TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 140, 2018, 3–21

Bartosz Kraszewski*

On an improvement of Carnot-like cycles devoted to turbines with isothermal expansion

Energy Conversion Department, Institute of Fluid Flow Machinery
Polish Academy of Sciences, Fiszera 14, 80-231 Gdańsk, Poland

Abstract

This paper presents a novel, procedure of calculations of simple thermodynamic cycles with the main isothermal expansion process. Calculation assumptions are based on the use of technologically advanced flow devices. Firstly the theoretical Ericsson cycle with description of isothermal process has been presented. Then the calculations for open gas cycle with external combustion chamber, realising Ericsson cycle with upper temperature of 1473 K were carried out, showing the possibilities of turbines with isothermal expansion and achieving record-breaking efficiency. The use of isothermal compression has also been considered. The second, supercritical cycle with organic medium and condensation process also shows predispositions to achieve record-high efficiency at upper cycle temperature of 573 K. For both cycles, graphs of linearized thermodynamic transformations were made as well as graphs of efficiency dependence on the pressure. At the end, unit work of turbines in cycles has been compared and discussed.

Keywords: Isothermal turbine expansion; Ericsson cycle; Open gas cycle; Organic cycle

Nomenclature

h	–	unit enthalpy
l	–	unit work
p	–	pressure
q	–	heat flux
s	–	unit entropy
T	–	temperature

*E-mail adress: bkraszewski@imp.gda.pl

Greek symbols

η	–	efficiency
π	–	pressure ratio

Subscripts

1, 2, 3...	–	parameters of medium in points of the cycle no. 1, 2, 3...
B	–	boiler
c	–	compressor
C	–	Carnot
CC	–	combustion chamber
$cond$	–	condenser
F	–	filter
gen	–	generator
in	—	inlet
out	–	outlet
p	–	pump
$mech$	–	mechanical
reg	–	regenerator
S	–	isentropic value
T	–	isothermal value
t	–	turbine

1 Introduction

Contemporary requirements for power plants, including ecological restrictions and the need to reduce the consumption of fossil fuels, are growing every year. Thus, there is still a need of improving power plants cycles efficiency. As Robert Szewalski has noticed, it is mainly achieved by increasing upper cycle temperature, T_{upper}, or using bottoming cycles – there is no meaningful influence on the bottom temperature, T_{bottom}, of the cycle [1]. The maximum possible efficiency of the cycle in given temperature division is described by Carnot cycle efficiency

$$\eta_C = 1 - \frac{T_{bottom}}{T_{upper}} . \tag{1}$$

Significant gain of efficiency can be also accomplished by using regeneration or steam overheating in the Rankine cycle – this procedure leads to 'carnotisation' of the cycle. All these improvements have been applied in the Szewalski cycle, which was analyzed in detail by Ziółkowski *et al.* [2,3]. Two different proposals for the use of organic cycles as bottoming cycles have been analyzed also by Kowalczyk *et al.* [4] and Mikielewicz [5]. A practical insight into the structure and measurements of organic Rankine cycle (ORC) microturbines has been presented by Klonowich *et al.* [6], Rusanov *et al.* [7].

While pure Carnot cycle is only theoretical and comparative cycle, there are other methods to achieve Carnot-close efficiency. A general, modern insight into the optimization of thermal cycles based on the Carnot cycle was presented by Feidt and Blaise [8]. Several practical proposals have been presented by Igobo and Davies [9], including piston engines, turbines and expanders. Ericsson turbine cycle, which is one of generalised Carnot cycles, has been described in book edited by Kosowski [10]. In the implementation of Ericsson turbine cycle the key problems are isothermal processes taking place in it, but already in the 1960s, Konorski analyzed the solution of supplying heat to the steam turbine steering blades, which may prove to be crucial in the construction of such turbines [11]. Schaffel and Szlek presented the results of calculations of the open gas cycle with isothermal expansion [12].

The aim of the paper is to bring closer the advantages resulting from the use of isothermal turbine expansion process and also to compare the cycles configurations. While the turbine with isothermal expansion is currently under development in The Faculty of Mechanical Engineering of Gdańsk University of Technology [13], it is worth introducing the procedure for calculating such cycles and their possibilities. The assumptions used in calculations have been based on technologically advanced flow devices like high-efficiency heat regenerators. The application of isothermal turbine expansion has been considered in three variants: open gas cycle with isothermal compression, open gas cycle without isothermal compression and supercritical organic cycle.

2 Isothermal compression and expansion processes

The thermodynamic cycles based on the processes of isothermal expansion and isothermal compression are designed to bring as close as possible the efficiency of the designed cycle to the efficiency of Carnot cycle shown in Fig. 1a. Figure 1b presents the Ericsson cycle [10,14] in which the following transformations occur: isothermal expansion, isobaric cooling, isothermal compression, isobaric heat supply. Attempts to implement it were made mainly in the form of piston motors, as shown in Fig. 2 [15].

The following analysis uses the assumption of isothermal expansion in the turbine, which is a construction problem difficult to implement. However, compressors with isothermal compression are already under advanced development [16], which is being achieved by internal cooling of the compressed medium.

