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Comparative analysis of effeciency of waste heat conversion in low-temperature Brayton cycle

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The paper discusses the feasibility, effectiveness and validity of a gas turbine power plant, operated according to the Brayton comparative cycle in order to develop low-potential waste heat (160°C) and convert it into electricity. Fourteen working fluids, mainly with organic origin have been examined. It can be concluded that low molecular weight working fluids allow to obtain higher power efficiency of Brayton cycle only if conversions without taking into account internal losses are considered. For the cycle that takes into account the compression conversion efficiency in the compressor and expansion in the gas turbine, the highest efficiency was obtained for the perfluoropentane working medium and other substances with relatively high molecular weight values. However, even for the cycle using internal heat recovery, the thermal efficiency of the Brayton cycle did not exceed 7%.

Keywords: Brayton cycle, working fluid, low temperature source, low temperature power plant

1. INTRODUCTION

The new energy policy of Poland by 2030 is aimed at improving energy efficiency, increasing energy security, developing renewable energy sources, including biofuels, developing competitive fuel and energy markets and reducing the impact of the generation of useful forms of energy on the environment. The basic directions of this policy correspond thematically with the main objectives of EU energy policy 3 × 20%.

The first point in the country's energy policy – improving energy efficiency – comprises multidirectional measures and one of them is to reduce emissions of waste energy, whose production is accompanied by all processes of energy conversion. Waste management should be considered on two main areas: technical possibility and economic viability.

At present, there are several methods of managing waste heat, for example: one described by Ling Bing et al. (2014) thermoelectric generators or one described by Saidur et al. (2012) turbocharger, through the use of various types of Stirling engines – presented in the paper (Sim et al., 2017), beta type or in a work (Bulinski et al., 2018) alpha type. Among the most popular and advanced technologies for waste heat utilisation are the vapour cycles: ORC power plants operated by subcritical Rankine cycle (Aboelwafa et al., 2018; In Seop et al., 2016; Mikielewicz and Mikielewicz, 2016; Sornek and Filipowicz, 2016; Wang et al., 2018) or according to the comparative cycle for supercritical parameters presented, e.g., by Landelle et al. (2017) and Li et al. (2015).

A separate group consists of, at the early stage of research, power plants with zeotropic fluids presented e.g. by Borsukiewicz (2017) or by Bamorovat Abadi (2015). The above mentioned vapour cycles incorporate

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two isentropic conversions and two isobaric ones, and if the conversions are implemented within a certain distance from the critical point, then such a solution is called the Brayton (Joule) gas cycle. Comparison of selected vapour and gas cycles, assuming that they are implemented at the same initial temperatures of the upper (T_{w1}) and the lower heat source (T_{c1}), are shown in Fig. 1.

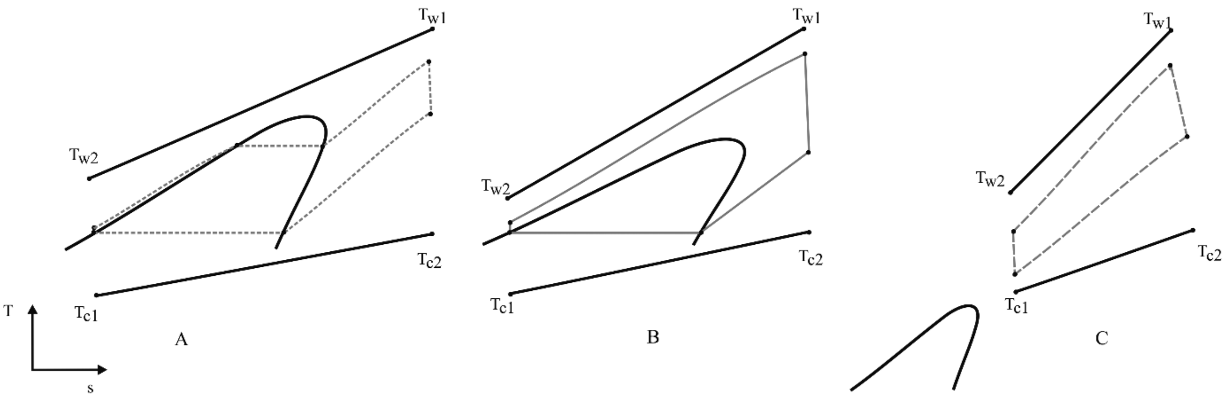

Fig. 1. Illustrative representation of cycles: A – vapour, subcritical, B – vapour, supercritical, C – single-phase vapour (gas) implemented at the same initial temperatures of the upper (T_{w1}) and the lower heat source (T_{c1})

