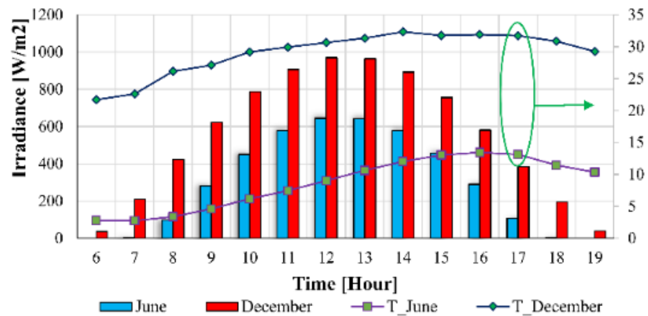

Figure 3 The solar irradiance and ambient temperature

The hourly repetitive water demand/consumption profiles used for a typical South African townhouse are illustrated in Figure 4. Water is supplied to users at 60 °C to eliminate bacterial growth, as recommended by Duffie and Beckman [16].

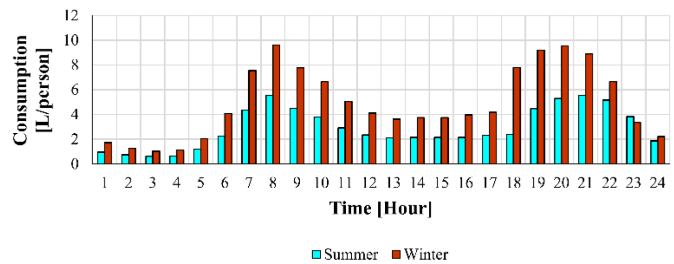

Figure 4 The average hourly water consumption for the winter and summer seasons in South Africa for a 4 occupant family [18]

COMPONENT SPECIFICATIONS

The ETHPC receivers consist of a borosilicate glass Dewar tube (ε_g= 90%) with a copper internal heat pipe operating with water as the internal fluid (saturation temperature of 25 °C). The dry type pressurized manifold is operated at a pressure of 400 kPa (saturation temperature of 143.65 °C). The manifolds' internal fluid is water, layered with a copper internal shell, polyurethane foam insulation and a stainless steel outer layer. The design specifications of the collector are shown in Table 3. The storage tanks have a volume capacity of 150 kg each with a U-Value of 0.16 W/m^2K [16].

The single-effect LiBr-H_2O absorption chiller theoretical specifications are indicated in Table 4. These parameters were selected to supply 4.678 kW heating and 4.505 kW cooling load, at a cooling COP of 0.784 to a residential space at start-up. The system has a start-up temperature of 75 °C and a minimum evaporative exit temperature of 4 °C.

Table 3 Solar collector specifications [21, 22]

Component	Parameter	Value	Unit
Glass tube	D_{o_o}	58	mm
	D_{o_i}	46.2	mm
	L	1800	mm
	Δx	2.2	mm
Absorber tube coating	ε_r	6	%
	α	94	%
Copper evaporator	D_o	8	mm
	Δx	1.2	mm
Copper condenser	D_o	14	mm
	L	57.5	mm
	Δx	1.2	mm
Manifold	D_i	60	mm
	Δx_{IT}	2.4	mm
	Δx_{INS}	100	mm
	Δx_{OT}	1.4	mm
	l_{pitch}	75	mm

Table 4 Absorption cycle specifications

Component	Parameter	Value		Unit
Desorber	UA	0.7		kW/K
	$\dot{m}_{11}$	0.2		kg/s
	T_{11}	75		°C
		Cooling	Heating	
Absorber	UA	0.7	0.7	kW/K
	$\dot{m}_{13}$	0.2	0.2	kg/s
	T_{13}	T_a	24	°C
Evaporator	UA	0.7	0.7	kW/K
	$\dot{m}_{17}$	0.2	0.2	kg/s
	T_{17}	21	T_a	°C
Condenser	UA	0.7	0.7	kW/K
	$\dot{m}_{15}$	0.2	0.2	kg/s
	T_{15}	T_a	24	°C
SHX	ε	0.64		-
Pump	$\dot{m}_1$	0.03		kg/s

Validation of Numerical Models

The model results that evaluates the thermal characteristics and performance of the solar collector was compared with results obtained from an experimental study by Jafarkazemi, et al. [23]. When the developed EES Program is solved for a 4 tube collector and using the environmental conditions used in their study, the obtained outlet temperatures are compared, as shown in Table 5. The model is accurate within ± 3% of the experimental values. As such, we can conclude that the model accurately predicts the performance of an ETHPC.

