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The influence of vapor superheating on the level of heat regeneration in a subcritical ORC coupled with gas power plant

SŁAWOMIR WIŚNIEWSKI*
ALEKSANDRA BORSUKIEWICZ-GOZDUR

West Pomeranian University of Technology, Department of Heat Engineering, al. Piastów 17, 70-310 Szczecin, Poland

Abstract The authors presented problems related to utilization of exhaust gases of the gas turbine unit for production of electricity in an Organic Rankine Cycle (ORC) power plant. The study shows that the thermal coupling of ORC cycle with a gas turbine unit improves the efficiency of the system. The undertaken analysis concerned four the so called "dry" organic fluids: benzene, cyclohexane, decane and toluene. The paper also presents the way how to improve thermal efficiency of Clausius-Rankine cycle in ORC power plant. This method depends on applying heat regeneration in ORC cycle, which involves pre-heating the organic fluid via vapour leaving the ORC turbine. As calculations showed this solution allows to considerably raise the thermal efficiency of Clausius-Rankine cycle.

Keywords: ORC; Heat regeneration, Gas turbine

Nomenclature

A	–	surface of heat exchanger, m^2
c_{ps}	–	the average specific heat of the exhaust gas, kJ/(kg K)
h_c	–	proper enthalpy of the organic fluid, kJ/kg
l	–	work per unit mass, kJ/kg
$\dot{m}$	–	mass flow of the substance, kg/s
N	–	power, W
p_c	–	pressure of the organic fluid, Pa

*Corresponding author. E-mail address: slawomir.wisniewski@zut.edu.pl

$\dot{Q}$	–	heat flux, kW
T	–	temperature, K
ΔN	–	power increase in cascade plant, kW
W1, W2	–	heat exchanger

Greek symbols

η	–	efficiency

Subscripts

d	–	supplied to the turbine unit
R	–	regeneration
s	–	exhaust gas
c	–	organic fluid
$c1$	–	initial temperature of vapour

1 Introduction

Unlimited access to energy is one of the factors determining civilization development. The time of the last decades was not only a period of great technological progress but also of great increase in energy consumption. The increased exploitation of the natural environment as well as diminishing recourses of conventional fuels have become a reason for increased interest in alternative and renewable energy sources. Apart from these, one of the more important directions in the energy sector development is to seek solutions that improve the efficiency of power generation in plants based on conventional energy carriers. The prospect of using up all these carriers enforces their rational and efficient utilization. The increase in efficiency of this usage can be achieved through improving the efficiency of installations which transform the energy of fossil fuels into electricity. Another way to increase this efficiency is to use waste heat from various kinds of technological processes.

In the paper have been analyzed the Organic Rankine Cycle (ORC) power plant system, with organic fluid as a working fluid, that uses waste heat of gases leaving the turbine unit, as the main heat source. Using this type of system is possible because of relatively high exhaust gas temperature, which, depending on the gas turbine unit characteristics, is exceeding 700 K [1]. Preliminary calculations as well as earlier studies [2] have shown that application of thermal coupling between a gas turbine unit and the ORC plant increases the power and efficiency of the system.

The method analyzed in the paper mainly concerns the ORC power plant system, which uses regenerative heating of the organic working fluid by the heat from the outlet vapour of the ORC turbine. As it was shown, heat recovery in ORC cycle allows for more efficient usage of the exhaust gases' from gas turbine.

2 Scheme description

The relatively high temperature of the exhaust gases leaving the gas turbine allows for further use of the aforementioned gases for energy generation purposes. This particular case shows the thermal coupling of gas turbine unit with an ORC plant. The gas turbine used in the analysis is available in one of the leading manufacturers' commercial offer [3]. Electrical power of the analyzed gas turbine is 5.25 MW, with efficiency reaching 30.5%. The turbine is powered by natural gas. According to the data directory the exhaust gases leaving the gas turbine have temperature of over 500 °C, with the mass flow rate at 20.8 kg/s.

