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Optimal boiling temperature for ORC installation

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Abstract In the paper a research on cost-effective optimum design boiling temperature for Organic Rankine Cycle utilizing low-temperature heat sources is presented. The ratio of the heat exchanger area of the boiler to the power output is used as the objective function. Analytical relations for heat transfer area as well power of the cycle are formulated. Evaporation temperature and inlet temperature of the heat source medium as well its mass flow rate are varied in the optimization method. The optimization is carried out for three working fluids, i.e. R 134a, water and ethanol. The objective function (economics profitability, thermodynamic efficiency) leads to different optimal working conditions in terms of evaporating temperature. Maximum power generation in the near-critical conditions of subcritical ORC is the highest. The choice of the working fluid can greatly affect the objective function which is a measure of power plant cost. Ethanol exhibits a minimum objective function but not necessarily the maximum cycle efficiency.

Keywords: ORC; Optimum design criteria; Fluid selection

Nomenclature

A	–	heat transfer surface, m^2
c_p	–	specific heat, J/kgK
C_{pf}	–	specific heat of ORC working fluid, J/kgK

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C_{pw}	–	specific heat of water, J/kgK
h	–	enthalpy, kJ/kg
h_{fg}	–	latent heat of evaporation, kJ/kg
k	–	overall heat transfer coefficient, W/m^{2}K
$\dot{m}$	–	mass flowrate, kg/s
s	–	entropy, kJ/kgK
W	–	power, W
t	–	temperature, °C
T_b	–	evaporation temperature, °C
T_s	–	condensation temperature, °C
Δt_m	–	logarithmic mean temperature difference, °C
x	–	quality
Q	–	rate of heat, W

Subscripts

$1, ...5$	–	state points
b	–	evaporation
f	–	ORC working fluid
p	–	constant pressure
w	–	waste heat

1 Introduction

The generation of power using industrial waste heat has been growing in the past years. Due to the increasing energy prices, it is becoming more and more economically profitable to recover even the low grade waste heat. The potential for exploiting waste heat sources from engine and power plant exhaust gases or industrial processes is particularly promising. An often used solution is the transformation of waste heat into electricity. For this a conventional steam turbine is a classic option. Other most widely used proposed solution is application of the organic Rankine cycle (ORC).

In ORC installation the working fluid is an organic substance better suited than water to undergo phase changes at lower temperatures of the heat source. The waste heat is used to produce vapor that is being expanded in the ORC turbine to generate electricity. A drawback to the use of steam is often the limited temperature level of the waste heat source. This puts a constraint on the maximum temperature and the evaporation pressure of the generated vapor in the ORC installation, and thus restricts the achievable electric efficiency of such power cycle. The low temperature heat source can either be of gaseous or liquid origin. The temperature of the heat source is a major determinant of the type of technology required to extract the heat and the uses to which it can be applied. Utilization

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of low-temperature heat sources has received much attention in a number of laboratories. In low-temperature systems there are large heat exchange surfaces required to extract required amount of energy. These factors emphasize the necessity of optimizing the cost effective design of power cycles.

In this paper a cost effective design optimization method is presented for ORC. In the analysis the cost of heat exchangers surface contributes largely to the power plant cost. Therefore, the performance of the power plant can be optimized using the optimization criterion similar to thosegiven in [1–3], where the ratio of total heat transfer area to total net power is considered to be the objective function. In our analysis the temperature of condensation is assumed as constant and equal to 30 °C. Hence the change of heat exchange area of condenser does not influence markedly the optimal boiling temperature in a cycle for the given working fluid. In the considered case evaporation temperature is varied in the optimization process. Also varied is the inlet temperature of the heat source medium as well as its mass flow rate. The optimum cycle performance is evaluated for various working fluids and compared. As the results of optimization show the correct selection of the working fluid can greatly affect the power plant cost as well as efficiency. The advantage here of the use of ORC is the fact that the low-temperature cycles better fill the area of wet vapor region in the lower temperature range and bring in such way these cycles closer to the ideal thermodynamical cycles. Secondly, the efficiency of the low-pressure (LP) turbine part is much lower in case of steam as working fluid than in the case of the low-boiling point fluid in ORC turbine. It is usually assumed that the efficiency of the last stage of the steam turbine is about 60% while the organic fluid turbine efficiency can be assumed at the level of 85%. Additionally, steam quality after expansion in the traditional steam cycle is low, but corresponding expansion in the ORC installation terminates at higher qualities of ORC working fluid wet vapor or even in the superheated region.

