

## Influence of heat recuperation in ORC power plant on efficiency of waste heat utilization

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**Abstract** The present work is devoted to the problem of utilization of the waste heat contained in the exhaust gases having the temperature of 350 °C. Conversion of the waste heat into electricity using a power plant working with organic fluid cycles is considered. Three Organic Rankine Cycle (ORC) power plant solutions are analysed and compared: a solution with the basic, single thermodynamic conversion cycle, one with internal heat recuperation and one with external heat recuperation. It results from the analysis that it is the proper choice of the working fluid evaporation temperature that fundamentally affects the maximum of the ORC plant output power. Application of the internal heat recuperation in the plant basic cycle results in the output power increase of approx. 5%. Addition of the external heat recuperation to the plant basic cycle, in the form of a secondary supercritical ORC power cycle can rise the output power by approx. 2%.

**Keywords:** Heat recuperation; ORC, Subcritical power plant; Supercritical organic cycle

### Nomenclature

$h$	–	specific enthalpy, kJ/kg
$l$	–	specific work, kJ/kg
$\dot{m}$	–	flow rate, kg/s
$q$	–	specific heat flow rate, kJ/kg
$\dot{Q}$	–	total heat flow rate, kW

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### Subscripts

- $d$  – dual fluid
- $n$  – organic fluid
- $s$  – waste heat

## 1 Introduction

Power plants based on the thermodynamic cycles with working fluids other than water, and called the Organic Rankine Cycle (ORC) power plants, are used in the increasing number of applications. There are several reasons for such popularity. One of the main reasons results from the situation that, for the heat sources with low or moderate temperatures (geothermal heat, waste heat), the ORC power plant enables, at relatively good economy, a practical conversion of heat into electricity. Another important reason is here created by the growing energy prices and CO<sub>2</sub> emission charges, which provides an economical justification for the waste heat utilization.

The ORC power plant operation, in its basic version, can be seen as an analogy to that of the traditional water steam power plants, [1–3]. There is here, however, a possibility to use various working fluids (mostly organic fluids) in the power plant cycle, and the choice of the working fluid depends on many parameters, [3–5]. The ORC power plant effectiveness and the design solutions of the power plant main components (turbogenerator, heat exchangers, pumps, etc.) result from the specific choice of the cycle working fluid.

In the present work, an attempt was made to assess the effectiveness of utilization of the waste heat carried by the exhaust gases with the temperature of  $T_{s1} = 350$  °C and mass flow rate of  $\dot{m}_s = 10$  kg/s. Three ORC power plant solutions were considered:

- A single cycle ORC power plant without heat recuperation, (*BR Version*),
- A single cycle ORC power plant with internal heat recuperation, (*RW Version*), and
- A dual cycle ORC power plant consisting of the main subcritical ORC cycle and of an additional supercritical ORC cycle, whereby the latter is powered by the heat of the superheated vapour leaving the turbine of the main ORC cycle and is thus an external recuperator for the main cycle, (*RZ Version*).

For all the above ORC power plant solutions the temperature of the exhaust gases leaving the waste heat exchanger was assumed to be at least at 120 °C (the presumption arises from the assumed condensing moisture temperature of the exhaust gases) and the condensation of the working fluid to be at 40 °C.

## 2 The single cycle ORC power plant without heat recuperation

The first solution to be analysed was the single cycle ORC power plant without heat recuperation, comp. Fig. 1. Toluene was selected as the working fluid for that cycle, because of its relatively high critical temperature (318.6 °C) and the possibilities of direct use in exhaust gases-working fluid heat exchanger (without thermal oil as intermediate fluid). The main disadvantage of toluene as a working fluid is its toxicity and low pressure (lower than the ambient) during condensation process.

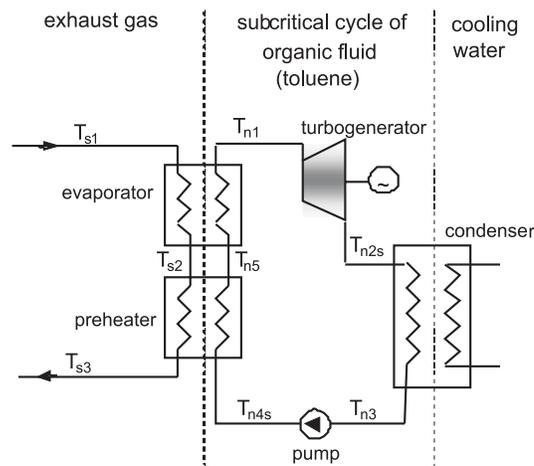


Figure 1. Scheme of the single cycle ORC power plant without heat recuperation (BR version).

