The efficiency of transcritical CO₂ cycle near critical point and with high temperature

Sindu Daniarta^{1,2*}, Attila R. Imre^{2,3}, and Piotr Kolasiński¹

¹Department of Thermodynamics and Renewable Energy Sources, Wrocław University of Science and Technology, Wybrzeże Wyspiańskiego 27, 50-370 Wrocław, Poland.

²Department of Energy Engineering, Faculty of Mechanical Engineering, Budapest University of Technology and Economics, Műegyetem rkp. 3, H-1111 Budapest, Hungary

³Centre for Energy Research, Department of Thermohydraulics, POB. 49, H-1525 Budapest, Hungary

Abstract. Efficiency is a key parameter used to assess the quality of operation of power generation systems and devices applied for converting one type of energy to the other. Although, in the end, an investment project is mainly evaluated by economic aspect. Furthermore, many researchers have been investigating the possible types of energy conversion systems and devices applied for power generation and utilizing different types of working fluids. This paper presents the inside into transcritical carbon dioxide (CO₂) cycle and the gradients of its efficiency. Transcritical CO₂ cycle (TCO₂C) here refers to a CO₂-based thermal power generation cycle absorbing heat from a heat source (ideally with constant pressure) till the supercritical state is reached. It is followed by an expansion to a sub-critical superheated or even two-phase (wet) state. As alternatives, trilateral flash cycle (TFC) and organic Rankine cycle (ORC) utilizing CO₂ are also introduced in this paper. The calculation in this study is computed based on MATLAB integrated with thermophysical properties like CoolProp and REFPROP, the mathematical models of the system are built and calculated with the same heat sink temperature of 224.41 K, and the heat source temperature is varied between 274.41 K and 500 K. At a certain temperature, the obtained result shows that the efficiency of the TCO₂C is lower than the efficiency of ORC. Another result proves that the quality of working fluid at the end of the expansion process significantly influences the efficiency of the cycle.

1 Introduction

Negative environmental impacts (assessed, for example, by ozone depletion potential (ODP) and global warming potential (GWP)), are crucial issues met during the selection of working fluid in modern power plants and other energy conversion systems which operating principle is based on the implementation of thermodynamic cycles (such as e.g., refrigeration or air-conditioning system). ODP is a measure of the degradation of the ozone layer by given fluid compared to reference working fluid, i.e., R-11 (trichlorofluoromethane). GWP is a parameter to measure contribution to global warming at a period of time (compared to carbon

* Corresponding author: sindu.daniarta@pwr.edu.pl

dioxide). Regarding these issues, many working fluids such as chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) are not considered as working fluid candidates in any process for further works. Some of them have been already or are currently phased out and forbidden to use by the Montreal and Kyoto Protocols, or the Kigali Amendment [1]. Accordingly, there is an increasing interest observed in research related to natural working fluids like alkanes, alkenes, or carbon dioxide (CO₂) which could be applied as working fluids in many processes and technologies [1 – 3].

Among natural working fluids, CO₂ is particularly attractive as it is a non-toxic, non-corrosive, inflammable, non-explosive substance, additionally featuring good thermophysical properties. This working fluid is abundant and affordable. Moreover, it might also be environmental-friendly (i.e., its ODP is equal to zero while GWP is equal to 1) compared to other refrigerant working fluids. All these reasons have been driving many researchers to proceed with scientific works on the physical properties of CO₂ and its potential application as a working fluid in thermodynamic cycles, like refrigerating systems [3, 4] and power plants (i.e., organic Rankine cycle (ORC), trilateral flash cycle (TFC), transcritical CO₂ cycle (TCO₂C), and supercritical CO₂ Brayton cycle) [5 – 10].