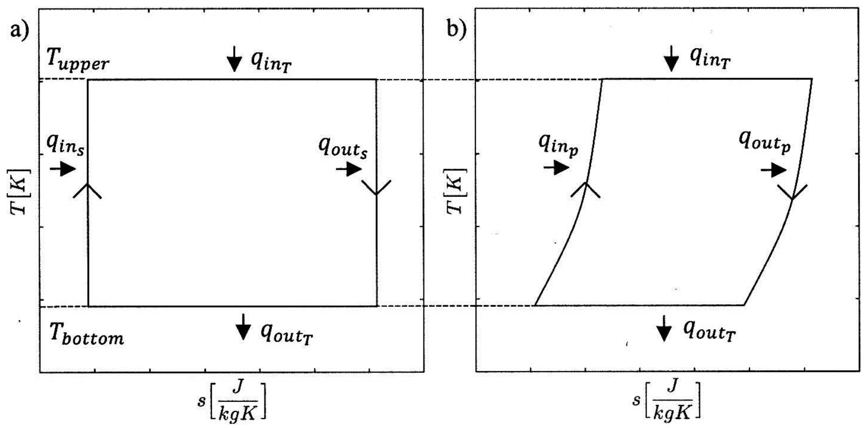

Figure 1: Thermodynamic transformation in temperature-entropy diagram: a) Carnot cycle, b) Ericsson cycle; $q_{in_{S/T/p}}$ – isentropic/isothermal/isobaric heat supply, $q_{out_{S/T/p}}$ – isentropic/isothermal/isobaric heat output.

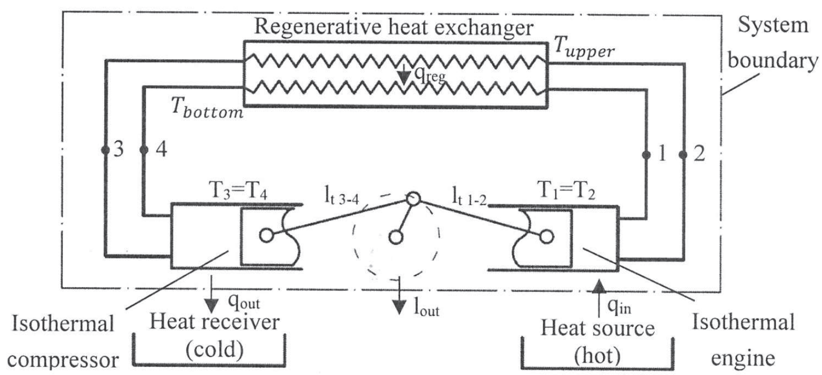

Figure 2: Scheme of Ericsson cycle realisation.

Basing on Fig. 3 and following equation [10,15]:

$$q_{1-2} = l_{t_{1-2}} + \triangle h_{2-1} \ , \tag{2}$$

where $\triangle h_{2-1}$ is unit enthalpy difference

$$\triangle h_{2-1} = h_2 - h_1 \ , \tag{3}$$

and assuming the heat of isothermal transformation as

$$q_{1-2} = \triangle sT = (s_2 - s_1)\, T_1 \ , \tag{4}$$

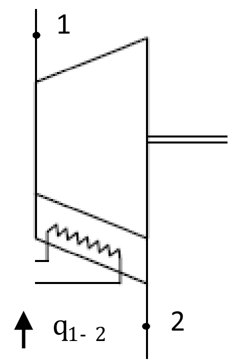

Figure 3: Scheme of turbine with isothermal expansion.

it is possible to define technical work on isothermal conversion in a turbine as

$$l_{t_{1-2}} = q_{1-2} - \triangle h_{2-1} \tag{5}$$

and with substitutions as

$$l_{t_{1-2}} = (s_2 - s_1)\,T - (h_2 - h_1)\ \ . \tag{6}$$

Analogously for the compressor

$$l_{t_{1-2}} = (s_1 - s_2)\,T - (h_1 - h_2)\,. \tag{7}$$

3 Open gas cycle with external combustion chamber

The realisation of the Ericsson cycle by gas turbine will be discussed first. The application of an external combustion chamber increases the flexibility in the area of used fuels and the thermodynamic stability of the parameters of the working medium, which will be air. The assumed upper cycle temperature of 1200 °C is easily reachable in combustion chambers, but difficult to achieve in turbine with additional heat supply – the usage of superalloys [17] and 3D print technology [18] should be considered. Furthermore, there are no premises of problem related to the chemical stability of air at this temperature [19,20]. Figure 4 shows an open gas cycle with an external combustion chamber and single regeneration. It also includes cooling the compressor and supplying heat to the turbine for isothermal transformations. The assumptions shown in Tab. 1 were used for the calculations of the cycle in the scheme below [14,21].