The thermal efficiency and the theoretical output power of the ORC power plant were determined from the relationships (1) and (2):

$$\eta_n^{th} = \frac{l_{ob}}{q_n}, \quad (1)$$

$$N_n = \dot{m}_n l_{ob} , \quad (2)$$

where:

- $l_{ob} = l_t - l_p$  – specific work of the cycle,
- $l_t = h_{n1} - h_{n2s}$  – specific work of the turbine,
- $l_p = h_{n4s} - h_{n3}$  – specific work of the pump,
- $q_n = h_{n1} - h_{n4s}$  – specific heat supplied to the cycle.

The working fluid mass flow rate  $\dot{m}_n$  was received from the evaporator energy balance equation:

$$\dot{m}_n = \frac{\dot{Q}_{s-evap}}{h_{n1} - h_{n5}} , \quad (3)$$

where:

$$\dot{Q}_{s-evap} = \dot{m}_s c_{ps} \Delta T_{evap} , \quad (4)$$

$$\Delta T_{evap} = T_{s1} - T_{s2} , \quad (5)$$

$$T_{s2} = T_{n1} + \Delta T_1 . \quad (6)$$

whereby  $\Delta T_1 = 10$  K was assumed.

The following effects were analysed for this ORC power plant solution:

- Influence of the toluene evaporation temperature  $T_{n1}$  on the power plant efficiency and output power,
- Effectiveness of the waste heat utilization and the resulting temperature  $T_{s3}$  of the exhaust gases leaving the power plant.

The respective approach is illustrated in Figs. 2a and 2b.

The exhaust gas temperature  $T_{s3}$  at the exit of the power plant was calculated from the preheater energy balance equation:

$$T_{s3} = T_{s2} - \frac{\dot{m}_n (h_{n5} - h_{n4s})}{\dot{m}_s c_{ps}} . \quad (7)$$

The total heat flow rate supplied to the power plant cycle was determined from the relationship:

$$\dot{Q}_s = \dot{m}_s c_{ps} (T_{s1} - T_{s3}) . \quad (8)$$

Selected results of the above calculations are given in Tab. 1.

From the analysis, as outlined in Fig. 2 and supported by the calculation data presented in Tab. 1, it results that the decrease in the working

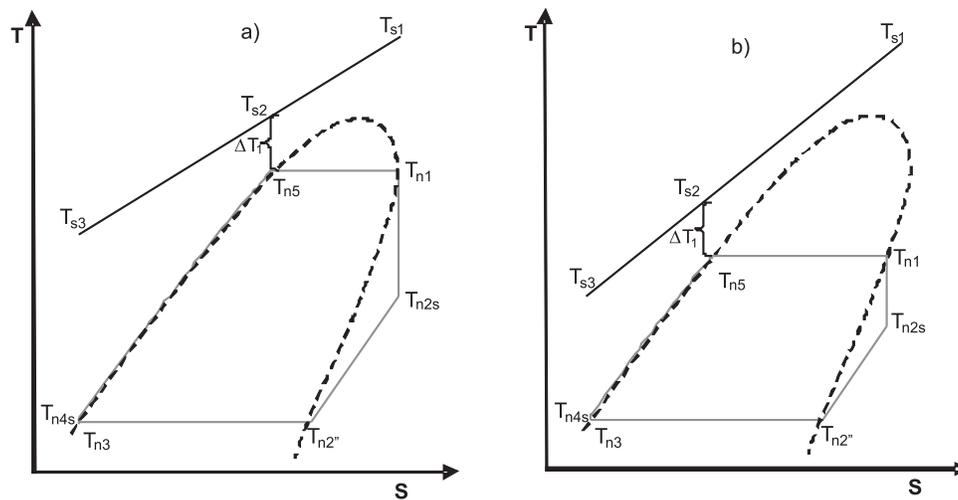


Figure 2. Thermodynamic cycles for the power plant without heat recuperation, at the working fluid evaporation temperature  $T_{n1}$  of: a) relatively high value, b) relatively low value.

