EUROSUN 2008
1st International Congress on Heating, Cooling, and Buildings
7th to 10th October, Lisbon - Portugal

245 - Modeling and experimental investigation of an Organic Rankine Cycle using scroll expander for small scale solar applications.

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Abstract

This study aims at developing and optimizing an Organic Rankine Cycle (ORC) for a small scale, decentralized parabolic trough system.
A numerical model the Organic Rankine Cycle is developed and an experimental study is carried out on an ORC prototype working with refrigerant HCFC-123, and whose heat sources consist in two hot air flows. The ORC model is built by connecting different sub-models: the heat exchanger models, a volumetric pump model and a scroll expander model. Measured performances of the ORC prototype are presented and allow the validation of the ORC model.

Keywords: Organic Rankine Cycle, Concentrating solar power, Parabolic trough

1. Introduction

Currently 1.6 billion people worldwide lack access to electricity [4]. Many communities will continue to lack access to centralized grid infrastructure due to remoteness or to the poor rates of return on investment in grid extension. Small scale, decentralized concentrating solar power can constitute a cost effective energy solution for remote places with a high solar irradiation.

Organic Rankine cycles (ORC) are particularly suitable for small scale parabolic trough systems, since they can generate electricity at a lower temperature than the conventional steam Rankine cycle.

Simulation models of ORC are necessary to optimize the operating conditions and the components of the cycle. Previous studies have demonstrated the influence of the working fluid thermodynamic properties on ORC performance [1, 6, 8]. However, until now, relatively few papers present a detailed simulation model for an ORC [4, 10, 12].

This paper presents both a numerical simulation model of an ORC and the results of an experimental study used to validate the simulation model. Special attention is paid to the expander, since it is a key component for global cycle performance.

2. Scope of the work

The Solar Turbine Group (STG) was founded for the purpose of developing and implementing a small scale solar thermal technology utilizing medium temperature collectors and an ORC to achieve economics analogous to large-scale solar thermal installations. This configuration aims at replacing or supplementing Diesel generators in off grid areas of developing countries, by generating clean power at a lower levelized cost (~$0.12/kWh compared to ~$0.30/kWh for Diesel [9, 11]). At the core of this technology is a solar thermal power plant consisting in a field of parabolic solar concentrating collectors and a vapor expansion power block for generating electricity (figure 1). An electronic control unit is added for autonomous operation as sub-megawatt scale plants cannot justify the staffing of operating personnel. The design tradeoffs for maintaining low costs at small scales, include operating at a lower cycle temperatures (<200□C) and using an ORC: lower temperatures enable cost savings in the materials and manufacture of the absorber units, heat exchangers, fluid manifolds and parabolic troughs.

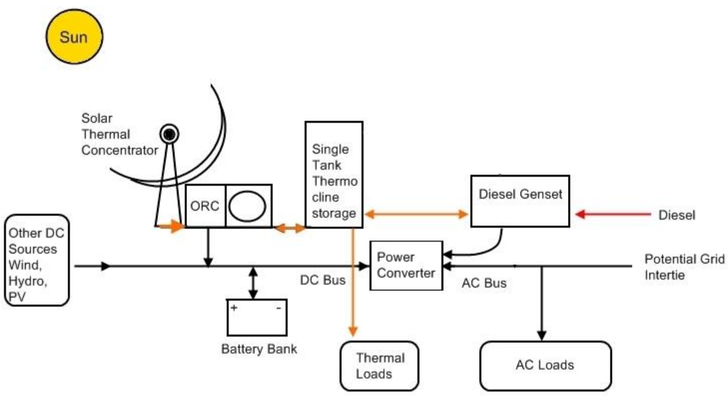

Figure 1 Solar ORC platform for off grid micro utility power generation

Figure 2 Solar ORC in field trials in Lesotho, southern Africa 2007 (Solar Turbine Group)

Because no thermal power blocks are currently manufactured in the kilowatt range a small-scale ORC has been designed for this application (Figure 2). The design is based on off-the-shelf components with little modification, such as HVAC scroll compressors (for the expander), and car power steering pumps.

