Revalorization of the Szewalski binary vapour cycle

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Abstract The aim of the paper is to revalorizate of the Szewalski binary vapour cycle by analysing thermodynamical and operational parameters of this cycle. This was carried out by accessible numerical computational flow mechanics codes using the step-by-step modeling of separate elements. The binary vapour cycle is providing steam as the working fluid in the high temperature part of the cycle, while another fluid – a low boiling point fluid of low specific volume – as the working substance substituting conventional steam over the temperature range represented by the low pressure steam expansion. The steam cycle for reference conditions has been assumed. Four working fluids in the low temperature part of the binary cycle such as propane, isobutane, ethanol and ammonia have been investigated.

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Nomenclature

h	–	specific enthalpy, kJ/kg
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, kW
$\dot{Q}_{chem}$	–	rate of chemical energy, kW
T	–	temperature, °C or K
$\dot{V}$	–	volume flow rate, m^3/s
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
ρ	–	density, kg/m^3
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
HEm	–	open feedwater heat exchanger
HP	–	high pressure
i	–	internal
in	–	inlet
IP	–	intermediate pressure
$loss$	–	loss
LP	–	low pressure
m	–	mechanical, condensate-steam mixture
ORC	–	organic Rankine cycle
out	–	outlet

OUT	–	outlet from cycle
P	–	pump
pip	–	pipelines
re	–	reference efficiency
reg	–	regeneration
RU	–	reference unit
s	–	steam, isentropic
t	–	technical
T	–	turbine
w	–	water
$1s, 2s, ...$	–	isentropic points of process
$1, 2, ...$	–	real points of process

1 Introduction

Currently, the binary vapour cycles are used in geothermal power plant although binary vapour sets [7,18,24,26] and cycles have been known since many years [22,23]. For example, mercury/steam cycle and steam/low boiling point fluid cycle were investigated already in some German pilot power plants [17,23]. Binary cycle technology was introduced in the last two decades in management of geothermal resources. Low enthalpy binary cycles, have been used in closed geothermal energy cycles, which are based on the organic Rankine cycle (ORC). Geothermal binary plants are relatively poor converters of heat into work – efficiencies are low, typically in the range of 0.08–0.12. Owing to cascade application if is possible to couple the ORC either to a conventional Rankine cycle or to another ORC, where the condenser of the first one acts as an evaporator of the next and so on [7,17].

A perspective design of binary cycle for the 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 can utilize the available waste heat, by the application of the concept of heat supply to ORC installation with use of water with temperature 90 °C as well as use of LP extraction of steam, as discussed in the recent works [14,29–30].

To attain a high efficiency there is a necessity 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 nominal value of efficiency is equal 0.4195 [25]. The steam turbine in thermal power plant consists of the large

and expensive low-pressure (LP) part which are still investigated and developed by scientists and energy utilities. The large dimension of the last stage blades of the LP turbines are a key to improving the steam turbine efficiency. An example of solution for the large and expensive LP part of turbine problem has been proposed by Szewalski [22,23], which will be discussed in the further part of the text.

The main aim of the present paper to present the analysis of operational and thermodynamic parameters of the Szewalski binary vapour cycle, realizing the steam cycle in the high pressure and medium pressure parts and the ORC instead of the 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 on the other one [15,23,30]. 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 of the configuration are a smaller amount of materials used and labour as well as reduction of erosive action of the working fluid on the blading system [22,23].

This paper analyses both 900 MWe supercritical power plant and the Szewalski binary vapour cycle using available computational flow mechanics (CFM) codes for the reference case without ORC and with consideration of the latter. CFM codes base on integral conservation equations of mass, momentum and energy [1–3,8,9,11,27–31]. In the Szewalski binary vapour cycle considered were four potential working fluids, namely propane, isobutene, ethanol and ammonia with respect to obtain a highest output and efficiency of the cycle.

2 The Szewalski binary vapour cycle

In this paragraph, the Szewalski binary vapour cycle is presented. The whole system consists of the traditional steam cycle and the organic Rankine cycle (ORC). 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 [22,23]. Comparison of the conventional steam cycle with the Szewalski concept of binary vapour cycle is presented in Fig. 1. So the modern steam turbine set of large output should be divided into two parts, between the intermediate pressure and low pressure casings (cylinders). Szewalski expected such division 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 [23]. At this pressure, steam leaving the steam turbine intermediate part of the set is being condensed, and it's latent heat is being transferred in the heat exchanger (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 enter the ORC turbine (T_{ORC}) and leaves on the other end to the ORC condenser (CON_{ORC}). After condensation the working fluid is fed back to the heat exchanger (HE_{ORC}), namely a condenser-boiler, by means of a boiler feed pump while the steam condensate is pumped back into the steam generator [22,23]. 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 connected by means of the heat exchanger (HE_{ORC}) which acts as the heat sink and condenser for steam and at the same time as a heat source, that is the boiler-generator, for the low temperature working fluid [22,23].