One of the frequently used energy conversion cycles, the ORC, seems to be a promising way for power generation compared to the conventional steam Rankine cycle because it utilizes organic working fluid instead of water. Therefore, it could be installed to utilize heat sources with various operating temperature ranges, even below 100 °C. As an alternative method, TCO₂C could be used. It refers to a CO₂-based thermal power generation cycle absorbing heat from a heat source (ideally with constant pressure) till the supercritical state is reached, followed by an expansion to a sub-critical superheated or even two-phase (wet) state. There are two important differences between ORC and TCO₂C; i.e., the working fluid is different (ORC uses an organic working fluid, while TCO₂C uses CO₂) and the isobaric heating step is proceeded in a different way (with evaporation for ORC and without evaporation for TCO₂C).

Several studies have been conducted to show how working fluid should be selected to obtain a favourable result. Comparison between ORC and TCO₂C [11] shows that TCO₂C has better economic performance. A comprehensive study about a comparison between Rankine cycle, transcritical cycle and combined cycle with the source temperature up to 500 °C was presented in a study [12] and reported that TCO₂C has a good performance when the temperature of heat sources ranges between 500 – 600 °C. Another comparative study between TCO₂C with Kalina cycle and ORC were presented in a study [13] and reported that TCO₂C outperforms Kalina and ORC for low-temperature heat sources.

Since many researchers believe that TCO₂C outperforms these other cycles, in this study, we will present the trendline of efficiency focusing on TCO₂C. As a comparison, ORC and TFC are also described in this article to understand the trendline of efficiency and the transition among the cycles. In this study, the temperature of heat sources is ranging between 274.41 K and 500 K. Obtained modelling results proved that under certain conditions, ORC – against all expectations - could outperform TCO₂C.

2 Systems, models and mathematical description

TCO₂C has a basic layout that is very similar to TFC and ORC, but the course of thermodynamic processes in the cycle is different. The working fluid in the transcritical cycle absorbs heat from heat sources under constant pressure (above the critical pressure) to reach a supercritical state (above the critical temperature) and then is expanded in the expander to superheated or even wet conditions (i.e., two-phase). For TCO₂C, TFC and ORC, the thermodynamic parameters of the working fluid in the endpoint of the heating (which is at the same time the initial point of expansion) differ. For TCO₂C, it is a supercritical fluid state

(neither liquid nor vapour), for TFC, it is a saturated liquid state, while for ORC, it is a saturated vapour state. The comparison between TFC, ORC, partially evaporated ORC (PE-ORC), superheated ORC, and TCO2C in terms of system layout and the course of thermodynamic processes (depicted in T - s diagrams) could be seen in Figure 1.

The difference between TFC and ORC showed in Figure 1 could also be indicated with the striped box area (visualizing the heat input) in the T - s diagram. This striped box means that ORC needs more heat to evaporate the working fluid, which is then expanded, while in TFC, that area (indicated in this case by dotted area), visualizes heat input needed only to flash the working fluid. It seems that the ORC could be interpreted as the sum of a TFC (dotted area) and a virtual Carnot cycle (stripped area). Being the Carnot cycle the thermodynamic cycle featuring the highest possible to obtain efficiency within the given temperature range, one could assume that ORC is more efficient than TFC. According to some recent studies, this is true only for cycles using wet or moderately dry working fluids, including CO₂ [14, 15].

Sometimes, the intermittent heat sources (e.g., solar thermal power, ocean thermal energy, waste heat, etc.) are found to be utilized for ORC power plant feeding, and this intermittent condition could significantly affect the thermodynamic process and operating conditions of the system. It is often found that the thermal power of these intermittent heat sources is not sufficient to evaporate the working fluid (i.e., the working fluid is not fully evaporated). Therefore, the expansion process starts from this partially evaporated or wet state (i.e., two-phase state or the condition where the working fluid quality is in the range of $0 < x_2 < 1$). It seems that this might lead to the serious issue for the performance of the expansion process and working conditions of the expander (e.g., the erosion of the turbine blade caused by droplet or corrosion of the machine's subassemblies). This process so-called PE-ORC (partially evaporated ORC) refers to Figures 1(c) and 1(e).

