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DEVELOPMENT AND EXPERIMENTAL VALIDATION OF AN ORGANIC RANKINE CYCLE MODEL

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ABSTRACT

This paper presents both a numerical model of an Organic Rankine Cycle (ORC) and an experimental study carried out on a prototype of such a cycle 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. This model is finally used to investigate potential improvements of the prototype.

INTRODUCTION

Organic Rankine Cycles (ORC) are particularly suitable for recovering energy from low-grade heat sources, such as waste heat in an industry process, exhaust gas or cooling system of an internal combustion engine, exhaust gas from a turbine, or heat produced by solar concentrators.

Past studies have shown the influence of working fluid thermodynamic properties on the ORC performances (Hung, 2001; Liu et al., 2004; Maizza et al., 2001). On the other hand, only few papers present simulation models of ORC (Kane et al., 2003; Yamamoto et al., 2001). However these ORC simulation models are necessary to optimize the operating conditions and the components of the cycle.

This paper presents and validates a numerical model of an ORC and an experimental study carried out on an ORC working with refrigerant HCFC-123.

MODELING OF AN ORGANIC RANKINE CYCLE

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.

Scroll expander model

The scroll expander model has been previously proposed by the authors and partly validated by tests with water steam (Lemort et al., 2006). In this model, the evolution of the fluid state through the expander is decomposed into the following steps (as shown in Fig. 1):

- cooling-down in the supply port of the expander (su ?   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).

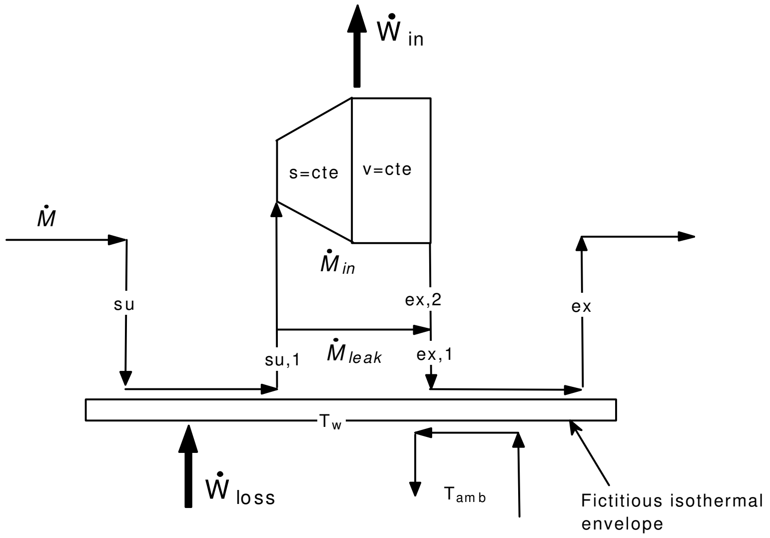


Fig. 1 Conceptual scheme of the expander model

Main parameters of the model are:

- the heat transfer coefficient AU_{su} at the expander supply (W/K);
- the heat transfer coefficient AU_{ex} at the expander exhaust (W/K);
- the heat transfer coefficient AU_{amb} between the expander and the ambience (W/K);
- the expander swept volume $V_{s,exp}$ (m^3);
- the expander internal built-in volume ratio $r_{v,in}$ (-);
- the expander equivalent leakage area A_{leak} (m^2);
- the mechanical losses torque T_m (N.m).

Condenser and boiler models

The condenser is modeled as a one-zone heat exchanger. As shown in Fig. 2, this model accounts for:

i) a pressure drop located at the condenser supply (calculated by considering an equivalent orifice, whose diameter is $d_{r,su,cd}$)
ii) ambient losses placed on water side at the condenser exhaust (calculated on the basis of a heat transfer coefficient $AU_{cd,amb}$ between the condenser and the ambience)