Because of the very small specific volume of the cold vapour, in fact smaller by orders of magnitude than the specific volume of steam at the same temperature level, the volume flow in the low-temperature turbine becomes quite small if compared with 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 [23]. For instance the low-pressure part of the conventional turbine unit of about 600 MWe, with three double-flow low-pressure cylinders (casings) , has an output of about one third of the whole while its initial cost is of the order of two thirds of the whole unit. If we succeed to lower the cost of this part, now comprising the cold vapour turbine and the heat exchanger, to one half of the cost of the conventional design, than the cost of the whole unit will be only two thirds of the previous one, and that counts markedly in the economic considerations bound with the real technical progress [22,23].
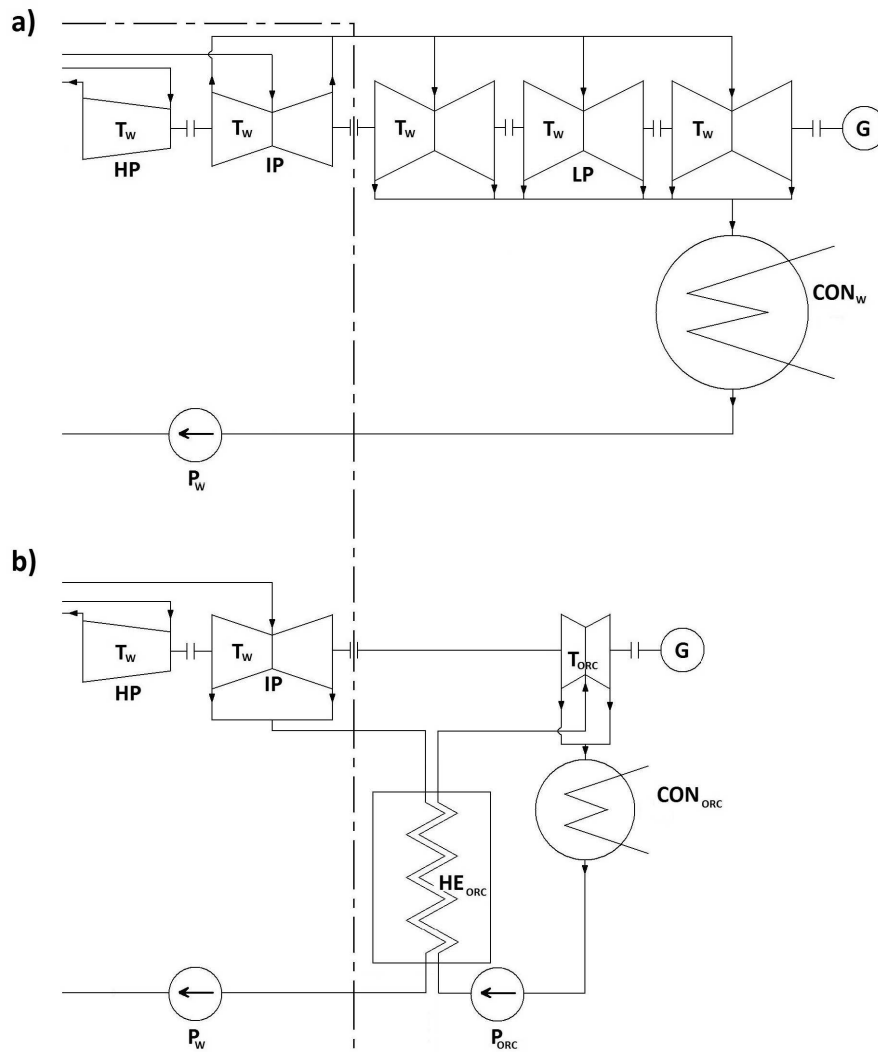
Figure 1: Schematic of the concept of the Szewalski binary vapour cycle b) comparison to conventional steam cycle a). LP – low-, IP –intermediate-, HP – high-pressure part of steam turbine, respectively, HE_{ORC} – regenerative heat exchangers, P_W – pump of water, G – generator, CON_W – condenser of 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 [23].

3 The low boiling point fluids

The performance of ORC systems highly depends on working fluids properties, which affects operating condition, environmental impact, system efficiency and economical viability [21,26]. In this work the attention was focused to the fact that the working fluid operates in the subcritical cycle. For the Szewalski binary vapour cycle important is the size of devices so the density of low boiling point fluid becomes crucial for working fluid [22,23]. A low density leads to higher volume flow rate: pressure drop in the heat exchangers is increased and the size of the expander must be increased. This has a huge impact on the cost of the system [4].
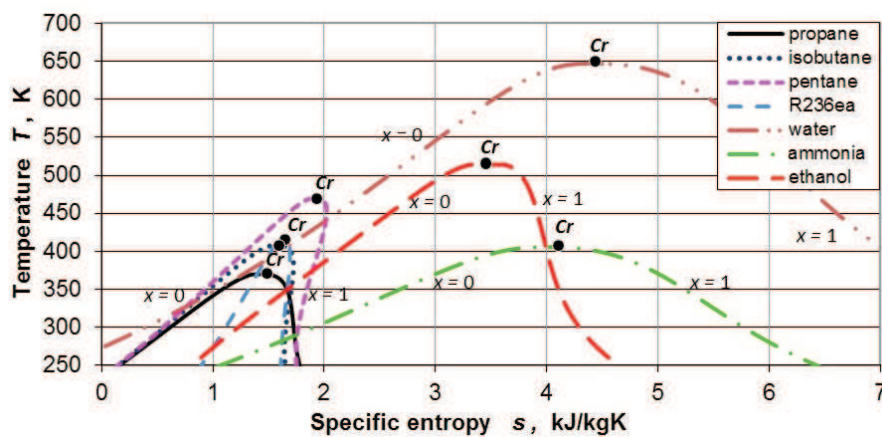

Figure 2: Diagram temperature – specific entropy, T-s, for chosen working fluids, [33].

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 the 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 [8,10,13,21]. In this paper investigated were four working fluids in the low temperature part of the binary cycle such as propane, isobutane, ethanol and ammonia (Fig. 2). The saturation vapour curve ($x = 1$) and the saturated liquid curve

$(x = 0)$ are separating the two-phase state and the single-phase state, which have been presented in Fig. 2. A 'wet' fluid shows a negative saturation vapour curve (ethanol, ammonia, water and propane), on the other hand, a 'dry' fluid shows a positive saturation vapour curve (pentane, isobutene and R236ea).

4 Main governing equation

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. Analysis of the thermodynamic cycle was performed with the use of CFM numerical codes. Mathematical models in CFM employ mass, momentum and energy balance equations in the integrated form (also called 0D or engineering form) [2,3,8,9,30,31]. In this paragraph, computational procedures for each component of the cycles are presented, namely: the turbines, pumps and heat exchangers. The output power and the efficiency calculations are also presented.

