Analysis of a solar assisted micro-cogeneration ORC system

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Abstract In this work, three solar assisted thermodynamic cycles for a micro-cogeneration system with a power output of 5 kW are studied. The thermodynamic cycles are based on the Organic Rankine Cycle (ORC) and the operating temperatures of solar thermal collectors are 80°C, 100°C–150°C and 200°–250°C, for cycles 1, 2 and 3, respectively. The main work objective was to model the selected cycles for optimisation according to the temperature range. The performance of several fluids was evaluated from a thermal and an economical point of view. The integration of the micro-cogeneration system with solar thermal collectors was evaluated and solar fractions were calculated for the climatic conditions of Almeria (Spain), Tunis (Tunis) and Cairo (Egypt).

Keywords micro-cogeneration; solar energy; organic Rankine cycle; solar fraction

Nomenclature

A	area [m^2]
c_p	fluid specific heat [J/kg/K]
f	solar fraction [-]
k	incidence angle modifier [º]
h	enthalpy [J/kg]
i	incidence angle [º]
I	incident radiation on collector surface [W/m^2]
$\dot{m}$	water mass flow rate [kg/s]
P	pressure [Pa]
Q	thermal energy [J, kWh]
$\dot{Q}$	thermal power [kW]
T	temperature [ºC]
x	vapour quality [-]
$\dot{W}$	electrical power [kW]

Greek letters

η	efficiency [-]
η_0	optical efficiency [-]

Subscripts

amb	ambient
annual	annual
boiler	boiler

col	collector
cond	condenser
elect	electrical
in	inlet
input	heat from the solar system and boiler
HX	heat exchanger
out	outlet
pump	pump
reg	regenerator
solar	solar
tank	thermal storage
water	water

1. Introduction

Existing large-scale thermal plants for power generation are usually located far away from centres of population. This prevents efficient utilization of a reasonable proportion of the waste heat produced. Moreover, current technology limits these power stations to a maximum efficiency of about 40%, which, after the transportation of electricity through the grid, is reduced to about 30% [1]. This means that vast quantities of fossil fuels are burnt with unwanted pollutants entering the atmosphere.

Solar radiation availability in the Mediterranean area (Europe and North Africa) is excellent when compared with other regions of the World. However, this resource has been poorly utilised. The use of solar energy with conventional energy sources, for combined heat and power (CHP) in buildings, reduces pollutant emissions and offers energy savings. This cogeneration strategy is a major objective of the European Union energy policy.

Micro-generation is the decentralized production of electricity, through different means (micro-turbines, fuel cells, Stirling engines, small internal combustion engines, PV cells) with an electrical power output up to 50 kW. Micro-cogeneration, or micro-CHP, is the combination of micro-generation with useful heat.

In this work, three solar assisted thermodynamic cycles for a micro-cogeneration system are studied. The thermodynamic cycles are based on the organic Rankine cycle (ORC) and the operating temperatures of solar thermal collectors are 80ºC, 100ºC–150ºC and 200º–250ºC, for cycles 1, 2 and 3, respectively. The micro-cogeneration system under analysis uses a micro-turbine and an electric generator with a power output of 5 kW. The main work objective is to model the selected cycles for optimisation according to the temperature range.

The performance of several fluids was evaluated from a thermal and an economical point of view, taking also into account fluid toxicity. The integration of the micro-cogeneration system with solar thermal collectors was evaluated and solar fractions were obtained for the climatic conditions of Almeria (Spain), Tunis (Tunis) and Cairo (Egypt).

2. Description of the solar assisted micro-cogeneration system

The micro-cogeneration system under analysis uses a micro-turbine and an electric generator with a power output of 5 kW. Such small turbines are recently available in the market, and may have typical overall efficiencies around 70%. Freepower [2] commercialises a combined heat and power (CHP) system that produces 6 kW of electricity; the micro-turbine efficiency is 73%, and the operating temperature is 165ºC at 11.6 bar (turbine inlet). Recently, Eneftech SA [3] has developed scroll turbines with an electrical power output of 5 kW, adapted to organic fluids, with global efficiencies of about 70%. Turbo-generator global efficiency is equal to the ratio of electricity output to fluid energy input (enthalpy change). This type of turbine was considered in this study.