In the discussed system, the authors applied preheating of the condensate in the ORC power cycle by vapour of the organic fluid leaving the vapour turbine. The method of thermal ORC system coupling with a gas turbine unit is shown in the diagrams presented in Figs. 1 and 2.

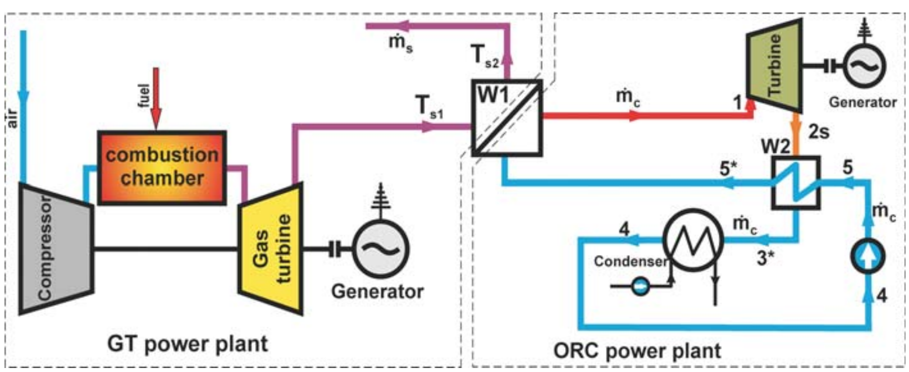

Figure 1. Thermal coupling of gas turbine unit with an ORC plant (variant 1).

The difference between these diagrams results from a different approach in determining the quantity of regenerative heat in the heat exchanger W2. In the first variant it was assumed that the entire flow of the organic fluid

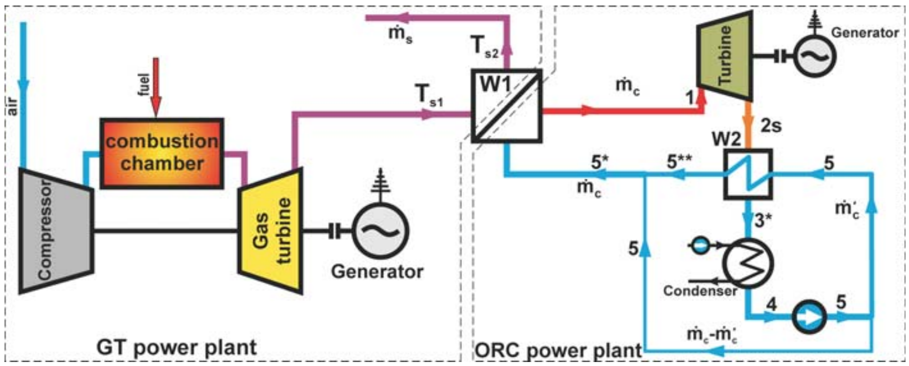

Figure 2. Thermal coupling of gas turbine unit with an ORC plant (variant 2).

leaving the condenser is heated in the W2 heat exchanger, while in the second variant only a part of this flow is heated. The difference is explained in a great detail in the section on the methodology of calculations.

In the ORC system under analysis it was assumed that the so called dry fluid is used as the working fluid. The parameters of the considered fluids are presented in the next section of the paper. Furthermore, it was assumed that only subcritical ORC cycles will be considered as is shown in T-s diagram in Fig. 3. It means that the maximum pressure of organic fluid is lower than its critical pressure p_{cr}.

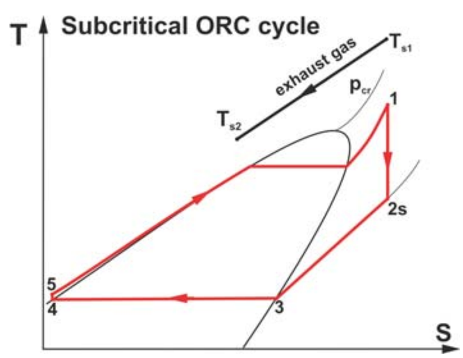

Figure 3. Cycle of thermodynamic changes in ORC power plant.

