TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 129, 2015, 51–75

PAWEŁ ZIÓŁKOWSKI[a] [2], TOMASZ KOWALCZYK[a,b], JAKUB HERNET[a] and SEBASTIAN KORNET[a,b]

The thermodynamic analysis of the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery

[a] *Energy Conversion Department, The Szewalski Institute of Fluid-Flow Machinery of the Polish Academy of Sciences, Fiszera 14, 80-231 Gdańsk, Poland*

[b] *Conjoint Doctoral School at the Faculty of Mechanical Engineering, Gdańsk University of Technology, Narutowicza 11/12, 80-233 Gdańsk, Poland*

Abstract

In this paper, thermodynamic analysis of the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases are presented. According to that purpose, the CFM (computation flow mechanics) approach has been used correctly. In this paper, traditional steam cycle, the bottoming organic Rankine cycle (ORC) and a system of waste heat recovery with use of water with temperature 90 °C have been analyzed. The Szewalski binary vapour cycle is providing steam as the working fluid in the high temperature part of the cycle, while another fluid – organic working fluid – as the working substance substituting conventional steam over the temperature range represented by the low pressure steam expansion. The steam cycle for reference conditions, the Szewalski binary vapour cycle, and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery have been comprised. Four working fluids in the low temperature part of binary cycle such as ammonia, propane, isobutene and ethanol have also been investigated. Moreover, the Szewalski cycle is a good resolution for proper using heat flux received from the exhaust gases heat regeneration system.

[2] Corresponding author. E-mail address: pziolkowski@imp.gda.pl

Keywords: Binary cycle; Waste heat; ORC, Thermodynamical analysis; Numerical analysis, CFM (Computational Flow Mechanics)

Nomenclature

l	–	specific work, kJ/kg
N	–	power, kW
$\dot{m}$	–	mass flow rate, kg/s
p	–	pressure, Pa
q	–	specific rate of heat, kJ/kg
$\dot{Q}$	–	rate of heat, heat energy flux, kW
$\dot{Q}_{chem}$	–	chemical energy flux, kW
T	–	temperature, $^{\circ}$C or K
W_d	–	fuel calorific value, kJ/kg

Greek symbols

Δp	–	pressure losses, MPa
ΔT	–	the temperature difference in the heat exchanger, K
η	–	efficiency, –
ζ	–	flow losses, –

Subscripts

B	–	boiler
con	–	condensation
cyc	–	cycle
D	–	deaerator
el	–	electrical
f	–	fuel
g	–	generator
Gr	–	gross
HE	–	heat exchanger
HEm	–	open feedwater heat exchanger
i	–	internal
IP	–	intermediate pressure
loss	–	loss
LP	–	low pressure
m	–	mechanical
ORC	–	organic Rankine cycle
out	–	outlet
OUT	–	outlet from cycle
P	–	pump
pip	–	pipelines

re	–	reference efficiency
RU	–	the reference unit
t	–	technical
w	–	water
1,2...	–	real points of process

1 Introduction

Currently, 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. Furthermore it is noticed that, the power generation from fossil fuels causes many environmental problems (e.g., global warming, air pollution, acid rain, etc.). 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 [3]. To attain the objectives outlined there it is necessary to build highly efficient power plants, capturing pollutions and converting industrial waste heat into electricity [21].

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 [3, 9, 10, 22, 23, 31, 40]. One of concept presented in works assumed design of binary cycle for increase of efficiency, using 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, by concept of heat supply to ORC installation with use water with temperature 90 °C as well as use of low-pressure (LP) extraction of steam, is discussed in works [40, 41]. Additionally, the mention system with using heat flux from the carbon dioxide (CO_2) capture installation is analyzed in the recent papers [23, 25].

Regarding power generation from low-grade heat source many thermodynamic cycles have been studied for low-temperature power generation, i.e., Kalina cycle [17, 21], binary cycle [14, 28, 34, 37] and ORCs (organic Rankine cycles) [21, 29, 36]. It should be added that there exist the incompatibility of steam power cycle with low-grade heat source because normal boiling point temperature of water is too high.

Nowadays, the binary vapour cycles are used in geothermal power plant although binary vapour sets [14, 27, 28, 34, 37] and cycles have been known for many years [32, 33]. For example, mercury/steam cycle and steam/low boiling point fluid cycle were investigated already in some pilot power plant [27, 33]. Binary cycle technology was introduced in the last two decades in geothermal fields. Low enthalpy binary cycles, have been used in closed geothermal energy cycles, which are based on the organic Rankine cycle. Geothermal binary plants are relatively poor converters of heat into work – efficiencies are low, typically in the range of 0.08–0.12. But owings to cascade application is possible to couple an ORC either to a conventional Rankine cycle or to another ORC, where the condenser of one acts as an evaporator of the next and so on [14, 27].

Bartnik [11] says that in a general case, the number of circulating media in hierarchic cycles can be arbitrarily large. An increase of the number of media with various temperatures of the operating range makes it possible to apply in a system higher range of the temperature increase between the upper and lower heat sources (environment). Thereby, exergy losses in the system are reduced and the production of electricity increases. The disadvantage of such a solution includes an increase of investment required to start the system. However, the loss of exergy stream in hierarchical j-cycle system comes as a consequence of mere increase of entropy streams of external heat source which are in contact with it (in practice we usually have to do with two sources) [11]. The examples of two sources hierarchic cycles were presented in [2,11,16,18–20,35,39]. Well known examples are common in combined cycle gas turbine [18, 19, 35, 39], gas-steam with fuel cells [19, 20], combined cycle gas turbine with total coal gasification [35], combined air/steam (e.g. , the Goliński-Jesionek multistage combined air/steam systems with external combustion) [16].

Srinivas and Reddy [30] examined the three power cycles viz. gas cycle (Brayton cycle), steam cycle (Rankine cycle) and organic Rankine cycle which were arranged in two possible methods to form a triple cycle and to increase its power generating performance. In one approach, all the three cycles are arranged in hierarchic series, the heat rejection of gas cycle is supplied to steam power plant having back-pressure turbine and the heat rejection of steam plant is supplied to ORC plant similar to a cascading. In second option, the two bottoming cycles, i.e., steam plant and ORC plant are arranged in parallel to gas turbine exhaust. In this connection, steam cycle works with high temperature heat recovery and ORC plant with low temperature heat recovery. Thermal efficiency and specific work of two triple cycles are compared with combined cycle (CC) power plant to draw the relative merits of two choices with a focus on compressor pressure ratio and gas turbine inlet temperature (GTIT). The results showed that the parallel arrangement in triple cycle offers greater benefit over the series arrangement [38]. Therefore the wok is focused on parallel configuration triple cycle results.

Angelino and Invernizzi [2] analyzed the performance of conventional cycles, such as steam-ORC system, Brayton cycle system (a closed nitrogen gas cycle) and CO_2 cycles. A binary steam–organic Rankine cycle at 550 °C has an efficiency of about 0.52, somewhat higher than that of a nitrogen Brayton cycle (0.507 at 700 °C). They said that carbon dioxide is recognized as an almost ideal medium for implementing single fluid condensation cycles. Additionally, they analyzed liquefied natural gas (LNG) as a source of usable cryogenic exergy for power cycles. With reference to the steam cycle in principle, it will be possible to assume a standard vacuum condensing cycle supplemented by a bottoming ORC unit. However since the low pressure turbine is the most expensive plant component it seemed more reasonable to 'cut' the steam cycle at a pressure slightly above one atmosphere and to commit to an ORC bottoming cycle the task of exploiting the full thermal potential from about 100 °C to the cryogenic LNG temperatures.

