

## **SMALL HYBRID SOLAR POWER SYSTEM: FIRST FIELD TEST RESULTS**

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### **ABSTRACT**

This paper presents field tests of an original concept of a small hybrid solar power plant integrating three technologies: hermetic volumetric scroll expander-generators installed in two superposed Organic Rankine Cycles (ORC), a (bio-)Diesel engine with heat recovery exchangers and a solar field made of two rows of sun following flat plate concentrators with vacuumed isolated collector tubes.

The basic idea of the concept is to exploit the synergy between equipment, use cheap and maintenance free expander-generators, guaranty power availability at all time and improve the efficiency of the engine when it has to operate alone at night time by converting the waste heat with the solar ORC. This type of hybrid power plant is intended for rural electrification purposes in developing countries or cogeneration in applications like heated swimming pools in other countries.

Pressurized hot water is used at this time as a thermal fluid in the collectors with HCFC123 in the topping cycle and HFC134a in the bottoming cycle.

The field tests have been performed during the summer 2001 in Lausanne (Switzerland) and the plant proved operationally reliable. However performance results (with exergetic efficiencies up to 45%) did not meet the expectations but measures to improve the concept have been identified.

### **INTRODUCTION**

In most developing countries, electricity grids are available mainly, and often only, in urbanized areas. At the beginning of this XXI<sup>st</sup> century, numerous rural areas

throughout the world still do not have access to a reliable source of electricity, fact which impairs their development. Even if development involves a complex problematic which cannot be solved only by access to electricity, its availability together with a rational use of it can have very positive effects. On the other hand, poor areas are most of the time isolated and decentralized electricity production seems to be the most rational solution, especially in large countries, like South-Africa for example. Moreover, this production mode avoids grid losses and transportation costs, and is suitable for waste heat utilization (co-generation).

Different concepts of small decentralized power plants currently exist, but for sustainability reasons solar energy is often considered especially in countries with a significant average solar radiation. However, the drawbacks of solutions relying only on solar supply are well-known and include, among others, the low density of incident energy requiring large collector areas and therefore high investment costs. Another problem is the production dependence on the meteorological conditions with the associated reduced power availability unless bulky and expensive storage systems are introduced. When using photovoltaics the latter drawback is often avoided by adding a fossil fuel engine, generally Diesel, resulting in two technologies in parallel with only a weak exploitation of the synergies between these equipments. However, technological developments integrating solar thermal power plant technologies with similar technologies fired by easily storable fuels open new and more interesting perspectives. The general idea in so-called Integrated Solar Fossil Cycle Systems (ISFCS) is to provide heat both from solar collectors and from a cogeneration power plant or engine, reducing then the above mentioned drawbacks and offering a way to gradually substitute fossil fuel by solar (Favrat, 1995; Allani et al., 1996). The most often mentioned ISFCS

concepts deal with high power production (up to several hundreds of MWe) and are reviewed in (Buck et al., 1998 or Kane et al. 2000). Earlier work is also reported in (Koai, Lior and Yeh, 1984).

However to meet the needs of rural areas in emerging countries, mini- ISFCS are required and one such concept has been designed at the Laboratory for Industrial Energy Systems (LENI) in the frame of a project called Solar Power System (SPS).

### SPS Concept

The prototype designed in Lausanne is composed by two rows of linear solar concentrators, two superposed ORC both equipped with hermetic volumetric scroll expander-generators and a Diesel cogeneration engine (15 kWe) with heat recovery on its exhaust gases as well as on its cooling water circuit. Therefore, heat sources at different temperature levels are available. One, at approximately 150°C, is pressurized hot water heated first by the solar collectors and then further heated by the exhaust gases of the engine. A second one is the engine cooling water at about 80°C (at present with a potential to go up to 95°C). In the single solar mode, the second source as well as the heat from the exhaust gases are no more available and the pressurized hot water is heated by the sun only.

