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Hierarchical cycles in the modern power system – exergy analysis under partial loads

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Abstract

The aim of this paper is to investigate thermodynamic efficiency of hierarchical thermodynamic cycles under partial loads by using of exergy analysis. Advanced hierarchical cycles are composed of few energy conversion systems, which can be powered by several different sources of energy jointly. This kind of cooperation gives opportunities to provide high efficient and clean conversion of fossil fuels to electricity. Moreover, power plant consisting of few traditional power system gives new possibilities of optimization under partial loads.

Keywords: Hierarchical cycles; Partial loads; Combine cycles

Nomenclature

B	–	exergy, kJ
b	–	specific exergy, kJ/kg

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E	–	energy, kJ
I	–	enthalpy, kJ
i	–	specific enthalpy, kJ/kg
L	–	work, kJ
N	–	power, kW
$\dot{m}$	–	mass flow rate, kg/s
r	–	specific evaporation heat, kJ/kg
S	–	entropy, kJ/K
s	–	specific entropy, kJ/K kg
T	–	temperature, K
$\bar{T}$	–	mean thermodynamic temperature, K
$\dot{Q}$	–	rate of heat, heat energy flux, kW
x	–	dryness fraction, –

Greek symbols

Δ	–	difference, –
Π	–	sum of entropy changes, kJ/K
δ	–	exergy loss, kJ
ξ	–	relative loss, –
η_b	–	exergetic efficiency, –

Subscripts

a	–	beginning of the process
b	–	end of the process
B	–	boiler
C	–	Carnot / condenser
C-R	–	Clausius–Rankine cycle
ch	–	chemical
EX	–	exhaust
F	–	fuel
G	–	electric generator
GT	–	gas turbine
HRSG	–	heat recovery steam generator
i	–	number of device/cycle
in	–	inlet
k	–	kinetic
m	–	mechanical
out	–	outlet
p	–	potential
s	–	isentropic proces
st	–	steam
ST	–	steam turbine
T-G	–	turbine – generator set
w	–	water
0	–	ambient
1,2...	–	points of process

1 Introduction

Rapid development of technology in the nineteenth and twentieth century had leaded to increased exploitation of natural resources. Hence in some local ecosystems had occurred imbalance [3]. With fear of global effects caused by intensive development of modern civilization, among others, in the European Union the emission limits of COx, NOx and SOx have been introduced [12]. In many countries of the Union, including Poland, to achieve the goals it has been necessary to prove a fundamental reform of the energy sector. Modernization of obsolete system based largely on coal technology has become insufficient. It is obvious that, in order to reduce emissions and improve energy conversion efficiency new technologies are needed. The primary, long-term goal is to provide sustainable and environmentally safe technology for natural resources exploitation.

The current state of knowledge about large-scale high efficiently and emission-free electricity generation is deficient. Currently, great amount of power plants use heat engines to convert chemical energy to electricity. However, the heat engines efficiency is subjected to limitation of the Carnot cycle efficiency. Due to current material capabilities and an ambient temperature, respectively approx. 1600 K and approx. 300 K, the Carnot cycle can achieve about 80% [7, 8, 33]. However, due to numerous thermodynamic limitations, in fact heat engines reach lower temperatures of high temperature reservoir, keeping at the same time low temperature of the low temperature reservoir [7, 33]. This applies steam cycles (Clausius-Rankine cycle). Opposite situation occurs in the gas turbine cycles (Joule-Brayton cycle). These limitations cause that the Clausius–Rankine cycle in supercritical applications currently reaches energetic efficiency up to 55% and Joule-Brayton J-B cycle of 40% [7, 33, 34]. Through hierarchical merging of the two thermodynamic cycles, the thermodynamic efficiency of combine cycles increases over 60% [7, 34, 42]. Moreover hierarchical cycles are particularly advisable for power plant with the CCS systems (carbon capture and storage) [6, 43, 44].