A review and comparison of the efficiency of various types of thermodynamic cycles available for medium temperature power plants was presented by Dunham and Iverson (2014). The paper analysed the possibility of applying the Rankine steam cycle, two-cycles (supercritical CO_2 and ORC) and the Brayton cycle (with and without heat regeneration) when solar energy is the driving source. Yoonhan et al. (2015) provided an overview of the literature in the field of the Bryton cycle working on the CO_2 working fluid. However, this work concerns the use of the system when powered from relatively high-temperature energy sources – with the idea of using the cycle in nuclear power plants, see also Li Ch et al. (2015). In the work (Bianchi et al., 2000) the increase in the efficiency of the cogeneration system by applying a reverse Brayton cycle was proposed, in this work the analysed energy source is exhaust gas stream at temperatures between 400 and 600°C.

As the use of power plants operated according to the comparative Brayton (Joule) cycle – in the case when the source of the driving energy is not fuel burned in the combustion chamber but other low or medium temperature sources – is not a well-known issue, that is why this study presents the results of a theoretical analysis of the comparative efficiency of the power plant working according to the Brayton cycle powered by a waste heat source at 160°C operating at various working fluids.

2. CALCULATION MODEL AND ASSUMPTIONS FOR CALCULATIONS

Technology of electricity production using the high-temperature gas turbine Brayton cycle, especially when the driving fuel is natural gas, has been well known and successfully used for years. That is why an installation diagram of both open and closed cycles will not be shown here. In the context of this study the analysis of the thermodynamic efficiency of the low-temperature, theoretical Brayton cycle without regeneration and with internal heat regeneration has been conducted, assuming that it is supplied from the heat source of the initial temperature $T_{w1} = 160$°C. The energy carrier is water at elevated pressure in order to avoid the process of water evaporation, with a mass flow of 10 kg/s. This way of energy supply is called an open source, as described in more detail in Borsukiewicz-Gozdur (2013) and Huixing et al. (2016) and is of particular importance in the performed thermodynamic analysis, as explained below. Figure 2 shows the cycles of thermodynamic conversions for the Brayton cycle without heat recovery (Fig. 2A) and with internal heat recovery (Fig. 2B), which is possible to carry out when the temperature of the working medium at the turbine outlet (T_{b4}) is higher than the temperature at the end of the compression process (T_{b2}). It should be emphasised that in the case of supplying the cycle with the heat flow carrier (and not with the fuel burned in the combustion chamber), the use of internal heat recovery increases the efficiency of the cycle while it does not affect the reduction of fuel consumption but only increases the final temperature of the heat carrier (T_{w2}), as shown in Fig. 2B. The calculations assume that the minimum temperature difference between the working medium and the heat carrier would be $\Delta T = 10$ K and in this way the highest temperature of the gaseous fluid at the turbine inlet T_{b3} is 150°C.