The main aim of the present paper is analysis of operational and thermodynamic parameters Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases. The Szewalski binary vapour cycle realizing the steam cycle in the high pressure and medium pressure parts and the ORC instead of the low pressure 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 [26,41]. 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 possible 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 [32, 33].

This paper analyzes both 900 MWe supercritical power plant, the Szewalski binary vapour cycle, the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases using available computational flow mechanics (CFM) codes for the reference case without ORC and with considerations of the latter. In both the Szewalski binary vapour cycle and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases there were considered four potential working fluids, namely propane, isobutene, ethanol and ammonia with respect to obtain highest output and efficiency of the cycle.

2 Idea of the Szewalski binary vapour cycle

In this paragraph, the Szewalski binary vapour cycle is presented. The whole system consists of the first traditional steam cycle and the second organic Rankine cycle. The Szewalski binary vapour cycle is providing steam as the working fluid in the high temperature part of the cycle, while another fluid – organic working fluid of low specific volume – as the working substance substituting conventional steam over the temperature range represented by the low pressure steam expansion (Fig. 1). The objective of this concept leads: 1) to significantly reduce the exhaust area of the turbine, and hence reduce the specific initial cost; and 2) to raise the power output attainable in a single turbine unit [32,33]. Comparison conventional steam cycle with the Szewalski concept of binary vapour cycle is presented on Fig. 1.

So the modern steam turbine set of large output should be cut up into two parts, between the intermediate pressure and low pressure casings (cylinders). Szewalski expected that this cut up into two parts at a pressure level of about 0.15 to 0.4 MPa, depending on the type of the low boiling point fluid for application in the low temperature turbine [33]. At this pressure, steam leaving the steam turbine intermediate part (IP) of the set is being condensed, and it is latent heat is being transferred in the heat exchanger ($\mathrm{HE_{ORC}}$) to the organic Rankine fluid (low boiling point fluid) in order to heat it up and then to evaporate at boiling temperature. The resulting cold vapours enters the ORC turbine ($\mathrm{T_{ORC}}$) and exhausts on the other end to ORC condenser ($\mathrm{CON_{ORC}}$). After condensation the working fluid is fed back to the heat exchanger ($\mathrm{HE_{ORC}}$), a condenser-boiler, by means of a boiler feed pump while the steam condensate is pumped back into the steam generator [32, 33]. Thus the binary steam/low boiling point fluid power cycle can be considered as being composed of two separate cycles with steam as the working fluid in the high temperature cycle, and a low boiling point fluid in the low-temperature one (Fig. 1b). These two cycles are joined together by means of the heat exchanger ($\mathrm{HE_{ORC}}$) which acts as the heat sink and condenser for steam and at the same time as the

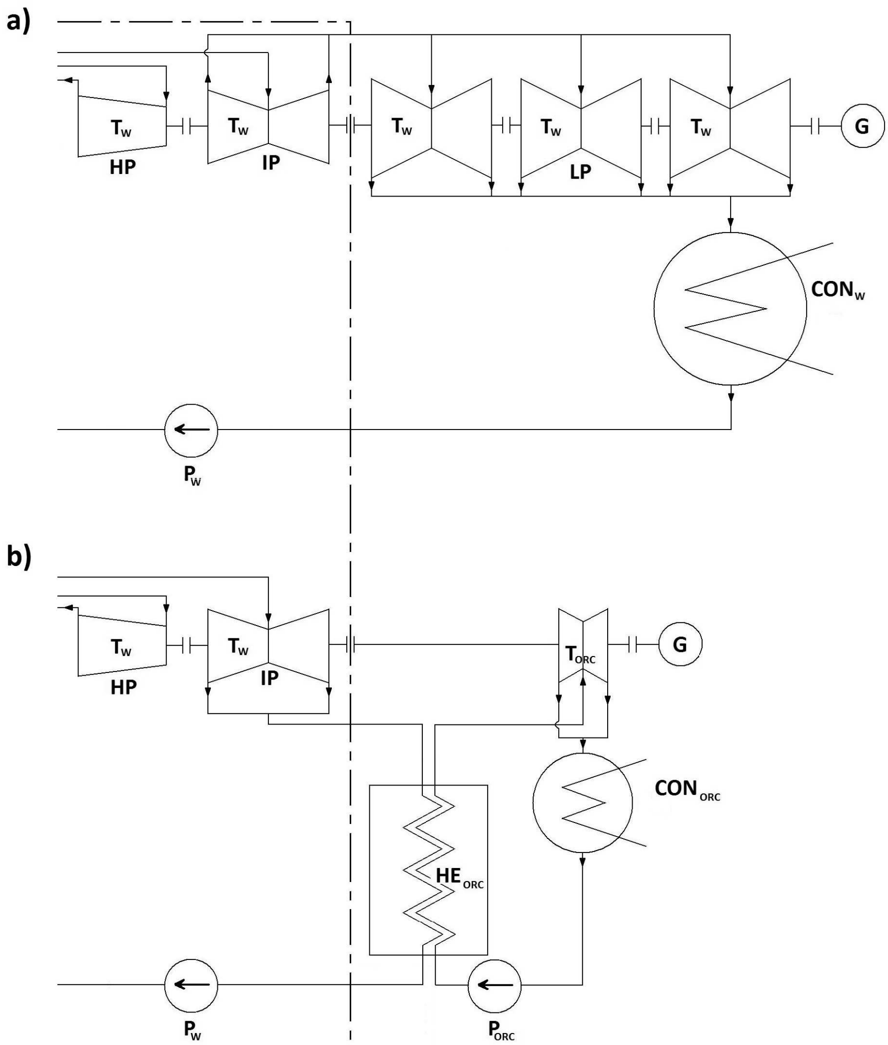

Figure 1: Schematic of the concept of the Szewalski binary vapour cycle b) comparison to conventional steam cycle a): LP – low-, IP – intermediate-, HP – high-pressure part of steam turbine, respectively, HE_{ORC} – regenerative heat exchangers, P_W – pump of water, G – electric generator, CON_w – condenser of water steam, T_w – turbine of steam, T_{ORC} – turbine of low boiling point fluid, P_{ORC} – pump of low boiling point fluid, CON_{ORC} – condenser of vapour of low boiling point fluid [33].

heat source, that is the vapour-generator, for the low temperature working fluid [32, 33].

Because of the very small specific volume of the cold vapour, in fact smaller by orders than the specific volume of steam at the same temperature level, the volume flow in the low-temperature turbine becomes quite small in comparison to the volume flow in the exhaust of the conventional condensing steam turbine. Hence, the low-temperature turbine of the binary cycle acquires only comparatively small dimensions [33]. However from work Ziółkowski *et al.* [38] we know that to cut up conventional steam turbine between intermediate pressure and low pressure casing is needed investigation with heat flux received from the flue gases heat regeneration system to increase ORC part of the Szewalski binary vapour cycle. This heat flux received from the system of waste heat regeneration can be used for heating of ORC liquid. Hence, the heat exchanger (HE_{ORC}) which acts as the condenser for water steam and at the same time as the evaporator of heated low boiling point fluid.