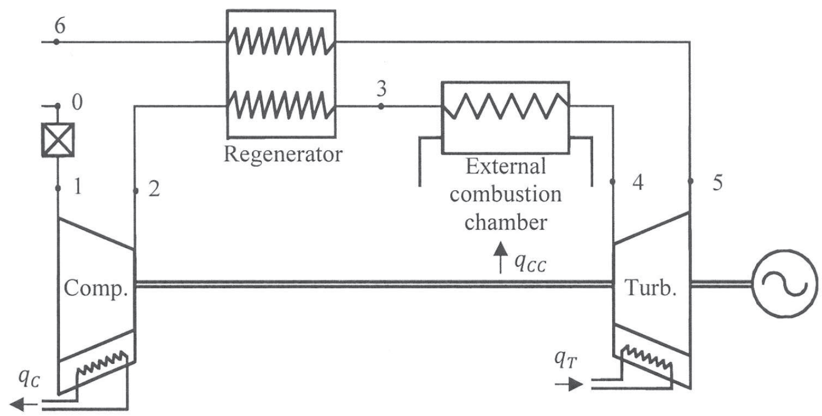

Figure 4: Open gas cycle with external combustion chamber: q_c – compressor outlet heat flux, q_{cc} – combustion chamber inlet heat flux, q_t – turbine inlet heat flux.

Table 1: Assumptions for the calculations of the open gas cycle.

Property	Symbol	Value	Unit
Atmospheric pressure	p_0	101325	Pa
Air temperature at compressor inlet	T_1	288	K
Air temperature at turbine inlet	T_4	1473	K
Final temperature difference in the regenerator	$\triangle T_{6-2}$	10	K
Pressure drop in the regenerator	$\triangle p_{reg_{\%}}$	2	%
Pressure drop in the combustion chamber	$\triangle p_{CC}$	4	%
pressure drop in the air filter	$\triangle p_F$	2	%
Compressor isentropic efficiency	η_C	90	%
Combustion chamber efficiency	η_{CC}	95	%
Mechanical efficiency	η_{mech}	99	%
Generator efficiency	η_{gen}	98	%

The tabular properties of air (and later for organic MM medium, hexamethyldisloxane) has been taken from CoolProp libraries through Matlab interface [25,26]. The energy demand for auxiliary equipment in this stage has been omitted.

At point No. 1 the pressure has been reduced by the loss in the filter

$$p_1 = p_0 \left(1 - \triangle p_F\right) \quad , \tag{8}$$

where p_0 is the atmospheric pressure, temperature T_1 has been assumed, and the enthalpy and entropy of the factor have been read from the tables:

$$h_1 = h\left(p_1, T_1\right) \ , \qquad s_1 = s\left(p_1, T_1\right) \ . \tag{9}$$

The pressure at point no. 2 is proportional to the pressure p_1

$$p_2 = p_1 \, \pi \tag{10}$$

and the temperature (assuming isothermal compression) does not change

$$T_2 = T_1 \ . \tag{11}$$

Enthalpy and entropy have been read from the pressure and temperature function:

$$h_2 = h\left(p_2, T_2\right) \ , \qquad s_2 = s\left(p_2, T_2\right) \ . \tag{12}$$

The enthalpy difference in compressor is equal to

$$\triangle h_{2-1} = h_2 - h_1 \ , \tag{13}$$

thus giving the isothermal compression work equal to

$$l_{1-2} = T_1 \ \left(s_1 - s_2\right) - \triangle h_{2-1} \ . \tag{14}$$

The pressure at point no. 3 has been reduced by the pressure loss in the regenerator

$$p_3 = \left(1 - \triangle p_{reg}\right) \, p_2 \ , \tag{15}$$

allowing to determine the pressure in point no. 4, reduced in turn by the pressure drop in the combustion chamber

$$p_4 = \left(1 - \triangle p_{CC}\right) \, p_3 \ . \tag{16}$$

Further, taking into account the assumed temperature T_4, the values of enthalpy and entropy have been read from the tables:

$$h_4 = h\left(p_4, T_4\right) \ , \qquad s_4 = s\left(p_4, T_4\right) \ . \tag{17}$$

At point no. 5, taking the cycle outlet pressure as atmospheric

$$p_6 = p_0 \ , \tag{18}$$

the pressure at regenerator outlet has been determined as greater than the cycle outlet pressure by the resistances in the regenerator

$$p_5 = \frac{p_6}{1 - \triangle p_{reg}} \tag{19}$$

and the temperature is constant due to the assumed isothermal expansion

$$T_5 = T_4 \ , \tag{20}$$

which allows reading enthalpy and entropy values from tables:

$$h_5 = h\left(p_5, T_5\right) \ , \qquad s_5 = s\left(p_5, T_5\right) \ . \tag{21}$$

Now, enthalpy difference in turbine is equal to

$$\triangle h_{5-4} = h_5 - h_4 \ , \tag{22}$$

thus giving the isothermal expansion work equal to

$$l_{4-5} = T_4 \ (s_5 - s_4) - \triangle h_{5-4} \ . \tag{23}$$

At point no. 6 the pressure p_6 has been assumed as atmospheric and the temperature as 10 K higher than the temperature after the compressor

$$T_6 = T_2 + \triangle T_{6-2} \ , \tag{24}$$

thus giving tabular values of enthalpy and entropy:

$$h_6 = h\left(p_6, T_6\right) \ , \qquad s_6 = s\left(p_6, T_6\right) \ . \tag{25}$$