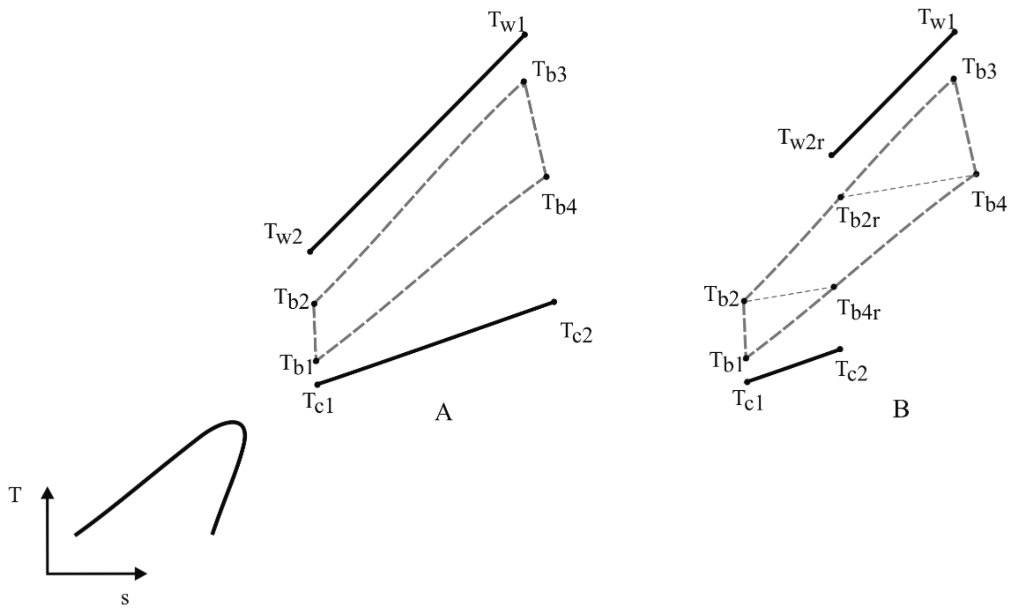

Fig. 2. Thermodynamic processes of the Brayton cycle A) without heat regeneration; B) with internal heat regeneration

Fourteen substances of different origins have been chosen for the calculation: organic and inorganic (also noble gas), having different critical temperatures (but the value of the critical temperature should not exceed 130°C), with different molecular weights. The basic parameters of working fluids are summarised in Table 1.

It was assumed that the lower pressure in the Brayton cycle is the same for all the fluids and it is $p_{b1} = 0.2$ MPa, whereas the upper pressure p_{b3} was determined from Equation (1):

$$\xi = \frac{p_{b3}}{p_{b1}} \tag{1}$$

where ξ is the pressure ratio. Within the scope of this work the calculations for pressure ratios in the range from 1.25 to 5 have been performed.

In order to carry out the analysis of the work efficiency of the power plant for the selected sixteen working fluids (presented in Table 1), the calculation of the efficiency and the power of the cycle have been performed.

The efficiency of the Brayton cycle was determined from Equation (2):

$$\eta_{th} = \frac{l_T - l_{Com}}{q_d} \cdot 100\% \tag{2}$$

In Equation (2), individual values were determined from the following relationships:

Table 1. Summary of working fluids and their selected parameters (source: own elaboration on the basis of Lemmon et al. (2013))

Working fluid	Chemical formula	Critical temperature °C	Critical pressure par MPa	Molar mass kg/kmol
Perfluorobutane	C4F10	113.18	2.3234	238.03
RC318	Cyclo-C_4F_8	115.23	2.7775	200.03
Trifluoroiodomethane	CF_3I	123.29	3.953	195.91
R218	$CF_3CF_2CF_3$	71.87	2.64	188.02
R227ea	CF_3CHFCF_3	101.75	2.925	170.03
R236fa	$CF_3CH_2CF_3$	124.92	3.2	152.04
Sulphur hexafluoride	SF_6	45.573	3.755	146.06
R23	CHF_3	26.143	4.832	70.014
Dimethylether	$(CH_3)_2O$	127.23	5.3368	46.068
N_2O	N_2O	36.37	7.245	44.013
Oxygen	O_2	−118.57	5.043	31.999
Ethylene	CH_2CH_2	9.20	5.0418	28.054
Helium	He	−267.95	0.22761	4.0026
Hydrogen	H_2	−240.01	1.2964	2.0159

- unit work in turbine:

$$l_T = (h_{b3} - h_{b4}) \cdot \eta_T \tag{3}$$

- unit work of compressor:

$$l_{Com} = (h_{b2} - h_{b1}) / \eta_{Com} \tag{4}$$

- unit heat supplied in the heater:

$$q_d = h_{b3} - h_{b2} \tag{5}$$

The values of enthalpy h_{b1}, h_{b2}, h_{b3} and h_{b4} have been defined using the database of thermodynamic properties of chemical substances Refprop 9.1 (Lemmon et al., 2013) according to the procedure:

- for the known temperature value $T_{b1} = 30°\text{C}$ and assumed lower pressure $p_{b1} = 0.2$ MPa the values of specific enthalpy h$_{b1}$ and specific s_{b1} were determined;
- as the conversion $b1$–$b2$ is an isentropic process, for a predetermined value of the entropy $s_{b1} = s_{b2}$ and the value of the upper pressure $p_{b2} = p_{b3}$ (pressure p_{b2} was calculated from Equation (1)), the following values were determined: specific enthalpy h_{b2} and temperature T_{b2};
- at the known pressure p_{b3} and temperature value at point $b3$ ($T_{b3} = 150°\text{C}$) the values of specific enthalpy h_{b3} and specific entropy s_{b3} were determined;
- on the basis of predetermined entropy at point $b3$ and the assumption that $s_{b4} = s_{b3}$ (conversion $b3$–$b4$ is isentropic) as well as the known pressure value at point $b4$ $(p_{b4} = p_{b1})$ the value of specific enthalpy h$_{b4}$ and temperature T_{b4} were determined.