Furthermore, to avoid this serious issue on the expander caused by the liquid droplets, some engineers and scientists tend to configure the ORC system by installing an additional heat exchanger (superheater) to heat up the fluid. Therefore, the working fluid at the outlet of this heat exchanger reaches the superheated condition. Then, this superheated working fluid (in the dry state) could be directly expanded. This process is described as a superheated ORC and its layout and T - s diagram could be seen in Figures 1(f) and 1(g), respectively.

As above mentioned, this study focuses on the TCO₂C that the layout and T - s diagram could be seen in Figures 1(h) and 1(i) respectively. Typically, the schematic layout of this process is similar to TFC (see Figures 1(a) and 1(h)) but the difference is the course of the thermodynamic processes (see Figures 1(b) and 1(i)). The working fluid in TCO₂C starts to be expanded in the transcritical, superheated or wet state. Since the expansion process proceeds from supercritical to superheated or subcritical state. Therefore, this process is a so-called transcritical cycle. In TCO₂C, only a heat exchanger is installed to change the liquid to supercritical fluid (see Figure 1(i)).

In this study, the cycle utilizes CO₂ as a working fluid, and the model is built and calculated using MATLAB. Thermophysical properties of the working fluid are obtained from CoolProp [16]. The temperature of the condenser is set constant at 224.41 K, and for the evaporator, the temperature is varied in the range of 274.41–500 K. In this study, the condenser is set below 0 °C to see the possibility of wider performance of CO₂ based power cycle. It is possible to have the heat sink at this condition by utilization of cold energy, such as the cooling potential from a regasification system of liquefied natural gas (LNG, which is needed to regasify from around -160 °C to 30 °C). Some articles have discussed the utilization of this cold energy as power generation [10, 17 – 20].

Fig. 1. The schematic layout and the course of the thermodynamic process in T - s diagram of (a – b) TFC, (c – d) ORC, (c – e) PE-ORC, and (f – g) superheated ORC, and (h – i) TCO₂C.

Fig. 2. The steps of modelling calculation in the T - s diagram for subcritical (a) TFC and (b) ORC

Fig. 3. The step of modelling calculation from initial calculation (1) to (i) depicted in the T - s diagram for ideal PE-ORC (purple lines), ORC (green lines), superheated ORC (blue lines), and TCO2C (red lines) where the condensation starts with the vapour quality of (a) $x_3 = 0.56$, (b) $x_3 = 0.60$, (c) $x_3 = 0.70$, (d) $x_3 = 0.80$, (e) $x_3 = 0.90$, and (f) $x_3 = 1.00$. For each figure, the method introduces the temperature increase in beginning of expansion from 2(1) to 2(i) with steps of 0.001 K and the conditions at the inlet and the outlet of the condenser are kept at the same, $3(1) = 3(i)$ and $4(1) = 4(i)$. The outlet pumps are varied from 1(1) to 1(i) based on the increase of desired pressure from 2(1) to 2(i).

In the presented study, the cycle is considered ideal to study the effect caused purely by the CO2. It means that the processes in the pump and expander are considered as isentropic and these proceeding in the evaporator and condenser are considered as ideal isobaric (without pressure losses). For the subcritical ORC and TFC, the model (see Figure 2) is developed only below the saturated curve, the so-called subcritical cycle.

Figure 2 illustrates the steps of modelling proceeded for fixed condenser temperature (224.41 K). The starting state for expansion for TFCs follows the saturated liquid curve, and for ORCs, it follows the saturated vapour curve until reaching a critical point. To obtain the

precise results, the temperature step for these TFC and ORC models is configured as 0.0001 K as there are small slopes near the critical point. Figure 2 also shows that the pump and the expander are modelled as isentropic. This method is similar to the methods used in studies [14, 15] for understanding the low and maximal efficiencies of ORC near the critical point.