To obtain enthalpy at point no. 3, the regenerator balance should be performed:

$$h_2 + h_5 = h_3 + h_6 \, , \qquad h_3 = h_2 + h_5 - h_6 \ , \tag{26}$$

which allows us to read temperature and entropy tabular values:

$$T_3 {=} T\left(p_3, h_3\right) \ , \qquad s_3 {=} s\left(p_3, h_3\right) \ . \tag{27}$$

The unit heat supplied to the circulation can be separated into heat supplied in the combustion chamber and for heating the turbine

$$q_{in} = q_{CC} + q_t \ , \tag{28}$$

which in turn can be written as

$$q_{in} = \frac{1}{\eta_{CC}} = \Big[(h_4 - h_3) + (s_5 - s_4) \ T_4 \Big] \ . \tag{29}$$

The net efficiency of the cycle has been determined as the ratio of the turbine work reduced by the compressor work to the heat supplied, including the generator and mechanical efficiency:

$$\eta_{net} = \frac{l_{4-5} - l_{1-2}}{q_{in}} \eta_{gen} \, \eta_{mech} \ . \tag{30}$$

Therefore, the temperature difference between the outlet of the heating medium and the inlet of the heated medium was monitored:

$$\triangle T_{5-3} = T_5 - T_3 \ . \tag{31}$$

A circuit without compressor cooling was also considered. In this case, the unit work of the compressor was calculated from the definition of compression isentropic efficiency:

$$h_{2\mathrm{s}} = h \left(p_2 , \ s_1 \right) , \tag{32}$$

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1} \, , \tag{33}$$

$$h_2 = \frac{1}{\eta_C} \ (h_{2s} - h_1) + h_1 \, , \tag{34}$$

$$l_{1-2} = h_2 - h_1 \ . \tag{35}$$

For comparison, Figs. 5 and 6 contain linear thermodynamic transformations for both variants of the cycle in the form of temperature-entropy (T-s) graphs. Table 2 presents the results for both variants.

Table 2: Assumptions for the calculations of the open gas cycle.

Parameter	Unit	Value	
		with compressor cooling	without compressor cooling
$\triangle h_{21}$	$\mathrm{J\,kg^{-1}}$	-2182.31	122738.99
$\triangle h_{54}$	$\mathrm{J\,kg^{-1}}$	-701.98	-157.11
$\triangle p_{F}-$	–	0.02	0.02
$\triangle p_{CC}$	–	0.04	0.04
$\triangle p_{reg}$	–	0.02	0.02
$\triangle T_{53}$	K	10.74	8.73
$\triangle T_{62}$	K	10.00	10.00
η_{gen}	–	0.98	0.98
η_{CC}	–	0.95	0.95
η_{mech}	–	0.99	0.99
η_{net}	–	0.72	0.65
η_{C}	–	–	0.90
h_1	$\mathrm{J\,kg^{-1}}$	414379.56	414379.56
h_2	$\mathrm{J\,kg^{-1}}$	412197.25	537118.55
h_{2s}	$\mathrm{J\,kg^{-1}}$	–	524844.65
h_3	$\mathrm{J\,kg^{-1}}$	1717663.41	1719523.67
h_4	$\mathrm{J\,kg^{-1}}$	1730604.18	1730059.31
h_5	$\mathrm{J\,kg^{-1}}$	1729902.20	1729902.20
h_6	$\mathrm{J\,kg^{-1}}$	424436.04	547497.08
l_{12}	$\mathrm{J\,kg^{-1}}$	194517.23	122738.99
l_{45}	$\mathrm{J\,kg^{-1}}$	931607.67	435722.73
p_0	Pa	101325.00	101325.00
p_1	Pa	99298.50	99298.50
p_2	Pa	992985.00	307825.35
p_3	Pa	973125.30	301668.84
p_4	Pa	934200.29	289602.09
p_5	Pa	103392.86	103392.86
p_6	Pa	101325.00	101325.00
π	–	10.00	3.10
q_{in}	$\mathrm{J\,kg^{-1}}$	993522.59	469580.27
s_1	$\mathrm{J\,(kg\,K)^{-1}}$	3851.98	3851.98
s_2	$\mathrm{J\,(kg\,K)^{-1}}$	3184.50	3882.37
s_3	$\mathrm{J\,(kg\,K)^{-1}}$	4950.58	5288.49
s_4	$\mathrm{J\,(kg\,K)^{-1}}$	4971.14	5307.39
s_5	$\mathrm{J\,(kg\,K)^{-1}}$	5603.06	5603.06
s_6	$\mathrm{J\,(kg\,K)^{-1}}$	3880.49	4226.48
T_1	K	288.15	288.15
T_2	K	288.15	409.94
T_3	K	1462.41	1464.42
T_4	K	1473.15	1473.15
T_5	K	1473.15	1473.15
T_6	K	298.15	419.94

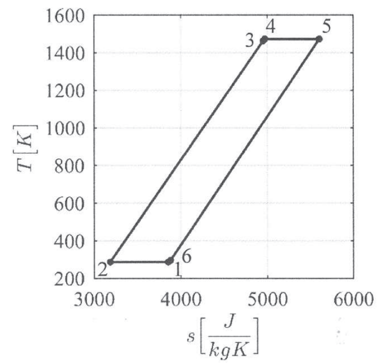

Figure 5: Linear T-s diagram for the variant with compressor cooling.

