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Exergy analysis of internal regeneration in supercritical cycles of ORC power plant

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Abstract In the paper presented is an idea of organic Rankine cycle (ORC) operating with supercritical parameters and so called dry fluids. Discussed is one of the methods of improving the effectiveness of operation of supercritical cycle by application of internal regeneration of heat through the use of additional heat exchanger. The main objective of internal regenerator is to recover heat from the vapour leaving the turbine and its transfer to the liquid phase of working fluid after the circulation pump. In effect of application of the regenerative heat exchanger it is possible to obtain improved effectiveness of operation of the power plant, however, only in the case when the ORC plant is supplied from the so called sealed heat source. In the present paper presented is the discussion of heat sources and on the base of the case study of two heat sources, namely the rate of heat of thermal oil from the boiler and the rate of heat of hot air from the cooler of the clinkier from the cement production line having the same initial temperature of 260 °C, presented is the influence of the heat source on the justification of application of internal regeneration. In the paper presented are the calculations for the supercritical ORC power plant with R365mfc as a working fluid, accomplished has been exergy changes and exergy efficiency analysis with the view to select the most appropriate parameters of operation of the power plant for given parameters of the heat source.

Keywords: ORC, Internal heat regeneration; Supercritical power plant; Exergy analysis

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1 Introduction

At present we can observe an intense development of the organic Rankine cycle (ORC) technology. Firstly, in the eighties of the past century such solutions were primarily considered in geothermal applications [1]. Nowadays of more significant importance is implementation of ORC installation in utilization of biomass energy or management of waste or post-process energy [2–4]. Different characteristics of sources render different solutions of ORC technology and one of the directions for its development is application of supercritical cycles.

2 ORC power plant with supercritical parameters and internal regeneration of heat

In ORC installation there can be used different working fluids. Some of them feature not necessarily high critical parameters, see Tab. 1, which enables construction of the power plant with supercritical parameters even at the supply of heat from the sources of low and medium temperature.

Table 1. Critical parameters for a selection of working fluids possible for application in ORC.

Working fluid	T_{cr} [°C]	p_{cr} [MPa]	Remarks
R227ea	101.7	2.926	Synthetic, fluorocarbon
Izobutane	134.7	3.640	Natural
R365mfc	186.9	3.266	Synthetic, fluorocarbon
MDM	290.9	1.415	Synthetic, silicon derivative

Due to the shape of saturation curves the working fluids can be divided into two groups, namely the wet fluids with the shape of saturated vapour line, $x = 1$, such as for water, or the so called dry fluids, where the shape is presented in Fig. 1. Use of the dry fluid enables for application of additional heat exchanger permitting for heat recovery from the exhaust vapour leaving the turbine and partial heating of liquid phase of working fluid. A schematic of ORC installation with the use of regenerator is presented in Fig. 2.

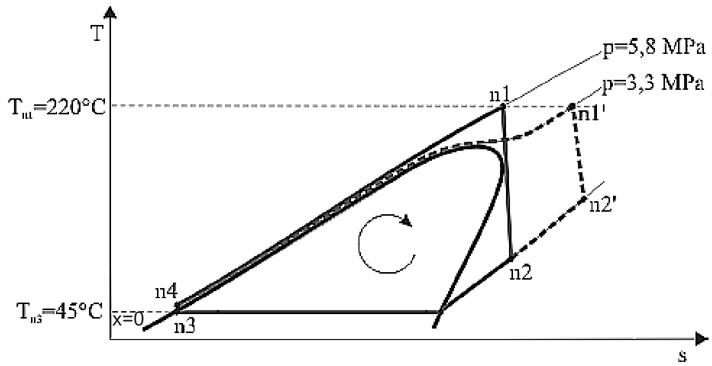


Figure 1. Shape of saturation curves in case of the R365mfc fluid with denoted two supercritical cycles for a minimum pressure (n1-n2-n3-n4) and maximum pressure (n1'-n2'-n3-n4).

