Increase of power and efficiency of the 900 MW supercritical power plant through incorporation of the ORC

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Abstract   The objective of the paper is to analyse thermodynamical and operational parameters of the supercritical power plant with reference conditions as well as following the introduction of the hybrid system incorporating ORC. In ORC the upper heat source is a stream of hot water from the system of heat recovery having temperature of 90 °C, which is additionally aided by heat from the bleeds of the steam turbine. Thermodynamical analysis of the supercritical plant with and without incorporation of ORC was accomplished using computational flow mechanics numerical codes. Investigated were six working fluids such as propane, isobutane, pentane, ethanol, R236ea and R245fa. In the course of calculations determined were primarily the increase of the unit power and efficiency for the reference case and that with the ORC.

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Nomenclature

h	–	specific enthalpy, kJ/kg
l	–	specific work, kJ/kg
N	–	power, kW
$\dot{m}$	–	mass flux, kg/s
p	–	pressure, Pa
q	–	specific heat, kJ/kg
$\dot{Q}$	–	rate of heat, heat energy flux, kW
$\dot{Q}_{chem}$	–	chemical energy flux, kW
t	–	temperature, °C
W_d	–	fuel calorific value, kJ/kg
x	–	vapour quality

Greek symbols

ΔT	–	temperature difference in the heat exchanger, K
η	–	efficiency, %
Π	–	compression/expansion ratio
ζ	–	flow losses

Subscripts

B	–	boiler
c	–	cold
con	–	condensation
cyc	–	cycle
D	–	deaerator
el	–	electrical
f	–	fuel
g	–	generator
Gr	–	gross
h	–	hot
HE	–	heat exchanger
HP	–	high pressure
i	–	internal
in	–	inlet
IP	–	intermediate pressure
$loss$	–	loss
LP	–	low pressure
m	–	mechanical, mixture
ORC	–	organic Rankine cycle
out	–	outlet
OUT	–	outlet from cycle
P	–	pump

pip	–	pipelines
re	–	reference efficiency
reg	–	regeneration
RU	–	reference unit
t	–	technical
T	–	turbine
w	–	water
1s,2s,...	–	isentropic points of process
1,2,...	–	real points of process

Superscripts

ext	–	extraction, bleed
w	–	rate of heat supplied by waste heat water
s	–	rate of heat supplied by the bleed steam
ORC	–	rate of heat supplied to ORC in heat exchanger HE1

1   Introduction

Both in the worldwide as well as Polish power engineering there is observed an increase of the public awareness and tendency to the sustainable development. In the case of the power sector that is related to production of electricity at the highest possible efficiency at minimum influence on the natural environment [1]. That issue has also been considered by the European Union in the Directive No. 2010/75/UE [7]. To attain the objectives outlined there it is necessary to build highly efficient power plants such as for example a new unit operating with supercritical parameters in Bełchatów Power Plant. The thermal cycle of that unit, featuring the capacity of 858 MWe, exhibits a high efficiency of conversion of chemical energy into electricity as the net efficiency of electricity production is exceeding 42% [22]. The carbon capture and storage (CCS) technologies are expected to reduce the perspective emissions of CO_2 in the units presently operated and those under-construction [2]. Therefore the large power perspective units operating with supercritical parameters could probably solve such arising problems as supplies of adequate amount of electricity as well as obeying the demanding standards of continuous reduction of carbon dioxide (CO_2) quotas emitted to atmosphere. On the other hand, however, there are numerous problems at operation of such large units and hence they should be scheduled for the base load operation by the National Grid [14], in order to sustain a high efficiency. In such case a proper supplement to the generation mix are clean gas technologies, which enable construction of smaller units with a similar efficiency as the coal-fired units *50 plus* [14,17].

Despite undoubtful advantages of the units with supercritical parameters there is a need for continuous search of new ways of increasing the efficiency. That is possible through the increase of the live steam parameters, use of waste heat, modification of the systems and replacing of low-efficiency subsystems with more modern ones [1,6,18–20,23,25]. The development of the supercritical units was accompanied by the progress in designing of elements and configuring the whole systems. Introduced were new materials with higher mechanical properties [1,5,20,23]. That allowed to increase the thermodynamic parameters of steam and contributed to reduction of the cycle losses, which overall transferred to the increase of units effectiveness by 7–8 pp [5].