As it is clearly shown in the figure above and in the carried out calculations the vapour leaving the ORC turbine is superheated and has relatively high temperature reaching 600 K. Therefore, there are realistic prospects of using the thermal energy contained in this vapour for preheating organic liquid (internal heat regeneration) which is shown in Figs. 1 and 2.

The liquefied organic fluid, after the preliminary heating in the W2 exchanger flows further to the W1 exchanger, which is the element joining the gas turbine unit with the ORC power plant. The organic fluid is then further heated, evaporated and superheated in the heat exchanger. In this exchanger, the exhaust gas leaving the gas turbine serves as the heating medium. The exhaust gases can be treated as a carrier of waste energy left after the process of electricity generation in the gas turbine unit.

3 The characteristic of ORC working fluids

Implementation of the proposed heat regeneration in ORC systems is only possible when using in ORC cycle a working fluid from the group of so called "dry fluids" [4]. In practice, this means that the expansion process of this fluid in the turbine (isentropic process) ends in the superheated vapour region. This was illustrated in the charts presented in Figs. 3 and 4.

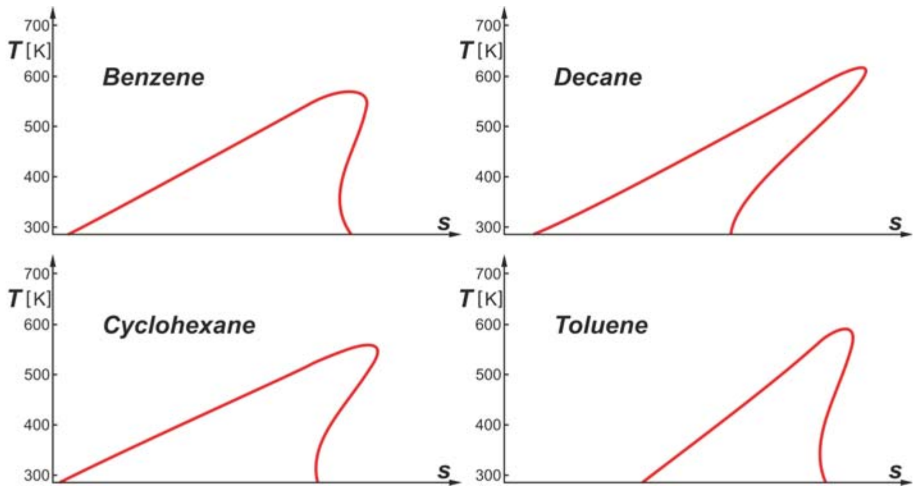

Figure 4. The shape of the saturation curves of the analyzed fluids (T-s diagram).

The authors of this paper analyzed the following four organic fluids possible for implementation in the ORC power plant cycle (the critical parameters of the fluids are given in brackets):

- Benzene ($T_{cr} = 562.05$ K, $p_{cr} = 4.89$ MPa),
- Cyclohexane ($T_{cr} = 553.64$ K, $p_{cr} = 4.07$ MPa),
- Decane ($T_{cr} = 617.65$ K, $p_{cr} = 2.10$ MPa),
- Toluene ($T_{cr} = 591.75$ K, $p_{cr} = 4.13$ MPa).

Figure 4 shows the actual saturation curves for the organic fluids being analyzed in T-s diagram. Because of the shape of these curves with the assumed working fluid condensing temperature of 308 K the expansion process for the fluid in the turbine for each of the analyzed cases ends in the superheated vapour region.

4 Calculation methodology

The parameters which characterize the gas turbine unit presented in this paper were taken from the manufacturer's database catalogue [3]. In calculations assumed was a constant temperature difference between the exhaust gases and the organic fluid in the heat exchanger at $\Delta T = 30$ K. It was also assumed that the condensation of the fluid in the ORC cycle takes place at the temperature of 308 K. Thermal and caloric parameters for the particular organic fluid in characteristic points of the cycle were based on the Refprop 7.0 database [5]. The more important equations used in the calculations are given bellow.