The economics of the ORC system is strictly linked to thermodynamic properties of the working fluid. A wrong selection of working fluid could lead to a less efficient and expensive plant generation unit. Some selection criteria have been put forward by various authors [2–11], incorporating thermodynamic properties, provided in literature but these do not have a general character. Authors also worked on the topic of working fluid selection and developed a thermodynamic criterion for wet fluids [12].

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2 Analysis

The schematic of the subcritical ORC and the temperature-entropy (T-s) diagram of the cycle are shown in Fig. 1. Heating medium, in the present case hot stream of water, is passed through the evaporator and is heating the organic working fluid with a low boiling point temperature. Vapor generated in the evaporator is used to drive the turbine. The working fluid leaving the turbine is then condensed and pumped back to the evaporator. The working fluid is then passed to the evaporator and completing the closed loop.

3 Theoretical analysis

In our case we analyze only the heat transfer area of evaporator, assuming that the surface of condenser is varying only with the change of mass flow rate. Temperature of condensation is assumed at 30 °C. During the optimization process the objective function is minimized by varying temperature of evaporation and inlet temperature of heat source. Steady state operation of the cycle is considered in the analysis. Saturated vapor is considered at the turbine inlet and saturated liquid is assumed at the condenser exit. Pumping work is neglected. For simplicity mean temperature drop in evaporator is assumed as arithmetic.

According to the notation presented in Fig. 1 the overall heat balance for evaporator is

$$\dot{m}_w(h_{1w} - h_{2w}) = \dot{m}_f(h_4 - h_1) \tag{1}$$

or when enthalpies are expressed through the product of specific heat and temperature:

$$\dot{m}_w C_{pw}(t_{1w} - t_{2w}) = \dot{m}_f \left[C_{pf}(t_{4f} - t_b) + h_{fg}(t_b)\right] . \tag{2}$$

Heat balance up to the pinch point in evaporator reads:

$$\dot{m}_w(h_{1w} - h_p) = \dot{m}_f h_{fg}(t_b) \tag{3}$$

or when enthalpies are expressed through the product of specific heat and temperature:

$$\dot{m}_w(h_{1w} - h_p) = \dot{m}_f h_{fg}(t_b) \tag{4}$$

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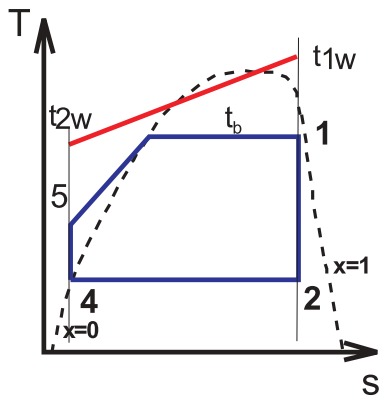

Figure 1. Temperature-entropy diagram of the ORC.

or

$$\dot{m}_w C_{pw}(t_{1w} - t_p) = \dot{m}_f h_{fg}(t_b) \ , \tag{5}$$

where

$$t_p = t_b + \Delta t_{\min} \ . \tag{6}$$

Dividing Eq. (3) by (6) we obtain:

$$\frac{t_{1w} - t_{2w}}{t_{1w} - t_p} = \frac{C_{pf}(t_b)(t_b - t_{4f}) + h_{fg}(t_b)}{h_{fg}(t_b)} \ . \tag{7}$$

Introducing (6) into (7) and rearranging we obtain:

$$t_{2w} = t_{1w} - (t_{1w} - t_b - \Delta t_{\min}) \left[\frac{C_{pf}(t_b)(t_b - t_{1f})}{h_{fg}(t_b)} + 1 \right] \ . \tag{8}$$

From (7) we know the ratio of mass flow rates of fluid in ORC system and heat source is

$$\frac{\dot{m}_f}{\dot{m}_w} = \frac{C_{pw}(t_{1w} - t_p)}{h_{fg}(t_b)} \ . \tag{9}$$