At present the solar collectors are designed to heat water up to 170°C, but higher temperatures using thermal oil instead of water are envisaged in order to increase the efficiency of the ORC. Pressurized water flows in vacuum insulated tubes which are at the focusing line of the concentrators. These are made of series of thin plate mirrors (CEP) of different width and fixed at calculated angle on linear supports in order to concentrate the solar radiation on the insulated tubes. These can be assimilated to Fresnel mirrors. The two rows of collector are oriented North-South with a tracking system from East to West. As mentioned above pressurized water has first been chosen for simplicity reasons. When total reliability of the concept will be proven, switch to thermal oil (allowing much higher temperatures at moderate pressure and avoiding freezing problems) is planned.

One option taken from the start has been to rely on fully hermetic cycles with hermetic expander-generators to avoid any shaft seals which could induce undesired leakage and long term maintenance problems. Furthermore cost reasons with the interest of using cheap large production components oriented the choice of turbine towards hermetic scroll compressors modified into

expanders. This latter option had been proven to work successfully in previous studies (Zanelli, Favrat 1994, Favrat, 1995; Kane et al., 1999). However those volumetric expanders have a range of efficient expansion ratios which is limited which constrains the cycle design. Accounting for those constraints and to maximize the efficiency of the plant and allow future extension to higher driving temperatures, it has been decided to implement two superposed ORC each with its own working fluid. In the present setup the High Temperature (HT) cycle uses HCFC123 and the Low Temperature (LT) cycle HFC134a. After having been pumped by a variable speed membrane piston pump into a plate evaporator, the topping cycle fluid is heated, evaporated and superheated (with pressurized water heat) before being expanded<sup>1</sup> in the HT scroll. Discharged vapor is then cooled, condensed and sub-cooled in a condenser-evaporator plate heat exchanger. Heat recovery is transferred to the bottoming cycle fluid for heating, evaporation and superheating. In hybrid mode, heat is also supplied to the HFC134a by the engine cooling water. Therefore, an additional plate heat exchanger, called preheater, is placed in series upstream of the evaporator of the bottoming cycle. The same type of pump and expander technologies are used for the bottoming cycle. The fluid is condensed in a plate condenser with cold water (7°C). Accounting for the heat recovered from the engine cooling network, the LT expander is oversized compared to the topping cycle one (8kWe versus 5kWe). The variable speed control of the pumps facilitate operation at part load.

To lubricate the bearings of the expanders and to avoid additional oil pumps, oil circulates with refrigerant from turbine outlet to the condenser, pump and evaporator at the outlet of which a separator is placed. The latter recovers oil to be injected within the hollow expander shaft, using the pressure ratio available. The efficiency of the separator doesn't need to be very high, as some amount of oil is desirable at the expander inlet to seal the inner gaps during expansion.

Pressure and temperature sensors are installed before and after each components of the plant and the values can be directly checked on a computer, with a special designed LabView Vi, allowing post-processing calculation of all the operation parameters. Figure 1 shows a schematic

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<sup>1</sup> During warm-up, vapor is by-passed

view of the plant (screen shot).

The cogeneration group showed in Figure 2 is composed of a Diesel engine, a shell-in-tube high temperature heat recovery exchanger for exhaust gases (leaving the engine at temperatures up to 650°C), and a preheater plate exchanger for the cooling water network. Regarding this cooling network, temperature at the engine outlet is kept constant at 82.5°C with a thermostatic valve regulating the water flow rate. The Diesel engine is a Lombardini LW 903 designed for industrial purposes. Its 3 cylinders in lines have a total capacity of 913 cm<sup>3</sup> and the engine is coupled with a three-phase asynchronous generator.

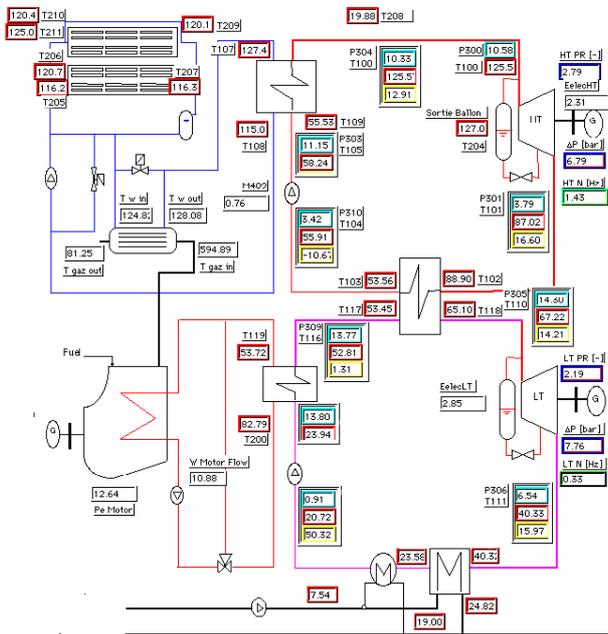


Figure 1. Schematic representation of the SPS power plant and typical parameters values (LabView screen shot).