The opposition to heat engines is a direct conversion of chemical energy into electricity. In this field fuel cells merit special attention. They are not subject to restrictions of Carnot cycle, so they can achieve high efficiency of chemical energy into electricity conversion. In large-scale electricity production the usage of solid-oxide fuel cell (SOFC) is taken into consideration [9, 24–26]. The main feature of the SOFC is high energy density per space dimension of the fuel cells. It is correlated with high operating temperature, which is necessary to provide adequate O^{2-} ions conductivity in electrolyte. High temperature gives the opportunity to use natural gas and perform a natural gas steam reforming inside the fuel cells. Usage of natural gas, which is common in industry, leads to CO_2 production in order to provide emission-free power plant capture and storage of CO_2 installation is necessary. High operating temperature of SOFC gives also opportunity of efficient application them in hierarchical thermodynamic cycles [9,24,26]. By topping the combine gas and steam cycle with the SOFC it is possible to achieve almost 70% of chemical energy to electricity conversion ratio [9,26].

On the one hand, maintenance of complex hierarchical cycle is complicated, because change of each parameter of each machine in the cycle influences onto whole power plant. On the other hand, this creates an opportunity to select the most suitable parameters and lead the maintenance under long-term partial loads in way that will provide better efficiency of the thermodynamic cycle and is more proper in meaning of safety of the machinery [19, 36]. Therefore, the aim of our paper is a critical review of the hierarchical cycles, especially from the partial loading point of view.

2 Hierarchical cycles diversity

Hierarchical cycles are a wide group of thermodynamic cycles, applied in power engineering, which consist of at least two subsystems generating electricity jointed together to achieve high energy conversion efficiency ratio [7, 8, 16]. The most common combination is gas turbine – steam cycle in so-called combined cycle [1,5,7,8,10,16–19,33,34,36,43,44]. To reach optimal efficiency of energy conversion in combined cycles high temperature of gas turbine exhaust is required to power supercritical steam cycles in most efficient applications. That means high inlet temperature to gas turbine, no regeneration heat exchangers and relatively low compression ratio is required. Furthermore,gas turbines have short start-up time and load-change time characteristics, which provides fast adaptation to the energy grid needs. On the other hand steam cycle can be effectively operating under long-term part loads thanks to combination of few governing methods and quite constant inlet temperature and outlet temperature and pressure [32, 33]. That provides good flexibility of the combine cycles [5,19,36]. In combined cycles HRSG (heat recovery steam generator) replaces conventional boiler and it reaches the highest efficiency in multipressure design [1, 33, 36]. However this is the most expensive solution for gas – steam cycles. Moreover in the combined cycles less regeneration heat exchangers in steam cycle are needed.

As each machinery, also combine cycles have some disadvantages. The main one is requirement of liquid or gas fuel to power the gas turbine, which is more expensive than solid fuels. Very often the fuel price difference is so high that smaller fuel consumption is not able to compensate this differences [33]. That makes the combined cycles advisable in countries which have their own natural gas resources. However further CO_2 limitation [12] may change this situation and make the combine cycles fuelled with natural gas or syngas form gasification process more profitable.

Another hierarchical combination of thermodynamic cycles, applicable in power engineering, is piston engine – gas turbine. This combination consists of turbocharged 2 or 4 stoke piston engine fuelled with oil or gas equipped with additional gas turbine. Usually few piston engines with turbocharger-bypass exhaust supply one gas turbine. This solution is close to idea of Joule-Brayton cycle where combustion chamber is replaced with the piston engine. Like in Joule-Brayton cycle, in opposite to typical turbocharger, only part of exhaust gases energy is directed to drive the compressor, rest of them are used to drive gas turbine and generate electricity [13, 14].

Different application of piston engines is a piston engine – steam turbine cycle. Piston engine is a source of heat at few levels of temperature. Cooling water and oil is used to preheating of water and hot exhaust gases to evaporate it. Although exhaust gases often reach even 400 °C which is enough to drive steam cycle, but often more appropriate is replace steam cycle with low boiling-point liquid cycles – the ORC (organic Rankine cycle) [13, 14].

Those two systems can be applied in one piston engine – gas turbine – steam cycle [13, 14]. However usage of piston engines is justified for local-scale electric generation or cogeneration because of low output power to mass and cubature of the machinery. Additionally, clean liquid or gas fuels of high calorific value and adequate octane or cetane number are needed to provide stable operating of engines.

Next combination of hierarchical coupling is a binary vapour cycle. Using steam as a reference point, two subgroups can be highlighted. In the first group, high boiling-point liquid is used to increase cycle temperature above the steam cycle temperature level. Example is mercury – steam cycle. Mercury vapour provides higher temperature under lower pressure than steam, which calls a topping of the system. Whereas steam cycle provides steam expansion in low pressure turbine down to ambient temperature. Total efficiency of energy conversion is higher than in conventional steam cycle of additional expansion of mercury vapour in mercury cycle and lower exergy loses in boiler respectively to lower temperature difference between flame and vapour [16, 37, 39].