A novel control strategy and electronic control system is required for the components discussed above to work in concert and in a maximally efficient manner. Among the functions to be managed is the control of individual components, such as Solar ORC fluid machinery, as well as directing the energy flows between these components, battery storage and AC loads. Optimization of the control strategy, a major objective of ongoing research, will be based on theoretical and experimental characterization of all system components, and will, in practice, rely on developing a control system with feedback from diverse parameters ranging from ambient temperature to the particular load profile to be matched.

The work presented in this paper focuses on the characterization of the ORC system by developing and validating a model, which will be used to select the best components, working fluid and control strategies for the solar Rankine engine.

3. Modeling

The ORC model is built by connecting the models of its different main components. A volumetric pump and a scroll expander models are considered since they are the technologies selected for the ORC prototype presented in this paper. All models are developed under EES [5] using semi-empirical parameters that are identified with experimental data.

3.1. Scroll expander model

The scroll expander model has been previously proposed by Lemort et al. [5] and partly validated by tests with steam. In this model, the evolution of the fluid through the expander is decomposed into the following steps (as shown in Fig. 3):

- Supply pressure drop (su→ su,1,1)
- Cooling-down in the supply port of the expander (su1,1 → su,1);
- Isentropic expansion from the supply pressure down to the adapted pressure imposed by the internal expansion volume ratio of the expander (su,1 → ad);
- Expansion at a fixed volume from the adapted pressure to the exhaust pressure (ad → ex,2);
- Mixing between suction flow and leakage flow (ex,2 → ex,1) and
- Cooling-down or heating-up in the exhaust port (ex,1 → ex).

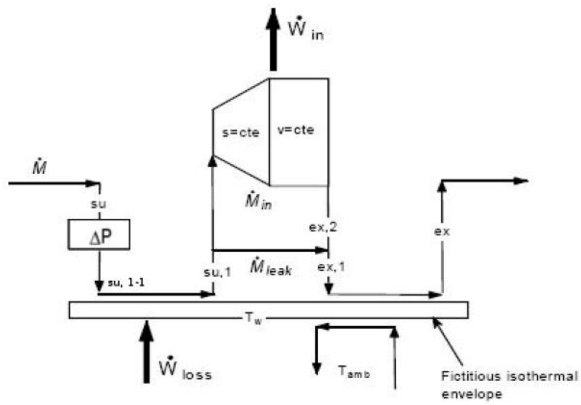

Figure 3 Conceptual scheme of the expander model

The model requires only nine parameters (heat transfer coefficients, friction torque, leakage area, pressure drop equivalent diameter). Those nine parameters, defined for a specific type of expander and for a specific working fluid, are determined on the basis of experimental data.

3.2. Condenser and evaporator models

The condenser and the evaporator are modeled by means of the *ε-NTU* method for counter-flow heat exchangers. The heat exchanger is subdivided into 3 zones, each of them being characterized by a heat exchange coefficient U. Figure 4 shows the 3-zones heat exchanger in the case of an evaporator.

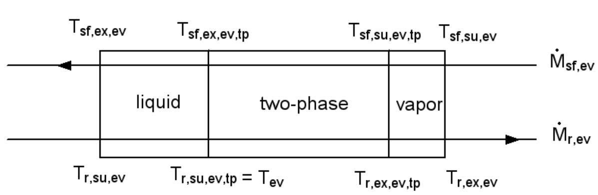

Figure 4 Three zones moving boundary evaporator model

The AU heat transfer coefficient is calculated by considering two resistances in series (secondary fluid and refrigerant sides). The nominal heat transfer coefficients are identified for a nominal flow rate. The coefficients are then obtained by: $h = h_n \cdot \dot{M}_n^{\ 0,8}$

This relationship can be justified by the Reynold's analogy for a turbulent flow through a pipe [2] by assuming that the fluid properties, not included in this expression, remain unchanged.

3.3. Pump model

The pump is modeled by its swept volume, its isentropic efficiency ($\eta_{pp,s}$) and its motor efficiency ($\eta_{pp,m}$). The two latter are supposed to be constant. The pump electrical consumption and the fluid flow rate are given by :

$$\dot{W}_{m,pp} = \frac{\dot{W}_{sh,pp}}{\eta_{pp,m}} = \frac{\dot{W}_{pp,s}}{\eta_{pp,s}.\eta_{pp,m}}$$

$$\dot{M}_r = \frac{\dot{V}_{s,pp}}{v_{r,su,pp}} = \frac{X_{pp}.\dot{V}_{s,pp,max}}{v_{r,su,pp}}$$

4. Experimental study

4.1. Test bench description

An experimental study is carried out on a prototype of ORC working with HCFC-123. The scroll expander is originally an oil-free open-drive air scroll compressor, adapted to operate in reverse. The heat source consists in a set of heat exchangers supplied with two hot air flows. The condenser is cooled by water (Figure 5).