In subsequent calculations, the value of unit heat input q_d was determined from Equation (5) and this allowed to determine the value of the mass flow of the working medium:

$$\dot{m}_b = \frac{\dot{Q}_d}{q_d} \tag{6}$$

whereas

$$\begin{aligned} T_{w2} &= T_{b2} + \Delta T \\ \dot{Q}_d &= \dot{m}_w \cdot c_{p,w} \left(T_{w1} - T_{w2}\right) \end{aligned} \tag{7}$$

and

$$T_{w2} = T_{b2} + \Delta T \tag{8}$$

The determined mass flow rate of the working medium was used to calculate the values of the turbine power N_T, compressor power N_{Com} and the power of power plant N_b:

$$N_T = \dot{m}_b \cdot l_T \tag{9}$$

$$N_{Com} = \dot{m}_b \cdot l_{Com} \tag{10}$$

$$N_b = N_T - N_{Com} \tag{11}$$

3. RESULTS OF CALCULATIONS AND ANALYSIS

In the first place, the results of calculations of the cycle efficiency and power were presented, in which all conversions were considered as ideal processes (isentropic conversions in turbine and compressor $\eta_T = 1$, $\eta_{Com} = 1$). Table 2 shows the values of specific enthalpy and temperature at the characteristic points of the cycle as well as the results of calculations of unit work for such parameters of the cycle operation (pressure ratio), for which the plant rating would be the highest. Table 3 presents the results of calculations showing the efficiency of the cycles without recovery and with internal heat recovery.

Table 2. Values of specific enthalpy and temperature at characteristic points of the cycle and values of unit work for the pressure ratio at which the highest efficiency was obtained ($T_{b1} = 30°C$, $T_{b3} = 150°C$, $\eta_T = 1$, $\eta_{Com} = 1$)

Working fluid	Pressure ratio [–]	Specific enthalpy in point kJ/kg				Temperature in point °C		Specific work of kJ/kg		
		h_{b1}	h_{b2}	h_{b3}	h_{b4}	T_{b2}	T_{b4}	turbine l_T	Com l_{Com}	cycle l_B
RC318	5	337.02	355.57	438.56	412.78	67.2	117.4	25.78	18.55	7.23
R227ea	5	348.96	371.46	453.17	422.78	67.7	113.2	30.39	22.50	7.89
R236fa	5	381.62	406.44	488.55	454.96	72.8	110.0	33.59	24.82	8.77
R23	2.75	399.86	438.65	497.74	450.75	82.3	94.2	46.99	38.79	8.2
N2O	2.25	472.06	522.12	584.14	524.6	86.4	87.2	59.54	50.06	9.48
Oxygen	1.75	275.37	323.1	387.26	330.26	81.8	89.2	57.00	47.73	9.27
Hydrogen	1.75	4003.8	4764.2	5737.2	4833.2	82.8	87.6	904.0	760.4	143.6
Helium	1.5	1580.1	1857.5	2203.6	1874.3	83.4	86.7	329.3	277.4	51.9

Table 3. Comparison of calculation results of the cycle operation efficiency with and without internal heat recovery ($T_{b1} = 30°C$, $T_{b3} = 150°C$, $\eta_T = 1$, $\eta_{Com} = 1$)

Working fluid	Pressure ratio [–]	Thermal efficiency, %		Power of cycle N_B, kW	Heat supplied, Q_d kW		Final temperature of heat carrier, T_{w2} °C	
		$\eta_{th,WR}$	$\eta_{th,R}$		without reg	with reg	without reg	with reg
RC318	5	8.71	14.99	302.09	3467.51	2015.62	77.2	111.9
R227ea	5	9.66	16.11	332.69	3445.36	2065.45	77.7	110.7
R236fa	5	10.68	15.57	345.08	3230.86	2216.47	82.8	107.1
R23	2.75	13.88	14.28	393.39	2834.81	2753.96	92.3	94.23
N2O	2.25	15.29	*)	407.35	2664.94	*)	96.4	*)
Oxygen	1.75	14.45	*)	412.51	2855.07	*)	91.8	*)
Hydrogen	1.75	14.76	*)	415.43	2814.84	*)	92.8	*)
Helium	1.5	15.00	*)	418.47	2790.64	*)	93.4	*)