A perspective design for increase of efficiency, utilizes low boiling point fluids in the installation cooperating with the supercritical power plant. As a result of such cooperation the organic Rankine cycle (ORC) can utilize the available waste heat [11,18,24]. Attracting attention is also implementation of the binary turbine, realizing the steam cycle in the high pressure and medium pressure parts and the ORC instead of the low pressure (LP) part. Heat transfer occurs in a cascade heat exchanger, which would be on one side the steam condenser and the generator of vapour of the low-boiling point fluid. Due to small specific volumes of the low boiling point fluid in comparison to steam it is possible to replace the large and expensive LP part of the turbine with a small ORC turbine. Following introduction of the low boiling point fluid as a working fluid it became possibile to significantly reduce the flow rate in the LP part of the turbine and hence to reduce the outlet area of the turbine as well as investment costs. Other advantages are smaller amount of materials used and labour as well as reduction of erosive action of the working fluid on the blading system [4,21].

There should also be remembered the key issue related to application of low-boiling point fluids. That is the vapour generation issue. The vapour generator should ensure preservation of a specified amount of vapour phase in the flow, as the complete evaporation of working fluid in consequence leads to the reduction of the mean heat transfer coefficient. In such case, due to existence of wet vapour in the flow it is necessary to install the vapour separator. After separation the liquid is directed to the evaporator, whereas vapour goes to the turbine to perform the ORC work. Sustaining of stable boiling conditions is related not only with the supply of adequate rate of heat but also requires sustaining a constant pressure distribution without any pulsation and variations [10]. Additionally it is important to

select possibly high boiling temperature, which is next the determinant of thermodynamic efficiency and the cost-effectiveness of vapour generator construction [11].

The objective of the present paper is analysis of operational and thermodynamic parameters of the 900 MWe supercritical power plant using available computational flow mechanics (CFM) codes for the reference case without ORC and with consideration of the latter. The CFM codes are scheduled for the solution of integral conservation equations of mass, momentum and energy [1,12,13,15–17]. The objective of the study is to utilize available capacity of 200 MWt of waste heat flux in the form of hot water with temperature of 90 ℃ by incorporation of ORC into the system. Considered were six potential working fluids, namely propane, isobutane, pentane, ethanol, R236ea and R245fa with respect to obtain the highest output and efficiency of the cycle.

2   Calculation model

The calculation model for the hybrid system of a supercritical plant with and without incorporation of ORC cycle is presented. Schematic of the concept of heat supply to ORC installation with the use of water with temperature 90 ℃ as well as use of LP extraction of steam is shown in Fig. 1. Next, there were defined formulas describing power and efficiency used in the analysis of the supercritical unit. In the further part of the text presented is a numerical model of the plant. At the end presented are parameters introduced for comparison of the effects of cooperation of the supercritical plant with the ORC.

2.1   Mathematical model of particular elements of the cycle

The thermal cycle of the supercritical plant as well as of ORC consists of vapour generator (heat exchanger), steam turbine, condenser and a circulation pump. In this paragraph, computational procedures for each component of the hybrid cycle are presented. Considered are also internal irreversibilities in the turbine and pump. Based on definition of internal irreversibility of steam turbine stage, η_{iT}, (also applicable for ORC) the power produced is determined. In Fig. 2 presented is enthalpy-entropy h-s diagram for the polytropic process taking place in the turbine. It is required to provide as input data for determination of electric power such parameters as generator efficiency, η_g, internal efficiency η_i, mechanical efficiency,
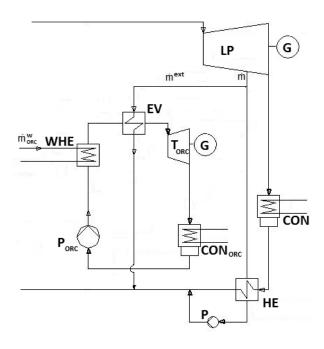
Figure 1. Schematic of the concept of heat supply to ORC installation with the use of water with temperature 90 °C as well as the use of LP extraction of steam: LP – low pressure part of steam turbine, HE – regenerative heat exchangers, P – pump of water, G – generator, CON – condenser of water steam, EV – evaporator, WHE - waste heat exchangers with use water with temperature 90 °C, T_{ORC} – turbine of organic fluid, P_{ORC} – pump of organic fluid, CON_{ORC} – condenser of organic fluid vapour [9].

η_m, and expansion ratio, Π. The steam features a known temperature at turbine inlet T, and the live steam mass flow rate $\dot{m}_s$.

As mentioned earlier the expansion process from state 1 to 2 (Fig. 2) in the steam turbine stage can be characterized by the definition of internal turbine efficiency according to relation [19]

$$\eta_{iT} = \frac{h_1 - h_2}{h_1 - h_{2_s}} \ , \tag{1}$$

where: h_1, h_2, h_{2_s} – fluid enthalpy in states (1), (2), and (2_s), Fig. 2.