The system is composed of two circuits: the primary circuit where the working fluid expands in the turbine, passes into a regenerator and condenses in the condenser, and the secondary circuit with the solar collectors. In the secondary circuit two possibilities were studied: solar collectors without thermal storage tank – Figure 1 – and solar collectors connected to a thermal storage tank – Figure 2. In both cases a boiler is considered for auxiliary energy supply (gas as energy backup) when necessary.

A heat exchanger transfers heat between the two circuits. The heat rejected in the condenser can be used for water or space heating/cooling.

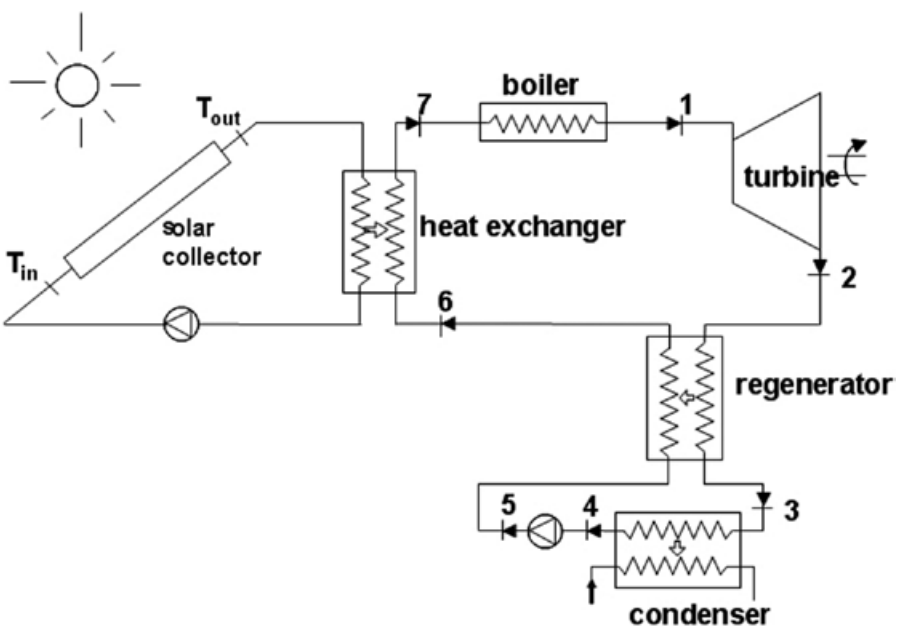

Figure 1. *Schematic representation of the micro-CHP system without thermal storage.*

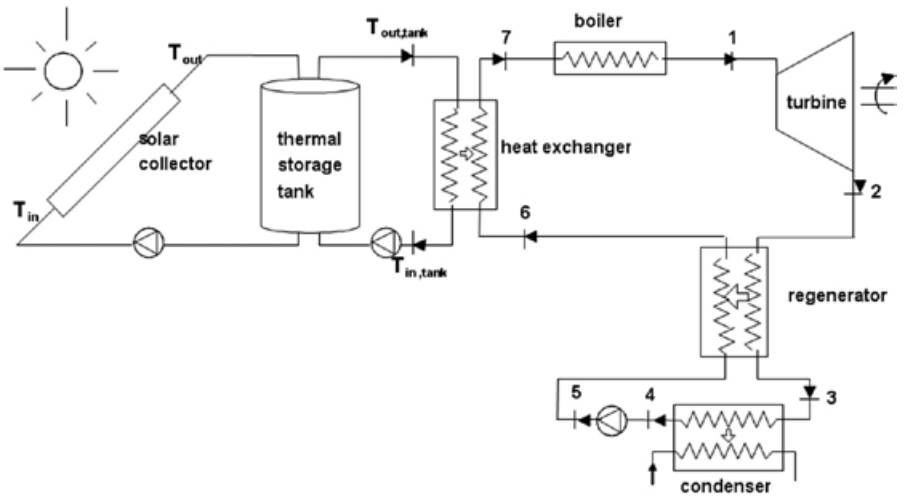

Figure 2. *Schematic representation of the micro-CHP system with thermal storage tank.*