Incoming heat flux from the exhaust gas in the heat exchanger W1 is given by:

$$\dot{Q}_s = \dot{m}_s c_{ps} \left(T_{s1} - T_{s2}\right) \ . \tag{1}$$

The average specific heat of the exhaust gas at temperatures from T_{s1} to T_{s2} was determined on the basis of tables [6].

The heat flux brought to the organic fluid in the heat exchanger may be calculated as:

$$\dot{Q}_c = \dot{m}_c \left(h_{c1} - h_{c2}\right) \ . \tag{2}$$

Using the energy balance equations (1) and (2) and assuming that the heat losses are negligibly small the mass flow of the organic fluid in ORC cycle was specified in the following form:

$$\dot{m}_c = \frac{\dot{m}_s c_{ps} \left(T_{s1} - T_{s2}\right)}{h_{c1} - h_{c5*}} \ . \tag{3}$$

The power of the Clausius-Rankine (C-R) cycle in ORC plant was calculated in accordance with the following relation (including the power needed for the pump):

$$N_{C-R} = \dot{m}_c \left(l_{1-2s} - l_{4-5}\right) = \dot{m}_c \left(h_{c1} - h_{c2s} - \left(h_{c5} - h_{c4}\right)\right) \ , \tag{4}$$

where:
l_{1-2s} – turbine work per unit mass,
l_{4-5} – pump work per unit mass.

The electrical power leaving the ORC plant generator is defined by the following relation:

$$N_{el} = \eta_i \eta_m \eta_g N_{C-R} \ . \tag{5}$$

The electrical power according to (5) was calculated by assuming the following values of individual performances: turbine internal efficiency $\eta_i = 0.80$; mechanical efficiency $\eta_m = 0.98$; generator's electrical efficiency $\eta_g = 0.97$ [7]. The system total electrical power (gas turbine unit + ORC power plant) was defined as the sum of electrical power generated in the same gas turbine unit and the power obtained through ORC plant cycle:

$$N_{el} = N_{el_c} + N_{el_z} \ . \tag{6}$$

The indicator ΔN of the coupled cycle power to the power of the gas turbine unit itself is defined as:

$$\Delta N = \frac{N_{el} - N_{el_z}}{N_{el_z}} 100 \ . \tag{7}$$

The efficiency of electricity generation in the coupled system (gas turbine unit + ORC power plant) was defined according to the following relation:

$$\eta_{el} = \frac{N_{el}}{\dot{Q}_d} = \frac{N_{el_c} + N_{el_z}}{\dot{Q}_d} \ . \tag{8}$$

Using the data sheets [3] it was determined that the supplied energy flux in the gas turbine unit is 17.21 MW.

The feature of the C-R circuit leads to the following relations between the organic fluid pressure values at specific points:

$$p_{c1} = p_{c5} = p_{c5*} = p_{c5**} \ \text{and} \ p_{c2s} = p_{c3} = p_{c4} = p_{c3*} \ . \tag{9}$$

Applying the regenerative W2 heat exchanger to the ORC power plant system makes it necessary to define the thermal and caloric parameters of the fluid as it enters and leaves this exchanger in order to determine the regenerated heat flux. Two methods of determining this heat flux, depending on the calculation variant, are given below.

First variant The diagram presented in Fig. 1 shows that the entire flow of the liquefied organic fluid is pre-heated in the W2 heat exchanger. Having this fact in mind the energy balance equation of the W2 heat exchanger, while neglecting the heat loss, takes the following form:

$$\dot{m}_c\, h_{c2s} + \dot{m}_c\, h_{c5} = \dot{m}_c\, h_{c3*} + \dot{m}_c\, h_{c5*}\ . \tag{10}$$

From Eq. (10) the enthalpy h_{c5*} can be determined of the organic fluid heated in the W2 exchanger (at the exchanger outlet).