The CFM code, relying to the internal efficiency of the turbine stage, η_{iT}, also determines the power production. In Fig. 3 presented is enthalpy-entropy $h-s$ diagram for the process taking place in the turbine. The input data include the internal efficiency of the turbine stage, η_{iT}, mechanical efficiency of turbine, η_m, and the steam mass flow rate, $\dot{m}_s$. Having known
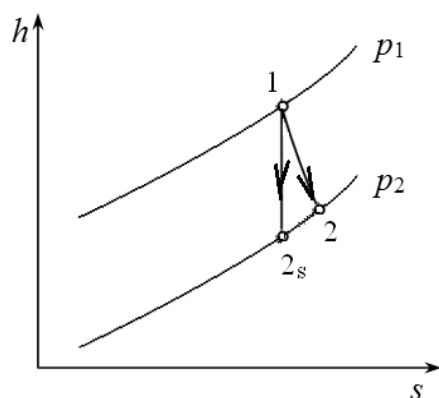

Figure 3: Isentropic expansion 1-2$_s$ in relation to the real expansion 1-2 from pressure p_1 to p_2.
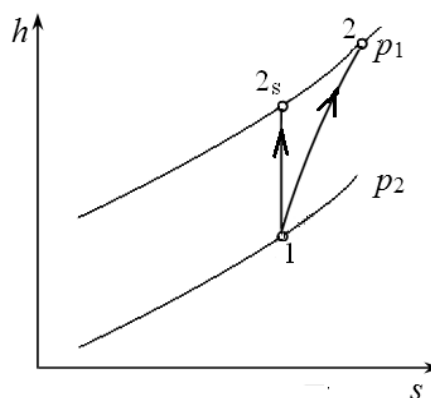
Figure 4: Isentropic pumping 1-2$_s$ and real pumping 1-2 from pressure p_1 to p_2.

the turbine inlet thermodynamic parameters, such as the pressure p_1, temperature T_1, and expansion ratio, π, the pressure p_2, can be calculated using formula

$$p_2 = \pi p_1 \,. \tag{1}$$

As mentioned earlier the expansion process from state 1 to 2 (Fig. 3) in the turbine stage is characterized by the turbine internal efficiency, defined as

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

where h_1, h_2, h_{2s} are the specific fluid enthalpy in state (1), (4), (2s), respectively.

The electric power, generated by the stage N_{elT} may be expressed by the formula [3,30]

$$N_{elT} = \eta_g \, \eta_{mT} \, \dot{m}_s (h_1 - h_2) \,, \tag{3}$$

where η_{mT} is the mechanical efficiency of the turbine, and η_g is the generator efficiency.

At the same time it must be mentioned that the present model uses the built-in tables of thermodynamical properties for the working fluid (for example: water, ethanol, ammonia, isobutene), available in CFM codes.

The electric power required to drive the pump, N_{elP}, for increasing the working fluid pressure is determined using the pump internal efficiency, η_{iP}, pump 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 water pump has a given temperature T_1, pressure p_1, and mass flow rate $\dot{m}_w$.

In Fig. 4 presented is the enthalpy entropy diagram $h - s$ for liquid pumping. The process of pressure increase in a pump from p_1 to p_2 can be characterized by the definition of internal pump efficiency η_{iP} according to relation:

$$\eta_{iP} = \frac{h_{2s} - h_1}{h_2 - h_1} \,, \tag{4}$$

where h_1, h_2, h_{2s} are the specific enthalpies of fluid determining the states (1), (4), (2s), respectively.

The consumed electric power by the pump N_{elP} can be expressed as

$$N_{elP} = \frac{\dot{m}_w \, (h_2 - h_1)}{\eta_{elm} \, \eta_{mP}} \,. \tag{5}$$

In CFM model of heat exchanger, the energy balance including all energy fluxes, need to be taken in to account. So the balance of rates of heat transferred in the heat exchanger (including the deaerator) can be written in the form as

$$\dot{Q}_{in} = \dot{Q}_{loss} + \dot{Q}_{out} \,, \tag{6}$$

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.
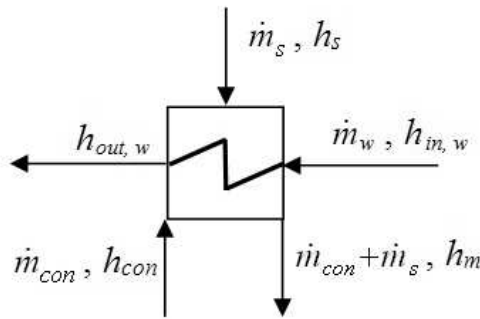

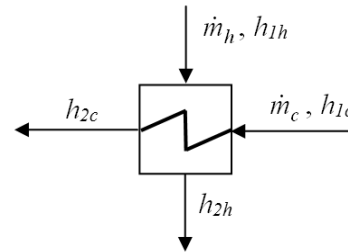Figure 5: Schematic of the open-feedwater heat exchanger.

Figure 6: Schematic of steam cooler and regenerative heat exchangers.

Equation (7) in case of the open-feedwater heat exchanger (Fig. 5) can be written as [3]

$$\dot{m}_s \, h_s + \dot{m}_{con} \, h_{con} + \dot{m}_w \, h_{in,w} = \dot{Q}_{loss} + (\dot{m}_{con} + \dot{m}_s) \, h_m + \dot{m}_w \, h_{out,w} \,, \tag{7}$$

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

Therefore the efficiency of the open feedwater heat exchanger is given by

$$\eta_{HEm} = \frac{\dot{Q}_{out}}{\dot{Q}_{in}} = \frac{\dot{m}_w \, (h_{out,w} - h_{in,w})}{\dot{m}_s \, (h_s - h_m) + \dot{m}_{con} \, (h_{con} - h_m)} \,. \tag{8}$$

Equation (7) in case of the closed feedwater heat exchanger, including the condenser, boiler and ORC heat exchangers (Fig. 6) can be written as

$$\dot{m}_h \, h_{1h} + \dot{m}_c \, h_{1c} = \dot{Q}_{loss} + \dot{m}_h \, h_{2h} + \dot{m}_c \, h_{2c} \,, \tag{9}$$

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

However, efficiency of noncontact regenerative heat exchanger is defined by

$$\eta_{HE} = \frac{\dot{m}_c \left(h_{2c} - h_{1c}\right)}{\dot{m}_c \left(h_{1h} - h_{1c}\right)} = \frac{h_{2c} - h_{1c}}{h_{1h} - h_{1c}} . \tag{10}$$

Analogical equations of the process taking place in the components of the cycle with water can be written for the case of ORC with low boiling point fluid.

