Energy 35 (2010) 1033–1039

Contents lists available at ScienceDirect

Energy

journal homepage: www.elsevier.com/locate/energy

Efficiency optimization potential in supercritical Organic Rankine Cycles

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ARTICLE INFO

Article history:
Received 30 October 2008
Received in revised form
19 May 2009
Accepted 9 June 2009
Available online 8 August 2009

Keywords:
ORC
Organic Rankine Cycle
Supercritical fluid
Low temperature heat recovery

ABSTRACT

Nowadays, the use of Organic Rankine Cycle (ORC) in decentralised applications is linked with the fact that this process allows the use of low temperature heat sources and offers an advantageous efficiency in small-scale concepts. Many state-of-the-art and innovative applications can successfully use the ORC process. In this process, according to the heat source level, special attention must be drawn to the choice of the appropriate working fluid, which is a factor that affects the thermal and exergetic efficiency of the cycle. The investigation of supercritical parameters of various working fluids in ORC applications seems to bring promising results concerning the efficiency of the application.

This paper presents the results from a simulation of the ORC and the optimization potential of the process when using supercritical parameters. In order to optimize the process, various working fluids are considered and compared concerning their thermal efficiency and the usable percentage of heat. The reduction of exergy losses is discussed based on the need of surplus heat exchanger surface.

1. Introduction

The difference between the Organic Rankine Cycle (ORC) and the classical Clausius Rankine Cycle is the use of organic working fluid instead of water-steam. ORC has a lot of advantages in applications in which low temperature heat sources are used (e.g. geothermal energy, solar desalination and waste heat recovery) [1]. One of the main challenges of the ORC is the choice of an appropriate working fluid and of the particular cycle design with which maximum thermal efficiency can be achieved. Bahaa et al. give already a good overview of thermal cycle efficiencies of ORCs for various working fluids at live steam temperatures of 100 °C both subcritical and supercritical [2].

The Critical Point (CP) of organic fluids is reached at lower pressures and temperatures compared with water. Therefore supercritical fluid parameters are easily realized. In the state-of-the-art applications, which are discussed nowadays, saturated or slightly superheated vapour is expanded in the turbine. However, the investigation of supercritical fluid parameters is of high importance, since it may lead to higher efficiencies making these plants even more attractive for waste heat applications. Analogue heat-pumps are operated at supercritical condition, which also leads to a better adaptation of the hot and cold stream leading to lower exergy losses for ORC [3].

2. Supercritical ORC process

Fig. 1 shows the process of a sub- and supercritical ORC in a T-s-Diagram for a constant superheated vapour temperature.

Even for constant superheated vapour temperatures, the heat input occurs at a higher average temperature level in the case of supercritical vapour parameters. In reality such big superheating as shown in the diagram would not be realized due to the tremendous heat exchange area needed due to the low heat-exchange coefficient of the gaseous phase [4].

The heat input to the working fluid of the ORC process is done usually with the help of thermal oil and is equal to:

$$\dot{Q}_{\text{Organic fluid}} = \dot{m}_{\text{ORC}}(h_3 - h_2) \quad (1)$$

h_1, h_2, h_3 and h_4 are the specific enthalpies according to Fig. 1.

The thermal efficiency of the cycle is defined as follows:

$$\eta_{\text{th}} = \frac{P_{\text{mech}}}{\dot{Q}_{\text{Organic fluid}}} \quad (2)$$

P_{mech} is the net mechanical power produced with the ORC process (which will be assumed as equal the net electrical power). This power output is analogue to the enthalpy fall in the turbine minus the enthalpy rise in the pump:

$$P_{\text{mech}} = \dot{m}_{\text{ORC}}[(h_3 - h_4) - (h_2 - h_1)] \quad (3)$$

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doi:10.1016/j.energy.2009.06.019

Nomenclature

A	area [m^2]
c_p	specific isobaric heat capacity [kJ/kg K]
$\dot{E}$	exergy flow [kW]
e	specific exergy [kJ/kg]
$\dot{m}$	mass flow [kg/s]
P	power [kW]
p	pressure [bar]
q	specific heat flow [kW/kg]
$\dot{Q}$	heat flow [kW]
s	specific entropy [kJ/kg K]
T	temperature [K]
t	temperature [°C]
U	heat transfer coefficient [W/m^2 K]
W	specific work [kJ/kg]
η	Efficiency [%]

In the case of supercritical process, the enthalpy fall $(h_{3'} - h_{4'})$ is higher than in the subcritical one, when on the other hand, the feed pump's additional specific work to reach supercritical pressure, which corresponds to the enthalpy rise $(h_{2'} - h_2)$, is very low.

Therefore, according to Eq. (1), the efficiency of the process is higher in the case of supercritical ORC parameters and this fact provides new frontiers in the investigation of ORC applications.