The method of determining the organic fluid enthalpy values in various points is given below. Parameters of point 3* were determined assuming that the vapour temperature when leaving the W2 exchanger is 5 K higher than the temperature of the liquefied organic fluid (ΔT_1 = 5 K). Temperature distribution in W2 exchanger have been calculated by using the method shown in [8]. For such defined temperature, with the pressure at node 3* equal to the condensation pressure the organic compound database enthalpy in point 3* was established from Refprop. The temperature T_{c5*} up to which the fluid in the W2 exchanger was heated has been defined on the basis of Refprop organic compound database (for the defined enthalpy of h_{c5*} and pressure $p_{c1} = p_{c5} = p_{c5*}$).

Second variant In the second calculation variant have been assumed that the heat transfer in the W2 exchanger between the vapour and the liquid of the organic fluid takes place at a constant temperature difference of $\Delta T_1 = \Delta T_2$ = 5 K. Temperature difference in the exchanger is shown in Fig. 6. Under this assumption, only a part of the earlier condensed compound can be heated in the W2 exchanger, the residual flux must be transferred outside the exchanger by an appropriate bypass. This is schematically illustrated in Fig. 7. In this case the thermal and caloric parameters of the organic fluid at the inlet and outlet from the W2 exchanger are known (they can be determined on the basis of the adopted temperature distribution in the exchanger, and on the basis of the Refprop organic compound database for proper evaporation and condensation pressure values of

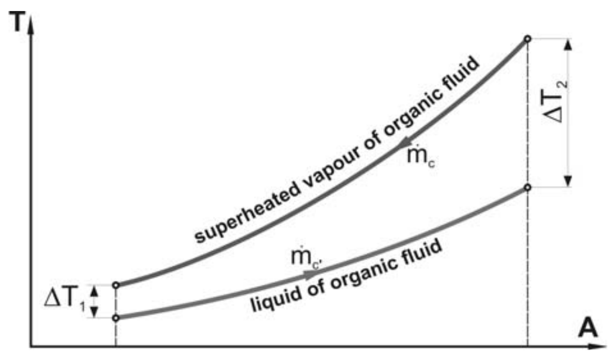

Figure 5. Temperature distribution in W2 exchanger for variant 1.

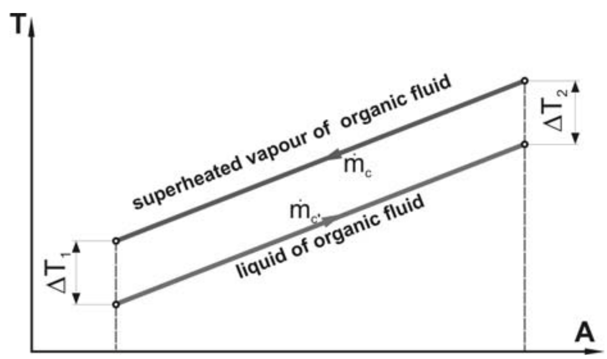

Figure 6. Temperature distribution in W2 exchanger for variant 2.

the organic compound). For the W2 exchanger presented above the energy balance, with neglecting the heat loss, is as follows:

$$\dot{m}_c\, h_{c2s} + \dot{m}_{c'}\, h_{c5} = \dot{m}_c\, h_{c3*} + \dot{m}_{c'}\, h_{c5**}\ . \tag{11}$$

An organic fluid flux was defined from the energy balance equation for W2 exchanger (Fig. 5). This flux can be transferred directly to W2 exchanger and can be determined from the following relation:

$$\dot{m}_{c'} = \frac{\dot{m}_c\,(h_{c2s} - h_{c3*})}{h_{c5**} - h_{c5}}\ . \tag{12}$$

Due to the fact that only part of the organic fluid flux is heated in the W2 exchanger there is a need of estimating the temperature and enthalpy of