Secondary binary vapour cycle consists of steam cycle as a top and ORC as bottom of the system [2,4,16,23,28–31,35,38,39,41,45–47]. However, steam turbine – ORC cycle does not increase thermodynamic efficiency, because range of the thermodynamic system is the same, it should be kept in mind that each additional link in the energy conversion chain in set range of temperature cause additional exergy losses. However usage of low boiling-point liquid cause minimization of low-pressure part of the steam turbine, outflow canals and condenser. Organic, high molecular mass fluid with a boiling point occurring at a lower temperature than the water-steam phase change has much lower specific volume than steam under low pressure. Benefits of this solution are:

- Decrease in cubature of the 'cold end' part of steam turbine, therefore it is possible to build a few GW energetic blocks [31, 38, 39].
- Avoidance of steam condensation at the last stages of turbine which conduct to [2, 16]:
  - higher internal efficiency of the low-pressure (LP) turbine under full and part loads;
  - higher reliability of the LP turbine under full and part loads.
- Increase of the power plant gross efficiency if additional stream of waste heat is used in ORC, e.g., heat stream from flue [4,28–30,35, 41,46–47].

Modernizations of solid fuel steam boilers have leaded to develop pressurized fluidized bed combustion (PFBC) system. PFBC systems are in fact combined gas and steam cycles where common gas combustion chamber is replaced by a pressurized fluidized bed boiler. PFBC divides into two minor subgroups: bubbling fluidized bed (BFB) and circulating fluidized bed (CFB). In both solutions, the main purpose of gas cycle is to provide an air flow for bed fluidization. Electricity is produced 'by the way'. Usage of gas turbine with internal burning of coil, or other fuel with amount of ashes, require high effective exhaust gases cleaning systems. Apart from some construction problems, main advantages of pressurized fluidized burning are [16,42]:

- Low flame temperature 750–950 °C which avoid of NOx production.
- Utilization of SOx during combustion process.
- High solid fuel burnout.
- Possibility of use low quality fuel.

Particularly issue is direct chemical energy to electricity conversion technology. There are two main subgroups of direct energy conversion devices which can be used in the hierarchical cycles. Firs group, called magnetohydrodynamic (MHD) generators, uses magnetohydrodynamic phenomenon to convert kinetic energy of electric conducting fluid moving through magnetic field into electricity. Source of high velocity fluid can be oxy-combustion of natural gas. In this situation, exhaust needs enough hot to provide ionization phenomenon. Alternatively, it is possible to use some additives to provide electric conductivity of exhaust gases. MHD generators have low efficiency but thanks to high temperature of exhaust they are good source of high quality heat in thermodynamic cycle merged with gas cycle or combined gas and steam cycle [27,40].

Another example of direct energy conversion is fuel cell. In opposite to MHD generators, fuel cells achieve higher efficiency than heat engines. Currently few types of fuel cells are under intensive developing, however for large-scale generation solid-oxide fuel cell (SOFC) has been taken into consideration. The main feature of SOFC is high operating temperature up to 1000 °C. High temperature is required to provide proper conditions for O^{2-} ions conductivity in the electrolyte, furthermore it allows to carry out the fuel reforming and shifting phenomena inside the cells. That enables to use a natural gas as a fuel. Moreover high exhaust temperature requires waste heat recovery system. Because efficiency of SOFC increases due to increase of operating pressure, hence the intuitional solution is to merge SOFC with Joule-Brayton gas cycle. Such systems can achieve 60-70% efficiency [9,24–26].

3 Energy approach in efficiency analysis

Thermodynamic analysis should involve technical and economical aspect as well. Those criteria are essential in design process of machines and systems because impose main restrictions – costs and technical possibilities. In process of modelling of thermodynamic cycles the main restrictions are on engine top temperature and pressure.

Exergy analyse approach, next to the enthalpy balance, includes also entropy balance. That leads to complex balance of irreversible phenomenon. In result in exergy approach ideal Carnot's cycle reaches 100% efficiency which is much more comfortable to comparison between thermodynamic cycles and machinery, because we can see the limit of theoretical perfection. During energy analysis, which is no less important, the Carnot's efficiency limit is hidden somewhere between analysed cycle efficiency and 100% which is an unreachable value even for ideal, reversible processes.