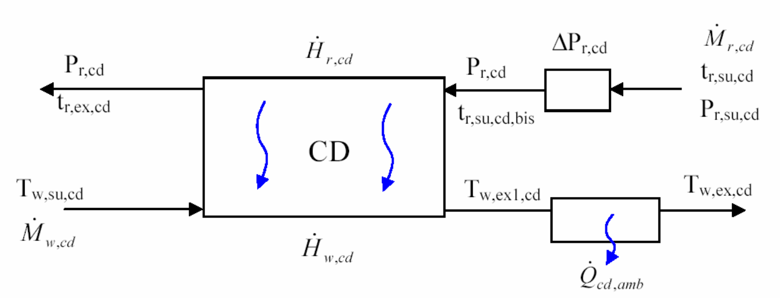


Fig. 2 Condenser model

The heat transfer is calculated by means of the *ε-NTU* method. The global heat transfer coefficient is calculated by assuming two resistances in series (water and refrigerant sides). As shown in Eq. (1), these two resistances could be calculated as a function of their respective mass flow rates (C_w and C_r are also parameters of the model)

$$AU_{cd} = \frac{1}{R_w + R_r} = \frac{1}{C_w \dot{M}_w^{0,8} + C_r \dot{M}_r^{0,8}} \tag{1}$$

The boiler modeling follows the same approach. However, one unique model is used here for each heat exchanger. The global model considers:

i) a pressure drop model at the HX1 supply and at the HX3 exhaust
ii) ambient losses placed on the air side at the HX1, HX2 and HX3 exhausts

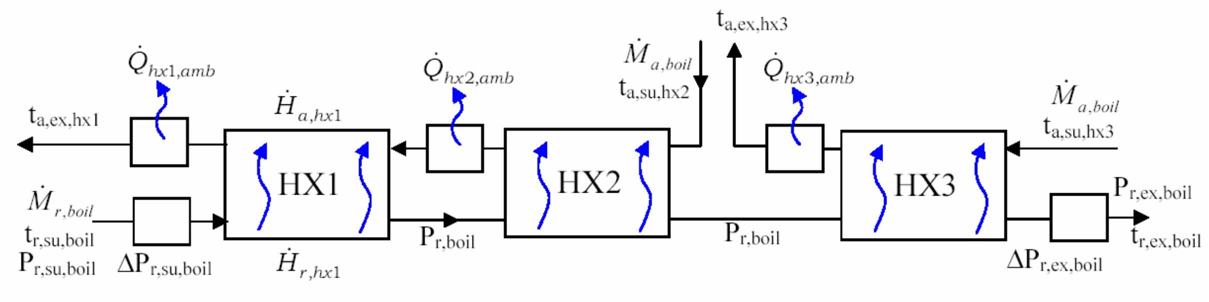


Fig. 3 Boiler model

Pump model

The volumetric pump is simply modeled by its swept volume and two supposed-to-be constant efficiencies: the isentropic (?$_{pp,s}$) and the motor (?$_{pp,m}$) efficiencies. The fluid heating-up introduced by the pump is neglected. The pump electrical consumption and the fluid swept flow rate are given by Eq. (2) and Eq. (3).

$$\dot{W}_{m,pp} = \frac{\dot{W}_{sh,pp}}{\eta_{pp,m}} = \frac{\dot{W}_{pp,s}}{\eta_{pp,s}.\eta_{pp,m}} \tag{2}$$

$$\dot{M}_r = \frac{\dot{V}_{s,pp}}{v_{r,su,pp}} = \frac{X_{pp}.\dot{V}_{s,pp,\max}}{v_{r,su,pp}} \tag{3}$$

Cycle model

The whole cycle is simulated by interconnecting the models of its different components (Fig. 4). The modeling makes appear the following "causalities":

I. For a given displacement, the pump imposes the refrigerant flow rate.
II. The boiler imposes the fluid overheating and the pump exhaust pressure.
III. Provided its rotational speed is fixed, the expander imposes the boiler exhaust pressure. The condenser supply temperature is also imposed by the expander.
IV. The condenser imposes the expander exhaust pressure and the pump supply pressure. The subcooling at the condenser exhaust is considered as an input of the model, since it is imposed by the refrigerant charge.