Organic fluids are desirable for low temperature applications, due to their high molecular weight and positive slope of the saturated vapour curve in the temperature-entropy plane, both attributes simplifying the design of the expander [4]. Several fluids were analysed for the power circuit: ammonia, n-pentane, HFE7100, methanol, cyclohexane, toluene, R245fa and water.

3. System modelling and results

3.1 Power circuit (primary circuit)

The inlet temperatures, point 1 in Figure 1, were 80ºC, 120ºC and 230ºC, respectively for cycles 1, 2 and 3. At the condenser outlet the fluid is at 45ºC (saturated liquid). Micro-cogeneration systems driven at low/medium temperatures (around 100ºC) generate an amount of heat which is much higher than the generated power, and thus coupling with a thermal heat pump may be ideal.

The efficiency of electricity production was calculated through equation 1. This efficiency represents the ratio between electrical power obtained from the micro-cogeneration system ($\dot{W}_{elect} - \dot{W}_{pump}$) and thermal power input in the boiler and solar system, through the heat exchanger ($\dot{Q}_{input}$).

$$\eta = \frac{\dot{W}_{elect} - \dot{W}_{pump}}{\dot{Q}_{input}} \tag{1}$$

Regeneration was considered if the temperature at the turbine outlet (point 2) was higher than the temperature at the pump outlet (point 5). A regenerator efficiency of 80% was considered:

$$\eta_{reg} = \frac{h_2 - h_3}{h_2 - h(P_2, T_5)} \tag{2}$$

The turbine inlet pressure was optimised to maximise electrical efficiency (equation 1). This optimisation allows us to answer the question: is it better to have saturated vapour or superheated vapour at turbine inlet? The optimisation was done with EES software [5].

Table 1 presents a comparison of different fluids for cycle 1. The turbine inlet and outlet pressures (P_1 and P_2), the ΔT of superheating at point 1, the heat required to obtain 5 kW of electricity (the heat from solar system and gas burner, ($\dot{Q}_{input}$), the condenser heat ($\dot{Q}_{cond}$), the efficiency (η), the specific vapour consumption (s.v.c.) and the quality at turbine outlet (x_{out}) are presented. The simulations were done with EES software. The specific vapour consumption gives an idea about system size.

It was found that wet fluids (water, methanol and ammonia) present the best thermal performances, although differences are not significant. However, the micro-turbine considered doesn't allow the use of wet fluids. Table 2 shows the prices and risks of the different working fluids.

As can be seen in Tables 1 and 2, within the dry fluids, toluene and cyclohexane present higher efficiencies, but are however, toxic and flammable. When price, efficiency and risk are taken into account, cycloheaxane can be considered as the best fluid for the power circuit (primary circuit).

Table 3 presents the results for cycle 2. When analyzing cycle 2, it was found that cyclohexane still led to the best performance results, for the same reasons as cycle 1, with an electricity generation efficiency of 12.4% ($\dot{Q}_{input}$ = 40.4 kW and $\dot{Q}_{cond}$ = 35.5 kW).

The choice of the fluid for cycle 3 was more complicated. For this cycle, where the operating temperatures of the solar thermal collectors are 200º–250ºC, it is necessary to consider that there are four fluids with a critical temperature below 230ºC: n-pentane, HFE7100, Ammonia and R245fa. Another constraint was considered, since according to the turbine manufacturer, the inlet pressure in point 1 is limited to 2500 kPa.