Let us define the exergy, according to [37], as a sum of usable part of internal and external energy

$$B = E_k + E_p + B_t \,, \tag{1}$$

where E_p, E_k describe potential and kinetic energy respectively, B_t is thermal exergy and it consists of two elements

$$B_t = \Delta_0 B + B_{ch} \,. \tag{2}$$

The $\Delta_0 B$ element describes the physical exergy, which includes pressure and thermal exergy differences between the substance thermodynamic state and ambient parameters, while the B_{ch} parameter describes chemical energy of the substance under ambient temperature and pressure.

Narrowing down our considerations to fluid-flow machinery with adiabatic insulation from the environment, we can assume that maximal technical work of the machinery is equal to thermal exergy drop of thermodynamic fluid which can be written as

$$-\Delta B_t = I_1 - I_2 + Q_0 \,, \tag{3}$$

where I_1, I_2 are the inlet and outlet enthalpy of the process, Q_0 is the amount of useless heat exchanged with the environment.

According to the entropy definition from the II law of thermodynamics thermal exergy drop can be described as

$$-\Delta B_t = I_1 - I_2 - T_0(S_1 - S_2) \,, \tag{4}$$

where: T_0 – ambient temperature, S_1, S_2 – inlet and outlet entropy respectively.

However, if chemical energy conversion of fuel in combustion chamber, boiler or fuel cell is taken into consideration, Eq. (4) must be applied to physical energy $\Delta_0 B$. Then Eq. (2) become

$$B_t = \Delta_0 I - T_0 \Delta_0 S + B_{ch}\,. \tag{5}$$

Procedure for the evaluation of chemical exergy depends on type of the reaction and the substrates. Full procedure for combustion process is presented [37].

To close the exergy balance of the thermodynamic process it is necessary to define exergy losses. To illustrate the balance equations, model of thermal engine is presented in Fig. 1.

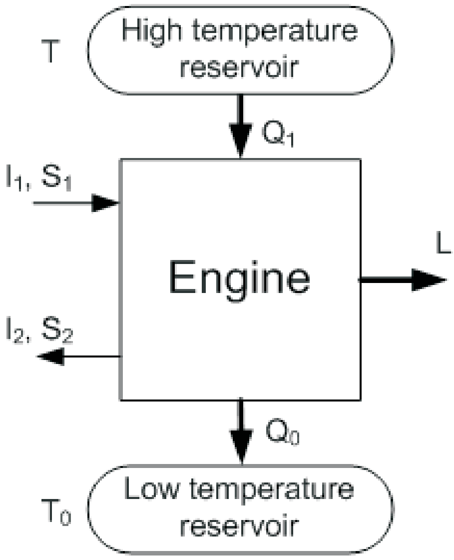

Figure 1: Model of thermal engine.

Proposed engine, e.g., closed Joule-Brayton cycle, in specified period of time uses thermal energy Q_1 gives mechanical work L and dumps worthless thermal energy Q_0. Additionally, coolant with enthalpy I_1 and entropy S_1 is heated to parameters I_2, S_2.

In general exergy losses in machinery can be define using energy balance equation of real and ideal process, respectively

$$L = Q_1 + I_1 - I_2 - Q_0\,, \tag{6}$$

$$L_{max} = Q_1 + I_1 - I_2 - Q_{0s}\,. \tag{7}$$

Using real and ideal processes work definition from (6) and (7) the exergy lossis defined as difference between them

$$\delta B = L_{max} - L = Q_0 - Q_{0s}\,. \tag{8}$$

According to the II law of thermodynamics increase in sum entropy of all bodies of the system is more than 0 and is described as

$$\Pi = -\frac{Q_1}{T} + S_2 - S_1 + \frac{Q_0}{T_0}\,. \tag{9}$$

In the ideal processthe sum of entropy increase equals zero, so,

$$0 = -\frac{Q_1}{T} + S_2 - S_1 + \frac{Q_{0s}}{T_0}\,. \tag{10}$$

By subtraction Eqs. 9 and 10 we have

$$\Pi\, T_0 = Q_0 - Q_{0s}\,. \tag{11}$$

Furthermore by inserting Eq. (11) to Eq. (8) we have defined exergy losses, as the Gouy-Stodola law