The performance of ORC systems and cycles highly depends on working fluids properties, which affects operating condition, environmental impact, system efficiency and economical viability [24, 33, 37]. In this work in the selection of working fluid the attention was focused to the fact that the working fluid operates in the subcritical cycle. For Szewalski binary vapour cycle gabarites of devices are important so density of low boiling point fluid becomes crucial for working fluid [32, 33]. A low density leads to higher volume flow rate: the pressure drops in the heat exchangers are increased and the size of the expander must be increased. This has a huge impact on the cost of the system [8].

The working fluid must have optimum thermodynamic properties at the range of working temperature and additionally satisfy several criteria, such as being nontoxic, environmentally friendly, nonflammable, economical, allowing a high use of the available energy from a heat source. For that reasons 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 [8, 21, 24]. In this paper four working fluids in the low temperature part of binary cycle such as propane, isobutane, ethanol and ammonia were investigated. However

Szewalski analyzed: ammonia, freons R-21, and R-114.

3 Definition of power unit and efficiency

In this paragraph, the output power and the efficiency calculations are presented. Firstly, gross electric power of the conventional steam power unit, $N_{el\,Gr}$, has been calculated on the basis of electric power produced by successive turbine stages N_{elT}, as follows [4, 38]:

$$N_{el\,Gr} = \sum N_{elT} \,. \tag{1}$$

Secondly, efficiency of gross production of electricity, $eta_{el\,Gr}$, has been defined as an ratio of electric power generated by the conventional steam power unit, $N_{el\,Gr}$, and the rate of chemical energy, $\dot{Q}_{chem} = \dot{Q}_B/\eta_B$, in the fuel [4, 38]

$$\eta_{el\,Gr} = \frac{N_{el\,Gr}}{\dot{Q}_{chem}} = \frac{N_{el\,Gr}}{\dot{m}_f W_d} \,, \tag{2}$$

where $\dot{m}_f$ is the mass flow rate of fuel, W_d – low calorific value, η_B – boiler efficiency; $\dot{Q}_{chem}$ – the rate of chemical energy in the fuel supplied to the boiler (B in Fig. 2), $\dot{Q}_B$ – rate of heat supplied to the cycle in the boiler.

Thermal efficiency of the cycle, η_{cyc}, 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:

$$\eta_{cyc} = \frac{\dot{Q}_B - \dot{Q}_{OUT}}{\dot{Q}_B} \,. \tag{3}$$

$\dot{Q}_{OUT}$ is given by the equation

$$\dot{Q}_{OUT} = \dot{Q}_{CON} + \sum \dot{Q}_{loss} + \dot{Q}_{pip} \,, \tag{4}$$

where $\dot{Q}_{CON}$ is the rate of heat removed from the condenser, $\sum \dot{Q}_{loss}$ – rate of heat losses in heat exchanger, and $\dot{Q}_{pip} = \dot{Q}_{pip_{01-02}} + \dot{Q}_{pip_{04-05}}$ – rate of heat losses in live steam pipelines and secondary steam pipelines. Rate of heat losses in steam pipelines, in line with Figs. 2–4, is calculated using formula from works [4, 38, 41].

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 conventional steam power unit, $N_{el\,Gr}$, to the rate of heat, $\dot{Q}_B$, required to produce steam in the boiler [4, 38] :

$$\eta_{re} = \frac{N_{el\,Gr}}{\dot{Q}_B} \,. \tag{5}$$

The efficiency of the organic Rankine cycle has been defined as a ratio of the specific technical work of the cycle $l_{t,\,ORC}$ to the specific rate of heat q_{ORC} supplied to the ORC [3, 30]:

$$\eta_{t,\,ORC} = \frac{l_{t,\,ORC}}{q_{ORC}} \,. \tag{6}$$

As a reference to the power of the entire unit the electric power of the steam plant, $N_{el\,Gr}$, was assumed. The power of the supercritical power plant cooperating with the ORC, $N_{el\,RU}$, has been determined on the basis of electric power produced in the particular stages of steam turbine, $N_{el\,T}$, and the power obtained from ORC, $N_{el\,ORC}$. The power obtained from the ORC may be expressed as:

$$N_{ORC} = \eta_m \eta_g N_{t,\,ORC} - N_{w,\,ORC} \,, \tag{7}$$

where: $\eta_m = \eta_{mT} = \eta_{mP}$ is the mechanical efficiency of the ORC turbine and the pump, $N_{w,\,ORC}$ – is the electrical power needed to drive the additional circulation pump in the system of heat recovery.

The gross power of the system incorporating the ORC, $N_{el\,RU}$, is the sum of both electricity generating units

$$N_{el\,RU} = N_{el\,Gr} + N_{ORC} \,. \tag{8}$$

The final reference efficiency, η_{re}, has been defined as a ratio of the power produced by conventional steam power unit $N_{el\,Gr}$ with the power of ORC, N_{ORC}, (as was mentioned in Eq. (8), $N_{el\,RU}$) to the rate of heat, $\dot{Q}_B$, required to produce vapour in the boiler [4, 38]:

$$\eta_{re} = \frac{N_{el\,RU}}{\dot{Q}_B} \,. \tag{9}$$

4 Numerical model of analyzed cycles

In this paragraph, both 900 MWe supercritical power plant, the Szewalski binary vapour cycle and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases numerical models are presented. All computations of the mentioned cycles have been performed using the basic principles of steam systems and thermodynamic phenomena modeling and algorithms for computing the properties of steam and low boiling point fluids. So calculations of the cycles have been accomplished for the nominal operation conditions using the CFM code [4–6,18,38–42]. The thermal cycle has been coded in on the basis of devices presented in previous work of authors [18,38–41] . Computational procedures for each component CFM codes belong to zero-dimensional models (0D), because it contains an algebraically integral formulation of typical balances: mass, momentum and energy. Additionally, CFM codes use mathematical tables for the fluid properties [3–6,15,18,38–42]. CFM type numerical tool gives a possibility to model combined two vapour cycles, what has recently been demonstrated in articles [5,18,38–41].

It should be added that, despite the large number of published articles on the analysis of the steam cycle and ORC most of them are limited to traditional resolution and none of them present a detailed analysis of the Szewalski hierarchic vapour cycle cooperating with a waste heat recovery. Several attempts for investigation have been recently presented with steam cycle [5, 7, 9, 10, 12, 13, 31, 42], organic Rankine cycle [21, 29, 37] and some with hybrid and binary steam/low boiling point fluid cycle [1,12–14,22,27,28,38,40,41].

4.1 Conventional supercritical power plant

At first numerical analysis was applied to the conventional supercritical steam power plant of the capacity of 900 MWe with the live steam parameters of 30.3 MPa/653 °C and secondary steam respectively of 6 MPa/672 °C [42]. A schematic of the conventional power plant for nominal condition with respective devices (B, HP, IP, LP, HE1–HE8, CON, P, D, G) has been presented in Fig. 2.