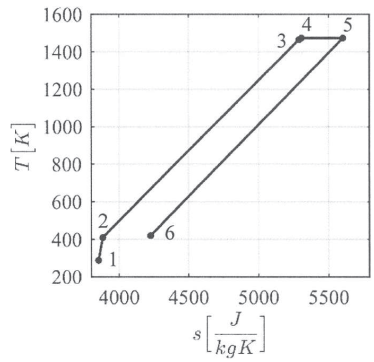

Figure 6: Linear T-s diagram for the variant without compressor cooling.

4 Cycle with an organic working medium

In the next step calculations for the cycle with isothermal expansion based on the organic Rankine cycle with one regenerator were made. As the working medium, low-boiling siloxane MM, described in Tab. 3, has been used. The properties of this organic fluid allow for advantageous performance of the turbine cycle in the set temperature range [22].

Table 3: Proporties of the MM organic medium [23].

ASHRAE[1] shortcut	Medium name	Chemical formula	Critical temperature	Critical pressure	Atmospheric press boiling point (P_{atm})
MM	hexamethyldisiloxane	$C_6H_{18}OSi_2$	518.69 K	1.939 MPa	448.65 K

[1] ASHRAE – American Society of Heating, Refrigerating and Air-Conditioning Engineers

In the cycle shown in Fig. 7, due to the condensation of the medium, a condenser and a conventional pump are required. The primary assumptions used for the calculations, extracted from the monographs [14,21,23] have been presented in Tab. 4. At point no. 1, the steam pressure p_1 has been accepted iteratively for the highest efficiency of the cycle. Therefore the temperature T_1 is assumed and enthalpy and entropy were read from the pressure and temperature function:

$$h_1 = h\left(p_1, T_1\right) , \qquad s_1 = s\left(p_1, T_1\right) . \tag{36}$$

Table 4: Assumptions for the calculations of the organic cycle.

Property	Symbol	Value	Unit
Upper cycle temperature	T_1	573	K
Condensation temperature	T_4	300	K
Final temperature difference in the regenerator	$\triangle T_{3-5}$	10	K
Pressure drop in the regenerator	$\triangle p_{reg}$	2	%
Pressure drop in the boiler	$\triangle p_B$	4	%
Pressure drop in the condenser	$\triangle p_{cond}$	5	%
Pump isentropic efficiency	η_P	80	%
Boiler efficiency	η_B	95	%
Mechanical efficiency	η_{mech}	99	%
Generator efficiency	η_{gen}	98	%

At point no. 4 the condensing temperature T_4 has also been assumed and the pressure has been defined as the saturation pressure for the assumed temperature

$$p_4{=}p_n\left(T_4\right)\ . \tag{37}$$

Enthalpy and entropy has thus been read from the pressure and temperature function:

$$h_4{=}h\left(p_4,T_4\right)\ ,\quad s_4{=}s\left(p_4,T_4\right)\ . \tag{38}$$

The pressure at point no. 5 is higher by some losses in the boiler and in the regenerator in relation to the required steam pressure:

$$p_6=\frac{p_1}{1-\triangle p_B}\,,\qquad p_5=\frac{p_6}{1-\triangle p_{reg}}\ , \tag{39}$$

whereas enthalpy results from the definition of pump efficiency – the ratio of isentropic enthalpy growth to real growth:

$$\eta_P{=}\frac{h_{5s}-\ h_4}{h_5-\ h_4}\,,\qquad h_5{=}\ \frac{1}{\eta_P}\ \left(h_{5s}-\ h_4\right)+h_4\ , \tag{40}$$

where the enthalpy of isentropic compression has been read from the pressure and entropy function

$$h_{5s}=h\left(p_5,\ s_4\right) \tag{41}$$

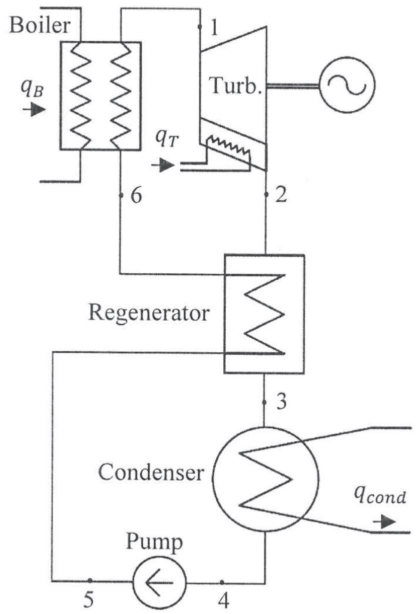

Figure 7: Closed cycle with organic medium: q_B – boiler inlet heat flux, q_T – turbine inlet heat flux, q_{cond} – condenser outlet heat flux.