$$\delta B = \Pi\, T_0\,. \tag{12}$$

To establish the impact of each machine exergy loss onto whole system helpful in calculation of relative exergy loss. It is a proportion of the exergy losses to the cycle driving exergy, given as a mass flow rate of fuel and unit fuel exergy [37]:

$$\xi = \frac{\delta \dot{B}_i}{\dot{m}_F b_F}\,, \tag{13}$$

where: $\dot{m}_F$ is the fuel mass low rate, b_F – specific fuel exergy, and $\delta\dot{B}_i$ – stream of exergy losses.

For example relative exergy losses in simply model of steam turbine – electric generator set ξ_{T-G} and condenser, ξ_C, are given by

$$\xi_{T-G} = \frac{\dot{m}_{st}\big[(s_{st} - s_C)T_0 + (i_{st} - i_C)(1 - \eta_m\eta_G)\big]}{\dot{m}_F b_F}\,, \tag{14}$$

$$\xi_C = \frac{\dot{m}_{st}(b_{st} - b_C)}{\dot{m}_F b_F} = \frac{\dot{m}_{st} x r}{\dot{m}_F b_F}\frac{T_{st} - T_0}{T_{st}}\,, \tag{15}$$

where: $\dot{m}_{st}$ – steam mass flow rate, η_m – turbine mechanical efficiency, η_G – electric generator mechanical and electrical efficiency, x – dryness fraction of turbine outlet steam, r – specific evaporation heat, T_{st} – steam absolute temperature, s_{st}, s_c – are respectively steam and condensate specific entropy, i_{st}, i_C – steam and condensate specific enthalpy, respectively and b_{st}, b_C – steam and condensate specific exergy.

Reaching the heart of the matter, estimating exergetic efficiency of the whole thermodynamic system should be considered, not only single machines. The most general and the simplest formula of exergetic efficiency refers to rate of driving exergy, $B_{N\,s}$, of reversible ideal thermodynamic process to driving exergy of real process B_N [15,23]:

$$\eta_b = \frac{B_{N\,s}}{B_N}\,. \tag{16}$$

From Eq. (16) arises that ideal Carnot cycle reaches exergetic efficiency of 100%. That is why, in general, exergetic efficiency of thermodynamic cycles can be interpreted as comparison of analysed cycle to the Carnot ideal cycle. More accurate equation depend on kind of physical process or, in case of thermodynamic cycle, on complexity of the cycle. For instance gross efficiency of steam boiler is give as

$$\eta_{b\,B} = \frac{\dot{m}_{st}\big(b_{st} - b_w\big)}{\dot{m}_F b_F} = \frac{\dot{m}_{st}\big[i_{st} - i_w - T_0(s_{st} - s_w)\big]}{\dot{m}_F b_F}\,. \tag{17}$$

However the exergetic efficiency of more complex model of real machinery or thermodynamic system can be easy estimated by balance of losses:

$$\eta_b = 1 - \sum_1^i \xi_i\,, \tag{18}$$

where i denotes number of devices, for simply model of Clausius-Rankine cycle the exergy balance is as follows:

$$\eta_{b\,C-R} = 1 - \xi_B - \xi_{T-G} - \xi_C = 1 - (1 - \eta_{b\,B}) - \xi_{T-G} - \xi_C\,. \qquad (19)$$

To demonstrate differences between exergetic and energetic analysis of steam power plant efficiency, according to example from [37], the exergetic and energetic efficiency of each element of simplified steam cycle is presented in Tab. 1. Low thermodynamic parameters of the cycle (pressure and temperature before turbine respectively: $40{\times}10^6$ Pa/450 °C) highlights the differences of presented approaches.

Table 1: Exergetic and energetic efficiency of selected steam cycle elements.

	Related losses ξ [–]			Efficiency η [–]
	Boiler	Turbomachinery	Condenser	Power Plant
Exergetic analysis	0,68	0.08	0.04	0.20
Energetic analysis	0.20	0.02	0.57	0.22

Significant differences between presented analysis approach occur in boiler and condenser, where the largest amount of heat is exchanging. That because $T_0\Delta S$ entropy element in added to enthalpy balance in exergy Eq. (4). Entropy balance shows rate of irreversibility for each process. In exergetic analysis of power plant the highest rate of irreversibility occurs during burning process. To reduce irreversibility of this phenomena it is advisable to pre-heat the feed water and increase live steam parameters [5, 22, 37].

