archives
of thermodynamics
Vol. **31**(2010), No. 3, 105–113
DOI: 10.2478/v10173-010-0017-6

Condensing power plant cycle — assessing possibilities of improving its efficiency

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Abstract This paper presents a method for assessing the degree of approaching the paper output of the Clausius-Rankine cycle to the Carnot cycle. The computations to illustrate its use were performed for parameters characteristic of the current state of development of condensing power plants as well as in accordance with predicted trends for their further enhancing. Moreover there are presented computations of energy dissipation in the machines and devices working in such a cycle.

Keywords: Condensing power plant; Efficiency improvement

1 Introduction

The pursuit for increasing the efficiency of energy conversion processes is an important task of research performed in many fields of science and engineering. It is motivated by the need of saving fuel reserves, reduction of losses and lessening the environmental burden. One of the major premises still is reduction of carbon dioxide emission. Nowadays the technologies using fossil fuels, mainly bituminous coal and lignite, prevail in the electric power generating industry [1–3]. Among these technologies the condensing power unit constitutes, beside the gas and steam system, the main component of the electric power generating facility. So, its improvement plays an essential part in increasing the thermodynamic efficiency, environmental friendliness

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and productivity of electricity production. A lot of research work activities were devoted to that issue [4–8], also in the context of integrating the CO_2-capture facilities with the condensing power unit's heat cycle [9,10].

The paper presents the issue of assessing the thermodynamic potential of a condensing power plant cycle. In quantitative assessments there was used the ratio of the cycle's efficiency determined with the use of mean temperature of heat feeding to and heat extraction out of the cycle to the cycle efficiency of the Carnot engine operating between the maximum and minimum temperatures of the condensing power plant cycle. Moreover there was assessed the influence of the efficiency improvement in the machines and devices on the efficiency of the whole installation.

2 Efficiency of condensing power plant cycle

Efficiency of power plant cycle with a specific production process structure may be determined taking into account the enthalpy-based or entropy-based analysis [11]. Further in assessing thermodynamic effects of cycle improvements we will use an entropy-based analysis. Using the symbols from Fig. 1 the cycle efficiency covering real processes is as follows [4]:

$$\eta_E^{(s)} = \frac{P^{(b)} - P_P}{\dot{Q}_{in}} = \left(1 - \frac{\bar{T}_{out}}{\bar{T}_{in}}\right)\left(1 - \frac{\bar{T}_{out}\bar{T}_{in}}{\bar{T}_{in} - \bar{T}_{out}}\frac{\dot{S}_{gen}}{\dot{Q}_{in}}\right)\frac{1}{1+\zeta_0} = \frac{\bar{\eta}_c\eta_j}{1+\zeta_0}\ , \tag{1}$$

with

$$\bar{\eta}_C = 1 - \frac{\bar{T}_{out}}{\bar{T}_{in}}\ , \tag{2}$$

and

$$\eta_j = 1 - \frac{\bar{T}_{out}\bar{T}_{in}}{\bar{T}_{in} - \bar{T}_{out}}\frac{\dot{S}_{gen}}{\dot{Q}_{in}}\ , \tag{3}$$

where:

$P^{(b)}$ – power output from the turbine, P_P – own needs of processes under control surface K_1;

$\dot{Q}_{in}$ – heat flux feeding the cycle;

$\bar{T}_{in}$ $=$ $\frac{h_0-h_3}{s_0-s_3}$, $\bar{T}_{out} = \frac{h_1-h_2}{s_1-s_2}$ – entropically averaging temperatures of feeding and extracting heat out of the cycle;

$\dot{H}_0$ – heat flux between the system and environment (heat losses to environment);

S_0 – entropy increase connected with the loss $\dot{H}_0$, while $\dot{S}_{gen}$ is total entropy generated in processes occurring under control surface K_1;

$\zeta_0 \quad = \quad \frac{\dot H_0 - \bar T_{out} \dot S_0}{\dot Q_{in}}$ – measure of efficiency reduction due to heat loss to environment.