*) Internal heat regeneration is not possible because $T_{b2} + \Delta T > T_{b4}$

The analysis of the data presented in Tables 2 and 3 and taken into account the molecular masses of the particular working factors (summarised in Table 1) reveal that for the working fluid with a lower molecular weight (hydrogen, helium) the unit work in the turbine and compressor is much higher than that for gases whose molecules have larger mass. For low molecular weight gases, higher power of cycle was obtained (at lower compression ratios) than that for gases with higher molecular weights. In addition, for low mass gases internal heat recovery cannot be used as the turbine gas temperature is only slightly higher than the compressor refrigerant temperature.

It is known that real conversion losses should be taken into account resulting from thermodynamic irreversibility. Their amount depends on many factors, and one of the most important ones is the efficiency of the machine through which the conversion is accomplished. These losses decrease with the development of technology. The efficiency of machines (turbines, compressors, heat exchangers) exerts a very significant impact on the efficiency of the Brayton cycle. In the next part of the paper the analysis results devoted to this issue are presented.

In Fig. 3 the resulting highest values of power of the Brayton cycle are shown, assuming that the conversions are performed without loss, and compared to the results obtained with the assumption of internal efficiency of the turbine and the compressor at the level of 0.9 for each of the machines.

As is seen from the data analysis shown in Fig. 3, taking into account the efficiency of devices implementing the Brayton cycle reduces the cycle power in a really significant way. In the case of low molecular weight gases, for example helium, the resulting cycle power, considering that the turbine and compressor efficiency, is only 11% of the theoretical efficiency for the isentropic processes in a cycle. For higher molecular weight gases, the reduction in power is not so great and e.g. for the working medium RC318 the cycle power in which losses are accounted for is equal to 35% of the ideal cycle power. For this reason, additional calculations of the Brayton cycle efficiency have been made using a variety of working fluids, although relatively high molecular weight substances were selected. Figure 4 shows the power calculation results of the Brayton closed cycle with various organic working fluids for pressure ratios from 1.25 to 5, assuming that the efficiency of the turbine and compressor is 90%.

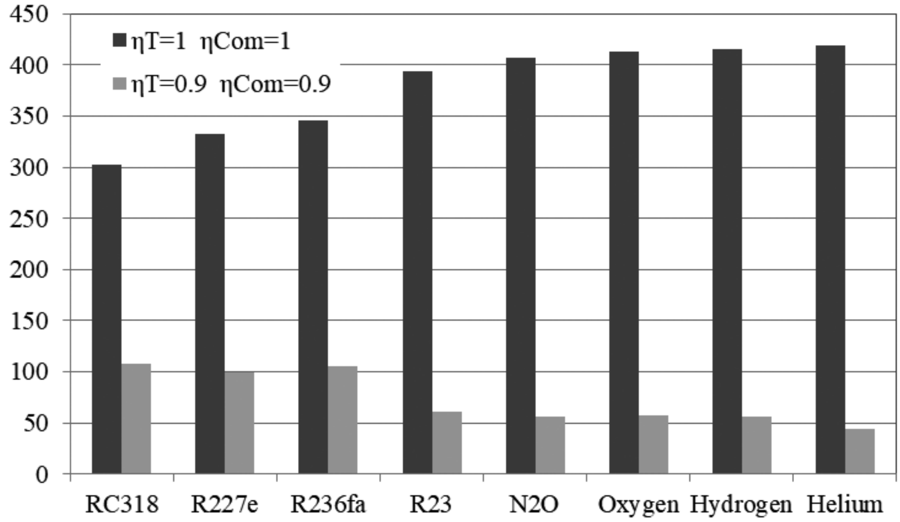

Fig. 3. Comparison of the Brayton cycle power values with different working fluids, for optimum pressure ratios, assuming different turbine and compressor internal efficiency