4 Part loads efficiency analysis of hierarchical cycles

Exergetic efficiency of each machine in the cycle has influence not only on power plant efficiency in meaning of Eq. (18) but also on others machines operational parameters, so also their efficiency. Obvious example is a gas turbine combustion chamber. Increase of exergy losses, for instance pressure droop, will decrease gas turbine pressure ratio and increase exhaust temperature. As result the HRSG exergetic efficiency will decrease. Therefore, it is not intended to analyse single machine of the system but to analyse all machines as a thermodynamic cycle [5, 10, 22, 37].

A particular issue is a part loads states analysis of thermodynamic cycles. Very often energy conversion efficiency is considered as result of design process for full load, and partial loads are considered as transient states which have to provide fast, and safe for the machinery, adjustment to the power grid requirement. In fact, in modern power systems, because of technical and economical limitations, almost each power plant has to be operated continuously and adapt to long-term load changes depended on time of day, holidays, seasons, etc. That makes the power plant part-load efficiency important issue from economic and ecological point of view [5, 11, 32, 33, 36].

Now, the aim of the chapter is to preliminarily analyse how efficiently hierarchical systems, in order to single cycles, can meet the needs of the modern power grid. To simplify the model, the analysis will based on hierarchical cycle limited to two thermodynamic cycles. Well-known combination is a combined gas and steam ideal cycle presented in Fig. 1.

Investigation of combined cycles flexibility during start-up and turn-off was conducted in [19], partial loads of steam and gas turbines in [11, 20, 21, 32, 33], dynamic regulation analysis of gas and steam turbines, separately, in [11] and of combined cycles in [36].

At the diagram of heat streams, in Fig. 2, output power and degradation of temperature are shown. That allows to understand the thermodynamic quality of the losses. For instant temperature difference between flame in combustion chamber T_1 and gas turbine inlet mean temperature $\overline{T}_{GT\,in}$ cause the biggest exergy loss. That because of irreversible of the heat exchange between combustion exhaust and compressed air, in which the biggest amount of entropy is produced. Loss of the thermodynamic potential is not given directly in simply energy balance equations, where stream of chemical energy in converted in to thermal energy with some losses related with exhaust and heat transfer from the machinery body to the environment. In this approach almost all energy from fuel combustion is given to the air stream at the turbine inlet. It describes the process in meaning of the amount of energy and not the quality. The importance of thermal energy quality onto energy conversion ratio, in heat engines, shows the equation for the cycle maximal output power, N_C, which is consistent with the ideal Carnot cycle efficiency. Using subscripts from Fig. 2 equation for maximal power N_C assumes the form