The power produced by ORC installation follows from the energy balance for the ORC cycle

$$W = \dot{m}_f(h_{1f} - h_{2f}) = \dot{m}_f \left[C_{pf}(t_{4f} - t_b) + h_{fg}(t_b) - C_{pf} t_{4f} - x h_{fg} \right] \ . \tag{10}$$

Introducing (10) into (11) we obtain power of ORC installation produced by unit of mass flow rate of heating medium:

$$\frac{W}{\dot{m}_w} = \frac{C_{pw}(t_{1w} - t_p)}{h_{fg}(t_b)} \left[C_{pf}(t_{4f} - t_b) + h_{fg}(t_b) - C_{pf} t_{4f} - x h_{fg} \right] \ . \tag{11}$$

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Another issue here is determination of the state 2 after expansion in turbine. Entropy of state 2 can be determined from the knowledge of entropy in state 1 (Fig. 1) and the change of entropy resulting from heating of the working fluid to saturation temperature in evaporator and finally conversion of liquid to the saturated vapor state 3, hence

$$s_1 = s_5 + \int_{T_5}^{T_b} \frac{c_{pf} dT}{T} + \frac{h_{fg}\left(t_b\right)}{T_b} \ . \tag{12}$$

Entropy of the state 5 is equal to entropy of the state 4, whereas it can be assumed that temperature of the state 5 is more or less equal to condensation temperature, i.e.

$$s_5 = s_4 \qquad T_5 \cong T_s \ . \tag{13}$$

In such case entropy of state 1 reads

$$s_1 = s_4 + \int_{T_s}^{T_P} \frac{c_p f dT}{T} + \frac{h_{fg}\left(t_b\right)}{T_b} = s_4 + \bar{c}_{pf} \ln \frac{T_b}{T_s} + \frac{h_{fg}\left(t_b\right)}{T_b} \ . \tag{14}$$

On the other hand the entropy of state 2 is:

$$s_2 = s_4 + \frac{h_{fg}\left(t_s\right) x}{T_s} \ . \tag{15}$$

We must remember also that $s_1 = s_2$. This allows to determine vapour quality x to be:

$$x = \bar{c}_{pf} \frac{T_s}{h_{fg}(t_s)} \ln \frac{T_b}{T_s} + \frac{T_s h_{fg}\left(t_b\right)}{T_b h_{fg}(t_s)} \ . \tag{16}$$

In order to find the ratio of heat transfer area to power produced by the cycle we have to calculate the heat transfer area of evaporator. The mean temperature drop of heat exchanged between heating source medium and working fluid in ORC installation is an average temperature drop in the section, in which fluid is warming up to saturation temperature and in boiling section. For simplicity the mean temperature drop is calculated as an arithmetic means value of outlet and inlet temperature drop on each section, hence:

$$\Delta t_m = \frac{t_p - t_b + t_{2w} - t_{1f}}{2} + \frac{t_{1w} - t_b + t_p - t_b}{2} \ . \tag{17}$$

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Heat balance for evaporator gives:

$$Q = \dot{m}_w C_{pw}(t_{1w} - t_{2w}) = k\Delta t_m A \ , \tag{18}$$

hence, the ratio of heat transfer area of evaporator to mass flow rate of heating medium is

$$\frac{Ak}{\dot{m}_w} = \frac{C_{pw}(t_{1w} - t_{2w})}{\Delta t_m} \ . \tag{19}$$

The ratio of heat transfer area of evaporator to power produced by the ORC system can be obtained dividing (20) by (12)

$$\frac{Ak}{W} = \frac{C_{pw}(t_{1w} - t_{2w})}{C_{pw}(t_{1w} - t_p)\Delta t_m} \frac{h_{fg}(t_b)}{\left[C_{pf}(t_{4f} - t_b) + h_{fg}(t_b) - C_{pf}t_{4f} - xh_{fg}\right]} \ , \tag{20}$$

and this relation is our optimization criterion.

4 Results of thermoeconomic optimization and discussion

As the work presented here is a part of the project aimed at utilization of waste heat in the reference power plant in the manner to produce electricity in ORC installation therefore the precondition of the study was to use the waste heat in the form of a stream of hot water at 90 °C, recovered from the exhaust gases. Such low enthalpy heat source is rather insufficient to produce a good quality vapor to feed the ORC turbine. That was the incentive to search for the ways of increasing the temperature of the vapor at ORC turbine inlet. In this paper presented will be calculations with heating the ORC using the bleed steam. The evaporation temperature of working fluid is optimizing. The condensation temperature has been assumed at the level of 30 °C.