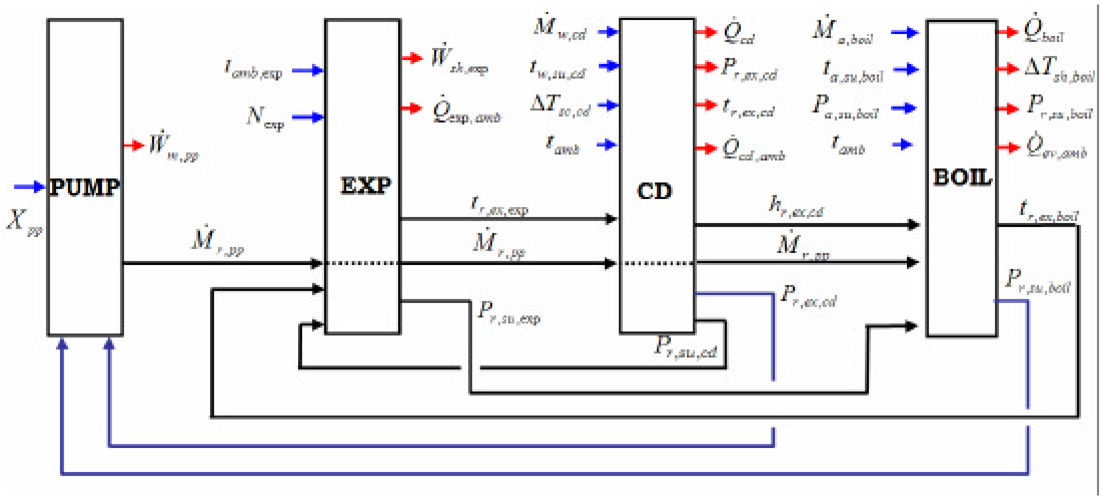


Fig. 4 Bloc diagram representation of the Rankine cycle model

EXPERIMENTAL STUDY AND VALIDATION OF THE MODEL

Description of the test bench

An experimental study is carried out on a prototype of ORC working with HCFC-123. The expander is originally an oil-free open-drive air scroll compressor, adapted in expander mode. Heat source consist in two hot air flows. The condenser is cooled by water.

The expander drives an asynchronous machine through a belt-and-pulley coupling. The asynchronous machine imposes the rotational speed of the expander. The expander mechanical power is determined by measuring simultaneously the rotational speed and the torque developed at the expander shaft. The refrigerant flow rate is determined from the energy balance over the condenser. The test bench is represented in Fig.5.

Description of the tests and reached performances

A series of 13 tests is carried out at a constant expander rotational speed (2200 rpm). A thermal power close to 15 kW is recovered at the boiler. Air enters into heat exchangers HEX2 and HEX3 at temperatures ranging respectively from 177°C to 189°C and from 137°C to 160°C. Its supply pressure doesn't exceed 8.5 bar and the expander exhaust pressure doesn't decrease under 1.5 bar. The expander develops a mechanical power ranging from 0.7 to 1 kW, with an overall isentropic effectiveness up to 65%.

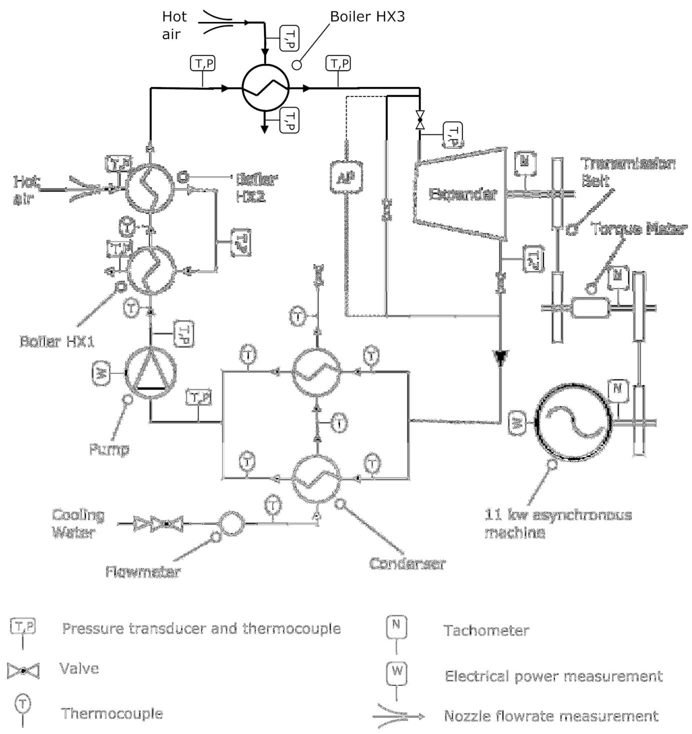
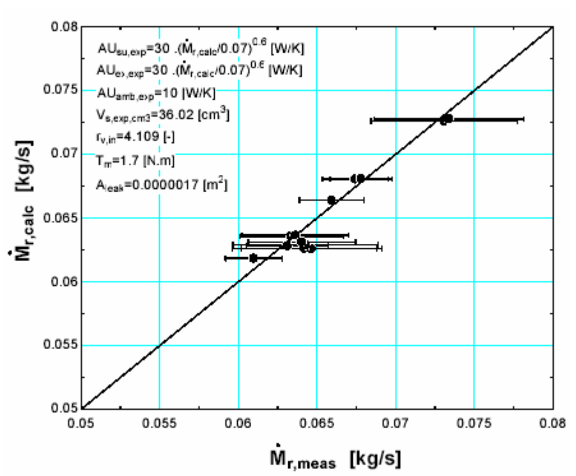