Table 1. Selected calculation results for the ORC power plant without heat recuperation.

$T_{n1}$ [°C]	$\eta_{th}$ [%]	$NC-R$ [kW]	$\dot{Q}_s$ [kW]	$\dot{m}_n$ [kg/s]	$l_{ob}$ [kW]	$T_{s3}$ [°C]	$T_{n2s}$ [°C]
300	30.8	231.1	1132.6	1.03	224.76	262.9	116.1
280	30.2	336.9	1611.6	1.56	216.53	226.0	113.6
260	29.4	433.0	2051.1	2.11	204.84	192.2	106.7
240	28.5	517.3	2450.5	2,70	191.32	161.5	98.5
220	27.3	587.1	2808.1	3.33	176.49	134.0	90.4
200	26.0	639.8	3121.9	3.98	160.65	109.9	81.7
180	24.4	672.1	3389.6	4.67	143.96	89.3	73.1
160	22.5	680.2	3609.6	5.38	126.40	72.4	64.8
140	20.3	660.9	3778.7	6.12	108.06	59.3	57.2

fluid evaporation temperature  $T_{n1}$  leads to a lower temperature  $T_{s3}$  of the exhaust gases leaving the power plant. This contributes towards the requested higher heat flow rate  $\dot{Q}_s$  that is supplied to the power plant cycle. As a consequence, in line with the increasing mass flow rate  $\dot{m}_n$  of the work-

ing fluid, the ORC plant output power  $N_{ORC}$  is growing. Despite the fact that the maximum in the plant output power was received for the toluene evaporation temperature of  $T_{n1} = 160$  °C this output power result, along with all ones for  $T_{n1} < 209$  °C (a lower part of Tab. 1), should be practically disregarded. This is because for all the respective calculation cases the temperature  $T_{s3}$  of the exhaust gases leaving the power plant drops below its assumed minimum value of 120 °C. Also, it should be noted that the decreasing value of  $T_{n1}$  results in the decreasing thermal efficiency of the power plant.

The contrast of arriving to the decreasing output power of the ORC power plant along with its increasing thermal efficiency (at constant temperature of the heat source) seems apparently to be false and illogical. However, if a more detailed consideration is offered to the relationship (2) that describes the plant output power as a product of the working fluid mass flow rate and of the specific work of the plant cycle then it can be noticed that, for a certain group of the working fluids, along with their decreasing evaporation temperature, the working fluid mass flow grows at higher rate than that of the decrease in the specific work of the plant cycle.

The relationships (3) and (4) are also worth to be analysed in more detail. The relation (3) was used to calculate the working fluid mass flow rate. The denominator of (3) is determined by the value of the working fluid evaporation enthalpy that, according to the phase diagram, is growing when the fluid evaporation temperature decreases. Now, in spite that the denominator of (3) shows an upward trend for the decreasing fluid evaporation temperature the total energy flow  $\dot{Q}_{s-*evap*}$  supplied to the evaporator is growing due to the increasing temperature difference  $\Delta T_{*evap*}$  of the exhaust gases upstream and downstream of the evaporator.

The working fluid vapour temperature  $T_{n2s}$  at the turbine outlet is another important quantity that should be given a notice. The values of  $T_{n2s}$  as presented in Tab. 1 are much higher than the fluid condensation temperature  $T_{n3}$ . A possibility to utilize the heat carried by the fluid vapour leaving the turbine is analysed in the the next sections of the present work.

### 3 The single cycle ORC power plant with internal heat recuperation

Heat carried by the turbine outlet vapour can be used, in line with one of commonly known methods, to preliminary warm-up of the working fluid in

the plant cycle. This is called an internal heat recuperation. A power plant installation incorporating the internal heat recuperation is outlined in Fig. 3, and the corresponding thermodynamic cycle diagram is presented in Fig. 4.