In Tab. 1 presented is a short characteristics of considered working fluids, possible for use in the ORC installation. In calculations the thermal and transport properties were taken from the newest database of physical properties of fluids Refprop9.0 [13]. In Tabs. 2 and 3 presented are the results of obtained calculations. Calculations have been accomplished using the MathCad15 software. Minimum pinch point temperature was assumed as $\Delta t_{min} = 5$ K, and the initial heat source temperature was a stream of hot water with inlet temperature $t_{w1} = 90$ °C.

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Table 1. Characteristics of considered working fluids.

Fluid	M	p_{cr}	T_{cr}	p_{cond} (30 °C)
	[kg/kmol]	[bar]	[°C]	[bar]
R134aa	102.03	40.593	101.06	7.7020
Water	18.015	220.64	373.95	0.042
C_2H_5OH	46.068	61.48	240.75	0.10555

Fluids selected for examination fall into the category of wet fluids. For the time being there are no considerations on how to increase the evaporation temperature of working fluid before the turbine. That will be considered later. In Figs. 2–7 presented are the results of calculations. First in Fig. 2 presented is the distribution of the outlet temperature of the heat source at varying evaporation temperature for three fluids. The objective of a correct utilization of the heat source is to obtain as low as possible temperature at outlet. From three fluids considered it seems that R134a is capable of reducing the temperature of waste heat source to the highest extent. Next attention is focused on Fig. 3 where the ratio of ORC working fluid flow rate to the one of the heat source with respect to evaporation temperature of working fluid is presented. As the working fluid in ORC installation is usually quite expensive and forms a significant expenditure in the plant cost we should look for the smallest amount of ORC fluid required for installation. That is obtained with water as working fluid, followed by ethanol. In case of R134a we have to apply three times more fluid that in case of ethanol. In Fig. 4 presented is the distribution of the power generated in ORC installation related to the flow rate of heat source fluid. Clear maxima can be observed for three fluids. Generally the highest power is obtained for R134a, which can also be seen for the highest evaporation temperature.

In Fig. 5 presented is the ratio of heat transfer area of evaporator to the mass flow rate of heating medium. All presented distributions exhibit a sharp maximum for temperature of about 57 °C. In Fig. 6 presented is the ratio of heat transfer area of evaporator related to the generated power. From the point of view of design principles we are interested in finding a minimum of that ratio. All three considered fluid exhibit such minimum. In case of water the relation (21) shows a very stochastic character and for the sake of present work its minimum is found at a temperature of 56 °C, whereas in case of other two fluids the minimum is less sharp and can be

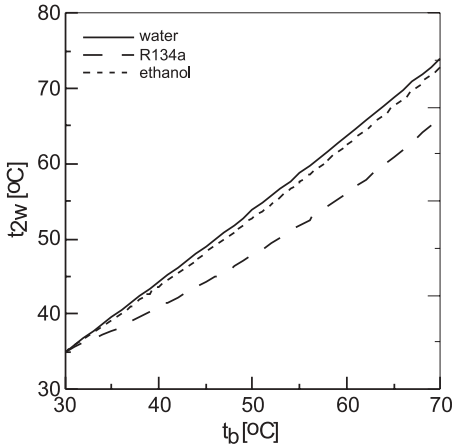

Figure 2. Relation of outlet temperature of the heat source with respect to evaporation temperature of working fluid.

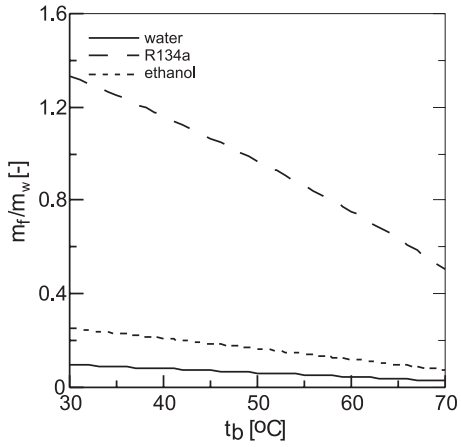

Figure 3. Relation of the ratio of ORC working fluid flow rate to heat source with respect to evaporation temperature of working fluid.