In the analyzing supercritical plant (Fig. 2) there are in operations the following fundamental devices, namely extraction-condensing turbine (HP, IP, LP – high-, intermediate- and low-pressure part of steam turbine, respectively) with generator (G) of the power of 900 MWe together with the coal-fired steam boiler (B) with the live steam the rate of 2200 t/h. In the system of the power plant there is also a series of other devices denoted in scheme, such as deaerator (D), low-pressure regenerative heat exchangers (HE1–HE4), high–pressure regenerative heat exchangers (HE5–HE7),

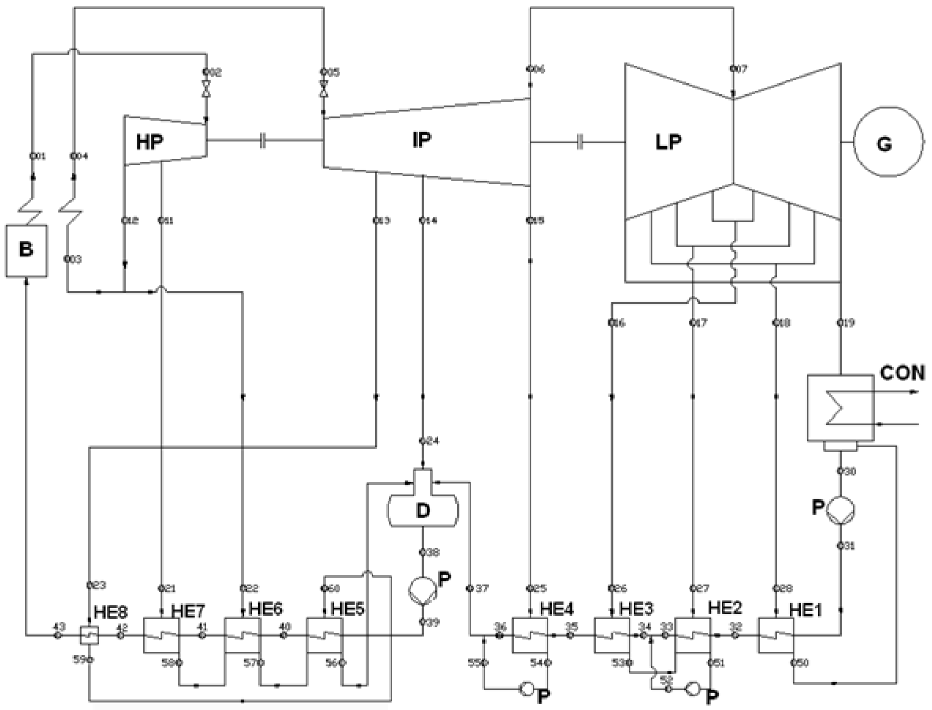

Figure 2: Schematic of the conventional supercritical power plant: B – boiler, HP, IP, LP – high-, intermediate- and low-pressure part, respectively, D – deaerator, HE1–HE4 – low-pressure regenerative heat exchangers, HE5–HE7 – high-pressure regenerative heat exchangers, HE8 – steam cooler, P – pump, G – generator, CON – condenser of steam. Basic steam cycle analyzed in work [42] is analogical to Fig. 1.a).

steam cooler (HE8), pump (P) and condenser of steam (CON). Additionally the temperature distribution in heat exchangers (HE1-HE8) were modeled, which allowed to obtain satisfactory thermodynamically parameters in characteristic nodes (21–60) of the cycle, which confirms the accuracy of the numerical model [5,38,40–42].

As it is shown in Tab. 1, the cycle efficiency of the 900.0 MWe – class power plant averages between $\eta_{cyc} = 0.5092$ and $\eta_{cyc} = 0.5091$ for the literature data and the software used during modeling process, respectively. Moreover, the gross electrical power of the supercritical power plant have been estimated at $N_{el\,Gr} = 899.5$ MWe by means of numerical analysis. Additionally, parameters in characteristic points (01–60 on Fig. 2) of the cycle have been estimated properly (Tab. 1). The obtained results should be regarded as satisfactory in spite of existing differences between results from numerical code and literature data.

4.2 The Szewalski binary vapour cycle

It has been assumed that HE_{ORC} – heat exchanger in Fig. 3, in the numerical model of Szewalski binary vapour cycle, is on one side the steam condenser and the generator of vapour of the low-boiling point fluid (Fig. 3). The level of condensate regeneration (HE) is constant hence temperature feeding the boiler is constant and equal $T_{43} = 310\,^{\circ}\mathrm{C}$. At the same time the rate of heat to the boiler does not change, in order to produce live steam with parameters presented in Tab. 1.

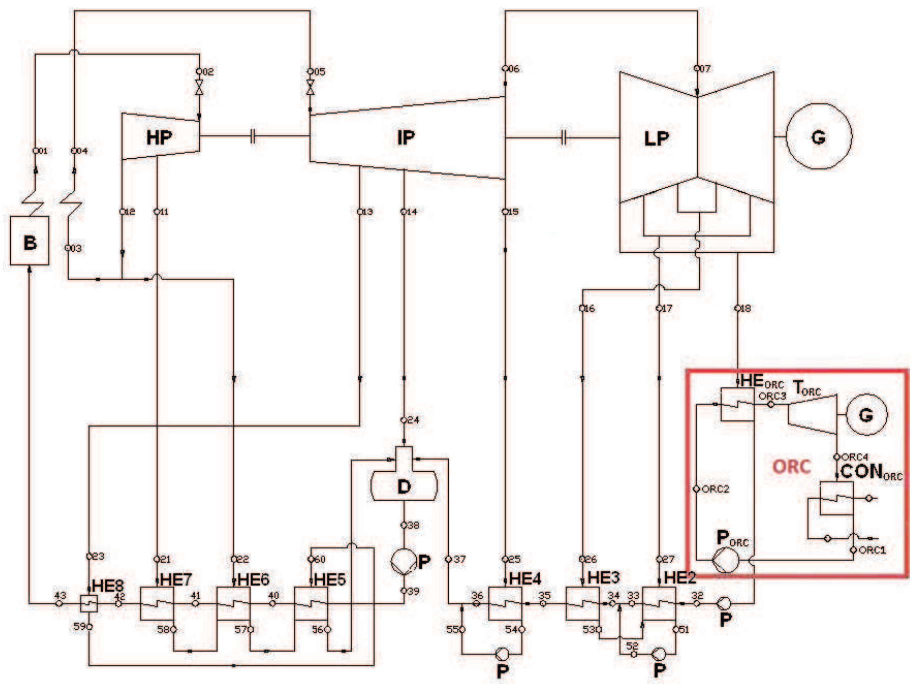

Figure 3: General schematic of the Szewalski binary vapour cycle, where additionally in comparison to Fig. 2: T_{ORC} – turbine of organic Rankine cycle, P_{ORC} – pump of organic Rankine cycle, CON_{ORC} – condenser of organic Rankine cycle, HE_{ORC} – heat exchanger, which is on one side the steam condenser and the generator of vapour of the low-boiling point fluid. Additional devices are marked in the frame. This scheme is analogical to Fig. 1.b) [38].

Table 1: Comparison of the input and output data from the model and data from literature [42].