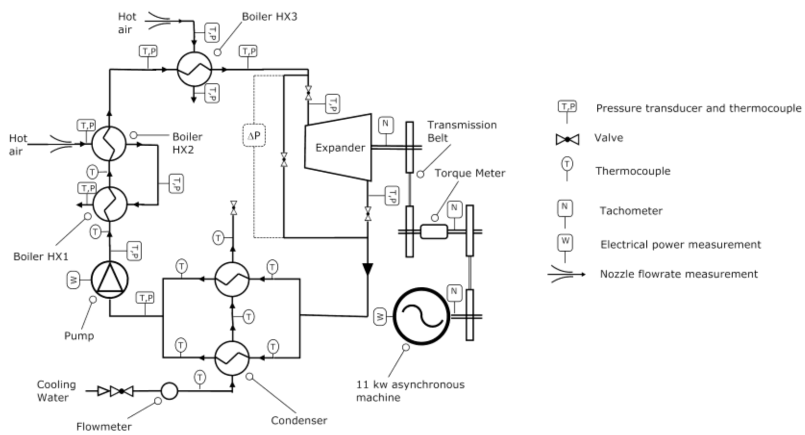

Fig. 5 Schematic representation of the test bench

4.2. Reached performances

A series a 39 steady-state performance points is achieved, by modifying the working conditions as indicated in the table below:

Working condition	Minimum value	Maximum value
First hot air source mean temperature	53.4 °C	86.4 °C
Second hot air source mean temperature	101 °C	163.2 °C
Air flow rate	0.071 kg/s	0.90 kg/s
Refrigerant flow rate	45 g/s	86 g/s
Condenser water flow rate	0.13 l/s	0.70 l/s
Condenser mean water temperature	13.2 °C	15 °C
Expander rotation speed	1771 rpm	2660 rpm

Three performance indicators are taken into account: the expander shaft power, the expander isentropic effectiveness, and the cycle efficiency.

The expander isentropic effectiveness is defined by: $$\varepsilon_s = \frac{\dot{W}_{sh}}{\dot{M}_r \cdot (h_{r,su,\exp} - h_{r,ex,\exp,s})}$$

And the cycle efficiency by: $$\eta_{cycle} = \frac{\dot{W}_{sh,\exp} - \dot{W}_{pp}}{\dot{Q}_{evap}}$$

A shaft power ranging from 0.38 to 1.82 kW is obtained, corresponding to a mechanical isentropic effectiveness ranging from 43 to 68% and a maximum cycle efficiency of 7.4%. The pressure ratio over the expander varied from 2.7 to 5.4.

4.3. Expander model validation

The parameters of the expander model are identified using test results. They are adjusted to fit the three model outputs (supply pressure, exhaust temperature, shaft power) to experimental data. The *input* variables of this calculation are: expander rotation speed, fluid flow rate, supply temperature and exhaust pressure. An error-objective function is defined, that should be minimized: this error function is a weighted sum of the relative errors for each *output*. It is defined as follows:

$$F = \sum_{i=1}^{39} \left[\left(\frac{T_{r,ex,\exp,meas} - T_{r,ex,\exp,pred}}{60} \right)_i^2 + \left(\frac{p_{r,su,\exp,meas} - p_{r,su,\exp,pred}}{p_{r,su,\exp,meas}} \right)_i^2 + \left(\frac{\dot{W}_{sh,\exp,meas} - \dot{W}_{sh,\exp,pred}}{2 \cdot \dot{W}_{sh,\exp,meas}} \right)_i^2 \right]$$

The parameters that minimize the objective function F are identified by means of a genetic algorithm. Given these parameters, the predicted and measured *outputs* can be compared:

- A maximum error of 3 K is reached for the prediction of the exhaust temperature.
- The supply pressure is predicted with a maximum relative error of 2.3%.
- The expander shaft power is predicted with a maximum deviation of 6% (Figure 6).