Gross electric power of the conventional steam power unit, N_{elGr}, has been determined on the basis of electric power produced by successive turbine stages N_{elT}, which can be written as [3,30]

$$N_{elGr} = \sum N_{elT} . \tag{11}$$

Efficiency of gross production of electricity, η_{elGr}, has been defined as a ratio of electric power generated by the conventional steam power unit, N_{elGr}, and the rate of chemical energy, $\dot{Q}_{chem} = \dot{Q}_B/\eta_B$, in the fuel [3,30]:

$$\eta_{elGr} = \frac{N_{elGr}}{\dot{Q}_{chem}} = \frac{N_{elGr}}{\dot{m}_f W_d} , \tag{12}$$

where: $\dot{m}_f$ – 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. 7), $\dot{Q}_B$ – 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 and removed from the cycle $\dot{Q}_{OUT}$ to the thermal power supplied to the cycle in the boiler

$$\eta_{cyc} = \frac{\dot{Q}_B - \dot{Q}_{OUT}}{\dot{Q}_B} . \tag{13}$$

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

$$\dot{Q}_{OUT} = \dot{Q}_{CON} + \sum \dot{Q}_{loss} + \dot{Q}_{pip} , \tag{14}$$

where: $\dot{Q}_{CON}$ – rate of heat removed from the condenser, $\sum \dot{Q}_{loss}$ – rate of heat losses in heat exchangers, $\dot{Q}_{pip} = \dot{Q}_{pip_{01-02}} + \dot{Q}_{pip_{04-05}}$ – rate of heat

losses in live steam pipelines and secondary steam pipelines.

The rate of heat losses in live steam pipelines, in line with Figs. 7–8, is calculated using formula [3,30,32]

$$\dot{Q}_{pip01-02} = \dot{m}_{s01}(h_{01} - h_{02}),\tag{15}$$

where: $\dot{m}_{s01}$ – mass flow rate of steam in the relevant point of the super-critical cycle (01), h_{01} – specific enthalpy of steam at inlet to the pipeline, h_{02} – specific enthalpy of steam at outlet from the pipeline. On the other hand, the rate of heat losses in the pipeline for the secondary steam are expressed as [3,30,32]

$$\dot{Q}_{pip04-05} = \dot{m}_{s04}(h_{04} - h_{05}),\tag{16}$$

where: $\dot{m}_{s04}$ – mass flow rate of steam in the relevant point of the super-critical cycle (04), h_{04} – specific enthalpy of steam at inlet to the secondary steam pipeline, h_{05} – specific 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 conventional steam power unit, N_{elGr}, to the rate of heat, $\dot{Q}_B$, required to produce steam in the boiler [3,30]:

$$\eta_{re} = \frac{N_{elGr}}{\dot{Q}_B}.\tag{17}$$

The specific work of the ORC cycle, $l_{t,ORC}$, has been estimated as a ratio of technical power, $N_{t,ORC}$, to the mass flow rate of ORC working fluid, $\dot{m}_{ORC}$ [3,30],

$$l_{t,ORC} = \frac{N_{t,ORC}}{\dot{m}_{ORC}},\tag{18}$$

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 heat, q_{ORC}, supplied to ORC has been determined as a ratio of the rate of heat supplied to the ORC, $\dot{Q}_{ORC}$, to the mass flow rate of working fluid $\dot{m}_{ORC}$ [3,30]:

$$q_{ORC} = \frac{\dot{Q}_{ORC}}{\dot{m}_{ORC}}.\tag{19}$$

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

$$\eta_{t,ORC} = \frac{l_{t,ORC}}{q_{ORC}} \ .$$ (20)

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 may be defined as

$$N_{ORC} = \eta_m \, \eta_g \, N_{t,ORC} \ ,$$ (21)

where $\eta_m = \eta_{mT} = \eta_{mP}$ is a mechanical efficiency of the ORC turbine and the pump.

The gross power of the system incorporating the ORC, N_{elRU}, is the sum of both electricity generating units:

$$N_{elRU} = N_{elGr} + N_{ORC} \ .$$ (22)

The final reference efficiency, η_{re}, has been defined as a ratio of the power produced by the conventional steam power unit, N_{elGr}, with the power of ORC N_{ORC} (as was mentioned in Eq. (21) N_{elRU}), to the rate of heat, $\dot{Q}_B$, required to produce vapour in the boiler [3,30]:

$$\eta_{re} = \frac{N_{elRU}}{\dot{Q}_B} \ .$$ (23)

5 Numerical model of Szewalski binary vapour cycle

Despite a large number of papers on the numerical analysis of the steam cycle and ORC most are limited to traditional resolution and none of them presents a detailed analysis of the Szewalski binary vapour cycle. Several attempts for investigation have been recently presented with steam cycle in supercritical units [3,5–6,20,32], organic Rankine cycle [10,13,16,19,21] and some with hybrid and binary steam/low boiling point fluid cycle [7,12,17,

18,29,30]. At first numerical analysis was applied to the conventional su-
percritical 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 [32]. A schematic of the conventional power plant for nomi-
nal condition with respective devices has been presented in Fig. 7.

In the analysis of the supercritical plant (Fig. 7) there are in operations
the following fundamental devices, namely extraction-condensing turbine
(HP, IP, LP -Ő 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 exchang-
ers (HE1–HE4), high-pressure regenerative heat exchangers (HE5–HE7),
steam cooler (HE8), pump (P) and condenser of steam (CON). Additionally
modeled were the temperature distributions in heat exchangers (HE1–HE8)
which allowed to obtain satisfactory thermodynamical parameters in char-
acteristic nodes (21–60) of the cycle, which confirms the accuracy of the
numerical model [3,29-30].