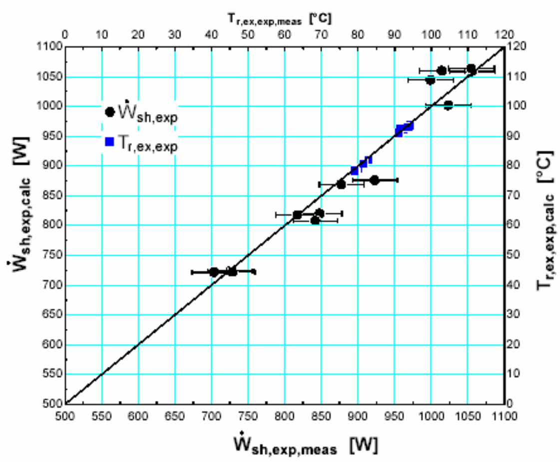
Fig. 5 Schematic representation of the test bench

Validation of the ORC model and sub-models

Expander and pump model validation. The validation of the expander model is carried out by imposing the supply and exhaust pressures, the supply temperature and the rotational speed as inputs of the model. Parameters are tuned in order to get equality between calculated and measured values of the flow rate, the mechanical power and the exhaust temperature. Fig. 6 (a) and (b) show the agreement between calculated and measured values after having identified the parameters (given in Fig. 6 (a)).



(a)

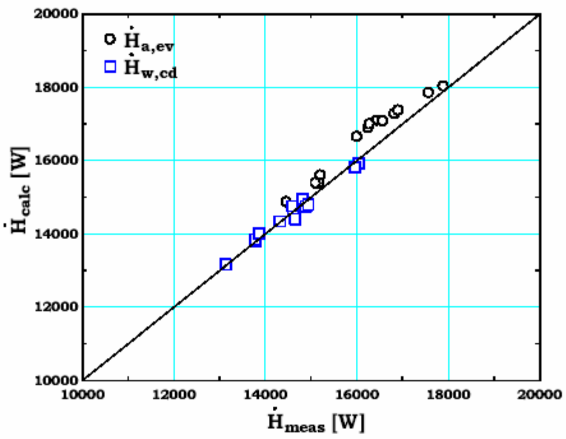


(b)

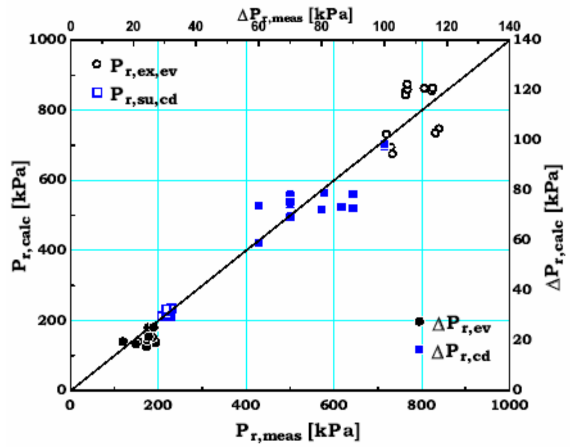
Fig. 6 Prediction of the refrigerant flow rate (a) and the expander shaft power and exhaust temperature (b)

According to the experimental measurements, the pump electrical consumption is of the order of 170W. In order to fit this value with the model, the efficiencies $\eta_{pp,s}$ and $\eta_{pp,m}$ are set to 0.4 and 0.5.

Heat exchangers models validation. The condenser and the boiler models are here validated on the basis of 13 tests. Fig. 8 shows the comparison between the main simulated and measured outputs of this model. For the condenser, a very good agreement is observed for the enthalpy flow rate and condensing pressure. For the boiler, the results can be considered as acceptable.



(a)



(b)

Fig. 7 Prediction of the condenser and boiler enthalpy flow rate (a) and pressure and pressure drop (b)

Cycle model validation. It appears that, after having been tuned, the global cycle model is able to reproduce fairly well, amongst other things, the expander power, supply and exhaust pressure and the refrigerant flow rate (Table 2).