The electric power, generated by the stage, N_{elT}, is determined from relation [19]:

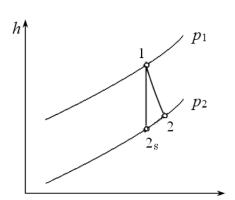$$N_{elT} = \eta_g \eta_{mT} \dot{m}_s (h_1 - h_2) \ , \tag{2}$$

Figure 2. Isentropic expansion 1-2$_s$ in relation to the real expansion 1-2 from pressure p_1
to p_2.

where: η_{mT} – mechanical efficiency of the turbine stage, η_g – generator efficiency.

The electric power consumed by the pump, N_{elP}, for increasing the working fluid pressure is determined using the pump internal efficiency η_{iP}, mechanical efficiency, η_{mP}, efficiency of conversion of electric energy into mechanical one, η_{elm} and pressure p_2, which must be ensured by the pump. The liquid at the inlet to the pump has a given temperature T_1, pressure p_1 and mass flow rate $\dot{m}_w$. In Fig. 3 presented is the enthalpy-entropy diagram h-s for liquid pumping.
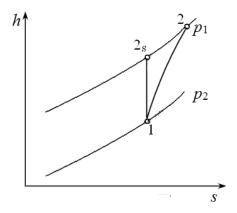

Figure 3. Isentropic pumping 1-2$_s$ and real pumping 1-2 from pressure p_1 to p_2.

The process of pressure increase (Fig. 3) in a pump from p_1 to p_2 is characterised by the pump internal efficiency, defined as [19]

$$\eta_{iP} = \frac{h_{2_s} - h_1}{h_2 - h_1} \,, \tag{3}$$

where: h_1, h_2, h_{2_s} – enthalpy of fluid determining the states (1), (2), and (2_s).

The consumed electric power by the pump N_{elP} is determined from relation [19]

$$N_{elP} = \frac{\dot{m}_w \left(h_2 - h_1\right)}{\eta_{elm}\eta_{mP}} \,. \tag{4}$$

The balance of rates of heat transferred in the heat exchanger (including the deaerator) can be written in the form

$$\dot{Q}_{in} = \dot{Q}_{loss} + \dot{Q}_{out} \,, \tag{5}$$

where: $\dot{Q}_{in}$ – rate of heat supplied to the exchanger, $\dot{Q}_{out}$ – rate of heat received from the exchanger, $\dot{Q}_{loss}$ – rate of heat lost to surroundings.
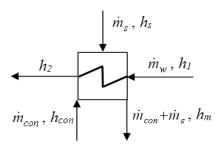

Figure 4. Schematic of the open-feedwater heat exchanger.

Equation (5) in case of the open-feedwater heat exchanger (Fig. 4) can be written as

$$\dot{m}_s h_s + \dot{m}_{con} h_{con} + \dot{m}_w h_1 = \dot{Q}_{loss} + (\dot{m}_{con} + \dot{m}_s)h_m + \dot{m}_w h_2 \,, \tag{6}$$

where: $\dot{m}_{con}$ – mass flow rate of condensate, h_s – steam enthalpy, h_{con} – condensate enthalpy, h_m – enthalpy of the condensate-steam mixture, h_1 – water enthalpy at inlet to heat exchanger, h_2 – water enthalpy at outlet

from heat exchanger. Therefore the efficiency of the open feedwater heat exchanger is

$$\eta_{HEm} = \frac{\dot{Q}_{out}}{\dot{Q}_{in}} = \frac{\dot{m}_w(h_2 - h_1)}{\dot{m}_s(h_s - h_m) + \dot{m}_{con}(h_{con} - h_m)} \ . \tag{7}$$

In case of the closed feedwater heat exchanger, including the condenser, boiler and ORC heat exchangers (Fig. 5.) Eq. (5) can be written as

$$\dot{m}_h h_{1h} + \dot{m}_c h_{1c} = \dot{Q}_{loss} + \dot{m}_h h_{2h} + \dot{m}_c h_{2c} \ , \tag{8}$$

where: $\dot{m}_c$ – mass flow rate of cold fluid, $\dot{m}_h$ – mass flow rate of hot fluid, h_{1h} – enthalpy of hot fluid at inlet to exchanger, h_{1c} – enthalpy of cold fluid at inlet to exchanger, h_{2h} – enthalpy of hot fluid at outlet from exchanger, h_{2c} – enthalpy of cold fluid at outlet from exchanger.
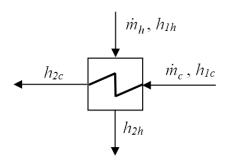

Figure 5. Schematic of steam cooler and regenerative heat exchangers.