Calculations of the cycle of the supercritical power plant has been ac-
complished for the nominal operation conditions using the CFM code [3].
The thermal cycle has been coded in on the basis of the devices presented
in Figs. 3–6. Parameters of the cycle have been assumed on the basis of the
guidelines from [32]. Part of the data has been presented in Tab. 1, which
has been next compared with the results of calculations of operational pa-
rameters.

As it is shown in Tab. 1, the cycle efficiency of the 900 MWe – class
power plant averages between $\eta_{cyc} = 0.5092$ and 0.5091 for the literature
data and the software used during modeling process, respectively. More-
over, the gross electrical power of the supercritical power plant have been
estimated at $N_{elGr} = 899.5$ MWe by means of numerical analysis. Addi-
tionally, parameters in characteristic points (01–60 in Fig. 7) of the cycle
have been estimated properly (Tab. 1). The obtained results should be re-
garded as satisfactory in spite of existing differences between results from
numerical code and literature data. It ought to be added that the target
values from [8] were respectively $N_{elGr} = 900.0$ MWe and $\eta_{elGr} = 0.4910$.

In the development of the numerical model of Szewalski binary vapour
cycle it has been assumed that the heat exchanger, HE_{ORC}, is on one side
the steam condenser and the generator of vapour of the low-boiling point

Table 1: Comparison of the input and output data from the model and data from literature [32].

| Parameter	Symbol	Unit	Literature data	Data from the model
Cycle efficiency	η_{cyc}	–	0.5092	0.5091
Gross electric power (at generator)	N_{elGr}	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)	η_{elGr}	–	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_{iLP1}	–	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	η_{iP}	–	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 W5	ζ_{59-60}	–	0.01	0.01
Feedwater flow losses through regenerative heat exchangers and vapour cooler	Δ_{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 MP and LP parts of turbine	ζ_{06-07}	–	0.01	0.01
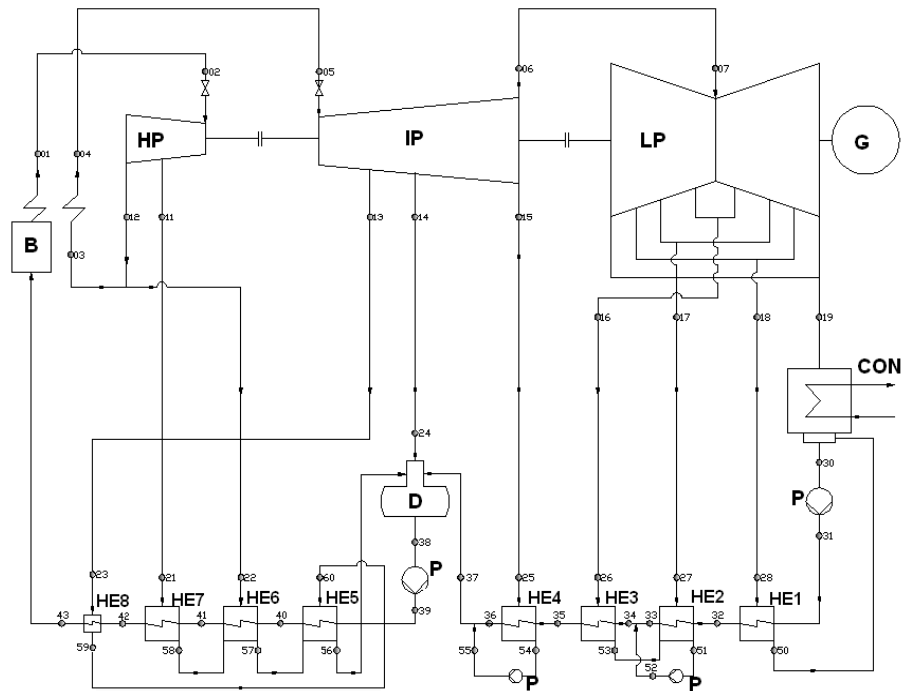
Figure 7: Schematic of the conventional supercritical power plant, where: B – boiler, HP, IP, LP – high, intermediate and low pressure part, respectively, D – deaerator, HE1-HE4 – low-pressure regenerative heat exchangers, HE5–HE7 – high-pressure regenerative heat exchangers, HE8 – steam cooler, P – pump, G – generator, CON – condenser of steam. Basic steam cycle analyzed in work [30] is analogical to Fig.1a.

fluid (Fig. 8) on the other one.

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 $\Delta T = 5$ K. In case of the ORC condenser there has been assumed temperature of the phase change the same as in the reference cycle, that is $T_{con} = 32.8\,^{\circ}$C. Moreover, the efficiencies of the elements ORC system were all set up as it is written below: 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 isn't supercooled after condensation as well in condenser of low boiling point fluid (CON$_{ORC}$) as in ORC heat exchanger (HE$_{ORC}$). The calculations of

the heat cycle had been done for the constant live steam parameters of 30.3 MPa/653 ºC (point 01 in Figs. 7 and 8). Characteristic points of the conventional steam cycle (01–60) and the ORC cycle (ORC1–ORC4) have been presented in Fig. 8.
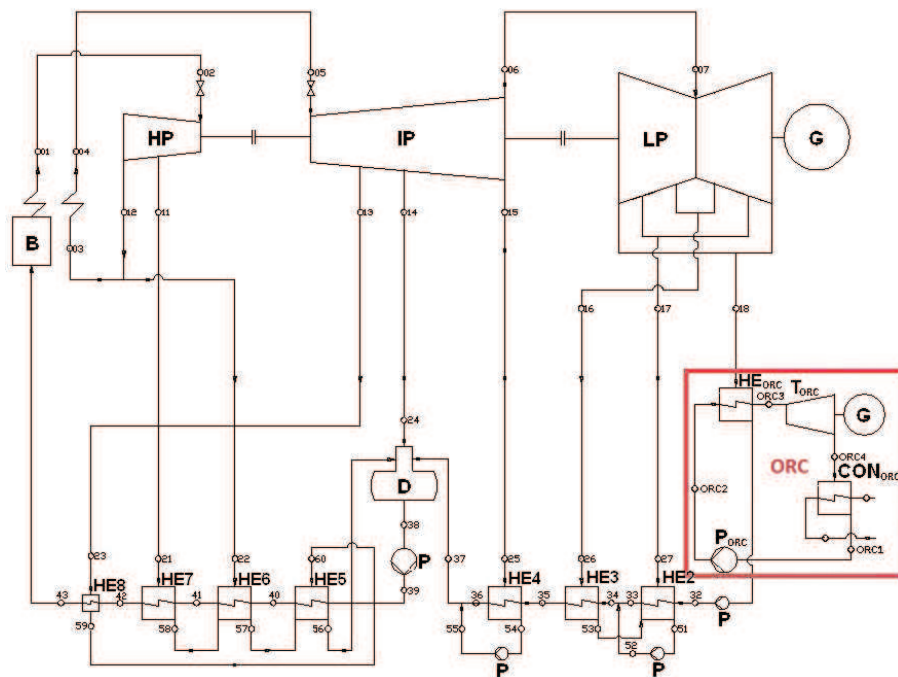