## Experimental results

ORC, solar concentrators and engine have been tested separately or only partially integrated during the year 2000 and detailed results are presented in (Larrain et al., 2000; Kane et al., 2001) and (Thély, 2000). During the summer 2001, the aim has then been to proceed with the integration on the field of the three technologies. The experimental analysis included the study of the cycles with a variable temperature heat source and in the various modes allowed by the integration of the engine.

In 2000, heat was supplied to the cycles by an oil heater at

a constant (adjustable) temperature. This is no more possible with solar concentrators, available heat at evaporator being given by the direct solar radiation (varying along the day) and the concentrator efficiency (sensitive to solar radiation). To facilitate the startup of the cycles, the first approach tested was to allow time for the preheating of the heat source, typically up to 150 to 160°C<sup>2</sup>. However for such temperatures, the required heat supply to heat, evaporate and superheat the HCFC123 is about 55 to 65 kWth (based on last year measurements). Such values turned out to be hard to achieve in the field because of a mismatch between the design nominal values of the power plant and of the solar field<sup>3</sup>. In such a starting mode the power demand being higher<sup>4</sup> than the supply, the water inlet temperature decreases with time, before reaching an equilibrium.

<sup>2</sup> With a direct solar radiation of 800 W/m<sup>2</sup>, the average rising temperature rate in closed loop from 25°C is about 2°C/min.

<sup>3</sup> This is due to a reduction in budget which did not allow the construction of the complete solar field as initially planned (100 m<sup>2</sup> instead of 160 m<sup>2</sup>)

<sup>4</sup> Although the speed of rotation of the volumetric scroll expanders could potentially be varied within a range from 50 to 110%, the choice was made here, for simplicity reasons to operate them at constant speed (direct connection to the grid).



Figure 2. The cogeneration unit

This transition period duration as well as the equilibrium temperature depend on the direct solar radiation and the working mode (solar or hybrid); For the different tests cited in this paper the allowed stabilization temperature is in the range of 115 to 135°C.

### Typical working days

The direct solar radiation varied along the summer and the best values were measured during the month of June with more than 850 W/m<sup>2</sup>. The output electrical power of the engine is around 12 kW<sup>5</sup> and approximately 11 kWt, respectively 20 kWt are recovered from the exhaust

gases, respectively from the engine cooling network. The main values obtained for two typical days are presented in the table 1 below<sup>6</sup>.

|  | 26.06.2001 | 26.09.2001 |
|--|------------|------------|
| Direct Solar Radiation (Average) [w/m <sup>2</sup> ] | 740        | 597        |
| Direct Solar Radiation (Pic) [w/m <sup>2</sup> ]     | 783        | 654        |
| Working Mode   | Solar      | Hybrid     |
| Working time [hours]                                 | 7.9        | 3.9        |
| Total Electricity production [kWh]                   | 33.3       | 67.5       |
| Turbines Electricity production [kWh]                | 33.3       | 19.3       |
| Diesel Consumption (l)                               | 0          | 18.8       |
| ORC Efficiency (Average) [%]                         | 10.2       | 7.93       |
| ORC Efficiency (Best) [%]                            | 16         | 8.5        |
| ORC Exergetic Efficiency (Average) [%]               | 44.0       | 43.1       |
| ORC Exergetic Efficiency (Best) [%]                  | 61.0       | 47.02      |
| Plant Global Efficiency (Average) [%]                | 4.9        | 15.59      |
| Plant Global Efficiency (Best) [%]                   | 7.3        | 16.29      |
| Fossil Efficiency (Average) [%]                      | -          | 35.3       |
| Fossil Global Efficiency (Best) [%]                  | -          | 36.7       |