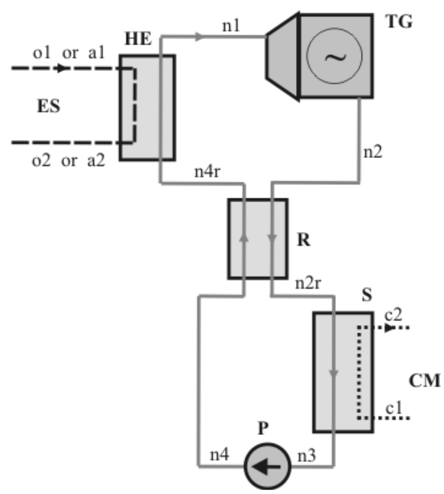


Figure 2. ORC power plant with internal regeneration (CM – cooling medium, ES – energy source, HE – supercritical heat exchanger, P – pump, R – regenerator, TG – turbogenerator).

3 Heat sources and their characteristics

As was mentioned in the introduction the most commonly used heat sources for driving ORC installations are biomass, geothermal energy and different types of waste and post-process heat. In relation to the type of energy source there are different heat carriers used in practice such as water, exhaust gases, thermal oils, air. Such heat sources have different characteristics, however from the point of view of thermal parameters they have been divided into two principal groups:

- Open heat sources, where the energy carrying fluid with initial temperature T_{s1} and the flow rate $\dot{m}_p$ supplies thermal energy to ORC. In such case the final temperature of heat source T_{s3} is a resultant value, and is not influencing further operation of installation.
- Sealed heat sources – where energy carrier supplies thermal energy to ORC plant and initial and final temperatures of the heat source T_{s1} i T_{s3} are predefined. In such case final temperature T_{s3} has a very significant impact on the effectiveness of operation of the heat source.

Generalising presented above examples we can conclude that in the case of the sealed heat source the heat potential $\dot{Q}_{source}$ is equal to the rate of energy supplied to the power plant $\dot{Q}_{\text{sup}}$ whereas in the case of open heat source the rate of energy supplied to the power plant $\dot{Q}_{\text{sup}}$ is usually smaller that the available one $\dot{Q}_{source}$ and only in the limiting case there can be a situation where the rate of available heat is delivered in total to the power plant installation.

In the frame of the present work considered is the influence of the type of heat source on optimal operation parameters of supercritical ORC power plant with internal regeneration. An example of the sealed heat source supplying the ORC installation is the rate of heat of thermal oil heated in the boiler due to fuel combustion. In the present study available heat transfer surface in the boiler enables for increase of oil temperature by 50 K and it has been assumed that the boiler capacity is 100 kW. An example of the open system is supplying of the ORC power plant with waste energy, namely air from the cement clinker cooling, which is emitted to atmosphere after passing the supercritical heat exchanger in ORC installation. In Tab. 2 presented are the characteristic quantities describing both considered heat sources, whereas in Fig. 3 presented is a sequence of thermodynamic processes of considered supercritical power plant for the vapour pressure $p_{n1} = 4.8$ MPa, where internal regeneration is also denoted.

It ought to be stressed that the primary assumption in the analysis is the postulation of the same value of the heat resource $\dot{Q}_{source}$ as well as the same initial temperature of energy carriers T_{o1} and T_{a1}. Value of the mass flowrate of air, being an example of the open heat source, has been

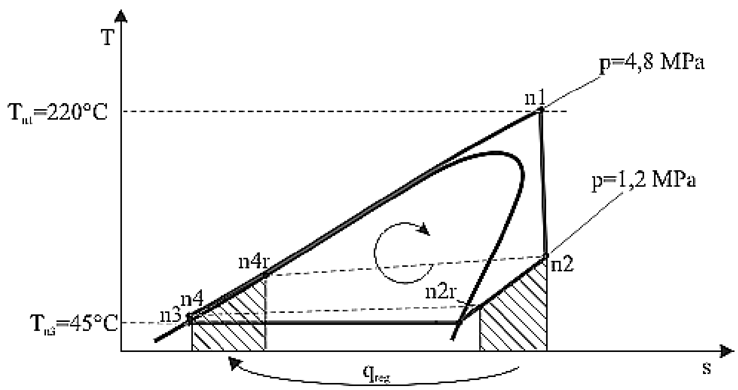


Figure 3. Supercritical ORC power plant with internal regeneration for the vapour pressure $p_{n1} = 4.8$ MPa.

Table 2. Characteristics of heat sources.