Figure 8: General schematic of the Szewalski binary vapour cycle , where additionally in comparison to Fig. 7: T_{ORC} – turbine of organic Rankine cycle, P_{ORC} – pump of organic Rankine cycle, CON_{ORC} – condenser of organic Rankine cycle, HE_{ORC} – heat exchanger, which is on one side the steam 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. 1b.

6   Results

The constructed numerical model of the analyzed Szewalski binary vapour cycle allowed to carry out a series of calculations in a wide range of variation of thermodynamic parameters of the system. The results of calculations have been presented 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 Figs. 9–13. The highest efficiency of organic Rankine cycle
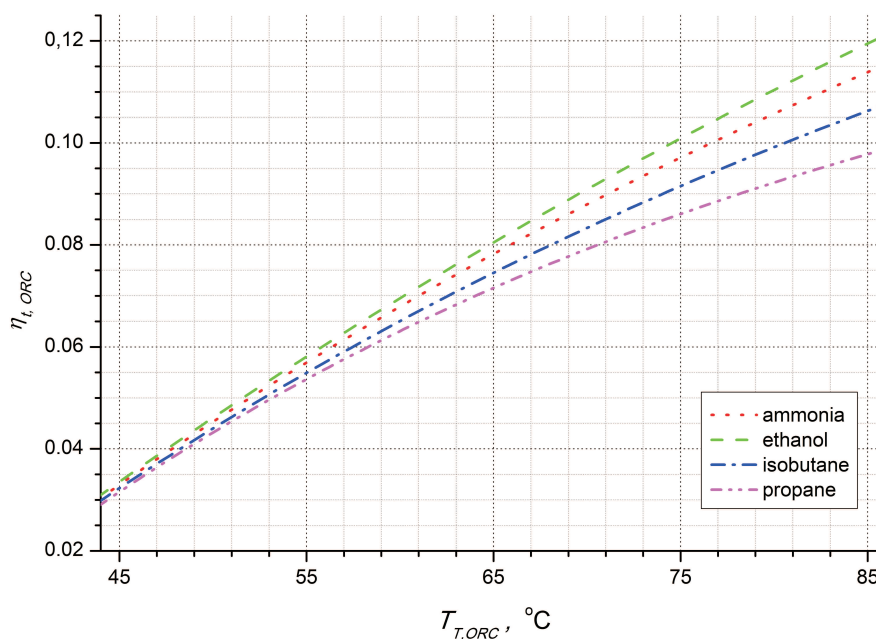

Figure 9: Efficiency of organic Rankine cycle, $\eta_{t,ORC}$, *vs.* temperature of vapour before turbine, $T_{T,ORC}$, and the type of low boiling point fluid.

under considered conditions is obtained in the case of ethanol and is equal to $\eta_{t,ORC} = 0.1195$ in specific saturation temperature $T_{T,ORC} = 85$ °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 the range of evaporation temperatures $T_{T,ORC}$ (hence temperature of vapour before turbine) are assumed between $T_{T,ORC} = 45$ °C and $T_{T,ORC} = 85$ °C. Corresponding values of pressure $p_{T,ORC}$ and the specific work of the ORC cycle, $l_{t,ORC}$, have been presented in Fig. 10. The effect of the increase of temperature of vapour before turbine, $T_{T,ORC}$, is clearly visible for the efficiency of organic Rankine cycle, $\eta_{t,ORC}$, (Fig. 9), pressure of vapour before turbine $p_{T,ORC}$ and the specific work of the ORC cycle, $l_{t,ORC}$, (Fig. 10) but, it should be mentioned, it is not identical for efficiency, specific work and pressure. The largest increase in efficiency – about 0.09 – occurs for temperature of vapour before turbine $T_{T,ORC} =$