Table1. Main values of the ORC cycles for 2 typical days

The cycles efficiencies are quite low, with averages of 10% in solar mode and around 8% in hybrid mode. First of all, it is important to notice that ORC are working at low temperature (heat source at about 130°C) and at that temperature the maximum theoretical efficiency (Carnot) is 25%. Moreover, the lowest efficiency value for the hybrid working mode is easily explained by the fact that a large part of heat power given to the cycles is supplied at low temperature (80°C) and only for the bottom cycle. In these conditions, the exergy efficiency is much more representative and reaches about 44%.

Important sources of losses are most of the heat exchangers and in particular the condenser-evaporator linking the two cycles. In fact, as it can be noticed in figure 3 and 4, the temperature difference between condensation and evaporation stage in this exchanger is about 20°C which is much too high. This can partly be explained by the large amount of oil mixed with the evaporating working fluid (134a), whose boiling temperature strongly increases in the dryout region of the evaporator (non linearity not represented in Figure 4).

One solution would be to introduce a falling film shell in tube evaporator instead of the plate evaporator with an

<sup>5</sup> This value could be easily increased by a better ventilation of the engine.

<sup>6</sup> All the efficiency given are calculated in subtracting pump powers to electricity production. The exact expressions are given at the end of the present paper

accurate and fine control of the liquid pump, the topping fluid 123 condensing inside the tubes.

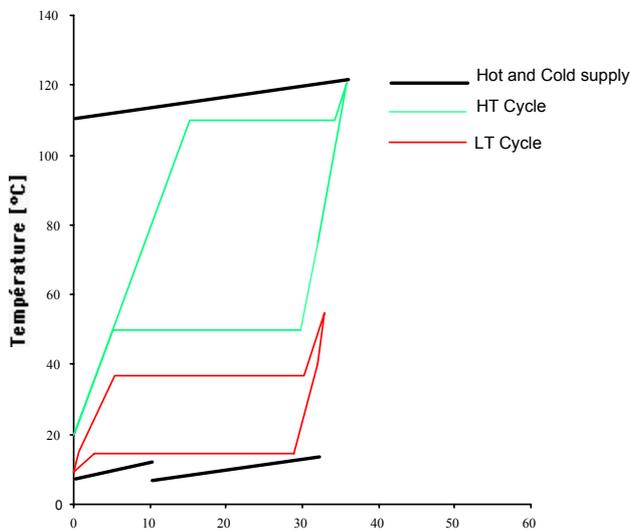


Figure 3. Heat-Temperature diagram of the ORC in hybrid mode

It can also be pointed out that the ORC are overdimensioned for the existing heat supply. Indeed, as shown in figure 5, both expanders are used at very part load, especially the LT one, in solar mode. The latter turbine was in fact designed to work with heat recovered from the engine cooling network. A compromise has to be found and alternatives include either a variable speed LT expander or the introduction of two LT expanders in parallel to better adjust the load.

A last remark regarding the cogeneration group and its role on the power plant, is that the preheater works in parallel-flow mode (as it can be noticed on Figure 4). This was done to facilitate the regulation of the temperatures of the cooling network at the engine outlet. The valve keeps a constant water temperature (82.5°C) whereas the heat exchanger being oversized results in a very small pinch (about 1°C). In counter-flow mode, the pinch would move at the ORC working fluid inlet, with potentially too high temperature variations, not ideal for the engine.

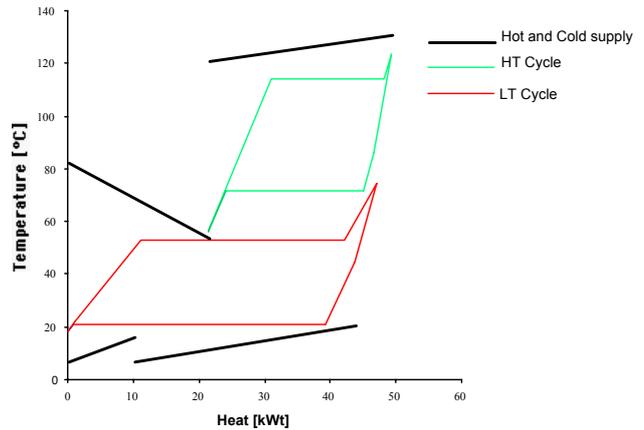


Figure 4. Heat-Temperature diagram of the ORC in solar mode

Another problem appeared during the test period and concerns the condenser. Indeed, probably due to variable cold water flow rate, the condensation pressure of LT cycle could not be maintained at 5 bars as it was first planned and realized during the laboratory test. This is also an important point to notice in order to improve the efficiency concept.