Energy source	Sealed heat source	Open heat source
	Rate of heat of thermal oil heated in boiler	Rate of heat of hot air from the clinkier cooler
Initial temperature	$T_{o1} = 260$ °C	$T_{a1} = 260$ °C
Heat capacity	$\dot{Q}_{source} = 100$ kW	$\dot{Q}_{source} = 100$ kW
Rate of supplied energy	$\dot{Q}_{\rm sup} = 100$ kW	Dependent on calculation case
Mass flow rate of energy carrier	Dependent on calculation case	$\dot{m}_a = 0.42$ kg/s
Final temperature	$T_{o1} = 210$ °C	T_{a2} dependent on calculation case

calculated from relation

$$\dot{m}_a = \frac{\dot{Q}_{source}}{h_{a1} - h_{a,ot}} , \tag{1}$$

where $h_{a,ot}$ is the air enthalpy in reference temperature in order to enable comparative analysis for all cases of calculations and h_{a1} is the inlet enthalpy of source.

4 Methodology of calculations

Thermal-hydraulic calculations have been accomplished for the case of the power plant with supercritical parameters and synthetic fluid R365mfc as working fluid. The installation was either supplied from the sealed or closed heat source. Energetical and exergetical analysis was carried out for the case of ORC installation and the characteristic quantities, on the basis of which comparisons were made, have been calculated from the following expressions:

- ORC power

$$N_{ORC} = N_T - N_P \ , \tag{2}$$

- thermal efficiency η_I

$$\eta_I^{ORC} = \frac{N_T - N_P}{\dot{Q}_{\text{sup}}} \ , \tag{3}$$

- exergetical efficiency η_{II}

$$\eta_{II}^{ORC} = \frac{N_T - N_P}{\dot{B}_{source}} \ , \tag{4}$$

where N_T, N_P are the turbine and pump power respectively, and $\dot{B}_{source}$ is the source exergy. Relevant relations have been presented in Tab. 3.

In order to calculate values of efficiencies and power it is necessary to:

- determine the value of specific enthalpies in characteristic points of ORC system (both for the case with and without regeneration),
- determine values of specific enthalpy for heat carries, namely air in case of open system and thermal oil for the system supplied from the open system.

In order to achieve that the RefProp9.0 [5] database of fluid thermodynamic and transport properties was used to determine respective enthalpies for the turbine inlet temperature of vapour T_{n1} = 220 °C and pressures p_{n1} ranging from 3.3 MPa (near-supercritical) to 6.8 MPa (maximum one enabling for vapour expansion in turbine to the superheated vapour), as well as condensation temperature of working fluid at 45 °C.

Subsequent step in the algorithm of calculations is determination of the flow rate of working fluid in the cycle. In the case of the sealed source (case of use of thermal oil), the mass flow rate of working fluid in ORC installation is calculated from the expression:

Table 3. Relations used in energetical and exergetical analysis.

Element of system	Energy analysis	Exergy analysis
Turbine	$N_T = \dot{m}_n\,(h_{n1} - h_{n2s})\,\eta_T$ (5)	$\dot{I}_T = \dot{B}_{n1} - \dot{B}_{n2} - N_T$ (6)
Condenser	$\dot{Q}_C = \dot{m}_n\,(h_{n2} - h_{n3})$ (7)	$\dot{I}_C = \dot{B}_{n2} - \dot{B}_{n3}$ (8)
Regenerator	$\dot{Q}_{REG} = \dot{m}_n\,(h_{n2} - h_{n2r}) = \dot{m}_n\,(h_{n4r} - h_{n4})$ (9)	$\dot{I}_{REG} = \dot{B}_{n4} + \dot{B}_{n2} - \dot{B}_{n4r} + \dot{B}_{n2r}$ (10)
Pump	$N_P = \dot{m}_n\,(h_{n4s} - h_{n3})\,/\eta_P$ (11)	$\dot{I}_P = N_P - \dot{B}_{n4} + \dot{B}_{n3}$ (12)
Supercritical heat exchanger	system without regeneration $\dot{Q}_{HE} = \dot{m}_n\,(h_{n1} - h_{n4})$ (13) system with regeneration $\dot{Q}_{HE} = \dot{m}_n\,(h_{n1} - h_{n4r})$ (14)	system without regeneration *) $\dot{I}_{HE} = \dot{B}_{n4} + \dot{B}_{o1} - \dot{B}_{o2} - \dot{B}_{n1}$ (15) system with regeneration *) $\dot{I}_{HE} = \dot{B}_{n4r} + \dot{B}_{o1} - \dot{B}_{o2} - \dot{B}_{n1}$ (16)
External loss exergy due to unused waste product	—	concerning only open source $\dot{I}_A = \dot{B}_{a2}$ (17)