Parameter	Symbol	Unit	Literature data	Data from the model
Cycle efficiency	η_{cyc}	–	0.5092	0.5091
Gross electric power (at generator)	$N_{el\,Gr}$	MW	900.00	899.49
Reference efficiency	η_{re}	–	0.51960	0.51958
Gross efficiency of production of electricity (for the case of hard coal)	$\eta_{el\,Gr}$	–	0.4910	0.4907
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 deaerator	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}	–	0.90	0.90
Internal efficiency of the group of stages of IP turbine	η_{iIP}	–	0.92	0.92
Internal efficiency of the group of stages of LP turbine	η_{iLP}	–	0.85	0.85
Internal efficiency of the last group of stages of LP turbine	η_{iLPl}	–	0.80	0.80
Boiler efficiency (hard coal)	η_B	–	0.945	0.944
Generator efficiency	η_g	–	0.988	0.988
Mechanical losses of turbine	ΔN_m	MW	0.9	0.9
Internal efficiency of pumps	$\eta_{i\,P}$	–	0.85	0.85
Efficiency of regenerative heat exchangers	η_{HEm}	–	0.995	0.995–0,996
Efficiency of vapour cooler	η_{HE}	–	0.995	0.994
Efficiency of deaerator	η_D	–	1.00	1.00
Flow losses in vapour pipelines to regenerative heat exchangers and vapour cooler	ζ	–	0.02	0.02
Flow losses in vapour pipeline from the vapour cooler to regenerative heat exchanger HE5	ζ_{59-60}	–	0.01	0.01
Feedwater flow losses through regenerative heat exchangers and vapour cooler	ζ_{loss}	–	0.01	0.01
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}	–	0.017	0.017
	ζ_{04-05}	–	0.017	0.017
Flow losses between IP and LP parts of turbine	ζ_{06-07}	–	0.01	0.01

In calculations of Szewalski binary vapour cycle, it has been assumed that minimum temperature difference between the evaporating low boiling point fluid and condensing steam is ΔT=5 K. In case of the ORC condenser temperature of the phase change is the same as in the reference cycle, T_{con} = 32.8 °C. Moreover, the efficiencies of the elements ORC system were all set up as following: turbine (T_{ORC}): internal $\eta_{iT} = 0.90$, mechanical $\eta_{mT} = 0.99$; pump (P_{ORC}): internal $\eta_{iP} = 0.85$, mechanical $\eta_{mP} = 0.99$; generator $\eta_g = 0.97$ and heat exchanger $\eta_{HE} = 0.98$. It has been additionally assumed that condensate is not supercooled after condensation as well in condenser of low boiling point fluid (CON_{ORC}) as in ORC heat exchanger (HE_{ORC}) [38]. The calculations of the heat cycle have been done for the constant the live steam parameters of 30.3 MPa/653 °C (point 01 in Figs. 2 and 3). Characteristic points of the conventional steam cycle (01–60) and the ORC cycle (ORC1–ORC4) have been presented in Fig. 3.

4.3 The Szewalski hierarchic vapour cycle cooperating with the system of heat recovery

In the development of the numerical model of the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases it has been assumed that HE_{ORC} – heat exchanger, is on one side the steam condenser and the generator of vapour of the low–boiling point fluid, however WHE – waste heat exchanger, is used for heating of ORC liquid (Fig. 4). As was mentioned before to cut up conventional steam turbine between intermediate pressure and low pressure casing is needed using heat flux received from the flue gases heat regeneration system to increase ORC part of the Szewalski binary vapour cycle. Hence, the heat exchanger (HE_{ORC}) which acts as the condenser for water steam and at the same time as the evaporator of heated low boiling point fluid.

Additionally it has been assumed that at the disposal is the theoretical rate of heat of water equal to 200 MWt from economizer (E – economizer are shown in Fig. 4.) which produces water with initial temperature of 90 °C from exhaust gases. The water is attaining its parameters in the waste heat recovery system where rate of heat is removed from the flue gases to water in economizer (E).

In calculations of ORC it has been assumed that minimum temperature difference between the heated working fluid and circulated water is $\Delta T = 5$ K, and temperature difference between the evaporating low boiling point fluid and condensing steam is $\Delta T = 5$ K. It has been additionally as-

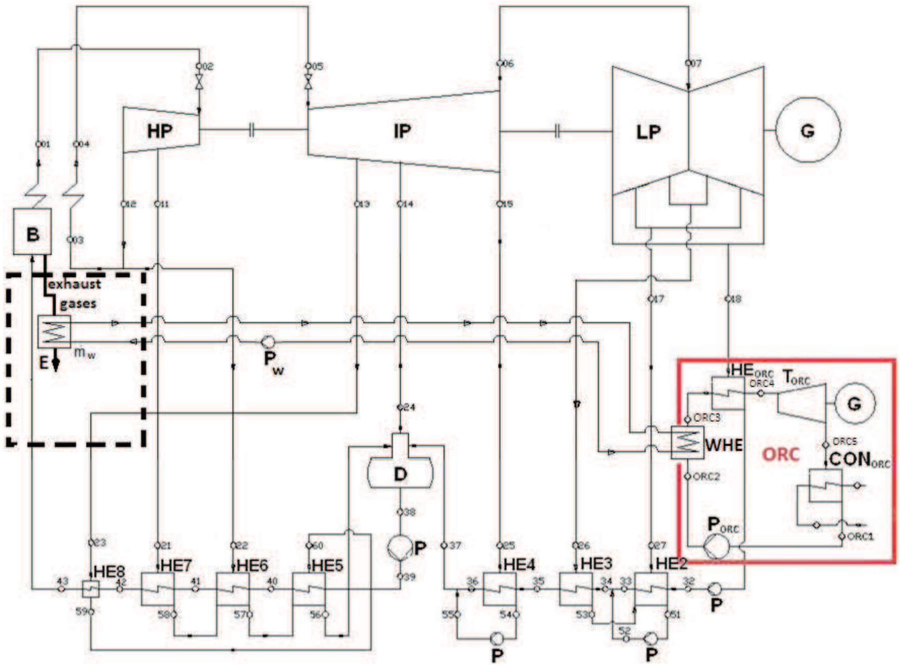

Figure 4: General schematic of the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases, where additionally in comparison to Figs. 2 and 3: WHE – waste heat exchanger with use water with temperature 90 °C, P_w – pump of circulated water, E – economizer which produce water with temperature 90 °C from exhaust gases. Additional devices in comparison to Fig. 2 are marked in the frame.

sumed that water, originally at temperature 90 °C reduces its temperature down to 50 °C. Moreover, the efficiencies of the elements of ORC system, temperature of condensing in CON_{ORC}, supercooling CON_{ORC} and HE_{ORC}, the live steam water parameters were all set up as it is written for the Szewalski binary vapour cycle. Characteristic points of the conventional steam cycle (01–60) and the ORC cycle (ORC1–ORC5) have been presented in Fig. 4.

5 Results

The constructed numerical model of the analyzed Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases has allowed carrying out a series of calculations in a wide range of variation of thermodynamic parameters of the system. The results of calculations have been developed in the tabular and graphical forms. Some of them are presented below.