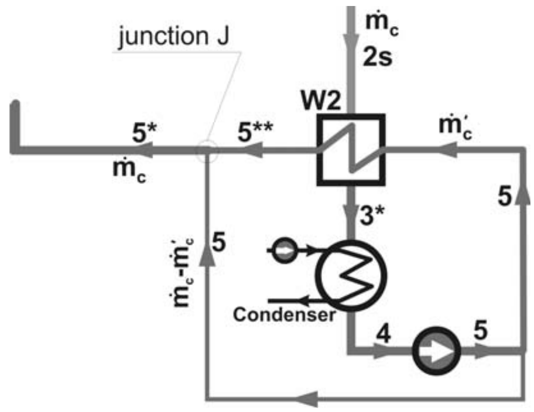

Figure 7. Conceptual diagram of the W2 regenerative heat exchange for variant 2.

the fluid, after mixing the flux leaving the exchanger and the flux directed through the bypass. The enthalpy of the mixed fluid flux was established basing on the energy balance equation of junction J shown in Fig. 7. The junction J energy balance is defined by the following equation:

$$\dot{m}_c h_{c5*} = \dot{m}_c^{,} h_{c5**} + \left(\dot{m}_c - \dot{m}_c^{,}\right) h_{c5} \ . \tag{13}$$

By using one of the calculation variants given above it is possible to determine the amount of heat supplied to the liquid of the organic fluid in W2 exchanger. After using the first variant we obtain the following relation:

$$\dot{Q}_R = \dot{m}_c \left(h_{c5*} - h_{c5}\right) \ . \tag{14}$$

In the second calculation variant the relation takes the form:

$$\dot{Q}_R = \dot{m}_c, \left(h_{c5**} - h_{c5}\right) \ . \tag{15}$$

In order to evaluate the influence of using heat regeneration on ORC power plant work efficiency, the authors of this paper compared the C-R circuit work efficiency with the efficiency in a cycle without heat regeneration. Dependencies which allow determining these efficiencies are presented bellow. In a plant without heat regeneration the cycle efficiency was determined from the relation:

$$\eta_{C-R} = \frac{l_{1-2s} - l_{4-5}}{q_d} = \frac{h_{c1} - h_{c2s} - \left(h_{c5} - h_{c4}\right)}{h_{c1} - h_{c5}} \ , \tag{16}$$

while in a plant with heat regeneration the cycle efficiency was defined by the expression:

$$\eta_{C-R}^{R} = \frac{h_{c1} - h_{c2s} - (h_{c5} - h_{c4})}{h_{c1} - h_{c5*}} \ . \tag{17}$$

5 Results of calculation

In Section 2 the system with two possible ways of heat regeneration in ORC cycle has been described. It was schematically shown in Figs. 1 and 2. In Tab. 1 example calculations for toluene, with the assumption of initial vapour temperature $T_{c1} = 673$ K are presented.

Table 1. Sample result of calculations for toluene.

ORC plant without heat regeneration	ORC plant with heat regeneration	
	Variant 1	Variant 2
Proper Enthalpy in various points of the circuit [kJ/kg]: $h_{c1} = 899.38$; $h_{c2s} = 605, 39, 47$; $h_{c3} = 265.89$; $h_{c4} = -141.06$; $h_{c5} = -137.16$		
—	$h_{c3*} = 271.81$ [kJ/kg] On the basis of equation (10): $h_{c5*} = 196.43$ [kJ/kg]	$h_{c3*} = 271.81$ [kJ/kg] $h_{c5**} = 312.13$ [kJ/kg] Using the relations (12) and (13): $h_{c5*} = 196.43$ [kJ/kg]
	Heat recovery unit [kW]	
	By relation (14): $\dot{Q}_R = 2191.79$	By relation (15): $\dot{Q}_R = 2191.79$
Thermal efficiency of the cycle		
By relation (16): $\eta_{C-R} = 0.280$	By relation (17): $\eta_{C-R}^{R} = 0.431$	

Table 2 summarizes the values of temperatures and enthalpy of the proper organic compound in various points of the cycle. These values were determined with the use of Refpop compound database, assuming that the evaporation temperature of the organic fluid is at 523 K, the condensation temperature is at 308 K and based on the presented methodology of calculation. Table 3 gives the basic values (power and electric efficiency) which characterize the coupled system of gas turbine unit and an ORC plant, as well as the power increase obtained in the effect of applying such coupling.