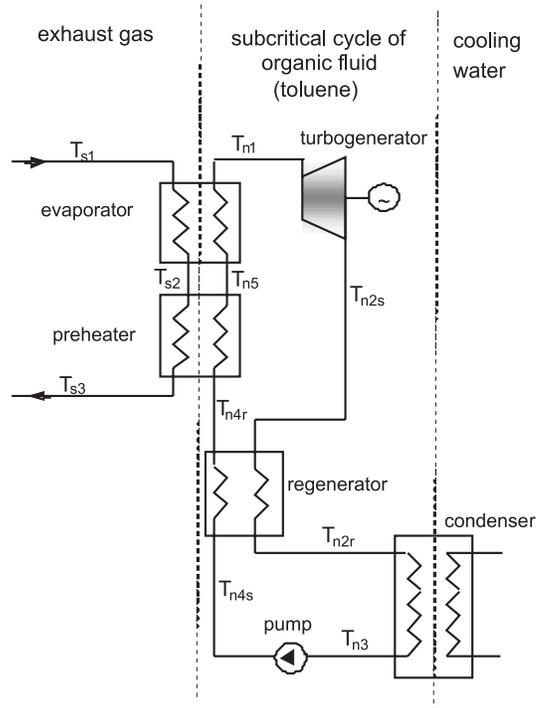


Figure 3. Scheme of the single cycle ORC power plant with internal heat recuperation (WR version).

The thermal efficiency of the ORC power plant incorporating the internal heat recuperation was calculated from:

$$\eta_r^{th} = \frac{l_{ob}}{q_{nr}}, \quad (9)$$

where the supplied specific heat is given by:

$$q_{nr} = h_{n1} - h_{n4r}. \quad (10)$$

The temperature  $T_{s3}$  of the exhaust gases leaving the power plant was de-

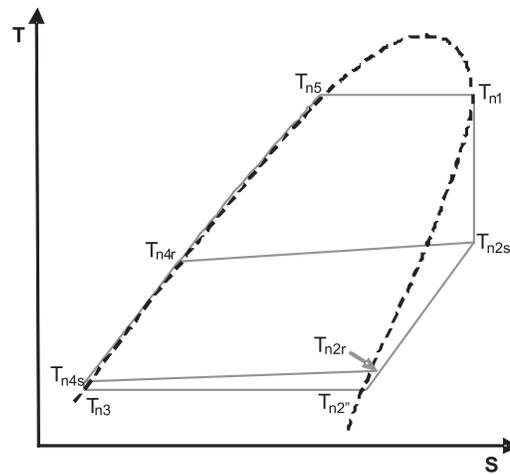


Figure 4. Thermodynamic cycle for the ORC power plant with internal heat recuperation.

terminated from the preheater energy balance equation:

$$T_{s3} = T_{s2} - \frac{\dot{m}_n (h_{n5} - h_{n4r})}{\dot{m}_s c_{ps}}, \quad (11)$$

and the remaining cycle parameters were calculated in analogy to the single cycle ORC power plant without heat recuperation.

The calculation results reflecting influence of the applied heat recuperation on the outlet temperature  $T_{s3}$  of the exhaust gases are presented in Fig. 5. In that picture, the toluene evaporation temperature of 209 °C, marked with point B and dashed line, denotes such value of the toluene evaporation temperature at which the highest plant output power is achievable for the plant cycle without heat recuperation, whereby the outlet temperature  $T_{s3}$  of the exhaust gases is still not below its minimum value of 120 °C. Application of the internal heat recuperation in the plant cycle enables that the respective toluene evaporation temperature can be decreased to the value of 196 °C, point A and dashed line, which results in a higher value of the plant output power. It appeared from the calculations that while the toluene evaporation temperature was decreased by 13 K the plant output power rose by 5%, approx. (i.e. by 24.9 kW).

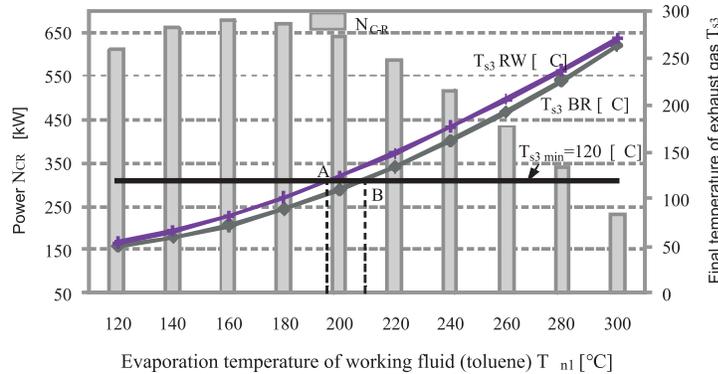


Figure 5. Diagram illustrating the influence of the working fluid evaporation temperature on the outlet temperature  $T_{s3}$  of the exhaust gases, for the ORC power plant (RW and BR versions).