Figure 3 shows the steps of modelling proceeded for PE-ORC, ORC, superheated ORC, and TCO2C where the vapour quality (i.e., the ratio of the mass of vapour to the total mass of working fluid) at the end of the expansion (x_3) is varied from 0.56 to 1.00. The TCO2C with $x_3 = 0.56$ (see Figure 3(a)) is developed to see the efficiency obtained in the very close vicinity of the critical point (due to the elevated density fluctuations in this region [21], crossing the critical point is technically almost impossible). The temperature of the condenser is fixed at 224.41 K, and the starting temperature of the expansion varies between 274.41 K and 500 K. Only the cycles formed below critical pressure are considered as TFC, PE-ORC, and ORC.

The boundary condition and calculating steps of this model are summarized in Table 1. The mathematical models calculated in MATLAB are described in Equations 1 – 7 for each step of the cycles. The efficiency of the cycles could be defined using Equations 8 – 12.

$$\Delta h_{\text{pre}} = (h_{\text{out,pre}} - h_{\text{in,pre}}) = \frac{\dot{Q}_{\text{pre}}}{\dot{m}_{\text{total}}} \quad (1)$$

$$\Delta h_{\text{eva}} = (h_{\text{out,eva}} - h_{\text{in,eva}}) = \frac{\dot{Q}_{\text{eva}}}{\dot{m}_{\text{total}}} \quad (2)$$

$$\Delta h_{\text{HE,sup}} = (h_{\text{out,HE,sup}} - h_{\text{in,HE,sup}}) = \frac{\dot{Q}_{\text{HE,sup}}}{\dot{m}_{\text{total}}} \quad (3)$$

$$\Delta h_{\text{HE,crit}} = (h_{\text{out,HE,crit}} - h_{\text{in,HE,crit}}) = \frac{\dot{Q}_{\text{HE,crit}}}{\dot{m}_{\text{total}}} \quad (4)$$

$$\Delta h_{\text{ex}} = (h_{\text{in,ex}} - h_{\text{out,ex}}) \eta_{\text{is,ex}} = \frac{\dot{W}_{\text{ex}}}{\dot{m}_{\text{total}}} \quad (5)$$

$$\Delta h_{\text{cds}} = (h_{\text{in,cds}} - h_{\text{out,cds}}) = \frac{\dot{Q}_{\text{cds}}}{\dot{m}_{\text{total}}} \quad (6)$$

$$\Delta h_{\text{pmp}} = (h_{\text{out,is,pmp}} - h_{\text{in,pmp}}) / \eta_{\text{is,pmp}} = \frac{\dot{W}_{\text{pmp}}}{\dot{m}_{\text{total}}} \quad (7)$$

Where h , $\dot{m}$, $\dot{Q}$, and $\dot{W}$, are specific enthalpy, mass flow rate, heat transfer rate, and power, respectively. The subscript of HE, pre, eva, sup, crit, ex, cds, pmp, in, and out represent heat exchanger, pre-heater, evaporator, superheater, critical, expander, condenser, pump, inlet, and outlet, respectively.

$$\eta_{\text{TFC}} = \frac{\dot{W}_{\text{ex}} - \dot{W}_{\text{pmp}}}{\dot{Q}_{\text{pre}}} \quad (8)$$

$$\eta_{\text{ORC}} = \frac{\dot{W}_{\text{ex}} - \dot{W}_{\text{pmp}}}{\dot{Q}_{\text{pre}} + \dot{Q}_{\text{eva}}} \quad (9)$$

$$\eta_{\text{ORC,sup}} = \frac{\dot{W}_{\text{ex}} - \dot{W}_{\text{pmp}}}{\dot{Q}_{\text{pre}} + \dot{Q}_{\text{eva}} + \dot{Q}_{\text{HE,sup}}} \tag{10}$$

$$\eta_{\text{TCO2C}} = \frac{\dot{W}_{\text{ex}} - \dot{W}_{\text{pmp}}}{\dot{Q}_{\text{HE,crit}}} \tag{11}$$

Table 1. The boundary condition and calculating steps of the model using CO2 as a working fluid with the temperature at the inlet to the expander varied between 274.41 K and 500 K.