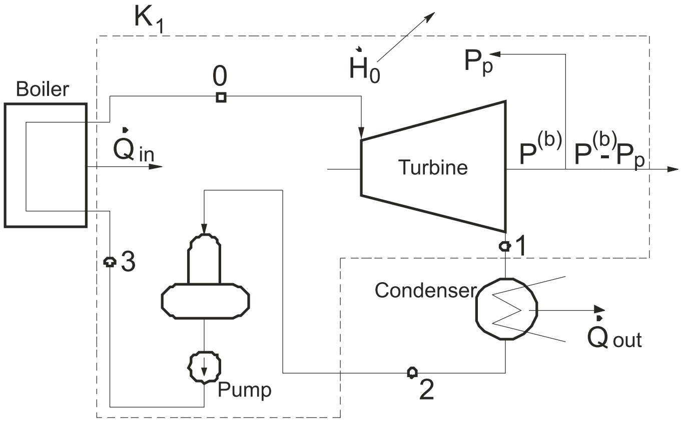

Figure 1. Diagram used for deriving the formula for energy efficiency of condensing power plant cycle.

Quantity $\bar\eta_C$, defined by (2), is the efficiency of the Carnot engine with temperatures $\bar T_{in}$ and $\bar T_{out}$, i.e. of upper and lower source respectively. Efficiency of the Clausius-Rankine cycle in case when the entropy changes in heat feeding and heat extraction processes is equal to each other. Quantity η_j may be referred to as indicator of the quality of energy machines and devices operating in a cycle. The following quantity

$$1 - \eta_j = \frac{\bar\eta_C - (1+\zeta_0)\eta_E^{(s)}}{\bar\eta_C} \tag{4}$$

shows potential possibilities of improving energy processes in machines and devices operating in closed system K_1 within a technology under investigation, Fig. 1.

Based on (1) it is possible to define an instantaneous energy efficiency of a power plant

$$\eta_{Eel1} = \frac{P_{el1}}{\dot m_f (LHV)} = \frac{\eta_B \bar\eta_c \eta_j \eta_{em}}{1+\zeta_0} \ , \tag{5}$$

where

$$P_{el1} = (P^{(b)} - P_P)\eta_{em} \ ,$$

and η_{em} denotes electromechanical efficiency, $\dot{m}_f$ is fuel mass flow rate, (LHV) represents lover heating value and η_B denotes boiler efficiency.

Expression (5) explicitly shows to what degree each of processes (devices) in a power plant dissipates energy fed to the system under investigation. By using (1) and (5) we compute the increase of heat rate caused by energy dissipation in the k-th process (device) of the power generating system

$$q_{loss}^{(k)} = \frac{\dot{Q}_{loss}}{P_{el1}} = \frac{1}{\eta_j - \bar{\eta}_C} \bar{T}_{out} \frac{\dot{S}_{gen}^{(k)}\left(1+\zeta_0\right)}{\dot{Q}_{in}} \ . \tag{6}$$

Having $q_{loss}^{(k)}$ we compute relevant values of $(\Delta\dot{m}_f(LHV))^{(k)}$ and $\Delta P_{el}^{(k)}$.

The value of $\bar{\eta}_C$ cannot exceed its upper limit that is the Carnot cycle's efficiency computed for maximum and minimum temperatures of a reference cycle for the condensing power plant (efficiency η_C). A condensing power plant potential may be traced by changing the live steam parameters, resuperheating parameters and feed water temperature.

3 Assessment of the Clausius-Rankine thermodynamic cycle potential

We assume the following ratio as a measure of thermodynamic potential of the Clausius-Rankine cycle

$$\bar{\eta}_C/\eta_C \ , \tag{7}$$

where:

η_C = $1 - T_0/T_n$ – the Carnot engine's cycle efficiency,

T_0 – highest temperature in cycle – live steam temperature,

T_n – lowest temperature in cycle.

Efficiency $\bar{\eta}_C$ has been defined by means of formula (2).