*) in case of air as heat carrier there ought to be used index ‘a’ instead of index ‘o’

- for the system without internal regeneration

$$\dot{m}_n = \frac{\dot{Q}_{\text{sup}}}{h_{n1} - h_{n4}}\,, \tag{18}$$

- for the system with internal regeneration

$$\dot{m}_n^r = \frac{\dot{Q}_{\text{sup}}}{h_{n1} - h_{n4r}}\,, \tag{19}$$

where in both cases the rate of supplied to the system heat is $\dot{Q}_{sup} = 100$ kW.

In the case of the open system (hot stream of air) in the case without regeneration the final temperature T_{a2} can be determined from the equation

$$T_{a2} = T_{n4} + \Delta T_{pp} \ , \tag{20}$$

where the enthalpy of air for temperature T_{a2} (at pressure $p_a = 0.2$ MPa) being determined from [5] which permits to determine the rate of energy supplied to the power plant Q_{sup} from the relation

$$\dot{Q}_{\text{sup}} = \dot{m}_a(h_{a1} - h_{a2}) \ . \tag{21}$$

Calculated from (21) value of the rate of heat supplied to the cycle permits to determine the flow rate of working fluid in the system from expression (18).

In case of the power plant supplied from the open heat source with application of internal regeneration there has been assumed the same mass flow rate of working fluid as in the case without regeneration. It has been assumed in calculations that the internal efficiency of turbine $\eta_t = 0.8$ and that of the pump $\eta_p = 0.7$ whereas the pinch point temperature difference is $\Delta T_{pp} = 15$ K. In the energy analysis particular changes of exergy have been determined from relations (6), (7), (10), (12), (14), (16) and (17) where

$$\dot{B}_{i,j} = \dot{m}_i b_{i,j} \ , \tag{22}$$

whereas values of specific exergy for air and R365mfc were calculated from the expression

$$b = h - h_{ot} - T_{ot}\,(s - s_{ot}) \tag{23}$$

with the use of the Refprop 9.0 software [5]. Specific exergy of thermal oil is calculated from Eq. (24), where c_{po} is the specific heat of thermal oil

$$b_{o,i} = c_{po,i}\left(T_{o,i} - T_{ot} - T_{ot}\ln\frac{T_{o,i}}{T_{ot}}\right) \ . \tag{24}$$

The reference level for exergy calculations has been assumed as $T_{ot} = 25$ °C, $p_{ot} = 0.1$ MPa.

The rate of source exergy, present in expression (4) has been calculated from the following relations:

$$\dot{B}_{source} = \dot{B}_{\text{sup}} = \dot{B}_{o1} - \dot{B}_{o2} \tag{25}$$

in case of the system supplied from the sealed source, and

$$\dot{B}_{source} = \dot{B}_{a1} \tag{26}$$

in case of the system supplied from the open source.

Attention ought also be focused on expression (17) which describes the external exergy loss related to not fully utilizing of the useful product. In case of supplying the ORC installation from the sealed heat source, namely with the hot thermal oil, such loss in the boiler is not present (there are other boiler losses, but these are not in the focus of the present work). On the other hand, in case of supplying the ORC from the open heat source such as with the hot air stream, value of that loss is dependent on the degree of temperature reduction of air in the supercritical heat exchanger. If the parameters of surroundings are assumed at the level of $T_{ot} = 25$ °C and pressure is $p_{ot} = 0.1$ MPa then emission of air from the installation having parameters greater than surroundings necessitates consideration of the external exergy loss $\dot{I}_A$.

5 Results of calculations

In the first instance presented have been selected parameters of states (Tab. 4) as well as values of specific exergy of working fluid (Tab. 5) in characteristic points of the cycle for different vapour pressures at turbine inlet. Mentioned above quantities does not depend on the kind of the heat source, but are a function of the power plant type (i.e. whether it is with or without regeneration) and in case with internal regeneration there appear two additional states resulting from application of the regenerative heat exchanger.