Table 1 *Thermal performance of several fluids for cycle 1*

Fluid	P_1 [kPa]	P_2 [kPa]	superheating ΔT [K]	$\dot{Q}_{input}$ [kW]	$\dot{Q}_{cond}$ [kW]	η %	s.v.c. [kg/Wh]	x_{out} %
Water	47.4	9.6	0	74.1	69.1	6.74	0.02175	95.63
n-pentane	364.7	136.7	0.26	75.3	70.4	6.64	0.1433	–
HFE 7100	184.8	60.7	0	76.1	71.2	6.57	0.4552	–
Methanol	177.7	43.7	0	74.3	70.0	6.73	0.04615	95.88
Cyclohexane	99	30	0.041	75.2	70.2	6.65	0.1317	–
Ammonia	4141	1782	0.1453	74.9	70.3	6.67	0.05114	92.02
Toluene	38.4	9.9	0.536	74.6	74.3	6.70	0.1249	–
R245fa	788.7	295.6	0	76.3	71.6	6.54	0.2823	–

Table 2. *Prices and risk of several fluids*

Fluid	Price* (€/litre)	Risk Statements
water	–	–
n-pentane	33	EF and T
HFE 7100	151	NR
Methanol	42	HF and T
Cyclohexane	65	HF and T
Ammonia**	185 (100 gr)	F and T
Toluene	79	HF and T
R245fa	n.a.	NR

EF: Extremely Flammable
HF: Highly Flammable
T: Toxic
NR: No Risk statements
*Prices offered by Sigma-Aldrich Company in March 2008
**Ammonia solutions have different prices and risk, depending on the solution

Table 3. *Thermal performance of several fluids for cycle 2*

Fluid	P_1 [kPa]	P_2 [kPa]	superheating ΔT [K]	$\dot{Q}_{input}$ [kW]	$\dot{Q}_{cond}$ [kW]	η %	s.v.c. [kg/Wh]	x_{out} %
Water	198.5	9.59	0	39.7	34.7	12.59	0.01136	91.94
n-pentane	896.9	136.7	0.41	41.1	36.3	12.14	0.07172	–
HFE 7100	504.2	60.67	0	41.6	36.7	12.02	0.2098	–
Methanol	630.5	43.7	0	40.4	35.8	12.39	0.02456	92.05
Cyclohexane	286.3	30	0.2408	40.4	35.5	12.36	0.0654	–
Ammonia	9116	1782	0	42.3	38.0	11.83	0.0333	76.41
Toluene	131	9.916	0.3247	40.1	35.5	12.45	0.06232	–
R245fa	1920	295.6	0	42.7	38.2	11.70	0.1477	–

Table 4 presents the results for cycle 3 for the fluids that are below a critical temperature of 230ºC.

For the fluids that are above a critical temperature of 230ºC, the performance increases with the turbine inlet pressure. Table 5 presents the results for these fluids when the pressure is fixed at 2500 kPa (the maximum according to the micro-turbine manufacturer); n-pentane presents the best result, with an efficiency of about 20%.

Taking all constraints into consideration (efficiency, toxicity, price, critical temperature and turbine inlet pressure) cyclohexane still led to the best performance results, with an electricity generation efficiency of 22.2% ($\dot{Q}_{input}$ = 22.5 kW and $\dot{Q}_{cond}$ = 17.6 kW).

3.2 Solar circuit (secondary circuit)

The solar circuit and its components were modelled with EES software coupled with the TRNSYS simulation program, [6].

Table 4. *Thermal performance of fluids that are below the critical temperature of 230ºC for cycle 3*

Fluid	P_1 [kPa]	P_2 [kPa]	superheating ΔT [K]	$\dot{Q}_{input}$ [kW]	$\dot{Q}_{cond}$ [kW]	η %	s.v.c. [kg/Wh]	x_{out} %
Water	2795	9.59	0	22.2	17.2	22.55	0.00611	84.63
Methanol	641	43.65	6	24.3	19.5	20.60	0.01478	83.42
Cyclohexane	2100	29.98	0.12	22.5	17.6	22.25	0.03077	–
Toluene	1229	9.9	0.08	22.0	17.2	22.70	0.02892	–

Table 5. *Thermal performance of fluids that are above the critical temperature of 230ºC for cycle 3 – P1 limited to 2500 kPa.*