The assessment has been performed for the technological structure the unit presented in Fig. 2. Computations were carried out for various parameters of primary and secondary steam as well as various feed water temperatures. Moreover there were performed computations for a system with two reheat stages and the same number of feedwater heaters, it was assumed that $T_n = \bar{T}_w$.

The economically and thermodynamically justified feed water temperatures in contemporary units range from 290 to 300 °C (an increase in feed water temperature causes the following increase $\bar{\eta}_c$, on the other hand it causes a decrease of temperature difference in the heat exchange process in boiler, effecting the increase of heat exchanging surface and increase of investment expenditure). In the analyses it was assumed that $t_{78} = 290\,°\mathrm{C}$. For supercritical units with steam parameters 35 MPa, 700/720 °C the feed water temperature of 350 °C was considered. In the computations of this case beside the aforementioned temperature also a feed water temperature of 300 °C was taken into account. The obtained results are presented in Tabs. 1a and 1b as well as in Fig. 3 [4].

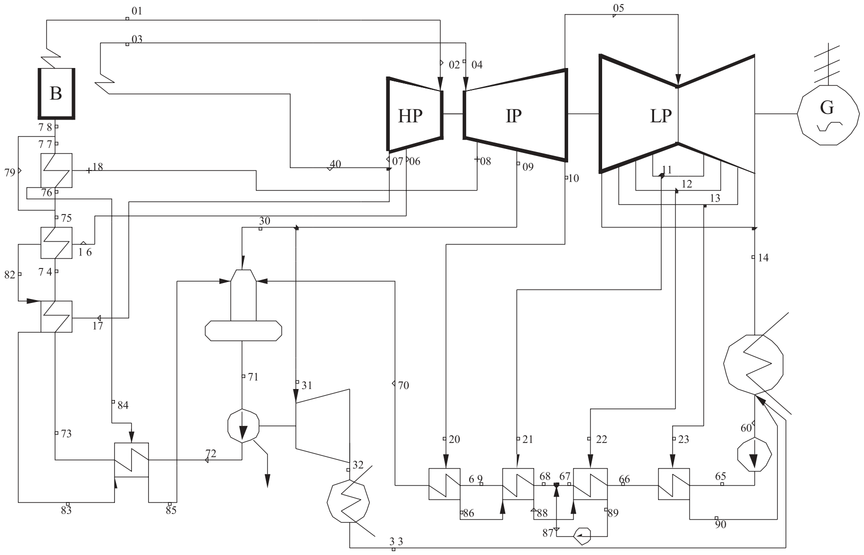

Figure 2. Reference diagram of condensing power plant cycle.

Moreover the tables present values of $(\bar{\eta}_C - \eta_0)/\bar{\eta}_C$, which is a measure of energy dissipation in the machines and devices of the heat cycle, where

$$\eta_t = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} \tag{8}$$

and $\dot{Q}_{in}$ is heat fed to the working fluid in boiler and in combustion exchangers.

Efficiency η_0 corresponds to $\eta_E^{(s)}$ in Eq. (4). The computations indicate an influence of all the parameters under consideration on the value of $\bar{\eta}_C/\eta_C$. A comparison of variants 1–4 shows that any increase of primary steam parameters should be accompanied by an increase of feed water temperature so as to obtain the same value of $\bar{\eta}_C/\eta_C$ (difference between 1 and 2 is

Table 1. Results of computations of condensing power unit's potential for different parameters of the cycle (see Fig. 2).

a)

Quantity	Unit	Parameters			
		27.5 MPa /560 °C /580 °C	27.5 MPa /560 °C /580 °C	27.5 MPa /580 °C /600 °C	27.5 MPa /600 °C /620 °C
p_{01}	MPa	27.5	27.5	27.5	27.5
t_{01}	°C	560	560	580	600
t_{03}	°C	580	580	600	620
t_{78}	°C	290	290	290	290
p_{14}	MPa	0.006	0.005	0.005	0.005
$\bar{T}_{in}$	K	675.4	675.4	682.18	688.8
$\bar{T}_{out}$	K	309.37	306.1	306.1	306.1
η_C	%	64.12	64.12	64.94	65.73
$\bar{\eta}_C$	%	54.19	54.68	55.13	55.56
η_0	%	48.81	49.28	49.76	50.21
$\bar{\eta}_C/\eta_C$	–	0.8452	0.8527	0.8489	0.8452
$(\bar{\eta}_C-\eta_0)/\bar{\eta}_C$	–	0.0993	0.0987	0.0974	0.0963
		1	2	3	4

b)