In Tab. 6 presented are the results of calculations of power and efficiency of the power plant together with the rates of regenerated heat and mass flow rates of working fluid R365mfc in case of ORC plant with and without regeneration and the heat supply in the form of the thermal oil (sealed source). As results from the analysis of data presented in Tab. 6 application of regenerative heat exchanger causes the increase of the efficiency and power of the plant. For different considered values of vapour pressure at turbine inlet the maximum power and efficiency was obtained for the pressure $p_{n1} = 4.8$ MPa in case of the system without regeneration and pressure $p_{n1} = 4.3$ MPa in case of a system with regeneration.

Table 4. Selected state parameters of working fluid in characteristic nodes of the cycle in relation to the live vapour pressure

T_{n1} [°C]	p_{n1} [MPa]	h_{n1} [kJ/kg]	$s_{n1} = s_{n2}$ [kJ/kgK]	h_{n2} [kJ/kg]	h_{n3} [kJ/kg]	h_{n4} [J/kg]	h_{n2r} [kJ/kg]	h_{n4r} [kJ/kg]
220	3.3	620.73	2.070	562.37	261.81	265.52	454.92	372.97
220	3.8	612.41	2.048	553.63	261.81	266.11	455.22	364.52
220	4.3	602.27	2.023	545.81	261.81	266.70	455.53	354.98
220	4.8	589.49	1.994	532.29	261.81	267.28	455.84	343.73
220	5.3	575.22	1.963	520.0	261.81	267.87	456.14	331.73
220	5.8	565.01	1.940	511.28	261.81	268.45	456.45	323.28
220	6.3	559.01	1.927	506.08	261.81	269.04	456.76	318.36

Table 5. Values of specific exergy of R365mfc in case with and without regeneration.

T_{n1} [°C]	p_{n1} [MPa]	b_{n1} [kJ/kgK]	b_{n2} [kJ/kgK]	b_{n3} [kJ/kgK]	b_{n4} [kJ/kgK]	b_{n2r} [kJ/kgK]	b_{n4r} [kJ/kgK]
220	3.3	103.18	34.29	0.92	3.60	13.13	20.91
220	3.8	101.39	31.81	0.92	4.03	13.16	19.30
220	4.3	98.56	29.17	0.92	4.45	13.18	17.55
220	4.8	94.43	26.38	0.92	4.87	13.21	15.56
220	5.3	89.50	23.51	0.92	52.9	13.23	13.62
220	5.8	86.05	21.51	0.92	5.71	13.26	12.49
220	6.3	84.18	20.46	0.92	6.14	13.29	12.04

In Tab. 7 presented are values of the plant power, rates of supplied and recovered heat as well as the mass flow rates of working fluid R365mfc in case of the power plant with and without regeneration at the heat supply using hot air (open source). Analysis of the results of calculations presented in Tab. 7 indicates that application of internal regeneration in case of supplying ORC plant from the open source (in relation to the case of the plant without regeneration) has the following implications:

- no increase of the plant power as the total rate of heat supplied to the plant with internal regeneration (from the stream of air and without regeneration) is equal to the rate of heat supplied from air to the plant without regenerator;

Table 6. Selected quantities characterising effectiveness of operations of supercritical power plant supplied from the sealed heat source.

		Power plant supplied from the sealed heat source										
		without regeneration					with regeneration					
T_{n1} [°C]	p_{n1} [MPa]	m_o [kg/s]	m_n [kg/s]	N_{C-R} [kW]	η_I [%]	η_{II} [%]	m_o [kg/s]	m_n [kg/s]	Q_{reg} [kW]	N_{C-R} [kW]	η_I [%]	η_{II} [%]
220	3.3	0.55	0.28	15.4	15.4	41.8	0.55	0.40	43.4	22.1	22.1	60.0
220	3.8	0.55	0.29	15.7	15.7	42.8	0.55	0.40	39.7	22.0	22.0	59.7
220	4.3	0.55	0.30	16.0	16.0	43.4	0.55	0.40	35.7	21.7	21.7	58.9
220	4.8	0.55	0.31	16.1	16.1	43.6	0.55	0.41	31.1	21.0	21.0	57.2
220	5.3	0.55	0.33	16.0	16.0	43.5	0.55	0.41	26.2	20.2	20.2	54.9
220	5.8	0.55	0.34	15.9	15.9	43.2	0.55	0.41	22.7	19.5	19.5	52.9
220	6.3	0.55	0.34	15.8	15.8	42.8	0.55	0.42	20.5	19.0	19.0	51.6

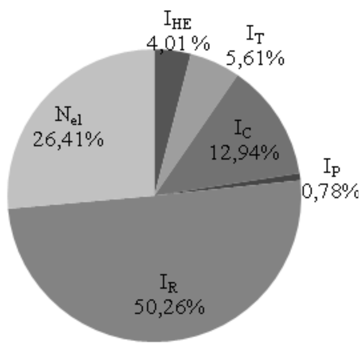
Table 7. Selected quantities characterising effectiveness of operations of supercritical power plant supplied from the open heat source.