Four working fluids have been considered in the study, namely propane, izobutane, etanol, and ammonia. Comparison of obtained results for every fluid is given in Fig. 5. The highest efficiency of organic Rankine cycle under considered conditions is obtained in the case of ethanol and is equal to $\eta_{\mathrm{t,\,ORC}} = 0.17$ in specific saturation temperature $T_{\mathrm{T,\,ORC}} = 117\,^{\circ}\mathrm{C}$. As was mentioned before in calculations of ORC it has been assumed that minimum temperature difference between the working fluid and condensing steam is $\Delta T = 5$ K. So range of evaporation temperatures $T_{T,\,ORC}$ (hence temperature of vapour before turbine) are assumed between $T_{T,\,ORC} = 45\,^{\circ}\mathrm{C}$ and $T_{T,\,ORC} = 117\,^{\circ}\mathrm{C}$. In the selection of working fluid the attention was focused to the fact that the working fluid operates in the subcritical cycle, so $T_{T,\,ORC}$ of propane is equal only $95\,^{\circ}\mathrm{C}$.

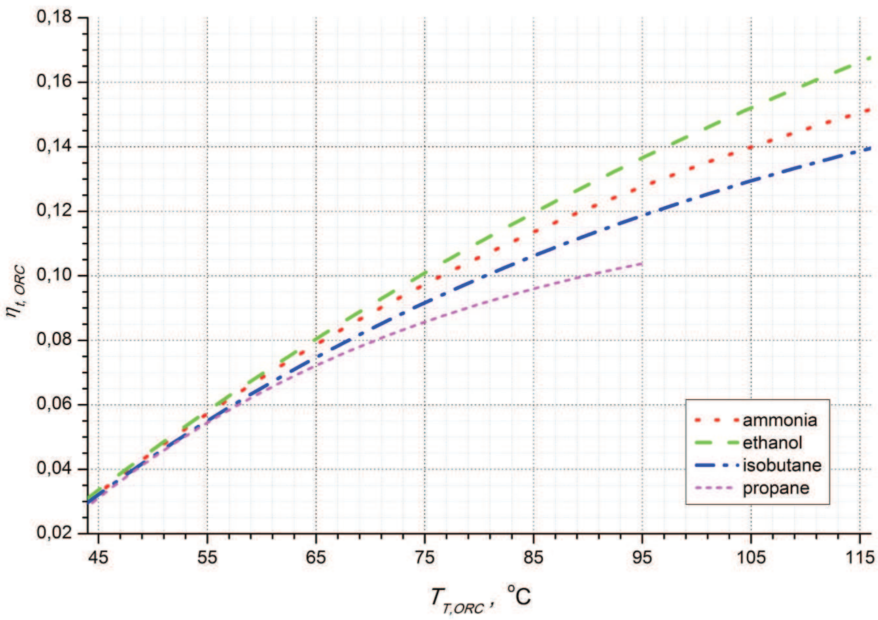

Figure 5: Efficiency of organic Rankine cycle $\eta_{t,\,ORC}$ vs temperature of vapour before turbine $T_{T,\,ORC}$ and the type of low boiling point fluid.

The effect of the increase of temperature of vapour before turbine, $T_{T,\,ORC}$, is clearly visible for efficiency of organic Rankine cycle, $\eta_{t,\,ORC}$, and the largest increase in efficiency – about 0.14 – occurs for temperature of vapour before turbine $T_{T,ORC} = 117\,^{\circ}\mathrm{C}$.

The calculation results of the influence of Szewalski binary vapour cycle [38], and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases on efficiency η_{re} and power $N_{el\,RU}$ of the whole system is presented in Fig. 6. The largest efficiency η_{re} in both analyzed cycles are achieved for relatively low values of temperature of condensing water $T_{con,W}$ (or temperature of vapour before turbine $T_{T,ORC}$). It should be mentioned that in heat exchanger (HE_{ORC}), which is on one side the steam condenser and the generator of vapour of the low–boiling point fluid, is assumed temperature difference which is equaled $\Delta T = T_{con,W} - T_{T,ORC} = 5$ K.

The rate of heat supplied to the boiler, $\dot{Q}_B$, was constant for all cases which have been investigated so the reference efficiency, η_{re}, depend only on power N_{elRU}. This was presented on Fig. 6 – curves of reference efficiency η_{re} and power N_{elRU} overlap each other.

As was mentioned before the modern steam turbine set of large output should be cut up into two parts, between the intermediate pressure and low pressure cylinders. Szewalski expected that this cut up into two parts at a pressure level of about 0.15 to 0.4 MPa, depending on the type of the low boiling point fluid for application in the low temperature turbine. As it was shown in the analysis from work [38], the power of the Szewalski binary vapour cycle, the optimal value of temperature of condensing water, $T_{con,W}$, for cut up into two parts is for ethanol 75 °C, for ammonia 65 °C, for isobutane 58 °C, for propane 55 °C, respectively. This temperature value $T_{con,W}$ corresponding with pressure of water condensation, $p_{con,W}$, equals: for ethanol 0.0386 MPa; for ammonia 0.025 MPa; for isobutane 0.0182 MPa; for propane 0.0158 MPa.

However for the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases (using waste heat) mention values are comparised with original idea. Hence, as it was presented in the Fig. 6, the power of the Szewalski hierarchic vapour cycle using waste heat, the optimal value of temperature of condensing water, $T_{con,W}$, for cut up into two parts is for ethanol 100 °C, for ammonia 84 °C, for isobutane 74 °C, for propane 67 °C, respectively. This temperature value $T_{con,W}$ corresponding with pressure of water condensation, $p_{con,W}$, equals: for ethanol 0.1014 MPa; for ammonia 0.056 MPa; for isobutane 0.037 MPa; for propane 0.027 MPa.

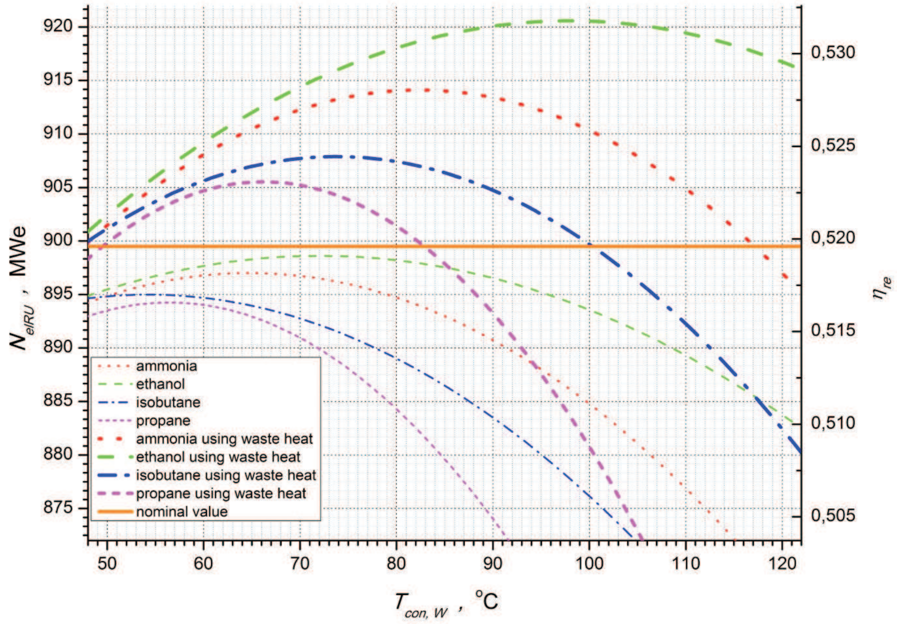

Figure 6: The power of the Szewalski binary vapour cycle [38], and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases (using waste heat) $N_{el\ RU}$ and the reference efficiency η_{re} vs temperature of condensing water $T_{con,\,W}$ and the type of low boiling point fluid. Comparison with nominal value of conventional steam cycle which presenting efficiency η_{re} and power $N_{el\ RU}$ (continuous line) supercritical steam plant from Fig. 2.