Efficiency of non-contact regenerative heat exchanger is given by

$$\eta_{HE} = \frac{\dot{m}_c(h_{2c} - h_{1c})}{\dot{m}_h(h_{2h} - h_{1h})} \ . \tag{9}$$

2.2 Definitions of power and unit efficiency

Gross electric power of the power unit has been determined on the basis of electric power produced by successive turbine stages, N_{elT}, which can be written as

$$N_{elGr} = \sum N_{elT} \ . \tag{10}$$

Efficiency of gross production of electricity has been defined as a ratio of electric power generated by the unit, N_{elGr}, to the rate of heat of chemical

energy, $\dot{Q}_{chem} = \dot{Q}_B/\eta_B$ (η_B – boiler efficiency) in the fuel [19]

$$\eta_{elGr} = \frac{N_{elGr}}{\dot{Q}_{chem}} = \frac{N_{elGr}}{\dot{m}_f W_d} \ , \tag{11}$$

where $\dot{m}_f$ — mass flow rate of fuel, W_d – low calorific value.

Thermal efficiency of the cycle is defined as a ratio of the difference between rate of heat supplied to the cycle in the boiler, $\dot{Q}_B$, and removed from the cycle, $\dot{Q}_{OUT}$, to the thermal power supplied to the cycle in the boiler

$$\eta_{cyc} = \frac{\dot{Q}_B - \dot{Q}_{OUT}}{\dot{Q}_B} \ . \tag{12}$$

The rate of removed heat, $\dot{Q}_{OUT}$, is defined as

$$\dot{Q}_{OUT} = \dot{Q}_{CON} + \sum \dot{Q}_{loss} + \dot{Q}_{pip} \ , \tag{13}$$

where $\dot{Q}_{CON}$ – rate of heat removed from the condenser, $\sum \dot{Q}_{loss}$ – rate of heat losses in heat exchangers, $\dot{Q}_{pip} = \dot{Q}_{pip01-02} + \dot{Q}_{pip04-05}$ rate of heat losses in live steam pipelines and secondary steam pipelines. Losses flux in live steam pipelines, in line with Fig. 6, are

$$\dot{Q}_{pip01-02} = \dot{m}_{s01}(h_{01} - h_{02}) \ , \tag{14}$$

where: $\dot{m}_{s01}$ – mass flow rate of steam in the relevant point of the supercritical cycle (01), h_{01} – enthalpy of steam at inlet to the pipeline, h_{02} – enthalpy of steam at outlet from the pipeline.
On the other hand heat losses in the pipeline for the secondary steam are defined as

$$\dot{Q}_{pip04-05} = \dot{m}_{s04}(h_{04} - h_{05}) \ , \tag{15}$$

where $\dot{m}_{s04}$ – mass flow rate of steam in the relevant point of the supercritical cycle (04), h_{04} – enthalpy of steam at inlet to the secondary steam pipeline, h_{05} – enthalpy of steam at outlet from the secondary steam pipeline.

Modeling of a combustion process in the boiler was not applied in the study and therefore additional indicator has been introduced, namely the reference efficiency, η_{re}, defined as a ratio of gross electric power of the unit, N_{elGr}, to the rate of heat, $\dot{Q}_B$, required to produce steam in the boiler

$$\eta_{re} = \frac{N_{elGr}}{\dot{Q}_B} \ . \tag{16}$$

2.3   Model of the supercritical plant

Several attempts for static simulation [2,8,18,20,23] have recently presented with steam cycle in supercritical units. This numerical analysis was applied to the supercritical plant of the capacity of 900 MWe with the live steam parameters of 30.3 MPa/653 °C and secondary steam respectively of 6 Pa/672 °C [8]. A schematic of the plant with respective devices (B, HP, IP, LP, HE1-HE8, CON, P, D, G) has been presented in Fig. 6.

In the supercritical plant there are in operation the following fundamental devices, namely turbine with open-feedwater heaters and generators of the power of 900 MWe together with the steam boiler fired with the coal at the rate of 2200 t/h. In the system of the power plant there is also a series of other devices denoted in Fig. 6.

Calculations of the cycle of the supercritical power plant has been accomplished for the nominal operation conditions using the CFM code [16]. The thermal cycle has been coded in on the basis of elements presented in Figs. 2–4.

At the same time it must be mentioned that the present model uses the built-in tables of thermodynamical properties for the working fluid, available in CFM, which accelerate significantly the computational time.

Parameters of the cycle have been assumed on the basis of the guidelines from [8]. Part of the data has been presented in Tab. 1, which has been next compared with the results of calculations of operational parameters. The differences in specific parameters are small and result from the assumptions of the study. The gross electric power and electrical efficiency has been estimated at the level of 899.49 MW and 49.07%.

