EXPERIMENTAL STUDY AND DYNAMIC MODELING OF A WHR ORC POWER SYSTEM WITH SCREW EXPANDER

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EXTENDED ABSTRACT

INTRODUCTION

In recent years, due to the increasing concern over energy shortage and global warming, the interest in low grade heat recovery from industrial processes has grown dramatically (IEA, 2010). Several studies have underlined the potential of small-capacity ORC power plants for waste heat recovery (WHR) applications (Verneau, 1979). For such systems accurate dynamic modeling represents an important tool in particular when control issues are considered (Colonna and van Putten, 2007) (Casella et al., 2013) (Quoilin et al., 2011). This paper presents a dynamic model of an ORC system, validated both in steady-state and transient conditions via experimental data from a 10 kW_e waste heat recovery ORC system with a screw expander.

EXPERIMENTAL SET-UP

ORC test-rig

A schematic layout of the test bench is shown in figure 1. The system has a nominal power of 10 kW_e and is based on a regenerative cycle. Solkatherm is selected as working fluid. The expander is originally a single screw compressor adapted to run in expander mode. Lubricating oil circulates through the cycle to lubricate the rotor, while the bearings are lubricated using a by-pass pipe directly from the pump to the expander. The expander is connected to an asynchronous machine whose speed is controlled by means of a four-quadrants inverter. A vertical variable speed multistage centrifugal pump is used to circulate the fluid through the system. An inverter allows controlling the pump rotational speed. The heat exchangers, all identical, are of the brazed plate type. No control system is implemented on the ORC test bench. Thermal oil, Therminol66, heated up in an electrical boiler, supplies the thermal energy necessary to evaporate the working fluid. A variable flow rate of glycol water (34% ethylene glycol) is used as heat sink in the condenser. The test stand is equipped with absolute pressure sensors and thermistors. The Solkatherm mass flow rate is measured by means of a Coriolis flow meter, installed at the turbo-pump outlet. The cooling loop is equipped with an ultrasonic flow-meter to measure the flow rate of glycol water while in the heating loop a pressure difference transmitter is used to calculate the Therminol66 mass flow.

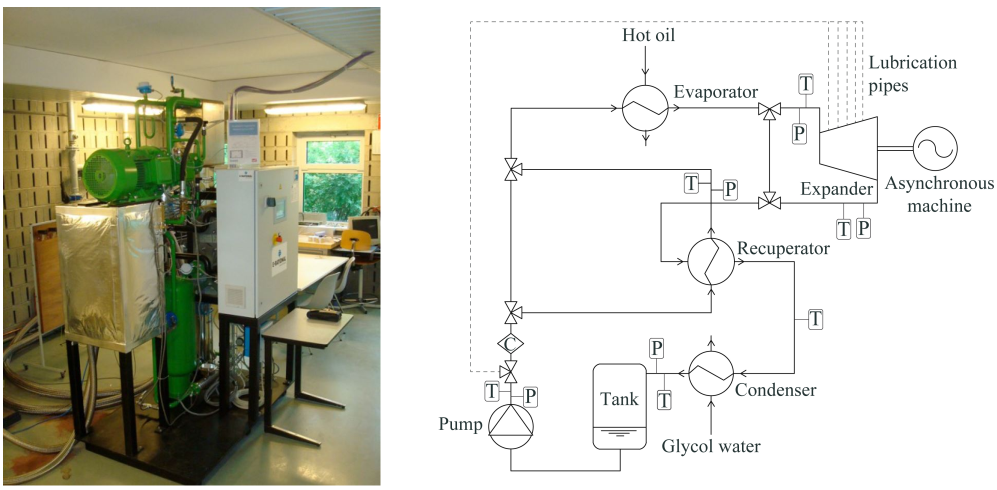
Description of the tests

In total, 120 steady-state experimental data points have been measured, covering a wide range of operating conditions (Table 1).