The calculation results clearly show that it is possible to found optimal value of process. It should be added that because of irreversibility in heat exchanger $\mathrm{HE_{ORC}}$ efficiency of Szewalski cycle is lower than nominal value of efficiency for conventional steam cycle, which is equal $\eta_{re} = 0.5196$ [38]. On the other hand, the increase of overall electricity production is observed for the Szewalski hierarchic vapour cycle using waste heat, and for all considered working fluids.

As well in Fig. 6 as in Fig. 7 was presented the reference efficiency η_{re} which has been defined by Eq. (5) for conventional steam cycle and by Eq. (9) for Szewalski binary vapour cycle, and for the Szewalski hierarchic vapour cycle using waste heat. The highest netto efficiency of Szewalski binary vapour cycle at the level of $\eta_{re} = 0.519$, $\eta_{re} = 0.518$ has been estimated for ethane and ammonia respectively. Much bigger top netto efficiency is obtain for the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery which it is estimated at the level of $\eta_{re} = 0.532$, $\eta_{re} = 0.528$ for ethane and ammonia respectively.

In the Fig. 7 the correlation of gross electric power of the conventional steam part to the power of ORC (see Eq. (8)), $N_{el\,Gr}/N_{ORC}$, in the Szewalski binary vapour cycle was presented. As it is shown in Fig. 7, $N_{el\,Gr}/N_{ORC}$ correlates to the type of low boiling point fluid. It is ought to add that cut up conventional steam turbine between intermediate pressure and low pressure casing is thermodynamically efficient with heat flux received from the flue gases heat regeneration system to increase ORC part of the Szewalski binary vapour cycle. This heat flux received from the flue gases heat regeneration system was used for heating of ORC liquid.

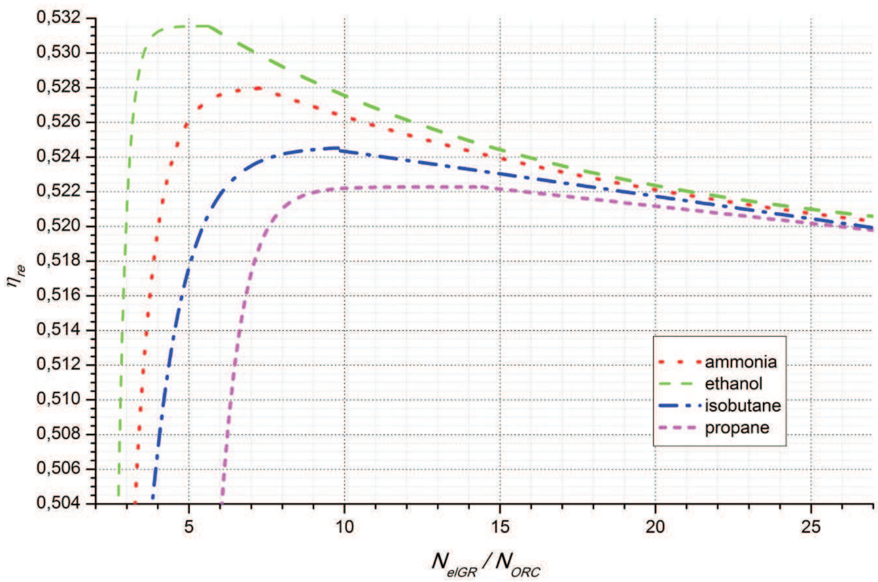

Figure 7: The reference efficiency η_{re} vs. $N_{el\,Gr}/N_{ORC}$ and the type of low boiling point fluid.

6 Conclusions

The numerical analysis has shown that electric power of the reference plant and the electrical efficiency have been estimated at the level of $N_{el\,Gr} = 899.49$ MWe and $\eta_{el\,Gr} = 0.4907$, respectively. It ought to be added that target values from [38, 42] were respectively $N_{el\,Gr} = 900.0$ MWe and $\eta_{el\,Gr} = 0.4910$.

As can be seen in all considered cases the highest ORC efficiency was obtained for the case of ethanol and is equal to $\eta_{t,ORC} = 0.17$ in specific saturation temperature $T_{T,ORC} = 117\,^{\circ}$C.

The highest netto efficiency of Szewalski binary vapour cycle at the level of $\eta_{re} = 0.5189$, $\eta_{re} = 0.5178$ have been estimated [38]. Moreover, the highest netto efficiency is obtain for the Szewalski hierarchic vapour cycle using waste heat which it is estimated at the level of $\eta_{re} = 0.532$, $\eta_{re} = 0.528$ for ethane and ammonia respectively. As it was shown in the thermodynamic analysis, the optimal value of temperature of condensing water $T_{con,W}$ for cut up into two parts is for ethanol 10 °C, for ammonia 84 °C, for isobutane 74 °C, for propane 67 °C, respectively. This temperature value $T_{con,W}$ corresponding with pressure of water condensation $p_{con,W}$ equals: for ethanol 0.1014 MPa, for ammonia 0.056 MPa, for isobutane 0.037 MPa, for propane 0.027 MPa. So, Szewalski hierarchic cycle using waste heat is good resolution for cooperating heat flux received from the flue gases heat regeneration system.

Analyzing both Szewalski binary vapour cycle, and the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery from exhaust gases very important are exergetic investigation. Part load characteristics of the conventional steam cycle as well as Szewalski binary vapour cycle is important information because in part load regime Szewalski binary vapour cycle can be more efficiency than traditional steam cycle.

Received 14 March, 2015

References

[1] Aneke M., Agnew B., Underwood C.: *Performance analysis of the Chena binary geothermal power plant.* Appl. Therm. Eng. **31**(2011), 1825–1832.

[2] Angelino G., Invernizzi: *Carbon dioxide power cycles using liquid natural gas as heat sink.* Appl. Therm. Eng. **29**(2009), 2935–2941.

[3] Badur J.: *Development of Energy Concept.* IMP PAN Publishers, Gdańsk 2009 (in Polish).

[4] Badur J.: *Five lectures of contemporary fluid termomechanics.* Gdańsk 2005 (in Polish).

[5] Badur J., Lemański M., Ziółkowski P., Kaczmarczyk O.: *Numerical analysis of the operating parameters of supercritical thermal cycle in power plant Jaworzno III with heat recovery system ORC.* Rep. IFFM PASci 4/2012, Gdańsk 2012 (in Polish).

[6] Badur J.: *Numerical modeling of sustainable combustion in gas turbines.* Rep. IFFM PASci, Gdańsk 2003 (in Polish).

[7] Badyda K.: *Mathematical model for digital simulation of steam turbine set dynamics and on-line turbine load distribution.* Transactions IFFM **126**(2014), 65–82.

[8] Bao J., Zhao L.: *A review of working fluid and expander selections for organic Rankine cycle.* Renew. Sust. Energy Rev. **24**(2013), 325–342.