2.4   Numerical model of hybrid system

In the selection of working fluid the attention was focused to the fact that the working fluid operates in the subcritical cycle. For that reason the most adequate fluids are hydrocarbons and fluorocarbons, as well as their mixtures. The compounds which can be selected for testing in ORC installations are methane, ethane, propane, butane, isobutane, n-pentane, isopentane, n-hexane, ethylene, propylene, n-heptane, n-octane, ethanol, carbon dioxide, nitrogen, ammonia, R236ea, R245fa, as well as a series of other fluids used for example in refrigeration technology [3].

In the development of the numerical model of cooperation of the 900 MWe power unit with ORC it has been assumed that at the disposal is the theo-
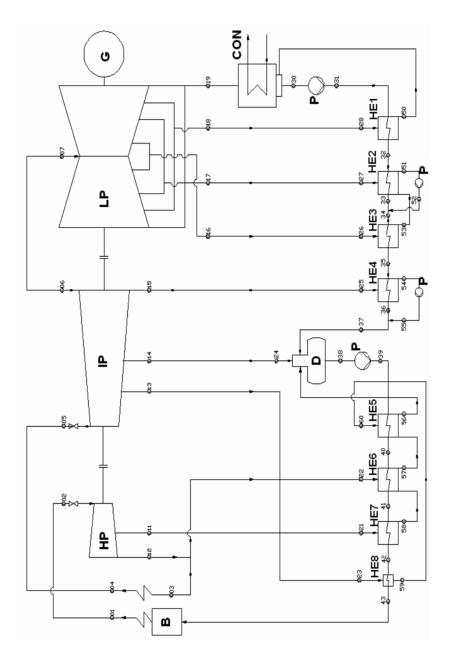
Figure 6. Schematic of the supercritical plant: B – boiler, HP, IP, LP – high, intermediate
and low pressure part, respectively, D – deaerator, HE1–HE4 – low-pressure
regenerative heat exchangers, HE5–HE7 – high-pressure regenerative heat ex-
changers, HE8 – steam cooler, P – pump, G – generator, CON – condenser
[19].

Table 1. Comparison of the input and output data from the model and data from literature [8].

Parameter	Symbol	Unit	Literature data	Data from the model
Cycle efficiency	η_{cyc}	%	50.92	50.91
Gross electric power (at generator)	N_{elGr}	MW	900.00	899.49
Reference efficiency	η_{re}	%	51.960	51.958
Gross efficiency of production of electricity (for the case of hard coal)	η_{elGr}	%	49.10	49.07
Temperature of live steam at outlet from the boiler	t_{01}	°C	653	653
Pressure of the live steam at outlet from the boiler	p_{01}	MPa	30.3	30.3
Temperature of live steam before the turbine	t_{02}	°C	650	650
Pressure the live steam before the turbine	p_{02}	MPa	30	30
Temperature of secondary steam at outlet from boiler	t_{04}	°C	672	672
Pressure of secondary steam at outlet from boiler	p_{04}	MPa	6	6
Temperature of secondary steam before turbine	t_{05}	°C	670	670
Pressure in degassifier	p_{24}	MPa	1.15	1.15
Pressure in condenser	p_{19}	MPa	0.005	0.005
Pressure after the condensate pump	p_{31}	MPa	2.2	2.2
Temperature of feeding water	t_{43}	°C	310	310
Internal efficiency of the group of stages of HP turbine	η_{iHP}	%	90	90
Internal efficiency of the group of stages of IP turbine	η_{iIP}	%	92	92
Internal efficiency of the group of stages of LP turbine	η_{iLP}	%	85	85
Internal efficiency of the last group of stages of LP turbine	η_{iLP1}	%	80	80
Boiler efficiency (hard coal)	η_B	%	94.5	94.4
Generator efficiency	η_g	%	98.8	98.8
Mechanical losses of turbine	ΔN_m	MW	0.9	0.9
Internal efficiency of pumps	η_{ip}	%	85	85
Efficiency of regenerative heat exchangers	η_{HEm}	%	99.5	99.5-99.6
Efficiency of vapour cooler	η_{HE}	%	99.5	99.4
Efficiency of deaerator	η_D	%	100	100
Flow losses in vapour pipelines to regenerative heat exchangers and vapour cooler	ζ	%	2	2
Flow losses in vapour pipeline from the vapour cooler to regenerative heat exchanger W5	ζ_{59-60}	%	1	1
Feedwater flow losses through regenerative heat exchangers and vapour cooler	Δ_{loss}	%	1	1
Pressure loss of circulation fluid in the boiler	Δp_{43-01}	MPa	4.2	4.2
Vapour pressure loss in secondary superheater	Δp_{03-04}	MPa	0.3	0.3
Flow losses in superheated steam pipelines	ζ_{12-03}	%	1.7	1.7
	ζ_{04-05}	%	1.7	1.7
Flow losses between MP and LP parts of turbine	ζ_{06-07}	%	1	1