Table 2 Comparison between (some) values measured and predicted by the ORC model

Test	$\dot{W}_{sh,exp}$ [W]	$\dot{W}_{sh,exp,meas}$ [W]	$\dot{M}_{r,pp}$ [kg/s]	$\dot{M}_{r,pp,meas}$ [kg/s]	$P_{r,su,exp}$ [Pa]	$P_{r,su,exp,meas}$ [Pa]	$P_{r,ex,exp}$ [Pa]	$P_{r,ex,exp,meas}$ [Pa]
1	996	1002	0,069	0,073	811935	835615	226166	230100
2	976	984	0,069	0,073	803147	827503	224831	228500
3	982	841	0,061	0,065	793125	761584	214257	226900
4	773	717	0,061	0,063	708406	729325	204173	226900
5	965	842	0,061	0,064	784815	757731	212473	227000
6	772	729	0,061	0,064	708070	728960	204331	227000
7	977	923	0,061	0,064	790906	763748	213903	218000
8	797	818	0,061	0,064	719521	727003	206276	208000

9	933	877	0,060	0,063	787868	764184	220169	219500
10	717	703	0,060	0,061	713568	717174	217291	219500
11	982	1024	0,068	0,066	823563	800923	234995	217700
12	998	1056	0,073	0,067	837805	817999	242465	217700
13	1009	1055	0,077	0,068	849990	819594	249662	217700

Potential improvements of the system

Some potential improvements of the system (components characteristics and operating conditions) are stressed (see Fig. 8), among which:

- Beneficial higher pressure ratio imposed to the expander. This can be achieved by reducing both the condenser exhaust pressure and pressure drop.
- Reduction of the subcooling at the condenser exhaust to the minimum required in order to prevent any risk of pump cavitation (adequate charge of refrigerant).
- Increase of the expander effectiveness by reducing both the mechanical losses and the internal leakage and by optimizing the internal built-in volume ratio $r_{v,in}$.

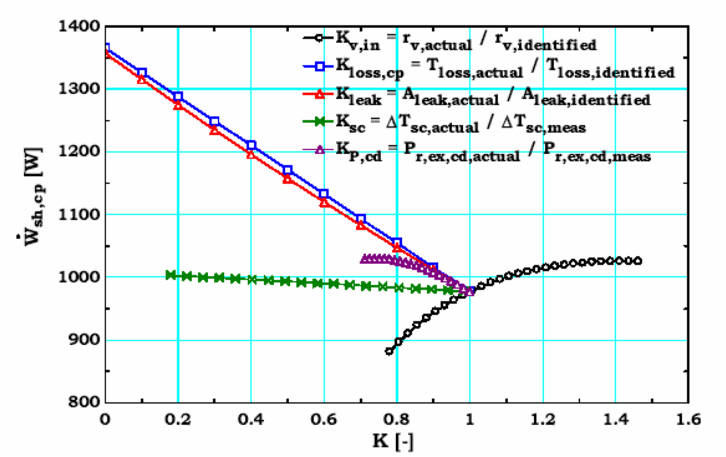


Fig. 8 Improvements of the cycle

CONCLUSIONS

This paper proposes a model of ORC involving a relatively limited number of parameters. The agreement between values predicted by the model and experimental results show a fairly good agreement (for the cycle model as well as for the different sub-models).
The ORC model is then used to evaluate the potential increase of the system performances following some improvements of its components and operating conditions.

The model could be easily generalized to other expanders and heat exchangers technologies, other working fluids and other heat sources and sinks than those considered in this study.

NOMENCLATURE

A	area	(m^2)	**Subscripts**	
AU	heat exchange coefficient	(W/K)	a	air
h	enthalpy	(-)	amb	ambient

$\dot{H}$	enthalpy flow rate	(W)	boil	boiler
$\dot{M}$	mass flow rate	(kg/s)	calc	calculated
N	rotational speed	(Hz)	cd	condenser
P	pressure	(Pa)	ex	exhaust
$\dot{Q}$	heat flux	(W)	exp	expander
$r_{v,in}$	internal volume ratio	(-)	leak	leakage
T	temperature	(°C)	m	mechanica l
T	torque	(N.m)	meas	measured
V_s	swept volume	(m^3)	pp	pump
v	specific volume	(m^3/kg)	r	fluid
$\dot{V}$	volume flow rate	(m^3/s)	sh	shaft
$\dot{W}$	power	(W)	su	supply
X	pump capacity ratio	(-)	w	water

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