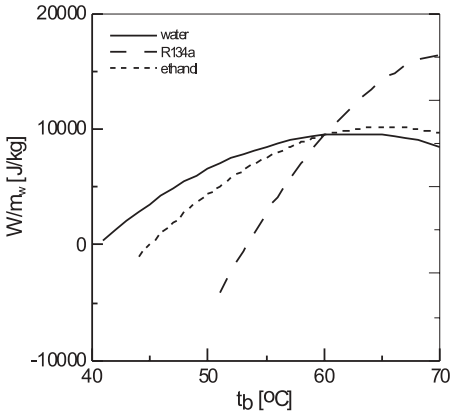

Figure 4. Relation of relative power to ORC fluid flow rate with respect to evaporation temperature of working fluid.

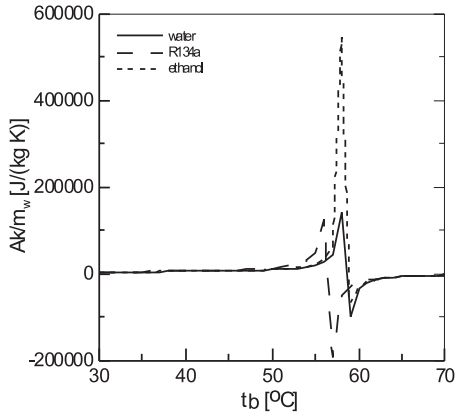

Figure 5. Relation of the ratio of heat transfer surface to ORC fluid flow rate with respect to evaporation temperature of working fluid.

found at temperatures of about 52 °C for ethanol, and 50 °C for water. In Fig. 7 presented is the distribution of vapour quality after expansion in turbine.

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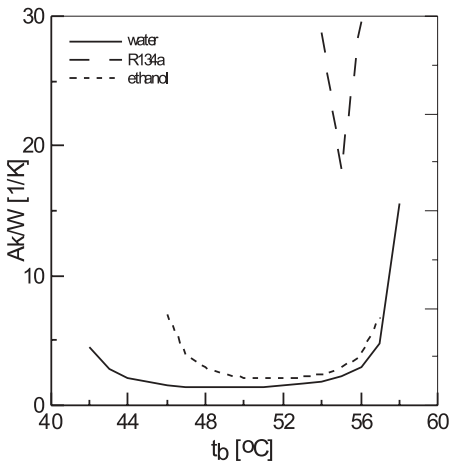

Figure 6. Relation of relative power to ORC fluid flow rate with respect to evaporation temperature of working fluid.

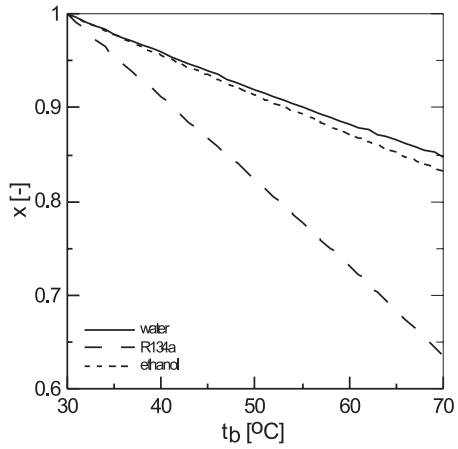

Figure 7. Relation of vapour quality after expansion in turbine with respect to evaporation temperature of working fluid.

5 Conclusions

In the paper presented is method of the optimal evaporation temperature results and heat recovery efficiency that mean power produced and area of heat exchanged engaged in this process. In the paper presented is a cost-effective optimum design boiling temperature for organic Rankine cycle utilizing low-temperature heat sources. The ratio of the heat exchanger area of the boiler to power output is used as the objective function. Analytical relations for heat transfer area as well power of the cycle are formulated. Evaporation temperature and inlet temperature of heat source medium as well its mass flow rate are varied in the optimization method. The optimization is carried out for three working fluids, i.e. R 134a, water and ethanol. The objective function (economy, profitability, thermodynamic efficiency) leads to different optimal working conditions in terms of evaporating temperature. Maximum power generation in near-critical conditions of subcritical ORC is the highest. The choice of working fluid can greatly affect the objective function which is a measure of power plant cost. Ethanol exhibits the minimum objective function but not necessarily the maximum cycle efficiency.

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