Quantity	Unit	Parameters			
		35 MPa /700 °C /720 °C	35 MPa /700 °C /720 °C /720 °C	35 MPa /700 °C /720 °C	35 MPa /700 °C /720 °C /720 °C
P_{01}	MPa	35	35	35	35
t_{01}	°C	700	700	700	700
T_{02}	°C	720	720	720	720
The $2^{n}d$ reheat stage steam temperature	°C	–	720	–	720
t_{78}	MPa	300	300	350	350
p_{14}	K	0.004	0.004	0.004	0.004
$\bar{T}_{in}$	K	734.31	757.04	752.09	774.94
$\bar{T}_{out}$	%	302.17	302.17	302.17	302.17
η_C	%	69.57	69.57	69.57	69.57
$\bar{\eta}_C$	%	58.85	60.08	59.82	61.00
η_0	%	52.78	54.29	53.10	54.58
$\bar{\eta}_C/\eta_C$	–	0.8459	0.8636	0.8598	0.8768
$(\bar{\eta}_C-\eta_0)/\bar{\eta}_C$	–	0.1031	0.0964	0.1123	0.1052
		5	6	7	8

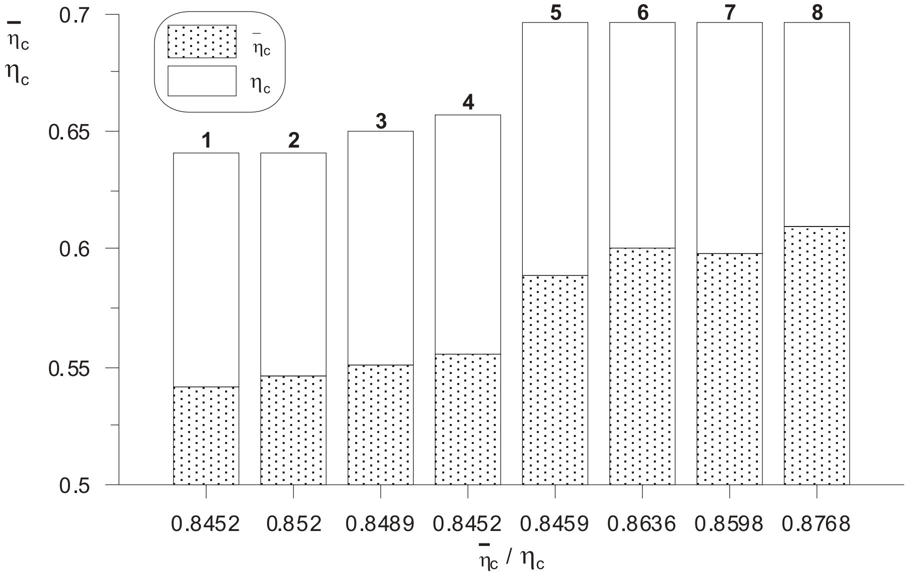

Figure 3. Values of $\bar{\eta}_C/\eta_C$, $\bar{\eta}_C$ and η_C for different parameters of the unit. Figures 1 to 8 correspond to columns marked in Tabs. 1a and 1b.