#### 4 The dual cycle ORC power plant (external heat recuperation)

Eventually, the dual cycle ORC power plant was selected as the next solution in the present comparative analysis of effectiveness of the low temperature heat utilization. In that solution, the turbine outlet vapour of the main ORC power plant cycle becomes the upper heat source for an additional, supercritical ORC power cycle. The respective ORC power plant installation is outlined in Fig. 6 and the applicable thermodynamic conversion cycles are shown in Fig. 7.

Five organic fluids with varying critical parameters were analysed as the working fluids for the supercritical cycle, and their critical parameters are listed in Tab. 2

The thermal efficiency and theoretical output power of the main, sub-critical ORC power plant cycle with toluene as the working fluid were calculated in analogy to the calculations for the ORC power plant without heat recuperation. For the supercritical power plant cycle the following relationships were used:

$$\eta_{td}^{th} = \frac{l_{obd}}{q_{dd}}, \quad (12)$$

$$N_d = \dot{m}_d l_{obd}, \quad (13)$$

where:

Table 2. Critical parameters of the working fluids selected for the supercritical cycle.

Working fluid	Critical pressure	Critical temperature
	[ °C]	[MPa]
R41	5.897	44.13
R125	3.618	66.02
R218	2.671	71.95
R143a	3.761	72.71
R32	5.782	78.11

$$\begin{aligned}
 l_{obd} &= l_{td} - l_{pd} && \text{specific work of the cycle,} \\
 l_{td} &= h_{d1} - h_{d2s} && \text{specific work of the turbine,} \\
 l_{pd} &= h_{d4s} - h_{d3} && \text{specific work of the pump,} \\
 q_{dd} &= h_{d1} - h_{d4s} && \text{specific work supplied to the cycle.}
 \end{aligned}$$

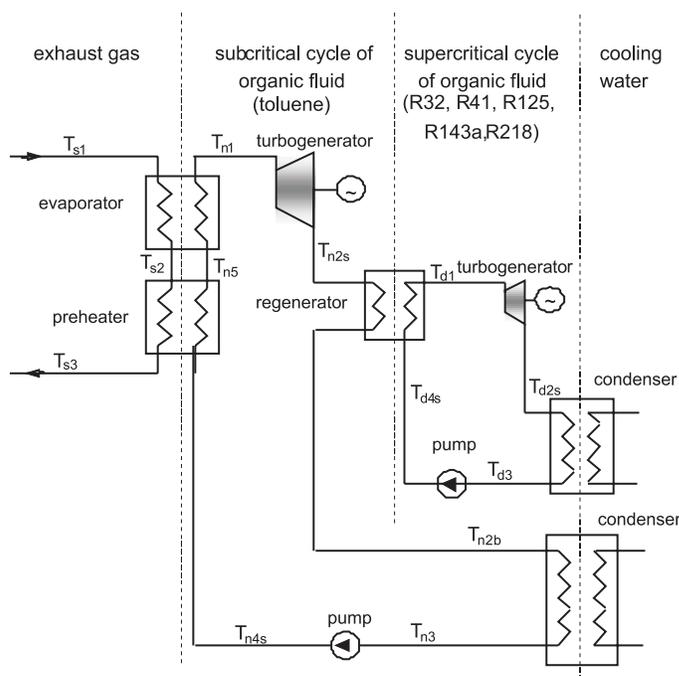


Figure 6. Scheme of the dual cycle ORC power plant with external heat recuperation (ZR version).