Fluid	P_1 [kPa]	P_2 [kPa]	ΔP above Pcrit [kPa]	$\dot{Q}_{input}$ [kW]	$\dot{Q}_{cond}$ [kW]	η %	s.v.c. [kg/Wh]	x_{out}
n-pentane	2500	136.7	–864	25.3	20.5	19.77	0.03666	–
Ammonia	2500	1782	–8833	112.3	107.4	4.45	0.0664	–
R245fa	2500	295.6	–1139	30.5	25.9	16.39	0.08704	–

The solar behaviour of the first 2 cycles (cycle 1 and cycle 2) was simulated for: Almeria (Spain), Tunis (Tunis) and Cairo (Egypt). The south facing collectors were associated in parallel and tilted 31º for Almeria and Tunis, and 25º for Cairo. The climatic data were obtained through Meteonorm, provided by TRNSYS.

Solar collector areas between 10 m^2 to 100 m^2 were studied. The fluid flowing in the collectors was water. When the thermal storage tank was used, it was modelled as a fully-mixed tank and the results were compared with a model using a thermally stratified tank. This sensible energy storage tank was modelled with TRNSYS software.

For the solar collectors, a water flow rate of 0.02 kg/s/m^2 was used and a storage volume equal to $50 \cdot A_{col}$ (in litres) was considered. A control system activates the circulation pump, so that water is circulated in the collectors only when the outlet temperature is higher than the storage temperature.

Flat plate collectors were considered for cycle 1, with efficiency parameters supplied by AES [7]. For cycle 2, a compound parabolic concentrator (CPC) was selected and the efficiency parameters were supplied by AO SOL [8].

For cycle 1, solar collector efficiency depends on the incidence angle modifier (k) – which is a function of incidence angle (i) – ambient temperature (T_{amb}), inlet water temperature (T_{in}) and incident radiation on collector surface (I):

$$\eta = k \cdot 0.8 - 2.5 \cdot \left(\frac{T_{in} - T_{amb}}{I} \right) \tag{3}$$

$$k = 1 - 0.136 \cdot \left(\frac{1}{\cos i} - 1 \right) \tag{4}$$

For incidence angles higher than 80º, solar contribution was set to zero.

For cycle 2, solar collector efficiency was calculated through, [8]:

$$\eta = \eta_0 - 1 \cdot \left(\frac{T_{in} - T_{amb}}{I} \right) \tag{5}$$

In this case, the dependence of optical efficiency (η_0) on incidence angle (i) is expressed by:

$$\begin{cases} \eta_0 = 0.61 + 0.002 \cdot i \quad for \quad i \in [0,35] \\ \eta_0 = 0.628 - 0.009 \cdot i \quad for \quad i \in [35,67] \\ 0 \quad for \quad i \in [67,180] \end{cases} \tag{6}$$

After obtaining the heat required to generate 5 kW of electricity ($\dot{Q}_{input}$), it is possible to determine the solar contribution by a thermal balance. Solar fraction (f) represents the percentage of energy input in the micro-cogeneration system that is due to solar energy. The fluid in the primary circuit is cyclohexane with water in the secondary circuit.

The monthly average solar fraction depends on the total heat transferred between fluids in the heat exchanger (Q_{HX}) and Q_{input}:

$$f = \frac{Q_{HX}}{Q_{input}} = \frac{Q_{HX}}{Q_{boiler} + Q_{HX}} \tag{7}$$

When the solar collectors are directly connected to the heat exchanger (without storage tank), Q_{HX} is equal to the useful energy gain in the solar collector (Q_{solar}). The useful solar input (instantaneous) is calculated through:

$$\dot{Q}_{solar} = \left(\dot{m} c_p \right)_{water} \left(T_{out} - T_{in} \right) \tag{8}$$

For cycle 1 and with cyclohexane as the fluid in the primary circuit, the power required to generate 5 kW of electricity ($\dot{Q}_{input}$) is equal to 75.2 kW (see Table 1). For cycle 2, $\dot{Q}_{input}$ is equal to 40.4 kW (see Table 3).