The cycle	The process in T-s diagram	T_2 (K)	T_3 (K)	x_3 (-)	Steps for $T_{in,ex}$ (K)
TFC	Figure 2(a)	$274.41 \leq T_2 \leq T_{crit}$	224.41	0.28 – 0.56	0.0001
ORC	Figure 2(b)	$274.41 \leq T_2 \leq T_{crit}$	224.41	0.56 – 0.83	0.0001
PE-ORC	Figure 3(a)	$274.41 \leq T_2 < T_{crit}$	224.41	0.56	0.01
ORC	Figure 3(a)	T_{crit}	224.41	0.56	-
TCO2C	Figure 3(a)	$T_{crit} < T_2 \leq 500$	224.41	0.56	0.01
PE-ORC	Figure 3(b)	$274.41 \leq T_2 < 303.94$	224.41	0.60	0.01
ORC	Figure 3(b)	303.94	224.41	0.60	-
Superheated ORC	Figure 3(b)	$303.94 < T_2 \leq 304.15$	224.41	0.60	0.01
TCO2C	Figure 3(b)	$304.15 < T_2 \leq 500$	224.41	0.60	0.01
PE-ORC	Figure 3(c)	$274.41 \leq T_2 < 298.59$	224.41	0.70	0.01
ORC	Figure 3(c)	298.59	224.41	0.70	-
Superheated ORC	Figure 3(c)	$298.59 < T_2 \leq 306.90$	224.41	0.70	0.01
TCO2C	Figure 3(c)	$306.90 < T_2 \leq 500$	224.41	0.70	0.01
PE-ORC	Figure 3(d)	$274.41 \leq T_2 < 281.73$	224.41	0.08	0.01
ORC	Figure 3(d)	281.73	224.41	0.08	-
Superheated ORC	Figure 3(d)	$281.73 < T_2 \leq 321.06$	224.41	0.08	0.01
TCO2C	Figure 3(d)	$321.06 < T_2 \leq 500$	224.41	0.08	0.01
Superheated ORC	Figure 3(e)	$274.41 \leq T_2 \leq 350.31$	224.41	0.09	0.01
TCO2C	Figure 3(e)	$350.31 < T_2 \leq 500$	224.41	0.09	0.01
Superheated ORC	Figure 3(f)	$274.41 \leq T_2 \leq 393.00$	224.41	1.00	0.01
TCO2C	Figure 3(f)	$393.00 < T_2 \leq 500$	224.41	1.00	0.01

Where η is the efficiency of the cycle. The formulas (see Equations 7 – 10) to calculate the efficiency are typically similar for all cycles, representing the ratio of power generation divided by the heat input. For the TFC, the heat input source is only the liquid pre-heater; for

a transcritical cycle, the situation is almost the same, although a pseudo-phase transition (pseudo-boiling) could happen, but it differs from traditional evaporation to a great extent [22, 23]. In the ORC there are two possibilities of the expansion depending on the expansion starting parameters. If the expanded working fluid is in a saturated vapour state, the heating is provided by a liquid pre-heater and an evaporator. If the expanded working fluid is dry (superheated) vapour, a third heat exchanger (superheater) is also needed (see Equation 10).

3 Results and discussion

The obtained modelling results are illustrated in the form of a cycle efficiency-expander inlet temperature diagram (i.e., the working fluid temperature at the expander inlet) which could be seen in Figure 4. Different x_3 values represent the vapour quality at the end of the expansion.

Figure 4 shows that the efficiency of the ORC and TFC are similar to the ones presented in Refs. [14, 15] for the subcritical cycles. However, the data is more precise near the critical point as the temperature steps are small, $\Delta T_{in,ex} = 0.0001$ K. This diagram also illustrates that there is maximal efficiency for ORC at a certain temperature. Moreover, for TFC, the maximal efficiency is found at the critical temperature.

Fig. 4. The area where TCO2C have lower efficiency than maximal ORC under subcritical. The marks in this figure refer to Figure 3.