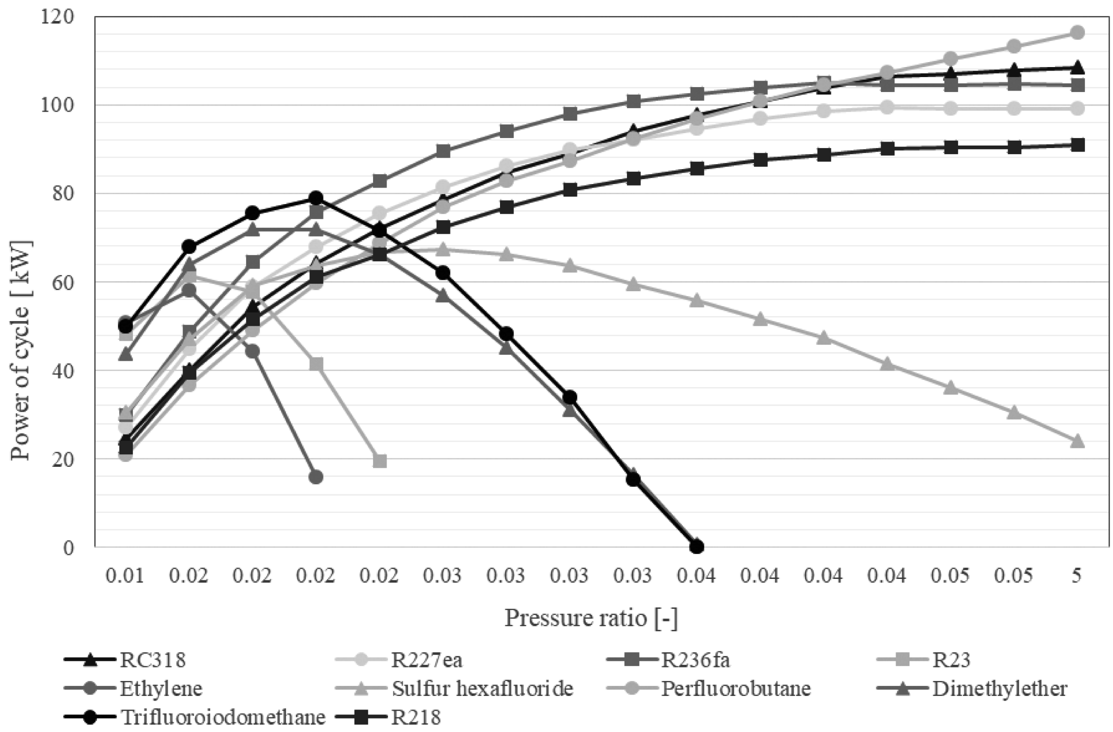

Fig. 4. The Brayton cycle power with various organic working fluids supplied from the source with a temperature of 160°C, assuming that internal efficiency of the turbine and compressor is $\eta_T = 0.9$, $\eta_{Com} = 0.9$, respectively

As can be seen from the analysis of the data presented in Fig. 4, for some of the fluids (R23, trifluoroiodomethane, ethylene, sulphur hexafluoride, dimethylether) such a pressure ratio can be indicated (usually low, ranging from 1.5 to 2.5) for which the plant rating is the highest. For the remaining substances the pressure ratio for which the power of the power plant would be the highest does not include the range of calculation (the ratio is higher than 5). The most advantageous working fluid in terms of the resulting plant rating is perfluorobutane – the substance with the highest molecular weight, among the considered substances.

As mentioned earlier, the use of internal recovery in the Brayton cycle powered by a heat carrier, from the so-called open heat source, does not affect the increase in the cycle power but reduces the amount of energy that is supplied to the cycle from the external source and thus increases the thermal efficiency as shown in Fig. 5. It also affects the increase in the final temperature of the heat carrier (Table 2, values in column T_{w2}), which may be significant when further use of the heat carrier (e.g. cogeneration) is planned.