The main objective of this study was to determine the influence of applying heat regeneration in ORC system on the C-R cycle thermal efficiency.

Table 2. Thermal and caloric parameters of the organic fluids.

Node of the cycle	Benzene		Cyclohexane		Decane		Toluene	
	T_c [K]	h_c [kJ/kg]	T_c [K]	h_c [kJ/kg]	T_c [K]	h_c [kJ/kg]	T_c [K]	h_c [kJ/kg]
1	623.0	823.9	623.0	912.2	623.0	732.2	623.0	782.2
	673.0	937.0	673.0	1062	673.0	884.6	673.0	899.4
	723.0	1053.2	723.0	1217.7	723.0	1043.1	723.0	1020.6
	773.0	1172.7	773.0	1379.9	773.0	1207.3	773.0	1145.8
2s	457.3	549.4	504.5	650.2	531.7	491.3	475.8	514.9
	506.7	633.6	555.8	772.8	580.0	622.0	524.6	605.4
	554.7	721.9	605.6	902.4	627.9	759.1	572.4	700.6
	601.6	814.3	654.6	1039	675.5	902.2	619.4	800.4
3	308.0	344.7	308.0	293.7	308.0	5.6	308.0	265.9
4	308.0	-81.5	308.0	-92.5	308.0	-349.9	308.0	-141.1
5	310.5	-75.0	310.8	-85.1	308.6	-348.2	309.5	-137.2
3*	313.0	350.2	313.0	300.4	313.0	14.0	313.0	271.8
5*	414.5	124.2	462.4	264.8	491.4	129.0	433.7	105.9
	453.0	208.4	504.7	387.4	523.0	259.8	473.4	196.4
	490.2	296.7	523.0	517.0	523.0	396.9	511.9	291.7
	523.0	389.1	536.5	653.5	556.9	539.9	523.0	391.4

The impact of this regeneration, as well as the impact of organic fluid superheating temperature T_{c1}, on C-R cycle thermal efficiency, for all analyzed fluids, is presented in the Fig. 8.

6 Summary

The calculations showed that the thermal coupling of ORC system with a gas turbine unit makes it possible to improve the work efficiency of the considered system. For each of the analyzed compounds the electrical efficiency and the power of the coupled system is higher than the power and efficiency of the gas turbine unit by itself. After analyzing the results summarized in Tab. 2 it can be concluded that for the parameters adopted the most beneficial results are achieved when ORC plant uses toluene as the working fluid. For this fluid even a 38% increase in power can be achieved.

Table 3. Values characterizing the work efficiency of the coupled system for the analyzed organic fluids.

	T_{c1} [K]	N_{el} [kW]	Δn [%]	η_{el} [–]
Benzene	623.0	6646.6	26.6	0.386
	673.0	6827.3	30.0	0.397
	723.0	7019.6	33.7	0.408
	773.0	7250.2	38.1	0.421
Cyclohexane	623.0	6353.8	21.0	0.369
	673.0	6486.7	23.6	0.377
	723.0	6805.6	29.6	0.395
	773.0	7184.5	36.8	0.417
Decane	623.0	6162.5	17.4	0.358
	673.0	6351.9	21.0	0.369
	723.0	6796.7	29.5	0.395
	773.0	7000.7	33.3	0.407
Toluene	623.0	6539.3	24.6	0.380
	673.0	6699.5	27.6	0.389
	723.0	6871.0	30.9	0.399
	773.0	7266.3	38.4	0.422