It was assumed that the heat exchanger thermal efficiency was constant (75%), operating in single-phase or two-phase mode. When using the storage tank, water and cyclohexane flow rates in the heat exchanger are the same.

Figure 3 shows the annual average solar fraction for different collector areas, for the 3 cities and for cycle 1. Due to utilization of the thermal storage tank, the system

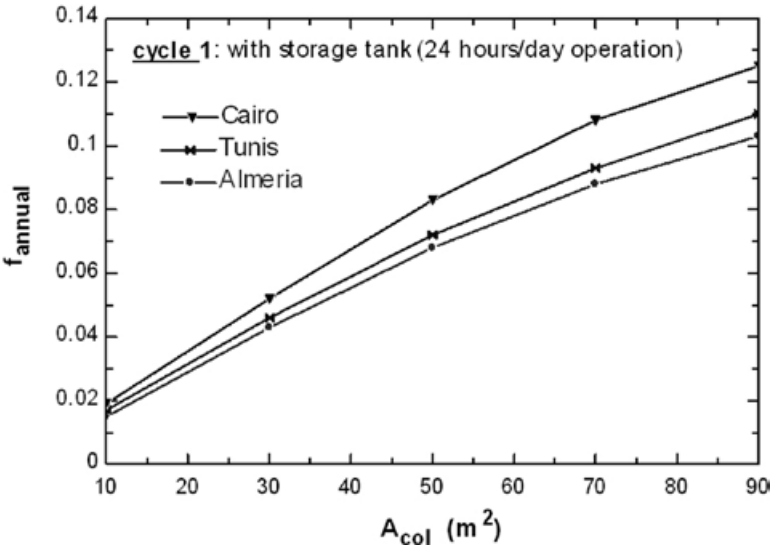

Figure 3. *Variation of average annual solar fraction with solar collector area for the 3 cities and with storage tank (24 hours/day operation), for cycle 1.*

may operate 24 h/day. In this case, the average solar fraction was calculated for a full year.

When the system operates without storage, operation from 9 to 19 h was considered (see Figure 4). In this case, the annual solar fraction is higher (about 3 times) when solar collectors are connected directly to the heat exchanger without a storage tank. But in this case, the micro-cogeneration system only operates during part of the day (when solar radiation is available). Thus, the total solar energy output is smaller, for lower requirements. The advantage of using thermal storage is that the system may operate for a longer period, with a higher total output. The annual solar fraction for different solar collector areas showed no variation when a tank with thermal stratification was considered.

Figure 5 shows the annual average solar fraction for different collector areas, with and without the storage tank, for cycle 2.

Comparing the 3 cities, Cairo has the highest solar fractions for cycle 1 and cycle 2.

4. Conclusion

The thermal performance of organic Rankine cycles generating 5 kW of electricity, and operated at 3 different inlet turbine temperatures, was simulated. The performance of several fluids was studied from a thermal and economical point of view, taking also fluid toxicity into account. Cyclohexane presented the best performance for cycles 1 and 2. For cycle 3, R245fa presents the best performance, although for