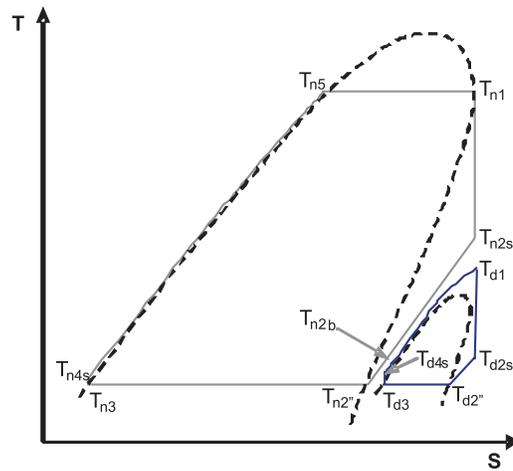


Figure 7. Thermodynamic conversion cycles for the ORC power plant with external heat recuperation.

The working fluid mass flow rate  $\dot{m}_d$  was determined from the evaporator energy balance equation

$$\dot{m}_d = \frac{\dot{Q}_d}{h_{d1} - h_{d4s}}, \quad (14)$$

where:

$$\dot{Q}_d = \dot{m}_n (h_{n2s} - h_{n2b}). \quad (15)$$

The calculated values of the supercritical cycle output power for all the selected working fluids are presented in Fig. 8 as function of the vapour pressure at the turbine inlet. The supercritical cycles were all powered by heat of the turbine outlet vapour of the main ORC cycle, with the vapour parameters of  $T_{n2s} = 85.6 \text{ }^\circ\text{C}$  and  $\dot{m}_d = 3.12 \text{ kg/s}$  being the outlet vapour parameters for the toluene evaporation temperature of  $209 \text{ }^\circ\text{C}$  in the main ORC power cycle (point B in Fig. 5).

As it can be seen from the calculation results presented in Fig. 8 the additional power attainable in the dual ORC power plant that incorporates an additional, supercritical power cycle can reach, in the best case, approximately half of the power surplus value that is achievable in the single cycle ORC power plant with internal heat recuperation.

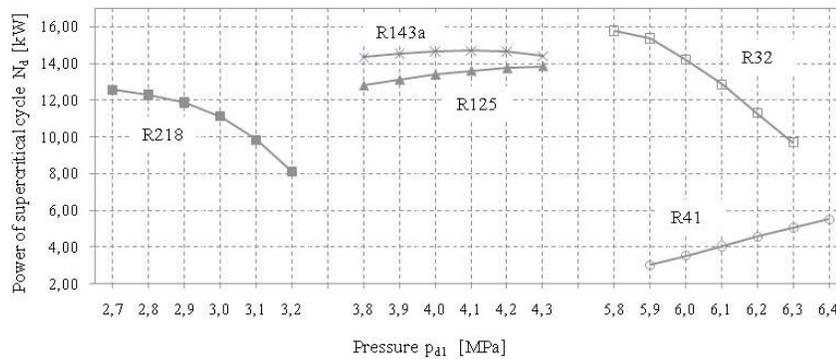


Figure 8. Output power of the supercritical power cycles for various working fluids as function of the turbine inlet vapour pressure.

## 5 Final remarks

Influence of the working fluid evaporation temperature on the thermal efficiency and output power of the ORC power plant was analysed. Simulation of the possibility to utilize heat carried by the turbine outlet vapour was made for two heat recuperation solutions. Internal heat recuperation in the main power plant cycle and external heat recuperation by means of addition of a low temperature, supercritical cycle were analysed. It resulted from the analysis that when toluene was used as the working fluid in the main power plant cycle then its turbine outlet vapour temperature depended on the adopted toluene evaporation temperature and can reach value as high as 116 °C, at  $T_{n1} = 300$  °C.

The calculations yielded the conclusion that the best effects in terms of reaching the maximum output power of the plant were achieved by selection of the proper evaporation temperature of the working fluid. By decreasing the toluene evaporation temperature from 300 °C to 209 °C the plant output power can be raised to the value that is more than 2.5 times greater (from 195.5 kW to 523.2 kW). Application of the internal heat recuperation in the plant cycle at its optimum design (for  $T_{n1} = 209$  °C) may bring additional 24.9 kW of the plant output power. On the other hand, application of an additional, supercritical power cycle appeared to be less effective and contributed only 13 kW of the extra output power. It should be also noted that the external heat recuperation by means of adding a supercritical power cycle results in the highest investment cost.

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