The influence of using heat regeneration in the ORC system on C-R cycle thermal efficiency is also analyzed. This analysis was conducted assuming that all fluids had the evaporation temperature of 523 K, and adopted a few temperature variants for superheating vapour. The calculations showed that thermal efficiency of the C-R cycle without heat regeneration is much lower than the efficiency of a cycle with heat regeneration. By adopting in this analysis a few superheating temperature values it was possible to determine its influence on the efficiency value. The obtained results of calculations show that in a cycle without heat regeneration, the increase of superheating temperature is not beneficial for the cycle efficiency, causing its decrease. Exactly the opposite situation occurs in the case of a cycle with regeneration. In this system, an increase of the initial temperature of vapour T_{c1} causes an increase in cycle's efficiency. This is clearly illustrated in the charts presented in Fig. 8. It is due to the fact that, in the case of circulation without heat regeneration the increase of vapour superheating temperature causes a significant increase of the applied heat flux with a much lower increase of the cycle's power. By contrast in the case of a cycle with heat

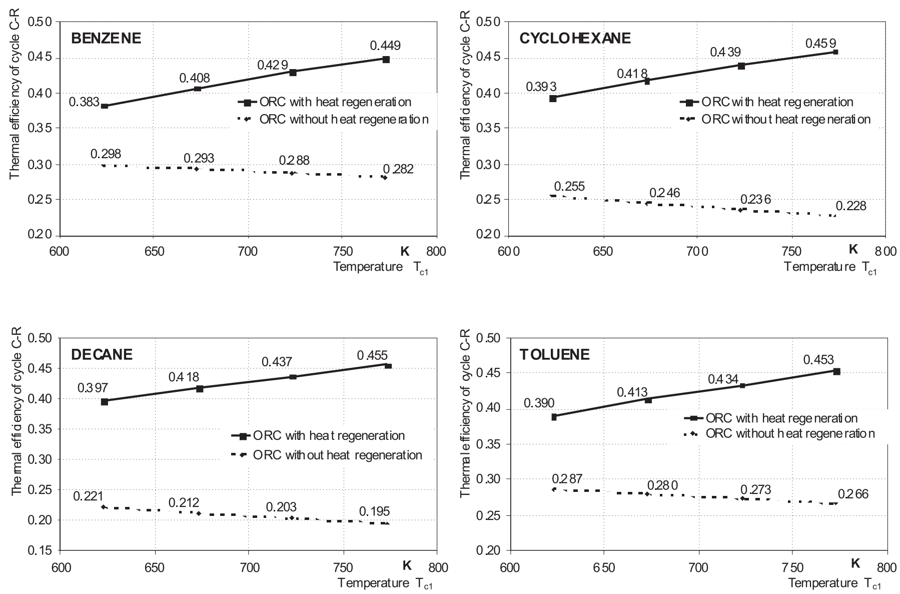

Figure 8. The influence of heat regeneration and the initial temperature T_{c1} of organic fluid vapour on C-R cycle thermal efficiency in an ORC.

regeneration, the increase of the vapour superheating temperature leads to an increase in temperature of the vapour leaving the turbine, therefore increasing the regenerated heat flux which in turn reduces the applied heat flux and directly increases the efficiency of the cycle.

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References

[1] Chmielniak T.: *Energy Conversion Technologies*. WNT, Warsaw 2008 (in Polish).

[2] Wiśniewski S., Borsukiewicz-Goadur A.: *Assessment of the effectiveness of operation of gas turbine coupled with the supercritical ORC power plant*. XII Forum of Power Engineers, Elektryka **64**, No. 336/2010, 131–132 (in Polsh).

[3] Industrial Gas Turbines – Siemens. www.siemens.com/energy.

[4] Borsukiewicz-Gozdur A., Nowak W.: *Comparative analysis of natural and synthetic refrigerants in application to low temperature Clausius-Rankine Cycle*. Energy **32**(2007), 344–352.

[5] NIST. Refprop 7.0, Standard Reference Database 23, Reference Fluid Thermodynamic and Transport Properties. National Institute of Standards and Technology, Gaithersburg 2002, MD, USA.

[6] Gogół W.: *Heat Exchange: Tables and Diagrams*. WPW, Warsaw 1979 (in Polish).

[7] Szargut J.: *Thermodynamics*. PWN, Warsaw 1998 (in Polish).

[8] Nowak W.: *The Theory of Recuperators* . WUPS, Szczecin 1993 (in Polish).