	Power plant supplied from the open heat source												
	without regeneration						with regeneration						
p_{n1} [MPa]	m_n [kg/s]	Q_{sup} [kW]	N_{C-R} [kW]	η_I [%]	η_{II} [%]	T_{a2} [°C]	m_n [kg/s]	Q_{sup} [kW]	Q_{reg} [kW]	N_{C-R} [kW]	η_I [%]	η_{II} [%]	T_{a2} [°C]
3.3	0.24	84.6	13.0	15.4	25.4	61.8	0.24	59.0	25.6	13.0	22.0	25.4	12.23
3.8	0.24	84.4	13.3	15.7	26.0	62.1	0.24	60.4	24.0	13.3	22.0	26.0	11.89
4.3	0.25	84.3	13.4	16.0	26.3	62.4	0.25	62.1	22.2	13.4	21.7	26.3	11.49
4.8	0.26	84.2	13.5	16.1	26.4	62.7	0.26	64.2	20.0	13.5	21.0	26.4	110.0
5.3	0.27	84.1	13.4	16.0	26.3	63.0	0.27	66.6	17.5	13.4	20.2	26.3	10.43
5.8	0.28	83.9	13.3	15.9	26.0	63.3	0.28	68.4	15.5	13.3	19.5	26.0	100.0
6.3	0.29	83.8	13.2	15.8	25.8	63.6	0.29	69.6	14.3	13.2	19.0	25.8	97.3

- increase of thermal efficiency due to the fact that the power of the plant remains the same whereas the rate of heat supplied to the installation from the external source is reduced;
- increase of final temperature of energy carrier fluid;
- decrease of the condenser heat transfer surface, as the rate of heat removed from the condenser is decreased, however we must remember

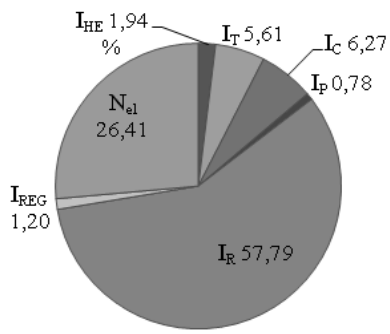
that in the system there is a necessity for additional heat exchanger where regeneration process takes place.

It ought also to be stressed that in the case of use of open heat sources the exergy analysis should be employed, as the use of merely of energy analysis does not result in a general description of the effectiveness of energy technology. An example of such analysis has been presented in Fig. 4, which contains the split of exergy rates in the ORC power plant supplied with the hot air stream and the cases with and without regeneration.



(a)

$p_{n1} = 4.8$ MPa, $\eta_I = 16.1\%$, $\eta_{II} = 26.4\%$



(b)

$p_{n1} = 4.8$ MPa, $\eta_I = 21.0\%$, $\eta_{II} = 26.4\%$

Figure 4. Split of exergy rates in the supercritical ORC power plant (a) without regeneration, (b) with regeneration.

As can be seen from the analysis of exergy losses on particular elements of the power plant presented in Fig. 4 application of the regenerator reduces exergy losses in the supercritical heat exchanger $\dot{I}_{HE}$ as well as in the condenser of working fluid $\dot{I}_C$. On the other hand, however, it renders the increase of exergy losses related to not using of the waste product $\dot{I}_R$ in the form of the flow rate of heat with temperature T_{p2}. In both cases obtained is the same power of the plant despite the fact that the case presented in Fig. 4b the value of thermal efficiency is higher by 6.1%.

6 Conclusions

Application of internal regeneration in ORC power plants supplied from the sealed heat sources provides measurable effects in the form of increase of power plant power and efficiency. In the case when the heat source is of the so called open type the application of internal regeneration does not produce the increase of the plant power and should only be considered when the elevated temperature of heat carrier fluid (in comparison to the temperature of that fluid in case of without internal regeneration) can be further utilized after performing its duties in the ORC installation.

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