85 °C. The largest pressure of vapour before turbine, $p_{T,ORC}$, occurs for temperature of vapour before turbine $T_{T,ORC} = 85$ °C, and for ammonia. It ought to be added that in the considered calculations the ammonia and ethanol seems to give much bigger the specific work of the ORC cycle, $l_{t,ORC}$, than isobutane and propane. In contrast to the above, the density of vapour at the outlet of ORC turbine, ρ_{ORC}, curves have an increasing or decreasing character depending on vapour quality, x, (Fig. 11). For $x = 1$ (isobutene) density of vapour at the outlet of ORC turbine, ρ_{ORC}, decreases with increase temperature of vapour before turbine, $T_{T,ORC}$, on the other hand for $x < 1$ density of vapour at the outlet of ORC turbine increases with increasing temperature of vapour before turbine. Values of volume flow rate at the outlet of ORC turbine, $\dot{V}_{Tout,ORC}$, increase due to an increase in the mass flux in ORC turbine. It ought to be noticed that the density of vapour at the outlet of ORC turbine and volume flow rate at the outlet of ORC turbine depend on parameters in the condenser of the organic Rankine cycle.
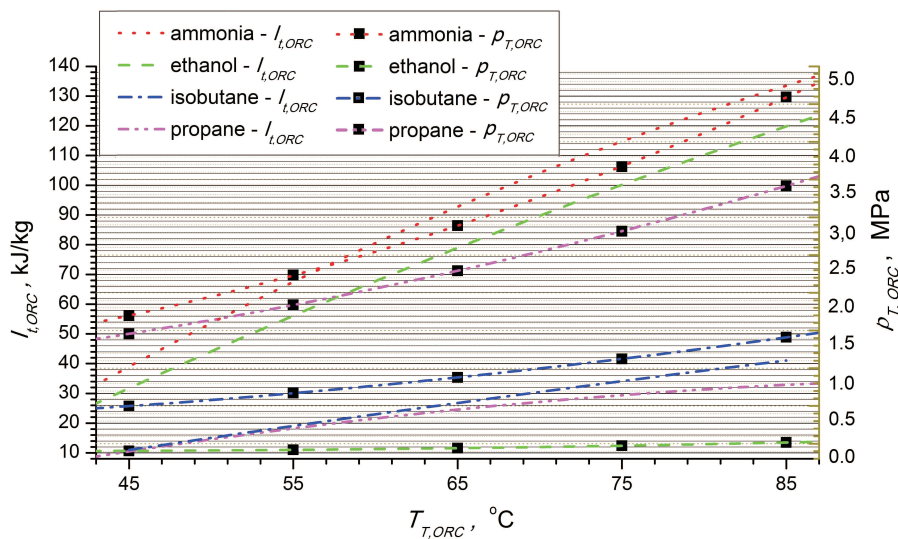

Figure 10: Pressure of vapour before turbine, $p_{T,ORC}$, and the specific work of the ORC cycle, $l_{t,ORC}$, *vs.* temperature of vapour before turbine, $T_{T,ORC}$, and the type of low boiling point fluid.

The results of calculations of the influence of Szewalski binary vapour cycle on efficiency, η_{re}, and power, N_{elRU}, of the whole system is presented in Figs. 12 and 13. It is interesting that the largest efficiency η_{re} are achieved
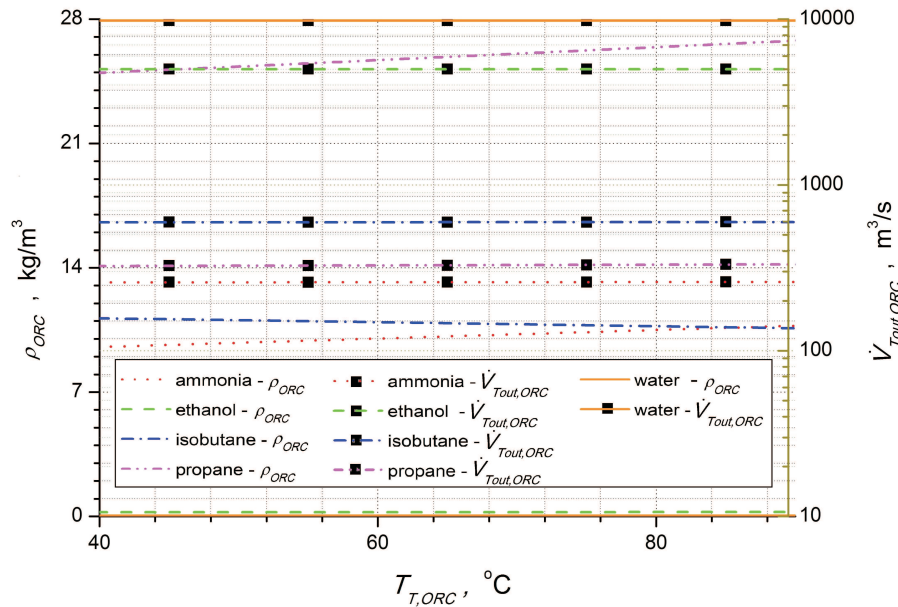
Figure 11: Density of vapour at the outlet of ORC turbine, ρ_{ORC}, and volume flow rate at the outlet of ORC turbine, $\dot{V}_{Tout,ORC}$, *vs.* temperature of vapour before turbine, $T_{T,ORC}$, and the type of low boiling point fluid.

for relatively low values of temperature of condensing water, $T_{con,W}$, or temperature of vapour before turbine. In the 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 equal $\Delta T = T_{con,W} - T_{T,ORC} = 5$ K. At the same time the level of condensate regeneration (HE8) in steam cycle is the same causing the constant value of temperature of water feeding the boiler to the level of $T_{43} = 310\,^\circ$C. It should be mentioned that the rate of heat supplied to the boiler equals $\dot{Q}_B = 1731.1$ MWt, in order to produce live steam with parameters presented in Tab. 1. As was mentioned the rate of heat supplied to the boiler was constant for all cases which have been investigated so the reference efficiency, η_{re}, depends only on power N_{elRU}. This was presented in Fig. 12 – curve of reference efficiency and power overlap each other.

