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Comparison of shell-and-tube with plate heat exchangers for the use in low-temperature organic Rankine cycles

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A B S T R A C T

Organic Rankine cycles (ORCs) can be used for electricity production from low-temperature heat sources. These ORCs are often designed based on experience, but this experience will not always lead to the most optimal configuration. The ultimate goal is to design ORCs by performing a system optimization. In such an optimization, the configuration of the components and the cycle parameters (temperatures, pressures, mass flow rate) are optimized together to obtain the optimal configuration of power plant *and* components. In this paper, the configuration of plate heat exchangers *or* shell-and-tube heat exchangers is optimized together with the cycle configuration. In this way every heat exchanger has the optimum allocation of heat exchanger surface, pressure drop and pinch-point-temperature difference for the given boundary conditions. ORCs with plate heat exchangers perform mostly better than ORCs with shell-and-tube heat exchangers, but one disadvantage of plate heat exchangers is that the geometry of both sides is the same, which *can* result in an inefficient heat exchanger. It is also shown that especially the cooling-fluid inlet temperature and mass flow have a strong influence on the performance of the power plant.

1. Introduction

Low-temperature geothermal heat sources are widely available [1], but the electricity production efficiency is low due to the low temperature. Many authors have tried to maximize the electricity production efficiency of organic Rankine cycles (ORCs) [2–4] by optimizing the cycle parameters (pressures, temperatures and mass flow rates). To perform such an optimization, simplifying assumptions are made about the components, but these assumptions can have a strong influence on the performance and the cost of the ORC.

As already explained in our previous work [5], this issue is already touched upon in the literature. Madhawa Hettiarachchi et al. [6] minimized the ratio of the total heat-exchanger surface and the net electrical power produced by the cycle, for a fixed heat-exchanger configuration. Quoilin et al. [7] developed an ORC model and plate heat exchanger models based on their experimental set-up, while neglecting the pressure drop in the heat exchangers. They used these models to predict the performance of their set-up in different working conditions. Shengjun et al. [8] maximized the performance of ORCs with shell-and-tube heat exchangers, while keeping the configuration of the heat exchangers fixed. Domingues et al. [9] optimized ORCs with shell-and-tube heat exchangers for vehicle exhaust waste heat recovery and investigated the effect of the number of tubes in the shell-and-tube heat exchangers on the performance of the ORC. Another approach was followed by Franco and Villani [10]. They divided the ORC in a system level and a component level. In a first step, the system level was optimized, followed by the optimization of the configuration of the components for the obtained optimal system configuration. An iteration between the optimization of both levels was needed to come to the final solution.

To obtain the global optimum configuration of the ORC, the system and the components should be optimized together so that the configuration of the components is optimal for the use in the cycle and so that the components are adjusted to each other. To perform such a system optimization, realistic models, which describe the performance of the components depending on geometric parameters, are needed.

In this paper models for heat exchangers are implemented and included in the system optimization. Both shell-and-tube heat exchangers and plate heat exchangers are discussed. In previous research [5] we have performed a detailed investigation of shell-

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Nomenclature

Greek	
β	corrugation angle (°)
Δp	pressure drop (Pa)
ΔT	temperature difference (°C)
η	efficiency (–)
Λ	corrugation width (m)
μ	dynamic viscosity (Pa s)
Φ	area enlargement factor (–)
ρ	density (kg/m^3)
θ	angle (°)
Roman	
a	corrugation amplitude (m)
A	area (m^2)
Bo	boiling number (–)
c_p	specific heat capacity (J/kg K)
d_o	tube outside diameter (m)
D	diameter (m)
e	specific exergy (kJ/kg)
G	mass velocity (kg/m^2 s)
h	heat transfer coefficient (W/m^2 K)
h	specific enthalpy (J/kg)
l_c	baffle cut length (m)
L_b	distance between baffles (m)
L_h	length of a plate (m)
L_p	distance between ports (m)
$\dot{m}$	mass flow (kg/s)
Nu	Nusselt number (–)
p	pressure (bar)
Pr	Prandtl number (–)
p_t	tube pitch (m)
$\dot{Q}$	heat flow (kW)
Re	Reynolds number (–)
T	temperature (°C)
W	plate width (m)
$\dot{W}$	mechanical power (kW)
X	corrugation parameter (–)
Sub- and superscripts	
0	dead state
1 – 9	number of the state
ac	acceleration
cycle	cycle
en	energetic
ex	exergetic
fr	frictional
h	hydraulic
id	ideal
in	inlet
max	maximum
min	minimum
net	nett
l	longitudinal
otl	outermost tubes
out	outlet
plant	plant
s	shell
source	heat source
t	transverse
tot	total
wf	working fluid

and-tube heat exchangers integrated in ORCs. The shell-and-tube heat exchangers are modeled with the Bell-Delaware method [11,12], which is a mature model and can be used for single-phase flow, condensing and evaporation.

The purpose of our research reported in this paper is to integrate plate heat exchangers in ORCs and to compare the result to that obtained with shell-and-tube heat exchangers. Martin [13] developed a model for plate heat exchangers with single-phase flow. This model is based on physical reasoning and many experimental data are used. Such generally applicable models do not exist for two-phase plate heat exchangers used as evaporators or condensers, although much research has been performed on the topic [14–20]. The authors of those references propose correlations for heat-transfer coefficient and pressure drop based on own experiments and these correlations are therefore only valid for the investigated cases. We shall utilize the correlations of Han et al. [16,17] for evaporation and condensation in plate heat exchangers, respectively. These papers correlate the performance of plate heat exchangers to many geometrical parameters.

This paper extends the work performed in Walraven et al. [5], in which ORCs with shell-and-tube heat exchangers are optimized for a reference case. The conclusion from that work is that it is optimal to use the 30° and 60° tube configurations in single and two-phase heat exchangers, respectively. In this paper, models for plate heat exchangers are added and ORCs with plate heat exchangers are compared to ORCs with shell-and-tube heat exchangers (with the optimal tube configuration). The influence of the heat-source-inlet temperature, heat-source-outlet temperature, total heat exchanger surface, cooling-fluid inlet temperature and the cooling fluid mass flow rate on the performance of the power plant are also investigated. The comparison between ORCs with the two different types of heat exchangers is performed in a wide range of parameters and for many fluids.

2. Organic Rankine cycle

Different types of ORCs exist and are simulated in this paper. The investigated cycles can be of the simple or recuperated type, be subcritical or transcritical and can have one or two pressure levels. Two examples are given in Fig. 1, in which the scheme of a single-pressure, recuperated ORC and a double-pressure, simple ORC are shown. All the possible heat exchangers (economizer, evaporator, superheater, desuperheater, condenser and recuperator) are shown in the figure, but are not always necessary.

In all configurations it is assumed that state 1 is saturated liquid and that the isentropic efficiencies of the pump and turbine are 80% and 85%, respectively. More information of the modeling can be found in previous work [4,5]. Instead of assuming a fixed pinch point temperature difference and ideal heat exchangers, models are used to calculate the heat transfer coefficients and pressure drops in each heat exchanger.

3. Shell-and-tube heat exchanger

The shell-and-tube heat exchanger type has already been studied in Walraven et al. [5]. Here we now recall some elements to support the later analysis with plate heat exchangers. TEMA E type