retical rate of heat of water equal to 200 MWt from economizer (economizer is shown in Fig. 7.) which produce water with initial temperature of 90 °C from exhaust gases. The water is attaining its parameters in the waste heat recovery system, where the rate of heat is removed from the flue gases to water in economizer (E). Such water can be used for the heating of ORC in waste heat exchanger (WHE) or can be used in heat regeneration in low-pressure heat exchangers (HE1). A novelty in the present work is the fact that the ORC is additionally being heated by the bleed steam in evaporator (EV) [9]. In the considered case the steam from extraction points has temperature $t^{ext} = 227$ °C and pressure $p^{ext} = 0.2469$ MPa. The corresponding condensation temperature of steam is 127 °C. That enables to heat the ORC working fluid to the temperature of $t_{T,ORC} = 122$ °C [19].

In calculations of ORC it has been assumed that the minimum temperature difference between the working fluid and condensing steam is $\Delta T = 5$ K. Assumed evaporation temperatures $t_{T,ORC}$ (hence temperature of vapour before turbine) and corresponding values of pressure $p_{T,ORC}$ have been presented in Tab. 2. In case of the ORC condenser there has been assumed temperature of the phase change the same as in the reference cycle, that is $t_{con} = 32.8$ °C. It has been additionally assumed that water, originally at temperature 90 °C reduces its temperature down to 50 °C. The mass flow rate of bleeding steam available for use in the ORC is available in the extent up to $\dot{m}^{ext} = 23$ kg/s. That corresponds to the rate of heat equal to $\dot{Q}^s_{ORC} = 51.1$ MWt.

2.5 Parameters used for comparison of hybrid system operating with different working fluids

The specific work of the cycle has been estimated as a ratio of technical power, $N_{t,ORC}$, to the mass flow rate of ORC working fluid, $\dot{m}_{ORC}$,

$$l_{t,ORC} = \frac{N_{t,ORC}}{\dot{m}_{ORC}} \ , \tag{17}$$

where technical power of ORC is a difference between power delivered by the turbine and consumed by the pump $N_{t,ORC} = N_{T,ORC} - N_{P,ORC}$. On the other hand the specific rate of thermal energy supplied to ORC is determined as a ratio of the rate of heat supplied to the ORC, $\dot{Q}_{ORC}$, to the mass flow rate of working fluid, $\dot{m}_{ORC}$,

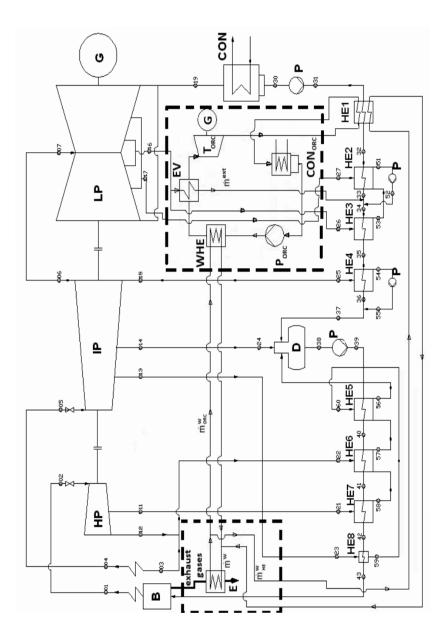$$q_{ORC} = \frac{\dot{Q}_{ORC}}{\dot{m}_{ORC}} \ . \tag{18}$$

Figure 7. General schematic of the supercritical power plant cooperating with ORC: B –
boiler, HP, IP, LP – high, intermediate and low pressure part, respectively, D –
deaerator, HE1–HE4 – low-pressure regenerative heat exchangers, HE5–HE7 –
high-pressure regenerative heat exchangers, HE8 – steam cooler, P – pump, G
– generator, CON – condenser of water steam, EV – evaporator, WHE - waste
heat exchanger with use water with temperature 90 °C, T_{ORC} – turbine of
organic fluid, P_{ORC} – pump of organic fluid, CON_{ORC} – condenser of organic
fluid vapour, E – economizer which produce water with temperature 90 °C
from exhaust gases. Additional devices are marked in the frame.