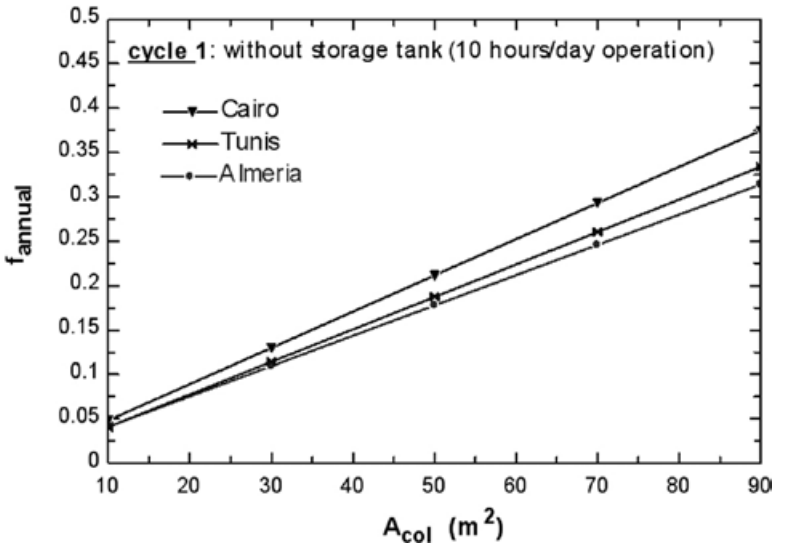

Figure 4. *Variation of average annual solar fraction with solar collector area for the 3 cities and without storage tank (10 hours/day operation), for cycle 1.*

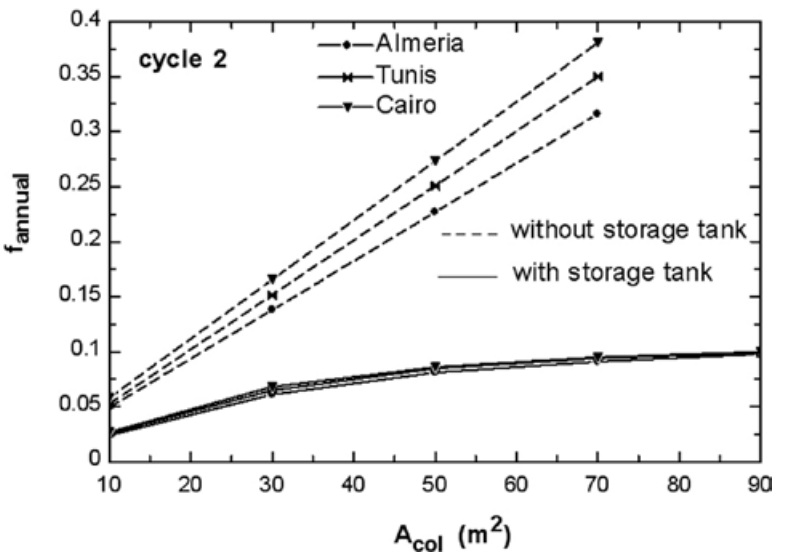

Figure 5. *Variation of average annual solar fraction with solar collector area for the 3 cities, with storage (24 hours/day operation) and without storage (10 hours/day operation), for cycle 2.*

a high pressure. If we limit the pressure to 2500 kPa, cyclohexane presents the best performance.

The integration of solar collectors, with and without a storage tank, in the micro-cogeneration system was analysed. The solar fraction for cycle 1 and cycle 2 was evaluated for Almeria, Tunis and Cairo. Comparing the 3 cities, Cairo presented the highest solar fraction for collector areas between 10 and 90 m^2. The advantage of using thermal storage with a boiler as a backup is that the system may operate for longer periods (up to 24 h/day).

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References

[1] A. C. Oliveira, C. Afonso, J. Matos, S. Riffat, M. Nguyen and P. Doherty, 'A Combined Heat and Power System for Buildings Driven by Solar Energy and Gas', *Applied Thermal Engineering*, 22 (2002), 587–593.
[2] Freepower Ltd, 2005, http://www.freepower.co.uk.
[3] Eneftech Innovation SA., Switzerland. www.eneftech.com.
[4] K. Koai, N. Lior and H. Yeh, 'Performance Analysis of a Solar-Powered/Fuel-Assisted Rankine Cycle with a Novel 30 hp Turbine', *Solar Energy*, 32(6) (1984), 753–764.
[5] S. A. Klein, 'Engineering Equation Solver', *F-Chart Software*, Middleton, USA 2007.
[6] TRNSYS, A Transient Simulation Program. User's Manual, Version 16, Solar Energy Laboratory, University of Wisconsin-Madison, 2005.
[7] AES, Tunisia, www.aes-tunisia.com/.
[8] AO SOL, Energias Renováveis S.A., Portugal, www.aosol.pt.