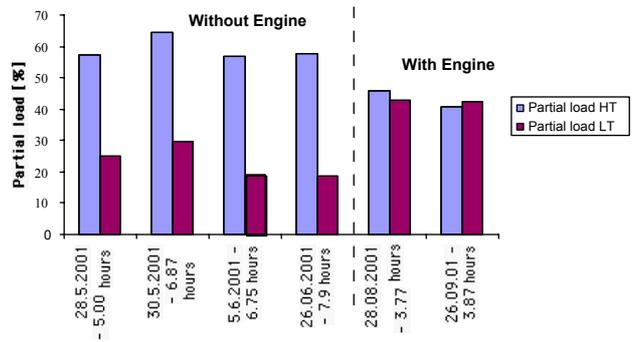


Figure 5: Part load average of the two turbines for 6 typical days

As it has been said, exergetic efficiency of the cycles is fair for this size of plant and with a significant potential for improvement. Figure 6 shows that in hybrid mode, this efficiency increases with the solar share. (ratio of the part of solar versus engine heat recovered).

This latter figure also illustrates the heat exergy limit (about 37 kWth and 115°C evaporator temperature) at which one of the turbine has to be disconnected. For lower values, the plant can still be maintained in operation but with the HT cycle only (and at very part load). The

LT expander is then bypassed and the bottoming cycle plays only a heat transfer role to the condenser.

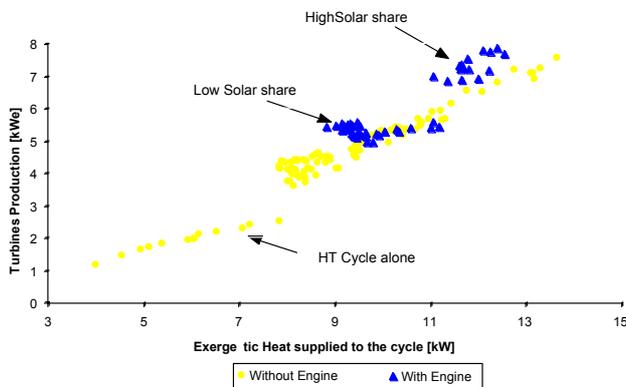


Figure 6: Turbines electricity production vs heat exergy provided to the cycles

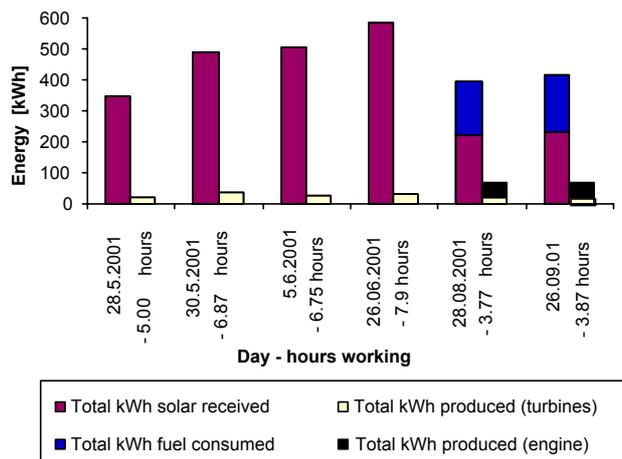


Figure 7: Total energy received and produced for 6 typical days

The latter figure summarizes the balances of energy in both working mode.