The efficiency of the heat-exchange system, which transfers the heat from the heat source (HS) to the organic fluid, is defined by the following Eq. (5):

$$\eta_{\mathrm{HEx}} = \frac{\dot{Q}_{\mathrm{Organic\ fluid}}}{\dot{Q}_{\mathrm{HS}}} \tag{4}$$

Finally, the efficiency of the whole system is defined as follows:

$$\eta_{\mathrm{System}} = \frac{P_{\mathrm{mech}}}{\dot{Q}_{\mathrm{HS}}} = \eta_{\mathrm{HEx}}\eta_{\mathrm{th}} \tag{5}$$

As the system efficiency is directly linked with the efficiency of the heat-exchange system, it is obvious that the aim is to maximise the transferred heat.

The above-presented efficiencies will be used for the qualitative analysis of the ORC applications, which will be described in this paper.

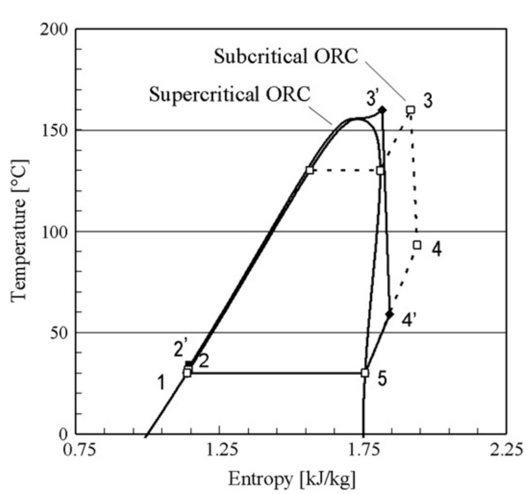

Fig. 1. Sub- and supercritical ORC. Example of R245fa.

3. The triangular cycle

It is widely known that the ideal thermal efficiency of a power generation cycle is the Carnot efficiency. However, this efficiency is linked with the fact that the heat input is realized in constant heat source temperature. Often authors apply the temperature of the heat source for the upper temperature to determine the maximum possible Carnot efficiency [5]. The efficiency of this ideal cycle is provided by the following widely known equation:

$$\eta_C = \frac{|W|}{q_{\mathrm{in}}} = \frac{q_{\mathrm{in}} - |q_{\mathrm{out}}|}{q_{\mathrm{in}}} = \frac{T_H - T_L}{T_H} = 1 - \frac{T_L}{T_H} \tag{6}$$

In power production cycles, like for example geothermal cycles, in the heat-exchange process, the heat source is cooled down [6] so that high Carnot efficiency and maximum amount of transferred heat to the process are competing objectives as can be seen in Fig. 2. On the other hand, the heat sink is heated to a temperature not very higher as the initial one [7].

In order to determine the maximum possible Carnot efficiency, this has to be taken into account by using the mean temperature of the heat transferred, which results in the model of the triangular (or trilateral) cycle, which is shown in Fig. 3 [8].

The work gained in the triangular cycle is presented in Fig. 4 and the thermal efficiency of the cycle is equal to:

$$\eta_T = \frac{T_H - T_L}{T_H + T_L} \tag{7}$$

In the literature, the triangular process has been compared with either refrigeration or heating cycles (e.g. supercritical CO_2 cycle) [9].

The ideal working process for a given heat source would be able to transfer the heat from the source at infinitesimal temperature difference, beginning at ambient temperature, which equals the condensation temperature of this process up to the heat source temperature. It is obvious that this process is impossible in reality but it serves as a benchmark for all other processes.

In processes like ORC, in which the heat input is not isothermal, the improvement of the process aims at the approach of the cycle to

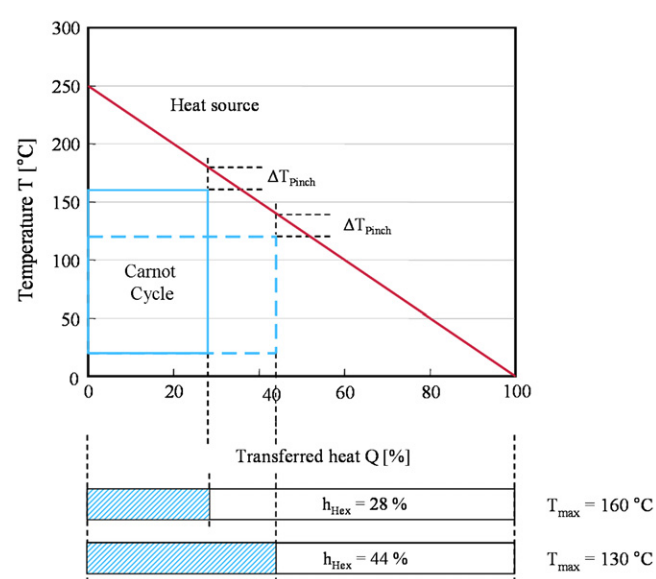

Fig. 2. Heat exchanger efficiency for cooled down heat source, for ideal (Carnot) cycle with $T_{max} = 130$ °C and 160 °C.