and similarly temperature and entropy also have been read from the appropriate pressure and enthalpy functions:

$$T_5 = T\left(p_5, h_5\right)\ , \quad s_5 = s\left(p_5, h_5\right)\ . \tag{42}$$

Notice that the pressure at point no. 3 is increased relative to the saturation pressure for the condensation temperature, due to losses occurring in the condenser

$$p_3 = \frac{p_4}{1 - \triangle p_{cond}}\ , \tag{43}$$

the temperature at the outlet of the heat exchanger has also been assumed to be higher than the inlet temperature of the heating medium by the difference of 10 K

$$T_3 = T_5 + \triangle T_{3-5}\ , \tag{44}$$

and enthalpy and entropy have simultaneously been read from the pressure and temperature functions:

$$h_3 = h\left(p_3,\, T_3\right)\ , \qquad s_3 = s\left(p_3, T_3\right)\ . \tag{45}$$

Now, at point no. 2 the pressure is increased due to losses occurring in the regenerator in relation to the pressure before the condenser

$$p_2 = \frac{p_3}{1 - \triangle p_{cond}} \ , \tag{46}$$

temperature results from the assumed isothermal expansion

$$T_2 = T_1, \tag{47}$$

and enthalpy and entropy have been read from the pressure and temperature function:

$$h_2 = T\left(p_2, T_2\right) \ , \qquad s_2 = s\left(p_2,\, T_2\right) \ . \tag{48}$$

Further, he enthalpy difference in the expansion equals

$$\triangle h_{2-1} = h_2 - h_1 \ , \tag{49}$$

thus giving the turbine work to be

$$l_{1-2} = T_1\left(s_2 - s_1\right) - \triangle h_{2-1} \ . \tag{50}$$

Yet, in point no. 6 the pressure has already been determined, while the enthalpy has been calculated from the thermal balance of the heat exchanger

$$h_6 = h_2 + h_5 - h_3 \tag{51}$$

and temperature and entropy has been read from the pressure and enthalpy function:

$$T_6 = T\left(p_6, h_6\right) \ , \qquad s_6 = s\left(p_6, h_6\right) \ . \tag{52}$$

In turn, unit heat supplied to the cycle has been calculated as

$$q_{in} = \frac{1}{\eta_B}\Big[\left(h_4 - h_3\right) + \left(s_5 - s_4\right) T_4\Big] \ . \tag{53}$$

The net efficiency of the cycle has been determined as the ratio of the turbine work reduced by the pump work, l_p, to the heat supplied, taking into account the efficiency of the generator, η_{mech}, and the mechanical one

$$\eta_{net} = \frac{l_{1-2} - l_p}{q_{in}} \, \eta_{gen} \, \eta_{mech} \ . \tag{54}$$

The temperature difference between the outlet of the heating medium and the inlet of the heated medium must be monitored by

$$\triangle T_{2-6} = T_2 - T_6 \ . \tag{55}$$

The results of calculations are presented in Tab. 5. The above calculation procedure is consistent with the classic design line of Robert Szewalski's heat circuits. Figure 8 presents linearized thermodynamic transformations on the T-s graph in combination with the medium saturation curve [1].

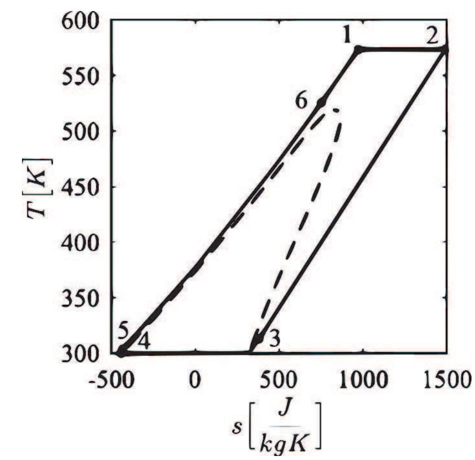

Figure 8: Temperature diagram depending on entropy for the organic cycle with isothermal expansion (solid line) in combination with the medium saturation curve (dashed line).

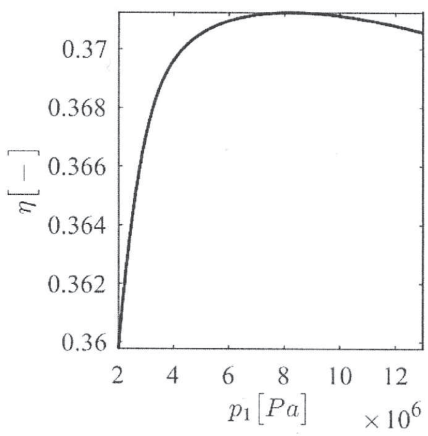

Figure 9: The dependence of the efficiency of the cycle on the fresh steam pressure.