## Conclusion

A new concept of mini hybrid solar power plant designed at LENI was field tested in Lausanne (Switzerland) during the summer 2001. The plant integrating three technologies (Linear Fresnel concentrators, Organic Rankine cycles and a Diesel engine) operated in a satisfactory way in various modes (from pure solar to hybrid with engine at full load). The hermetic scroll expander-generators

equipped with a special in-shaft oil injection system performed adequately and reliably in spite of the strong variations of thermodynamic conditions. Fluctuations of solar radiation can be coped with the adaptation of the mass flow rates of the ORC working fluids and a reasonable range of vapor superheating can be maintained. Of course, the variation of solar radiation makes the cycles work at part load, which is detrimental for the efficiency.

Low ORC and global efficiencies (exergetic efficiencies below 45%) have been obtained and can mainly be explained by the mismatched nominal design between the ORC expanders and the solar field. This resulted in too low heat supply inducing too low evaporation temperatures of the topping cycle. Another reason is the low efficiency of the condenser-evaporator heat exchanger. A new design of the evaporator-condenser is being studied and new laboratory tests with a higher heat rate thermal oil source are being planned for the winter 2001-2002. Other improvement possibilities include better and less expensive pumps, variable speed expanders, direct evaporation in the collectors, higher temperatures of the topping cycle, etc.

The concept of ORC with hermetic scroll expander-generators can be applied to a whole range of heat recovery applications. Two single cycles are presently being installed to increase the efficiency of biogas engines in a green waste fermentation plant in Geneva, converting only the engine cooling energy in a first approach.

The field experience gained is being used to improve the automatic control of such plants which is currently underway. Tests replacing Diesel fossil fuel by biodiesel can also be imagined, resulting in a fully renewable hybrid power plant.

## Acknowledgments

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## Equations

Different efficiencies given in table 1 are calculated with the following expressions:

$$\varepsilon_{\text{ORC}} = \frac{\dot{E}_{\text{elecORC}} - \dot{E}_{\text{ORCpump}}}{\dot{M}_{\text{pw}} \cdot (h_{\text{inpw}} - h_{\text{outpw}}) + \dot{M}_{\text{cw}} \cdot (h_{\text{incw}} - h_{\text{outcw}})}$$

$$\eta_{\text{ORC}} = \frac{\dot{E}_{\text{elecORC}} - \dot{E}_{\text{ORCpump}}}{\dot{M}_{\text{pw}} \cdot (k_{\text{inpw}} - k_{\text{outpw}}) + \dot{M}_{\text{cw}} \cdot (k_{\text{incw}} - k_{\text{outcw}})}$$

$$\varepsilon_{\text{plant}} = \frac{\dot{E}_{\text{elecORC}} - \dot{E}_{\text{ORCpump}} + \dot{E}_{\text{elecengine}}}{\dot{Q}_{\text{directsolar}} + \dot{M}_{\text{fuel}} \cdot \text{LHV}}$$

$$\varepsilon_{\text{fossil}} = \frac{\dot{E}_{\text{elecORC}} - \dot{E}_{\text{ORCpump}} + \dot{E}_{\text{elecengine}}}{\dot{M}_{\text{fuel}} \cdot \text{LHV}}$$

With:

$\varepsilon_{\text{ORC}}$ : ORC efficiency [%]

$\eta_{\text{ORC}}$ : ORC Exergetic Efficiency [%]

$\varepsilon_{\text{plant}}$ : Plant Global Efficiency [%]

$\varepsilon_{\text{fossil}}$ : Fossil Global Efficiency [%]

Symbols:

$\dot{E}$ : Power [W]

$\dot{M}$ : Mass flow rate [kg/s]

$h$ : Specific enthalpy [J/kg]

$k = h - T_a \cdot s$ : Massflow specific exergy [J/kg]

$s$ : Specific entropy [J/kg $^{\circ}$ K]

$T_a$ : Atmospheric temperature [ $^{\circ}$ K]

$\dot{Q}$ : Heating power [W]

LHV: Low Heating Value [J/kg]

Subscripts

pw: Pressurized water

cw: Engine cooling water

in: Heat exchanger inlet

out: Heat exchanger outlet

directsolar: Direct solar radiation

elecORC: Cumulated for both expanders

pumpORC: Cumulated for both pumps

elecengine: Engine electricity

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