The efficiency of the organic Rankine cycle is a ratio of the unit technical work of the cycle, l_{tCR}, to the unit rate of heat, q_{ORC}, supplied to the ORC

$$\eta_{t,ORC} = \frac{l_{t,ORC}}{q_{ORC}}. \tag{19}$$

As a reference to the power of the entire unit assumed was the electric power of the steam plant, N_{elGr}. The power of the supercritical power plant cooperating with the ORC, N_{elRU}, has been determined on the basis of electric power produced in the particular stages of steam turbine, N_{elT}, and the power obtained from ORC, N_{ORC}. The power obtained from the ORC is defined as

$$N_{ORC} = \eta_m \eta_g N_{t,ORC} - N_{w,ORC} \; , \tag{20}$$

where: η_m – is mechanical efficiency of the turbine and the pump, $N_{w,ORC}$ – is the electric power required to drive the additional circulation pump in the system of heat recovery.

The gross power of the system incorporating the ORC is the sum of both electricity generating units

$$N_{elRU} = N_{elGr} + N_{ORC} \; . \tag{21}$$

Defined also was the final reference efficiency as a ratio of the power produced by plant with the ORC, N_{elRU}, to the rate of heat, $\dot{Q}_B$, required to produce vapor in the boiler [19]

$$\eta_{re} = \frac{N_{elRU}}{\dot{Q}_B} \; . \tag{22}$$

3 Results

Six working fluids have been considered in the study, namely propane, izobutane, pentane, etanol, R245fa and R236ea. Comparison of obtained results for every fluid is given in Tab. 2. The highest efficiency under considered conditions is obtained in the case of ethanol and is equal to $\eta_{t,ORC} = 17.76\%$.

As mentioned earlier the flow rate of steam from the bleed heating the ORC was equal to $\dot{m}^{ext} = 23$ kg/s. The results of calculations on the influence of ORC together with the regeneration system in low-pressure heat exchangers on the operation of supercritical plant is presented in Tab. 3.

The best results are obtained for the case of R245fa, as in such case the highest increase of efficiency and the power of the whole plant is observed.

Following the introduction of the hybrid system and the use of heat from the extraction points at the flow rate of steam of $\dot{m}^{extr} = 23$ kg/s the ORC uses the rate of heat equal to $\dot{Q}^s_{ORC} = 51.1$ MWt. At the same time the level of condensate regeneration (HE) reduces causing the decrease of temperature feeding the boiler to the level of $t_{43} = 303.93$ °C. At the same time the rate of heat to the boiler increases to $\dot{Q}_B = 1749.2$ MWt, in order to produce live steam with parameters presented in Tab. 1.

Table 2. Comparison of the ORC parameters.

Parameter	Unit	ORC working fluid					
		propane	isobutane	pentane	ethanol	R245fa	R236ea
$p_{T,ORC}$	Pa×10^5	36.00	22.50	9.46	4.59	31.0	24.8
$t_{T,ORC}$	°C	87.3	106.4	122.0	122.0	106.3	122.0
p_{con}	Pa×10^5	11.60	4.47	0.90	0.13	5.77	2.67
x	–	0.926	1.000	1.000	0.934	0.992	1
$l_{t,ORC}$	kJ/kg	30.69	53.60	79.19	187.34	17.33	29.25
q_{ORC}	kJ/kg	304.77	404.05	503.50	1054.64	152.48	201.67
$\eta_{t,ORC}$	%	10.07	13.27	15.73	17.76	11.37	14.50

It ought to be noticed that the cycle with propane as a working fluid consumes the highest total rate of heat, $\dot{Q}_{ORC}$, to heat and evaporate the low boiling point fluid from amongst analysed ORC. It should also be added that propane utilizes the highest rate of heat from the stream of hot waste water, $\dot{Q}^w$, originally at temperature 90 °C. All working fluids are exposed to the same rate of heat from the extraction points equal to $\dot{Q}^s_{ORC} = 55.1$ MWt, as well as the condensate of steam in all cases consumes the same rate of heat from regeneration $\dot{Q}_{HE1} = \dot{Q}^w_{HE1} + \dot{Q}^{ORC}_{HE1} = 89.4$ MWt. Therefore of principal importance for operation of the entire unit is the rate of heat $\dot{Q}^w_{ORC}$, which can be removed from the waste heat.

4   Conclusions

As can be seen in all considered cases the increase of overall electricity production is observed for all considered working fluids.
Electric power of the reference plant together with the electrical efficiency

Table 3. Selected results of calculations of the influence of ORC presence on the super-critical cycle at $\dot{m}^{extr} = 23$ kg/s.