Performance	η_{cycle} (%)	η_{exp} (%)	η_{pump} (%)	T_{eva} (°C)	ΔT_{sc} (°C)	ΔT_{sh} (°C)	PP_{ev} (°C)	ΔP_{LP} (bar)	ΔP_{HP} (bar)
Min	2.2	27.3	12.3	119.3	9	1	0.1	0.06	$0.4{\cdot}10^{-3}$
Max	11.3	56.35	20	125	26	29	0.7	0.17	0.09

Table 1: Min/Max performance achieved during measurement campaigns. η_{cycle} cycle efficiency, η_{exp} isentropic effectiveness of the expander, η_{pump} effectiveness of the pump, T_{eva} temperature at the outlet of the evaporator, ΔT_{sc} condenser subcooling, ΔT_{sh} evaporator super-heating, ΔP_{LP} and ΔP_{HP} low and high pressure drop respectively.

The Solkatherm flow circulating in the test rig is imposed by varying the pump rotational speed. The condensation pressure is varied by modifying the flow of glycol water. The dynamic response of the unit is analyzed applying a step function to the pump rotational speed. The value of the applied perturbation must be large enough to make the response detectable when compared to noise.



(a) (b)

Figure 1: 1(a) Side view of the ORC test bench. 1(b) Schematic layout and sensors position of the ORC test rig.

ORC POWER SYSTEM MODEL AND VALIDATION

Steady state modeling

Performance curves for the different components of the ORC system have been developed based on the steady-state experimental data points, since they are considered appropriate to be implemented into a dynamic model (Quoilin, 2011). The expander is modeled by its isentropic efficiency and its filling factor. These two parameters are expressed as a function of the rotational speed, the inlet pressure and the pressure ratio using empirical performance curves proposed by (Declaye et al., 2013). The parameters of these curves are identified with the experimental data using a non-linear optimization algorithm. The turbo-pump is characterized by its isentropic effectiveness and the delivered mass flow rate. The heat exchangers are modeled by 4 heat transfer coefficients (three for the liquid, two-phase and vapor zones on the refrigerant side and one for the secondary fluid side). Inconsistency during the transition between two heat transfer coefficients is avoided by means of a smooth transition function. The recuperator model is based on the condenser and evaporator model considering a single fluid phase on both side of the exchanger. It is assumed that the pressure drops in both the high pressure and low pressure lines are lumped on the vapor sides of both lines. This hypothesis is valid only if the pressure drops are relatively small in the heat exchangers, which is the case in the present test rig (see table 1). From experimental data all the exchangers show a too small pinch point (smaller than temperature sensor uncertainty), which is likely due to the large exchange area of the components. For that reason it was not possible to validate the exchangers model.

Dynamic modeling

The dynamic model of the ORC power system has been built using the *ThermoCycle* library, an open-source Modelica library recently developed by the Energy Systems Research Unit at the University of Liege. In the proposed model the organic fluid properties are computed with the external program CoolProp, linked to Modelica through an appropriate package. In this work all the simulations are performed using the commercial program Dymola (Dynamic Modeling Laboratory). The heat exchangers are modeled by means of a discretized 1-D model (Quoilin, 2011). Since the time constants characterizing the turbo-pump and the screw expander are large compared to the heat exchangers, the performance curves described above are implemented. In order to consider the damping effect of the tank after the condenser a liquid receiver model is implemented based on the assumptions of thermodynamic equilibrium at all times. Non-condensible gases are considered in the tank by taking into account their partial pressure.

Dynamic validation

A transient trend of the power block is obtained by varying with a step change the pump rotational speed while keeping constant the expander rotational speed. During the experiments the control of the external heating circuit ensures a constant temperature of the thermal oil at the inlet of the evaporator. Neither the thermal oil flow of the evaporator nor the cooling water glycol flow are modified during the transients. It has been ensured that the power plant was operating in steady-state condition before any change is imposed to the system.

CONCLUSIONS

In this work, a dynamic model of low-capacity ORC system has been developed, calibrated, and validated with experimental data. The test rig exhibits good thermodynamic performance for the imposed temperature levels (about 11% efficiency for a heat source of 125 °C) and allowed measuring a large amount of operating conditions. The model validation shows a good agreement between measurements and predictions, both in steady-state and transient conditions. The only parameter that could not be predicted by the model is the sub-cooling since it mainly depends on the amount of non-condensing gases present in the cycle. Finally, this paper demonstrates that the *ThermoCycle* library can reliably be used for the modeling of such thermodynamic systems, allowing for instance the implementation and simulation of various control strategies.

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