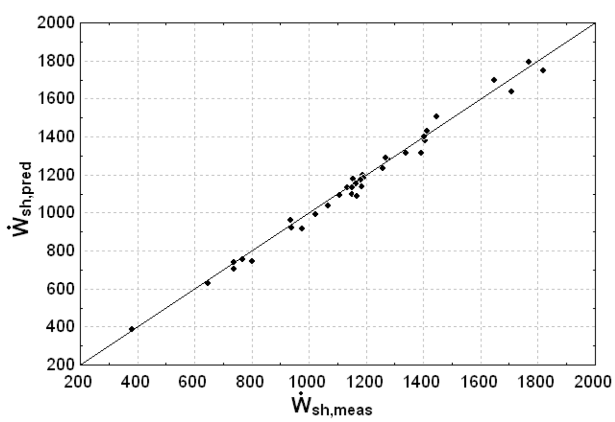

Fig. 6 Predicted vs measured expander shaft power

4.4. Heat exchanger model validation

In order to tune the heat exchanger model with the experimental data, the heat transfer coefficients need to be identified. For one exchanger, four heat transfer coefficients are defined: one for each zone on the refrigerant side, and one for the secondary fluid.

In the evaporator, two different heat exchangers were used. In total, eight parameters were therefore necessary to describe the set of exchangers in series.

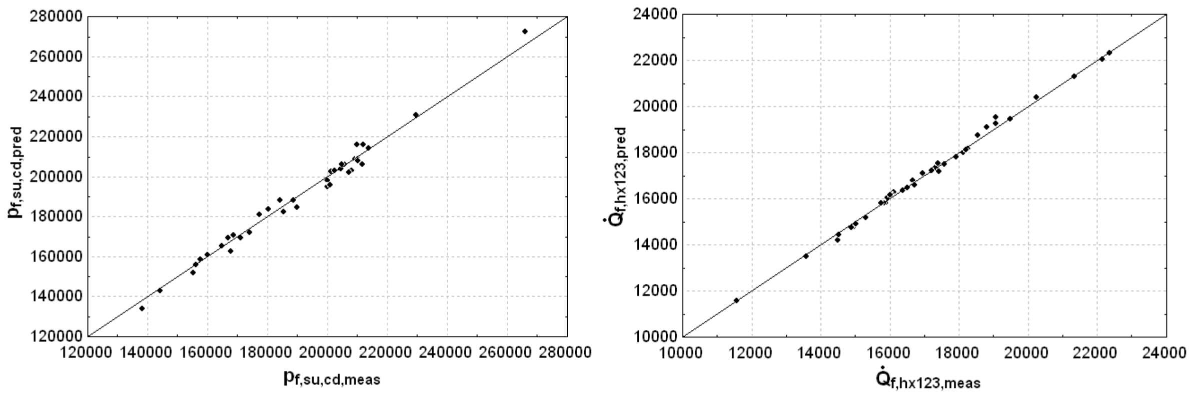

Fig. 7 Condenser predicted supply pressure and evaporator predicted power

For given supply and exhaust temperature conditions, the condenser model predicts its pressure. Figure 7 shows that this pressure is predicted with a relative error of about 3%.
In the evaporator, the pressure is imposed by the expander and the feed pump. Given its supply temperature and the saturation pressure, the model predicts the heat flow and the exhaust temperature. Figure 8 shows that the heat flow is predicted with an error lower than 2%.

4.5. Validation of the global model of the cycle

Once the models of each component of the circuit are validated, these models are interconnected to simulate the whole system, as shown in Figure 8.