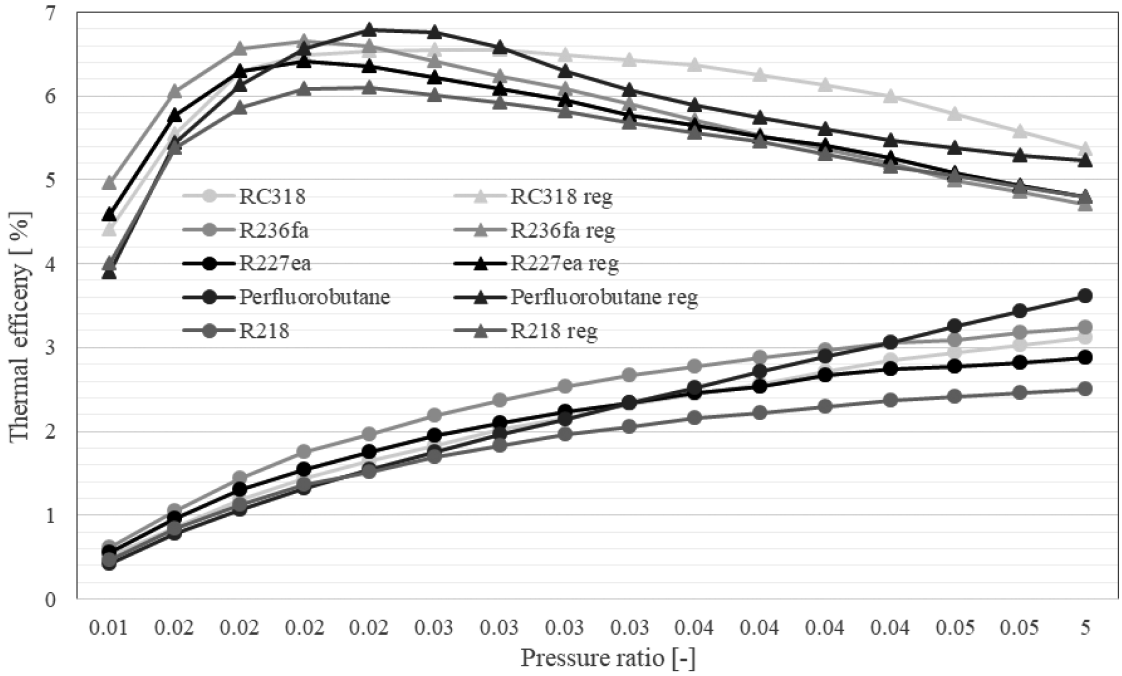

Fig. 5. Efficiency of the organic Brayton cycle with and without the use of internal heat regeneration, supplied from a source of temperature 160°C ($\eta_T = 0.9$, $\eta_{Com} = 0.9$)

4. SUMMARY AND CONCLUSIONS

The paper discusses the feasibility, effectiveness and validity of the use of the gas turbine power plant, operated according to the Brayton comparative cycle in order to develop low-potential waste heat (160°C) and convert it into electricity.

On the basis of the cycle performance analysis it can be concluded that low molecular weight working fluids allow to obtain higher power efficiency of the low-temperature Brayton cycle only if the conversions without taking into account internal losses are considered. For the cycle that takes into account the compression conversion efficiency in the compressor and expansion in the gas turbine, the highest efficiency was obtained for the perfluoropentane working medium and other substances with relatively high molecular weight values even for the cycle using the internal heat recovery, the thermal efficiency of the Brayton cycle does not exceed 7%. With today's technology even such efficiency would be difficult to obtain as there are no machines: compressors and turbines operating on the organic substances, listed in Table 1, and it is difficult to predict what internal efficiencies they would be characterised by (operating at a pressure range of 0.2–1 MPa). It is worth noting that the efficiency of the theoretical Brayton cycle is comparable to the theoretical efficiency of the ORC (Organic Rankine Cycle). However, the fact of taking into account the efficiency of machines performing particular cycles decreases the efficiency of the Brayton cycle much more than is the case of the ORC. As a summary statement, it can be added that 20 years ago the ORC technology – dedicated to the management of waste heat and the heat of renewable resources – was considered to be completely unprofitable and ineffective while at present it has become commercially available. It can be assumed that also low and medium temperature organic and inorganic gas turbine power plants will find their place in industrial and energy applications, with the development of technology, especially in the field of fluid-flow machines.

SYMBOLS

c_p	specific heat at constant pressure, J/(kg·K)
h	specific enthalpy, kJ/kg
l	cycle work per unit mass, kJ/kg
$\dot{m}$	flow rate, kg/s
N	power, kW
p	pressure, MPa
$\dot{Q}$	rate of heat transferred, kW
q	heat per unit mass, kJ/kg
s	specific entropy, kJ/(kg·K)
T	temperature, °C

Greek symbols

ΔT	temperature difference, K
η	cycle efficiency, %
ξ	pressure ratio, [–]

Subscripts

b	concerning working fluid in Brayton cycle
$b1$	state after cooling of working fluid
$b2$	state after compression
$b3$	state before expansion machine
$b4$	state after leaving the turbine
c	concerning cooling medium
Com	concerning compressor
cr	critical point
d	input heat
T	concerning turbine
w	concerning heat carrier (water)

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