[9] Bartela Ł., Skorek–Osikowska A., Kotowicz J.: *Integration of supercritical coal–fired heat and power plant with carbon capture installation and gas turbine.* Rynek Energii **100**(2012), 3, 56–62 (in Polish).

[10] Bartela Ł., Skorek–Osikowska A., Kotowicz J.: *Thermodynamic, ecological and economic aspects of the use of the gas turbine for heat supply to the stripping process in a supercritical CHP plant integrated with a corbon capture installation.* Energ. Convers. Manage. **85**(2014), 750–763.

[11] Bartnik R.: *Thermodynamic fundamentals for production of electric power in hierarchical j–cycle system.* Transactions IFFM **126**(2014), 141–151.

[12] Espatolero S., Cortés C., Romeo L.M.: *Optimization of boiler cold-end and integration with the steam cycle in supercritical units.* Appl. Energ. **87**(2010), 1651–1660.

[13] Feidt M.: *Thermodynamics of energy systems; a review and perspectives.* J. Appl. Fluid Mech. **5**(2012), 2, 85–98.

[14] Ghasemi H., Paci M., Tizzanini A., Mitsos A.: *Modeling and optimization of a binary geothermal power plant.* Energy **50**(2013), 412–428.

[15] Głuch J.: *Selected problems in determining an efficient operation standard in contemporary heat and flow diagnostics.* Pol. Marit. Res. S1/2009, Gdańsk (2009), 22–27.

[16] Gnutek Z: *On the Goliński-Jesionek multistage combined air/steam systems with external combustion.* Trans. IFFM **126**(2014), 33–54.

[17] Horbaj P., Braunmiller G., Taus P.: *On a environmentally friendly supply of energy.* Transactions IFFM **124**(2014), 221–230.

[18] Jesionek K., Chrzczonowski A., Ziółkowski P., Badur J.: *Enhancement of the Brayton cycle efficiency by water or steam utilization.* Transactions IFFM **124**(2012), 93–109.

[19] Kowalczyk T., Badur J.: *Hierarchical cycles in power engineering – a review of the literature.* Rep. IFFM PASci 861/2014, Gdańsk 2014.

[20] Lemański M., Karcz M.: *Performance of lignite-syngas operated tubular Solid Oxide Fuell Cell.* Chem. Process Eng. **23**(2007), 1–24.

[21] Le V.L., Feidt M., Kheiri A., Pelloux-Prayer S.: *Performance optimization of low–temperature power generation by supercritical ORCs (organic Rankine cycles) using low GWP (global warming potential) working fluids.* Energy **67**(2014), 513–526.

[22] Łukowicz H., Kochaniewicz A.: *Analysis of the use of waste heat obtained from coal-fired units in Organic Rankine Cycles and for brown coal drying.* Energy **45**(2012), 203–212 .

[23] Mikielewicz D., Barleta Ł., Ziółkowski P., Wajs J., Mikielewicz J.: *Operation of the 900 MW power plant with the ORC supplied from three heat sources.* In: Aktualne Zagadnienia Energetyki, Vol. II (K. Wójs, P. Szulc, Eds.), Wrocław University of Technology Press, 2014, 327–342.

[24] Mikielewicz D., Mikielewicz J.: *A thermodynamic criterion for selection of working fluid for subcritical and supercritical domestic micro CHP.* Appl. Therm. Eng. **30**(2010), 2357–2362.

[25] Mikielewicz D., Ziółkowski P., Wajs J., Mikielewicz J.: *Combined operations of 900 MW power plant with the ORC through the bleed steam extraction point and CO_2 recovery system.* In: Heat Transfer Renewable Sources of Energy 2014, Academic Publishing House ZUT Szczecin, 2014, 380–386.

[26] Mikielewicz J.: *On perspective ideas of future turbines by prof. Robert Szewalski.* In: Proc. of a Jubilee Session devoted to 100th Anniversary of Birthday of prof. Robert Szewalski, Gdańsk 2002 (in Polish).

[27] Rosyid H, Koestoer R, Putra N, Nasruddin, Mohamad, Yanuar A.: *Sensitivity analysis of steam power plant–binary cycle.* Energy **35**(2010), 3578–3586.

[28] Shokouhmand H., Atashkadi P.: *Performance improvement of a single, flashing, binary, combined cycle for geothermal power plants.* Energy **22**(1997), 7, 637–643.

[29] Song X., Shu-ying W., Dong-sheng Z.: *Slag–washing water of blast furnace power station with supercritical organic Rankine cycle.* J. Cent. South Univ. **20**(2013), 737–741.

[30] Srinivas T., Reddy: *Study on power plants arrangements for integration.* Energy Convers. Manage. **85**(2014), 7–12.

[31] Stępczyńska K., Łukowicz H., Dykas S.: *Calculations of ultra-supercritical coal fired 900 MWe power unit of with waste heat recovery.* Arch. Energ. **62**(2012), 155–164 (in Polish).

[32] Szewalski R.: *Actual problems of development of energetical technology. Enhancement of unit work and efficiency turbine and power unit.* Ossolineum, Wrocław 1978 (in Polish).

[33] Szewalski R.: *The binary vapour turbine set of great output, its concept and some basic engineering problems.* Transactions IFFM, 42–44(1969), 119–140.

[34] Tchanche B., Lambrinos Gr., Frangoudakis, Papadakis G.: *Low-grade heat conversion into power using organic Rankine cycles – a review of various applications.* Renew. Sust. Energ. Rev. **15**(2011), 3963–3979.

[35] Topolski J.: *Combustion diagnosis in combined gas-steam cycle.* PhD thesis, The Szewalski Institute of Fluid-Flow Machinery PASci, Gdańsk 2002 (in Polish).

[36] Vankeirsbilck I., Vanslambrouck B., Gusev S., De Paepe M.: *Organic Rankine cycle as efficient alternative to steam cycle for small scale power generation.* HEFAT 2011 8th Int. Conf. on Heat Transfer, Fluid Mechanics and Thermodynamics, July 11–13, 2011, Pointe Aux Piments, Mauritius.

[37] Vélez F., Segovia J.J., Martín M.C., Antolín G., Chejne F., Quijano A.: *A technical, economical and market review of organic Rankine cycles for the conversion of low-grade heat for power generation.* Renew. Sust. Energ. Rev. **16**(2012), 6, 4175–89.

[38] Ziółkowski P., Hernet J., Badur J.: *Revalorization of the Szewalski binary vapour cycle.* Arch. Thermodyn. **35**(2014), 3, 225–249.

[39] Ziółkowski P., Lemański M., Badur J., Nastałek L.: *Power augmentation of PGE Gorzow gas turbine by steam injection – thermodynamic overview.* Rynek Energii **98**(2012), 161–167.

[40] Ziółkowski P., Mikielewicz D.: *Thermodynamic analysis of the supercritical 900 MWe power unit, co–operating with an ORC cycle.* Arch. Energ. **62**(2012), 165–174 (in Polish).

[41] Ziółkowski P., Mikielewicz D., Mikielewicz J.: *Increase of power and efficiency of the 900 MW supercritical power plant through incorporation of the ORC.* Arch. Thermodyn. **34**(2013), 4, 51–71.

[42] *The reference cycle for the activities within the Strategic Project Contract SP/E/1/67484/10.* Institute of Machinery and Power Equipment, Silesian University of Technology, Gliwice 2011.