Some marks (A, B, C, D, E, and F) in Figure 4 refer to the marks in Figure 3 that indicates the transition of one process to the other. For example, the efficiency curve of the simulated model from D-1-0 to D-1-1 ($274.41 \leq T_2 < 281.73$) in Figure 4 is obtained based on PE-ORC with $x_3 = 0.80$ referring to Figure 1(d) that the condenser is at a constant temperature

of 224.41 K. When the expansion starts from the temperature of 281.73 K (see Figure 1(d)), the process is ORC as expansion starts with initial vapour quality of $x_2=1$. Furthermore, at the temperature range of $281.73 < T_2 \leq 321.06$, the process will be superheated ORC as an additional heat exchanger is needed to heat up the working fluid until it reaches a superheated state. The differences between PE-ORC, ORC, and superheated ORC are described by the evaporation process and superheating. Using the same method in Figure 1(d), the process will be TCO2C ($T_2 > 321.06$) that is seen in Figure 4 (indicated with blue dot lines).

The modelled result also found that at a certain temperature, the efficiency of TCO2C is lower than ORC (maximal efficiency) as is indicated with the cyan area highlighted in Figure 4. It is plausible to expect that the maximal efficiency of ORC should play a key role in thermal power plant design. Moreover, the transcritical cycles going close to the critical point should be avoided because of the low efficiency (compared to ORC maximal efficiency) and uncertainties in supercritical conditions.

Fig. 5. The cycle efficiency-expander inlet temperature diagram valid for the temperature range of 344 – 422 K. The marks E-1-1 and F-1-1 refer to Figures 3(e) and 3(f).

The position of the curves in Figure 4 illustrates that the superheated ORC cycle might not be a good choice to utilize low-temperature heat sources because of the efficiency, economic aspect, etc. However, many researchers tend to design the ORC with a superheater to avoid the serious issue caused by the liquid droplets that could damage the turbine blade, and cause erosion. The selection of proper expanders has to be taken into full consideration because it plays a significant role in the choice of layout of the cycle and the design of thermodynamic processes. Nowadays, volumetric expanders tend to be a promising technology to be employed in low-temperature power systems [24]. The experimental test proved that this type of expander could handle the wet condition. These expanders offer a simple design, low level of noise and vibrations, low rotational speed and compactness.

Considering these devices as prime movers might be favourable for ORC and other energy conversion systems in the future.

Fig. 6. The cycle efficiency-expander inlet temperature diagram, the mark refers to Figure 3.

The cycle efficiency-expander inlet temperature diagram plotted for the working fluid temperature range of 344 K – 420 K is illustrated in Figure 5. It seems that all efficiencies are changing smoothly. However, many crossings are visible in the diagram. It means that depending on the maximal cycle temperature, one could always find a cycle, which could outperform the others and the best cycle could differ for different temperatures. For example, below the temperature of ca. 386 K, the efficiency of the transcritical cycle with $x_3=0.80$ outperforms the others. However, above the temperature of 386 K, the transcritical cycle with the wet expansion $x_3=0.56$ has the best efficiency. Due to the special shape of CO₂ isobars in the supercritical region, crossing of efficiency-lines (representing different x -values) exist.

The full results of this study could be seen in Figure 6 where the efficiency of superheated ORCs with $x_3=1.00$ at the end of the expansion process is illustrated. It is reported that the efficiency of this cycle is the lowest among the reported ones.

4 Conclusion

In this paper, the trendline of the efficiency for TFC, ORC, PE-ORC, superheated ORC, and TCO₂C has been introduced and presented by using a diagram of cycle efficiency-expander inlet temperature. This study could be used as consideration in the preliminary design stage of CO₂ based thermal power plant. It is reported that transcritical cycles with maximal cycle temperature close to the critical one should be avoided because they have lower efficiency than the maximal efficiency of subcritical ORC with smaller maximal cycle temperature (see Figure 4). At a certain temperature range of inlet expansion, TCO₂C with the expansion

process ending at $x_3=0.56$ (see Figure 6), $x_3=0.70$ (see Figures 4 and 6), and $x_3=0.80$ (see Figures 5 and 6) outperform the other types of the cycles reported in this study. Since there are many types of expanders (turbines and volumetric expanders), the results of the present study could be also used during the selection of proper expanders that have to be taken into consideration to obtain the desired operating conditions of the power plant.