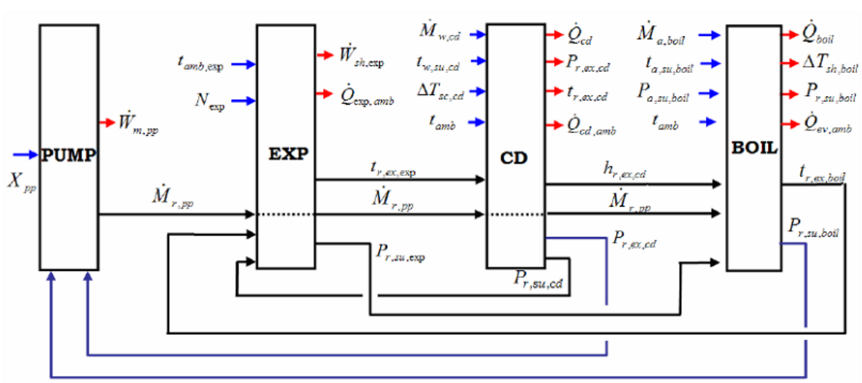

Fig. 8 Block diagram of the global model of the cycle

The following “causalities” are observed:

- the pump imposes the refrigerant flow rate,
- the expander imposes its supply pressure and exhaust temperature,
- the evaporator imposes the refrigerant superheating and the pump exhaust pressure,

- the condenser imposes the expander exhaust pressure and the pump supply pressure.

This simulation model is not fully predictive, because the subcooling at the exhaust of the condenser is defined as a model input. In order to predict this subcooling, a refrigerant charge model would have to be included.

Figure 9 shows the prediction of the output shaft power for the global cycle model. All measurements are predicted within a 10 % accuracy. Errors of each model are indeed cumulated, which leads to a lower accuracy for the global model than for the individual components.

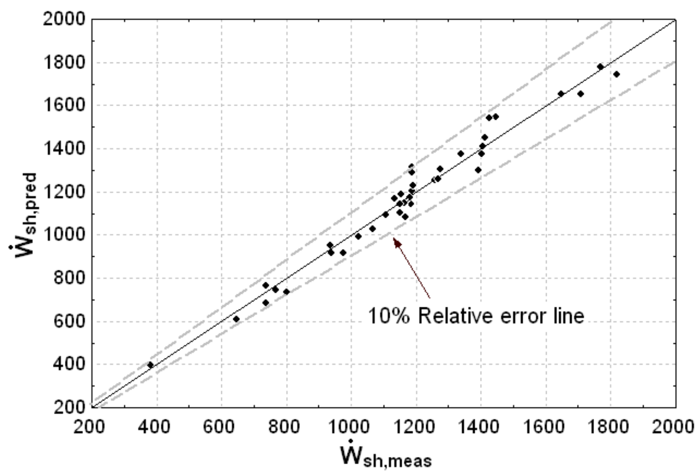

Fig. 9 Predicted vs measured output power with the global model

5. Conclusion

This paper proposes a semi-empirical model of an ORC involving a relatively limited number of parameters. The comparison between predicted values and experimental results show a fairly good agreement (for the cycle model as well as for the different sub-models).

The experimental study carried out shows a good expander isentropic effectiveness and demonstrates the viability of utilizing a mass-produced compressor as an expander in a small scale ORC. This represents an important step towards realizing the cost reductions that would make a kilowatts-sized Solar ORC economical for developing countries.

That the overall cycle efficiency remains limited is partly explained by the low temperature of the heat source and by a low pump efficiency. The former can be rectified by using higher temperature heat sources, while the latter can be addressed by selecting of a pump optimized for the pressure ratio and flow rate of the ORC.

Future work will focus on the integration of the ORC model into a global model including the solar collector in order to size the system, define a control strategy, and optimize the working conditions and the components.

Nomenclature

A	area	(m^2)
AU	heat exchange coefficient	(W/K)
F	objective function	(-)
h	convective heat transfer coefficient	(W/m^2K)
$\dot{M}$	mass flow rate	(kg/s)
N	rotational speed	(rpm)
p	pressure	(Pa)
$\dot{Q}$	heat flux	(W)
T	temperature	(°C)
T	torque	(N.m)
V_s	swept volume	(m^3)
v	specific volume	(m^3/kg)
V_s	swept volume	(m^3)
$\dot{V}$	volume flow rate	(m^3/s)
$\dot{W}$	power	(W)
X	pump capacity ratio	(-)

Subscripts

a	air
amb	ambient
cd	condenser
ex	exhaust
exp	expander
f	working fluid
hx	heat exchanger
meas	measured
n	nominal
pp	pump
pred	predicted
r	refrigerant
s	isentropic
sf	secondary fluid
sh	shaft
su	supply
w	water

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