5 Conclusions

As shown in the above calculations, the application of the isothermal expansion process in a turbine can bring tangible benefits in the form of record-high efficiency with relatively simple power plant systems. For an open air gas system with an external combustion chamber and a regenerator, the efficiency of 72% has been achieved with compressor cooling (isothermal compression) and 65% without compressor cooling. While for the first variant the optimum compression ratio was 3.1, for the second variant it has been limited to 10 – the thermodynamically optimal pressure equals much more, but with a slight increase in efficiency, as shown in Fig. 10. Such a large difference in optimal compression ratios results from both isothermal compression and expansion processes – in the case of compressor cooling, increasing the compression ratio increases the technical work of the compressor, resulting in a proportionally increased technical work of the tur-

Table 5: Assumptions for the calculations of the organic cycle.

Parameter	Unit	Value
$\triangle h_{2-1}$	$\mathrm{J\,kg^{-1}}$	114702.65
$\triangle p_{CC}$	–	0.04
$\triangle p_{cond}$	–	0.05
$\triangle p_{reg}$	–	0.02
$\triangle T_{2-6}$	K	47.55
$\triangle T_{5-3}$	K	10.00
η_{gen}	–	0.98
η_B	–	0.95
η_{mech}	–	0.99
η_{net}	–	0.37
η_P	–	0.80
h_1	$\mathrm{J\,kg^{-1}}$	470387.17
h_2	$\mathrm{J\,kg^{-1}}$	585089.82
h_3	$\mathrm{J\,kg^{-1}}$	100310.17
h_4	$\mathrm{J\,kg^{-1}}$	-148278.20
h_5	$\mathrm{J\,kg^{-1}}$	-134878.10
h_{5s}	$\mathrm{J\,kg^{-1}}$	-137558.12
h_6	$\mathrm{J\,kg^{-1}}$	349901.55
l_{12}	$\mathrm{J\,kg^{-1}}$	181065.16
l_P	$\mathrm{J\,kg^{-1}}$	13400.10
p_1	Pa	7700000.00
p_2	Pa	6506.61
p_3	Pa	6376.48
p_4	Pa	6057.65
p_5	Pa	8184523.81
p_6	Pa	8020833.33
q_{in}	$\mathrm{J\,kg^{-1}}$	438161.50
s_1	$\mathrm{J\,(kg\,K)^{-1}}$	973.66
s_2	$\mathrm{J\,(kg\,K)^{-1}}$	1489.70
s_3	$\mathrm{J\,(kg\,K)^{-1}}$	384.19
s_4	$\mathrm{J\,(kg\,K)^{-1}}$	-440.88
s_5	$\mathrm{J\,(kg\,K)^{-1}}$	-432.03
s_6	$\mathrm{J\,(kg\,K)^{-1}}$	753.08
T_1	K	573.15
T_2	K	573.15
T_3	K	313.46
T_4	K	300.00
T_5	K	303.46
T_6	K	525.60

bine (widening horizontally the T-s chart – the upper pressure increases while bottom isobar stays the same).

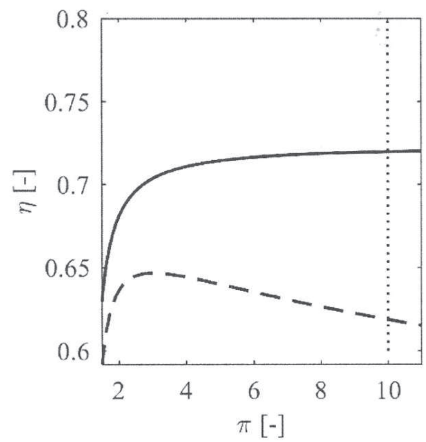

Figure 10: Dependence of the efficiency from compression ratio for open cycles; continous line – isothermal compression, dashed line – non-isothermal compresion, dotted line – assumed maximum.

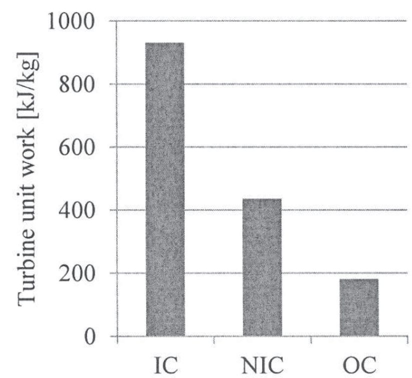

Figure 11: Listing of unit work of turbines with isothermal expansion; IC – isothermal compression, NIC – non-isothermal compression, OC – organic cycle.

The system of a steam power plant with a low boiling organic medium MM with one regenerator and an external combustion chamber, due to application of the isothermal expansion in the turbine allows us ,to achieve efficiencies of 37% at a relatively low temperature. The optimal fresh steam pressure was approx. 7.7 MPa, as shown in Fig. 8. It is worth paying attention to the adopted upper cycle temperature of 593.15 K which is the approx. temperature of chemical stability of the MM medium [24].

Due to the very high regeneration ratio and assumed low temperature pinch points in both cycles, the use of high efficiency heat exchangers should be taken into account. In the variants considered, a turbine unit work was compared as shown in Fig. 11. The criterion of a minimal decrease in the unit enthalpy of the turbine stage is not reliable at isothermal expansion – an individual approach in the design of stages is required. The analysis did not take into account the supply of auxiliary equipment, however, pumps operation at such an intense heat exchange can have a very significant impact on the final efficiency of the power plant.