As was mentioned before the modern steam turbine of large output should be divided into two parts, between the intermediate pressure and low pressure cylinders. Szewalski expected that this division into two parts should be at the pressure level of about 0.15 to 0.4 MPa, depending on the
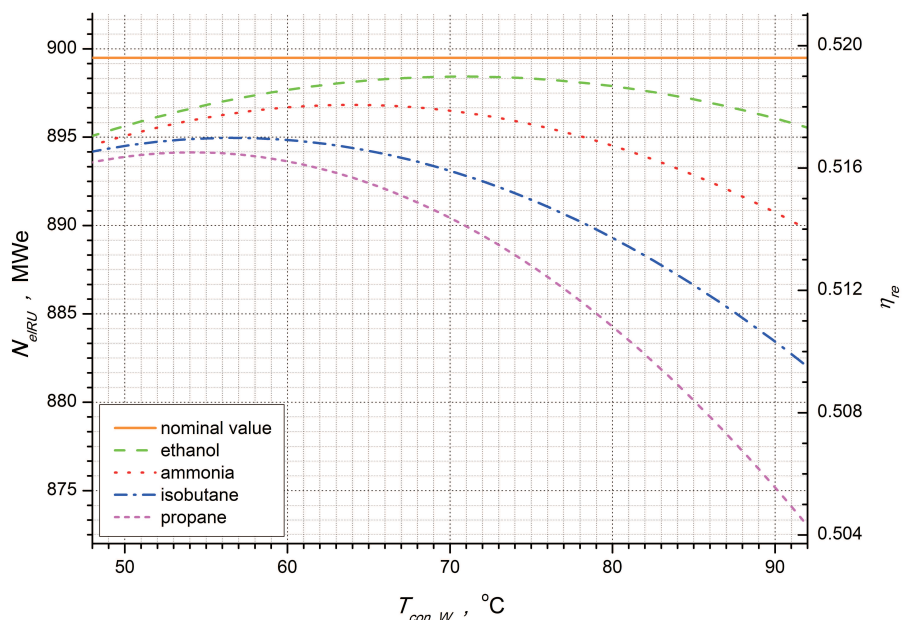
Figure 12: The power of the Szewalski binary vapour cycle, N_{elRU}, and the reference efficiency, η_{re}, vs. temperature of condensing water, $T_{con,W}$, and the type of low boiling point fluid. Comparison with nominal value of conventional steam cycle which presenting efficiency, η_{re}, and power, N_{elRU}, (continuous line) supercritical steam plant from Fig. 7.

type of the low boiling point fluid for application in the low temperature turbine. As it was shown in the analysis, the optimal value of temperature of condensing water for the split into two parts is for the case of ethanol 75 °C, – ammonia 65 °C, – isobutane 58 °C, – 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; –ammonia 0.025 MPa; – isobutane 0.0182 MPa; – propane 0.0158 MPa.

The calculation results clearly show that it is possible to found the optimal value of the process but due to irreversibilities in heat exchanger HE_{ORC} efficiency of the Szewalski cycle is lower than the nominal value of efficiency for the conventional steam cycle, which is equal to $\eta_{re} = 0.5196$. As well in Fig. 12 as in Fig. 13 the reference efficiency, η_{re}, was presented which has been defined by Eq. (16) for conventional steam cycle and by Eq. (22) for Szewalski binary vapour cycle. The highest netto efficiency of Szewalski binary vapour cycle at the level of $\eta_{re} = 0.5189$, and 0.5178 has

been estimated for ethane and ammonia respectively.

In Fig. 13 the correlation of gross electric power of the conventional steam part to the power of ORC (see Eq. (21)), N_{elGr}/N_{ORC}, in the Szewalski binary vapour cycle was presented. As it is shown, N_{elGr}/N_{ORC} correlates to the type of low boiling point fluid, so to split the conventional steam turbine between intermediate pressure and low pressure casing the investigation is needed with respect to the 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 can be used for heating of ORC liquid. Hence, the heat exchanger, HE_{ORC}, which will be acting as the condenser for water steam and at the same time as the evaporator of heated low boiling point fluid and won't be work as the heater ORC liquid.

It ought to be added that CFM type numerical tool gives a possibility to model combined two vapour cycles, which has recently been demonstrated in articles [9,27–31].
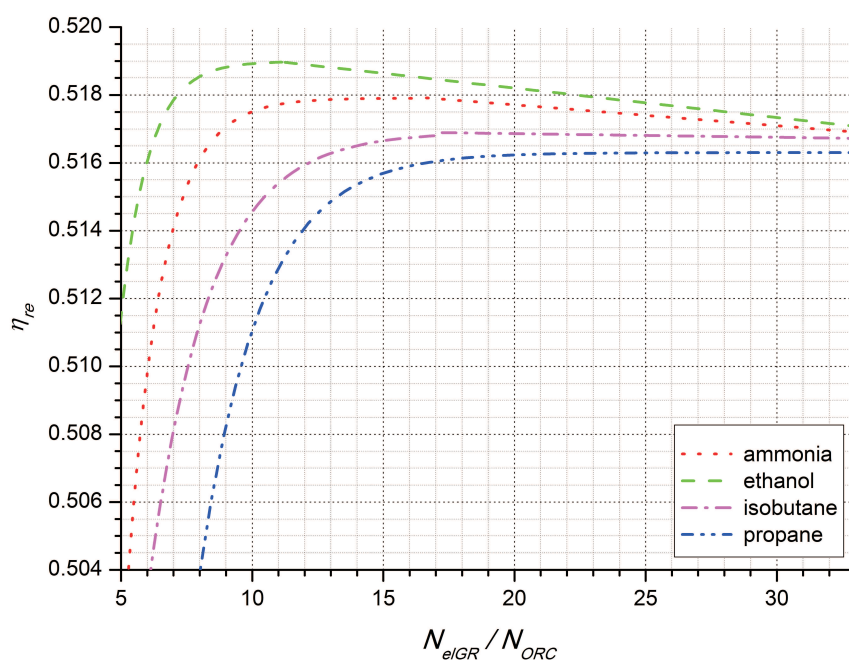

Figure 13: The reference efficiency, η_{re}, vs. N_{elGr}/N_{ORC}, and the type of low boiling point fluid.

7 Conclusions

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

As can be seen in all considered cases the highest ORC efficiency was obtained for the case of ethanol and is equal to $\eta_{t,ORC} = 0.1195$ in specific saturation temperature $T_{T,ORC} = 85\,^{\circ}\mathrm{C}$.

The highest netto efficiency of Szewalski binary vapour cycle at the level of $\eta_{re} = 0.5189$, and $\eta_{re} = 0.5178$ have been estimated. It should be added that in the considered calculations the ethanol and ammonia obtain the highest efficiency for Szewalski binary cycle. As it was shown in the thermodynamic analysis, the optimal value of temperature, $T_{con,W}$, for the turbine division into two parts is for ethanol 75 °C, for ammonia 65 °C, for isobutane 58 °C and 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; respectively.

The Szewalski cycle can be a good resolution for using heat flux received from the flue gases heat regeneration system. 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.

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