Likewise, the model developed for evaluating the thermal characteristics and performance of the absorption cycle is validated by comparing the models results with the results obtained with a simulation investigation from ASHRAE [24].

Table 5 Comparison between experimental and model results

Time of day	T_{Inlet}	$T_{experimental}$	T_{model}
10:00:00 AM	44,4	47,74	47,7
10:38:00 AM	49,4	53,03	52,5
14:20:00 PM	50,9	55,57	55,3
14:45:00 PM	42	47,04	46,8

Table 6 Absorption model vs. simulation results

	COP	$\dot{Q}_e$ [kW]	$\dot{Q}_a$ [kW]	$\dot{Q}_c$ [kW]	$\dot{Q}_d$ [kW]	P_L [kPa]
Model	0.683	2193	3049	2352	3208	0.686
ASHRAE	0.68	2148	2984	2322	3158	0.697
Error %	0.54	2.09	2.18	1.29	1.58	1.49

The model's values are accurate to within 2.18% of the simulation values. It is concluded that that the model accurately predicts the performance of the absorption cycle. Therefore, it follows that the model is acceptable for study purposes and accurately illustrates what is expected in practical application.

TRENDS AND RESULTS

This investigation is accomplished by analyzing only the hourly tank temperatures recorded from the array of parallel connected ETHPC's. Considered collector parameters are shown in Table 7.

Table 7 Collector arrays investigation parameters

Parameter	Value	Unit
Slope angle	30°	Degrees
Manifold mass flow rate	0.03	kg/s
Receiver array	25	Tubes
3 Array absorption area	9.80	m^2
4 Array absorption area	13.06	m^2
5 Array absorption area	16.33	m^2

Figure 5 and 6 shows the periods when the system is active as a multi-purpose solar thermal system (SWH & SSC/SSH), the absorption machine limits has been met, and as such indicating the values on and above the boundary line, or only supplying hot water to occupants (SWH), below the boundary line.

In both these figures, it is illustrated that the 25-tube receiver collector mounted with 5 arrays produces the highest tank temperatures and are able to operate the absorption cycle during both seasons without failing, yielding tank temperatures at 111.75 °C and 95.18 °C for the summer and winter seasons, respectively. However, this system configuration can produce temperatures hazardous to residential users, presenting the risk of scalding, overheating and early corrosion of the components. For these reasons, a system producing lower temperatures is preferred such as that of the 25-tube collector with 4 arrays. This system produced a cooling capacitance of 5.94-7.3 kW at a cooling COP of 0.78-0.8 in winter and 4.24-5.94 kW at a cooling COP of 0.77-0.73 in summer.