Received in March 2018

References

[1] Szewalski R.: *Current problems in the development of energy technology.* PAS, Warszawa 1978.

[2] Ziółkowski P., Kowalczyk T., Hernet. J, Kornet S.: *The thermodynamic analysis of the Szewalski hierarchic vapour cycle cooperating with a system of waste heat recovery.* Trans. Inst. Fluid-Flow Mach. **129**(2015), 51–75.

[3] Ziółkowski P., Kowalczyk T., Kornet S., Badur J.: *On low-grade waste heat utilization from a supercritical steam powerplant using an ORC-bottoming cycle coupled with two sources of heat.* Energ. Convers. Manage. **146**(2017), 158–173.

[4] Kowalczyk T., Ziółkowski P, Badur J.: *Exergy losses in the Szewalski binary vapor cycle.* Entropy **17**(2015), 7242–7265.

[5] Mikielewicz J. *Utilisation of bleed steam from power plant to increase saturation temperature in organic Rankine cycle.* Trans. Inst. Fluid-Flow Mach. **126**(2014) 21–32.

[6] Klonowicz P., Borsukiewicz-Gozdur A., Hanausek P., Kryłłowicz W., Brüggemann D.: *Design and performance measurements of an organic vapour turbine.* Appl. Thermal Energ. **63**(2014), 297–303.

[7] Rusanov R., Jędrzejewski Ł., Klonowicz P., Żywica G., Lampart P., Rusanov A.: *Design and performance study of a small-scale waste heat recovery turbine.* Trans. Inst. Fluid-Flow Mach. **133**(2016), 145–162.

[8] Feidt M., Blaise M.: *A new three objectives criterion to optimize thermomechanical engines model.* In: Proc. 1st Int. e-Conf. on Energies, 2014.

[9] Igobo I.N., Davies P.A.: *Review of low-temperature vapour power cycle engineswith quasi-isothermal expansion.* Energy **70**(2014), 22–34.

[10] Kosowski K., Łuniewicz B., Obrzut B.: *Ericsson Cycle.* In: Steam and Gas Turbines, 2nd Edn., (K. Kosowski, Ed.). ALSTOM Power LLC, Elbląg 2007.

[11] Konorski A.: *Steam drying in condensing turbines by internal heating using own steam.* Trans. Inst. Fluid-Flow Mach. **11-12**(1962), 63–120.

[12] Schaffel N., Szlek A.: *Thermodynamical analysis of isothermal gas turbine cycles.* Arch. Energ. **37**(2007), 1-2, 171–180.

[13] Kosowski K., Cieślński A., Piwowarski M., Stępień R., Włodarski W.: *CHP cycles with reciprocating engines and microturbines.* In: Proc. 12th RDPE Conf., Warsaw 2015.

[14] Goliński J.A., Jesionek K.J.: *Air and Steam Power Plants.* Maszyny Przepływowe: Vol. 31, Wyd. IMP PAN, Gdańsk 2009 (in Polish).

[15] Pudlik W.: *Thermodynamics*, (3rd Edn.). Gdańsk University of Technology. Gdańsk 2011 (in Polish).

[16] Jin Young Heo, Min Seok Kim, Seungjoon Baik, Seong Jun Bae, Jeong Ik Lee: *Thermodynamic study of supercritical CO_2 Brayton cycle using an isothermal compressor.* Appl. Energy **206**(2017), 1118–1130.

[17] Harada H.: *High temperature materials for gas turbines: The present and future.* Proc. Int. Gas Turbine Cong. 2003 Tokyo, November 2–7, 2003.

[18] Siebert M.: *Breakthrough with 3D printed gas turbine blades.* Siemens, https://www.siemens.com (access 20 Dec. 2017).

[19] Bauer E.: *Physics of high-temperature air. Part 1, Basics.* Inst. Def. Anal. USA, 1990.

[20] Capitelli M., Colonna G., Gorse C., D'Angola A.: *Transport properties of high temperature air in local thermodynamic equilibrium.* Eur. Phys. J. **11**(2000), 279–289.

[21] Perycz S.: *Steam and Gas Turbines.* Ossolineum, Wrocław 1992 (in Polish).

[22] Kraszewski B.: *Thermodynamic analysis of high efficiency turbine cycles with organic media.* Eng. thesis, Gdańsk University of Technology, Gdańsk 2015 (in Polish).

[23] Piwowarski M.: *Selection of the Working Medium and Favorable Values of Basic Microturbine Design Parameters for Energy Conversion from Low Temperature Sources.* Wyd. IMP PAN, Gdańsk 2013 (in Polish).

[24] Preissinger M., Brüggemann D.: *Thermal stability of hexamethyldisiloxane (MM) for high-temperature organic Rankine cycles (ORC).* Energies **9**(2016), 3, 183–194.

[25] http://www.coolprop.org

[26] https://www.mathworks.com/products/matlab.html