Since the condenser in this study was set at 224.41 K, it is possible to have the heat sink below 0 °C by utilizing the cold energy from LNG or other refrigerant fluids. The utilization of this cold energy as power generation is good in the sustainable point of view.

Part of the research reported in this paper and carried out at BME has been supported by the NRDI Fund (TKP2020 NC, Grant No. BME-NC) based on the charter of bolster issued by the NRDI Office under the auspices of the Ministry for Innovation and Technology.

References

1. B.K. Sovacool, S. Griffiths, J. Kim, and M. Bazilian, *Renew. Sustain. Energy Rev.* **141**, 110759 (2021).
2. A. Chesi, F. Esposito, G. Ferrara, and L. Ferrari, *Appl. Energy* **135**, 274 (2014).
3. B. Dai, S. Liu, K. Zhu, Z. Sun, and Y. Ma, *Energy* **122**, 787 (2017).
4. A. Yu, W. Su, X. Lin, N. Zhou, and L. Zhao, *Energy Convers. Manag.* **221**, 113214 (2020).
5. S. Khatoon, S. Ishaque, and M.-H. Kim, *Energy Convers. Manag.* **232**, 113895 (2021).
6. R. Chen, M. Romero, J. González-Aguilar, F. Rovense, Z. Rao, and S. Liao, *Energy Convers. Manag.* **232**, 113870 (2021).
7. W. Su, Y. Hwang, S. Deng, L. Zhao, and D. Zhao, *Energy Convers. Manag.* **165**, 720 (2018).
8. L. Pan, B. Li, W. Shi, and X. Wei, *Appl. Energy* **253**, 113608 (2019).
9. D. Zhao, R. Zhao, S. Deng, L. Zhao, and M. Chen, *Energy Convers. Manag.* **226**, 113578 (2020).
10. S. Daniarta and A.R. Imre, *Period. Polytech. Mech. Eng.* (2020).
11. M. Li, J. Wang, S. Li, X. Wang, W. He, and Y. Dai, *Geothermics* **50**, 101 (2014).
12. C. Li and H. Wang, *Appl. Energy* **180**, 707 (2016).
13. F. Meng, E. Wang, B. Zhang, F. Zhang, and C. Zhao, *Energy Convers. Manag.* **195**, 1295 (2019).
14. A. Mohammed Ahmed and A.R. Imre, *Pollack Period. Pollack* (2021).
15. A.M. Ahmed, L. Kondor, and A.R. Imre, *Energies* **14**, (2021).
16. I.H. Bell, J. Wronski, S. Quoilin, and V. Lemort, *Ind. Eng. Chem. Res.* **53**, 2498 (2014).
17. I.-H. Choi, S. Lee, Y. Seo, and D. Chang, *Energy* **61**, 179 (2013).
18. O. Koku, S. Perry, and J.-K. Kim, *Appl. Energy* **114**, 250 (2014).
19. R.F. García, J.C. Carril, J.R. Gomez, and M.R. Gomez, *Energy* **105**, 16 (2016).
20. B.B. Kanbur, L. Xiang, S. Dubey, F.H. Choo, and F. Duan, *Renew. Sustain. Energy Rev.* **79**, 1171 (2017).
21. A.R. Imre, A. Groniewsky, G. Györke, A. Katona, and D. Velmovszki, *Period. Polytech. Chem. Eng.* **63**, 276 (2019).

22. D.T. Banuti, J. Supercrit. Fluids **98**, 12 (2015).
23. A.R. Imre, C. Ramboz, U.K. Deiters, and T. Kraska, Environ. Earth Sci. **73**, 4373 (2015).
24. P. Kolański, *Układy Organic Rankine Cycle (ORC) Małej i Średniej Mocy Wykorzystujące Rozprężarki Wyporowe* (Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław, 2021).