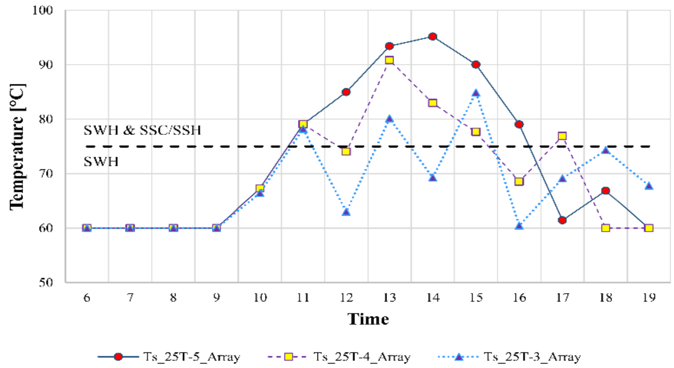

Figure 5 Dynamic performance trend: June

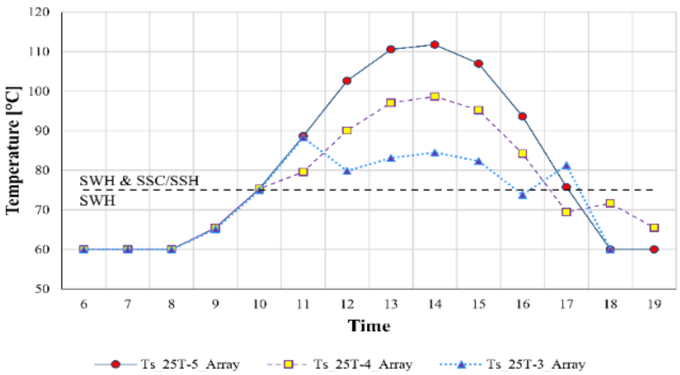

Figure 6 Dynamic performance trend: December

In Figure 7 the 25 collector with 3 array is shown. It can be visualized that the temperature curve fluctuates with the heat demand variations when the system is operating in the SWH or SWH & SSC/SSH zone. Although the water consumption load profile remained the same ($\dot{Q}_{Demand}$), the total usage load ($\dot{Q}_{Usage}$) increased in periods in which the absorption machine is operable. In these periods the tank temperature for the following collector loop (T_{s-n}) decreases due to the netto usage surpassing the collector arrays' net useful heat ($\dot{Q}_u$), obtaining a negative storage capacitance ($\dot{Q}_s$).

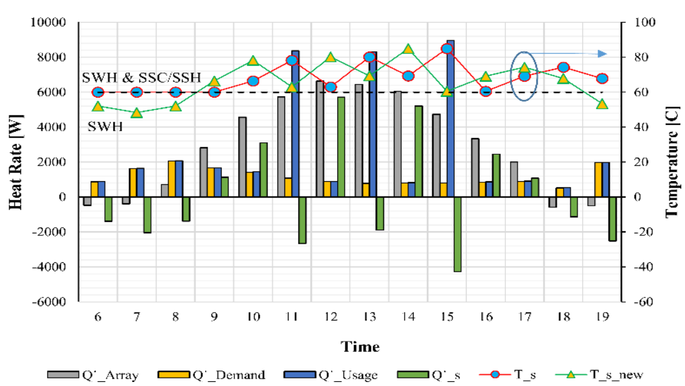

Figure 7 Hourly storage capacitance and temperature drops observed in a multi-purpose solar thermal system in June

CONCLUSION

In this investigation, a numerical model was developed and implemented in Engineering Equation Solver to investigate the dynamic performance trends of a multi-purpose solar thermal system. To validate this model for evaluating the thermal characteristics and performance of the solar collector and the absorption cycle, the results from this study have been compared with results obtained from previous literature and found within the good agreement to produce reliable results.

The tank temperature curves fluctuating in the SWH and SWH & SSC/SSH zones was observed in regards to the usage load variations when the absorption cycle is operable. It was observed that the 25-tube collector with 5 parallel connected array configurations produced the highest tank temperatures and would be able to operate the multi-purpose solar thermal system in both seasons. However, temperatures hazardous to residential users might develop. Considering this hazard, the 25-tube collector with 4 arrays was viewed to be more acceptable to utilize.

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References

[1] REN21, "Renewable energy 2014 global status report," Paris: REN21 Secretariat 978-3-9815934-2-6, 2014.

[2] Centre for Renewable and Sustainable Energy Studies and AEE-INTEC. (2014, 2015/03/09). *The South African solar thermal technology road map* [A discussion document]. Available: www.solarthermalworld.org/sites/gstec/modules/pubdlcnt/pubdlcnt.php?file=http://www.solarthermalworld.org/sites/gstec/files/story/2014-12-05/solar-thermal-road-map-working-document-3-nov-2014.pdf&nid=63774

[3] ESKOM. (2015, 2015/04/09). *ESKOM Fact sheet -Demand side management-air-conditioning facts*. Available: www.eskom.co.za/.../DSM_0002AirConditioningFactsRev7.pdf

[4] CNBAfrica. (2014, 2015/02/12). *MTN Launches Africa's first solar cooling system*. Available: http://www.cnbcafrica.com/news/technology/2014/07/11/mtn-launches-africa%E2%80%99s-first-solar-cooling-system/

[5] Bvumbe T. J. and Inambao F. L., "Operational evaluation of the performance of a solar powered absorption system in Pretoria," *Journal of Energy in Southern Africa* Vol. 24,2012 pp. 26-32.