a result of a pressure change in a condenser). The same conclusion may be drawn for cases 5–8. Assumption of a feed water temperature of 300 °C for parameters 35 MPa / 700 °C / 720 °C essentially decreases the solutions energy efficiency relative to η_C. To a lesser degree it concerns the situation with parameters 35 MPa / 700 °C / 720 °C / 720 °C. From the point of view of $\bar{\eta}_C/\eta_C$ systems 2 and 8 have the parameters best configured. Despite the high parameters of steam in cases 5 and 7 there are obtained relatively small values of ratio $\bar{\eta}_C/\eta_C$. It follows from that that the cases require a better selection of cycle parameters. Interesting conclusions have been drawn from an analysis of values of

$$(\bar{\eta}_C - \eta_0)/\bar{\eta}_C ,$$

being a measure of energy dissipation in the machines and devices of a technological profile corresponding with such a power plant cycle. For the parameters presented in Tab. 1a there is a potential of improving efficiency of the machines and devices of the order of 0.1 and in the cases under consideration it decreases when the steam parameters increase. After assuming the data presented in Tab. 1b as starting point we will observe an increase of the potential. Hence one can conclude that units with a temperature of 700/720 °C should be optimised from the point of view of technological structure and characteristics of devices (maximization of efficiency η_0). A detailed analysis of influence of the parameters on efficiency of a condensing power unit is presented in [4].

4 Conclusions

The performed analysis allows an assessment of the current state of bringing the Clausius-Rankine cycle nearer to the Carnot cycle.The primary steam's and resuperheated steam's parameters are not the only ones increase of which has an effect on ratio $\bar{\eta}_C/\eta_C$. To achieve an increase of the ratio it is necessary to ensure an appropriate configuration of the whole system (including an appropriate selection of feed water temperature).

Dissipation of energy in machines and devices of the cycle was the second quantity under investigations. Based on the computations performed it is difficult to present an unambiguous interpretation. Yet, it is possible to state that the configuration of the power unit as presented in Fig. 2 needs optimisation for high parameters of steam. This is because in the case the energy dissipation in the machines and devices is relatively high (see the results in columns 7 and 8 of Tab. 1b).

The presented data supplemented with additional computations from [4] can be useful in assessing undertakings aimed at improving the Clausius-Rankine cycle.

Acknowledgements The results presented in this paper were obtained within Ordered Project PBZ-MEiN-4/2/2006 financed from resources assigned for science in 2007–2010.

Received 15 June 2010

References

[1] *Key World Energy Statistics 2009.* IEA 2010.

[2] Majchrzak H.: *Polish state energy policy until 2030.* Presentation at the plenary session of Power Engineering Problems Committee of PASci, Warsaw, April 2009 (in Polish).

[3] *Prognosis for fuel and energy demand unitl 2030. Supplement 2 to the project of the directive of Polish state energy policy until 2030 (version 3).* Ministry of Economy, 26.02.2009 (in Polish).

[4] Chmielniak T., Ziębik A. (eds.): *Supercritical cycles for coal-fired systems.* Silesian University of Technology Publishers, Gliwice 2010 (in Polish).

[5] Chmielniak T., Łukowicz H., Kochaniewicz A.: *Directions for the increase of efficiency of contemporary power units.* Rynek Energii **79**(2008), 6, 14–20 (in Polish).

[6] Chmielniak T., Łukowicz H.: *Determination of cycle parameters for the Polish coal-fired supercritical unit.* In: 1st Conference on Modern Energy Technologies and Equipment, Cracow 2007 (in Polish).

[7] Mohrbach L., Stolzenberger C.: *Ultra Super Critical Coal Generation.* IERE 2008 Florida Workshop *More kWh's . . . . Less Carbon.*

[8] Gierschner G.: *COMTES700 — on track towards the 50plus Power Plant.* New Build Europe 2008, Düsseldorf, 4–5 March 2008.

[9] *CO_2 Capture Ready Plants.* Technical Study, Report No. 2007/4.

[10] Chmielniak T., Łukowicz H., Kochaniewicz A.: *Analysis of supercritical systems of steam plants with heat recovery from the exhaust gases.* Archives of Energetics XXXVIII (2008), No. 2, 31–39 (in Polish).

[11] Chmielniak T.: *Energy conversion technologies.* WNT, Warsaw 2008 (in Polish).