Parameter	Unit	Working fluid					
		Propane	Isobutane	Pentane	Ethanol	R245fa	R236ea
$\dot{m}_{ORC}$	kg/s	395.0	208.2	146.4	62.1	730	410.5
$N_{t,ORC}$	MW	12.12	11.16	11.59	11.63	12.65	12.01
N_{elRU}	MWe	916.00	915.05	915.48	915.52	916.52	915.90
ΔN_{elRU}	MWe	16.51	15.56	15.99	16.03	17.03	16.41
η_{re}	%	52.366	52.312	52.337	52.339	52.396	52.360
$\Delta\eta_{re}$	%	0.406	0.352	0.377	0.379	0.436	0.400
$\dot{Q}_{ORC} = \dot{Q}_{ORC}^s + \dot{Q}_{ORC}^w$	MWt	120.4	84.2	73.7	65.5	111.3	82.8
$\dot{Q}_{ORC}^w$	MWt	65.3	29.1	18.6	10.4	56.2	27.7
$\dot{Q}_{HE1} = \dot{Q}_{HE1}^w + \dot{Q}_{HE1}^{ORC}$	MWt	89.4	89.4	89.4	89.4	89.4	89.4
$\dot{Q}_{HE1}^w$	MWt	89.4	85.4	81.3	89.4	89.4	83.2
$\dot{Q}_{HE1}^{ORC}$	MWt	0.0	4.0	8.1	0.0	0	6,23
$\dot{Q}^w = \dot{Q}_{ORC}^w + \dot{Q}_{HE1}^w$	MWt	154.7	114.5	99.9	99.8	145.6	110.9
$\dot{m}^w$	kg/s	920	680	594	594	866	666

have been estimated at the level of $N_{elGr} = 899.49$ MWe and $\eta_{elGr} = 49.07\%$. It ought to be stressed that target values from [8] were respectively $N_{elGr} = 900$ MWe and $\eta_{elGr} = 49.10\%$. Additionally modeled were the temperature distributions in heat exchangers which allowed to obtain satisfactory thermodynamical parameters in characteristic nodes of the cycle, which confirms the accuracy of the numerical model [19]. Analysis of the operation of the supercritical plant cooperating with ORC enabled to establish the increase of the power of the whole unit ΔN_{elRU} and the reference efficiency $\Delta\eta_{re}$ for respective fluids by the following numbers:

- R245fa – $\Delta N_{elRU} = 17.3$ MWe and $\Delta\eta_{re} = 0.436\%$,
- Propane – $\Delta N_{elRU} = 16.51$ MWe and $\Delta\eta_{re} = 0.406\%$,
- R236ea – $\Delta N_{elRU} = 16.41$ MWe and $\Delta\eta_{re} = 0.400\%$,
- Ethanol – $\Delta N_{elRU} = 16.03$ MWe and $\Delta\eta_{re} = 0.379\%$,
- Pentane – $\Delta N_{elRU} = 15.99$ MWe and $\Delta\eta_{re} = 0.377\%$,
- Isobutane – $\Delta N_{elRU} = 15.56$ MWe and $\Delta\eta_{re} = 0.352\%$.

It ought to be added that the lowest ORC efficiency was obtained for the case of propane and is equal to $\eta_{t,ORC} = 10.07\%$, whereas the highest one is

for the case of ethanol $\eta_{t,ORC} = 17.76\%$. The highest rate of heat received from water heated to temperature 90 °C is supplied to the installation with propane as working fluid $\dot{Q}^w = 154.7$ MWt, whereas the smallest amount of rate of heat is taken by a system with ethanol $\dot{Q}^w = 99.8$ MWt.

It seems that in the considered calculations the propane seems to utilise most of available waste heat despite not having best electricity conversion efficiency. In the condensate regeneration process there is being used the heat flux $\dot{Q}_{HE1} = 89.4$ MWt. Reasuming, R245fa seems to be the most appropriate medium which could have been used in the analysed cycle, because it ensures the highest power and efficiency increase, which is $\Delta N_{elRU} = 17.03$ MWe and $\Delta\eta_{re} = 0.436\%$, respectively. Moreover, this working fluid utilized about 75% of total heat flux received from the flue gases heat regeneration system. On the basis of the achieved results it might be assumed that the best utilized rate of heat, in the thermodynamic way, comes from the rate of waste heat while using the medium of a low heat of vaporization at the specific saturation temperature $t_{T,ORC}$. To utilize 100% of total heat flux received from the flue gases heat regeneration system in subsequent calculations considered will be additional extraction point to aid the one considered in the paper. It is envisaged that then the ethanol will become the best performing working fluid.

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