[6] Franchini G., Notarbartolo E., Pedovan L. E., and Perdichizzi A., "Modelling, Design and Construction of a Micro-Scale Absorption Chiller," *Energy Procedia*, Vol. 82,2015 pp. 577-583.

[7] Florides G. A., Kaligirou S. A., Tassou S. A., and Wrobel L. C., "Design and Construction of a LiBr-Water absorption machine," *Energy Conversion and Management*, Vol. 44,2002 pp. 2483-2508.

[8] Florides G. A., Kaligirou K. A., Tassou S. A., and Wrobel C., "Modelling, simulation and warming impact assessment of a domestic-size absorption solar cooling system," *Applied Thermal Engineering*, Vol. 22,2002 pp. 1313-1325.

[9] Atmaca I. and Yigit A., "Simulation of solar-powered absorption cooling system," *Renewable Energy*, Vol. 28,2003 pp. 1277-1293.

[10] Arsalis A. and Alexandrou A. N., "Parametric study and cost analysis of a solar heating-and-cooling system for detached single family households in hot climates," *Solar Energy*, Vol. 117,2015 pp. 59-73.

[11] Fong K. F. and Lee C. K., "Investigation of separate or integrated provision of solar cooling and heating for use in low-rise residential building in subtropical Hong Kong," *Renewable Energy,* Vol. 75,2014 pp. 847-855.
[12] Stoecker W. F. and Jones J. W., *Refrigeration & Air Conditioning*, 2 ed. Singapore: Jay's Publishers services, Inc, 1982.
[13] Eastop T. D. and McConkey A., *Applied Thermodynamics for engineering Technologists*, 5 ed. England: Pearson Education Limited, 1993.
[14] Herold K. E., Radermacher R., and Klein S. A., *Absorption chillers and heat pumps,* 2 ed. Florida: CRC Press, 2016.
[15] Cengel Y. A. and Ghajar A. J., *Heat and mass transfer : Fundamentals and applications*, 4 ed. New York: Mc Graw Hill Companies,Inc., 2011.
[16] Duffie J. A. and Beckman W. A., *Solar engineering of thermal processes*, 4th ed. Canada: John Wiley & Sons, Inc., 2013.
[17] ASHRAE, *ASHRAE Handbook, Heating, Ventilating and Air-Conditioning Applications (SI Edition)*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 2007.
[18] Meyer J. P. and Tshimankinda M., "Domestic hot water consumption in South African townhouses," *Energy Convers.,* Vol. 39,1996 pp. 679-684.
[19] South African Weather Service, "Global, beam and diffuse irradiance of Pretoria, South Africa (January 1957- December 1987) ", ed, 2015.
[20] South African Weather Service, "Meteorological data of Pretoria UNISA, South Africa (January 2009 -December 2014)," ed, 2015.
[21] SOLARRAY. (2015, 2015/07/22). *High pressure solar water heater catalogue*. Available: http://solarpowergeyser.co.za/high-pressure-solar-water-heater-150-l/
[22] ITS Solar. (2015, 2015/07/21). *Evacuated tube specifications*. Available: http://www.itssolar.co.za/download.php?file=evac/ITS-10-15-20-Evacuated-Tube-Specifcations.pdf
[23] Jafarkazemi F., Ahmadifard K., and Abdi H., "Energy and exergy of heat pipe evacuated tube solar collectors " *Thermal Science,* Vol. 20,2016 pp. 327-335.
[24] ASHRAE, *ASHRAE Fundamentals Handbook*. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc., 2001.