$$N_C = \dot{Q}_1 \frac{T_1 - T_0}{T_1} \; . \tag{20}$$

Stream of exergy losses, according to Eq. (12), for gas – steam cycle will

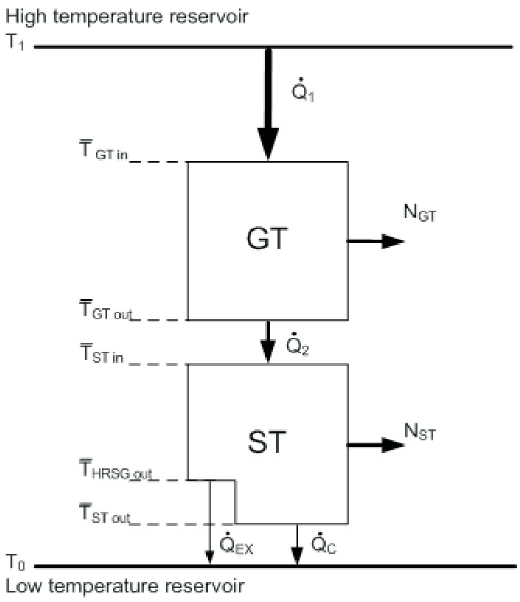

Figure 2: Diagram of hierarchical combined gas-steam cycle: GT – gas turbine, ST – steam turbine, T_0 – ambient temperature, T_1 – combustion temperature, $\overline{T}_{GT\,in}, \overline{T}_{GT\,out}, \overline{T}_{ST\,in}, \overline{T}_{ST\,out}$ – steam and gas turbines inlet and outlet mean thermodynamic temperatures, respectively, $\overline{T}_{HRSG\,out}$ – outlet mean thermodynamic temperatures of the HRGS exhaust gases, $\dot{Q}_1$ – heat flow from the combustion process to the gas turbine, $\dot{Q}_2$ – heat flow from the gas turbine exhaust to the steam cycle working fluid, $\dot{Q}_C$ – heat flow from the steam turbine condenser to the environment, $\dot{Q}_{EX}$ – heat flow from the HRSG exhaust gases to the environment, N_{GT} – gas turbine output power, N_{ST} – steam turbine output power.

be defined as

$$\begin{aligned} \delta\dot{B} = &T_0 \left(\dot{Q}_1 \frac{T_1 - \overline{T}_{GT\,in}}{T_1 \overline{T}_{GT\,in}} + \dot{Q}_2 \frac{\overline{T}_{GT\,out} - \overline{T}_{ST\,in}}{\overline{T}_{GT\,out} \overline{T}_{ST\,in}} + \right. \\ &\left. + \dot{Q}_{EX} \frac{\overline{T}_{HRSG\,out} - T_0}{\overline{T}_{HRSG\,out} T_0} + \dot{Q}_0 \frac{\overline{T}_{ST\,out} - T_0}{\overline{T}_{ST\,out} T_0} \right) \, , \end{aligned} \tag{21}$$

where mean thermodynamic temperature $\overline{T}$ in general is defined as [7,8]:

$$\overline{T} = \frac{\int_{S\,b}^{S\,a} T(S)dS}{S_b - S_a} = \frac{I_b - I_a}{S_b - S_a} \ . \tag{22}$$

Total power of the combined cycle can be estimated as a subtraction of ideal Carnot cycle output power, N_C, and the exergy losses, $\delta\dot{B}$,

$$\begin{aligned} N =& N_C - \delta\dot{B} = \dot{Q}_1 \frac{T_1 - T_0}{T_0} - T_0 \Bigg(\dot{Q}_1 \frac{T_1 - \overline{T}_{GT\,in}}{T_1 \overline{T}_{GT\,in}} + \\ & + \dot{Q}_1' \frac{\overline{T}_{GTout} - \overline{T}_{ST\,in}}{\overline{T}_{GTout}\overline{T}_{ST\,in}} + \dot{Q}_{EX} \frac{\overline{T}_{HRSG\,out} - T_0}{\overline{T}_{HRSG\,out} T_0} + \\ & + \dot{Q}_0 \frac{\overline{T}_{ST\,out} - T_0}{\overline{T}_{ST\,out} T_0} \Bigg) \end{aligned} \tag{23}$$

According to Eq. (23) thermodynamic efficiency of ideal cycle can be held at the same level under partial loads. Provided that temperatures must be held at the design point, that means that power adjustment must be realized by control of inlet heat stream, $\dot{Q}_1$, which is equivalent to working fluid mass flow rate adjusting. However, this is not possible in real machinery, particularly in power engineering applications where shaft rotation speed must be constant, because change of working fluid mass flow rate cause change of the velocity field which leads to increase of energy dissipation. Those losses are partly reduced because of specific volume increase due to pressure decrease, but their become significant during operation in conditions far from the design point [20,21,32,33].

In power engineering gas turbine power adjustment is based on flow throttling at the inlet to compressor or, in case of multishaft turbo-sets, by changing rotation speed of compressor – turbine set, turbine – electric generator set has constant rotation speed [11,33]. However flow throttling, as well as decreasing of compressor rotation speed, decrease compression ratio in the cycle, there by exhaust temperature increase. Though the working fluid mass ratio regulation is coupled with fuel stream control, if the exhaust temperature is held at the design point, the turbine inlet temperature will decrease. In practise gas turbine inlet temperature is held at the design point to reduce thermal stress in first stages of the turbine during regulation process. Theoretically, by analysis of Eq. (23), it would be advisable to use regulated recovery heat exchanger to decrease gas turbine exhaust temperature under partial loads. It would move the additional entropy production, caused by partial load, from HRSG to gas cycle and decrease entropy production in combustion chamber [33, 34]. However, additionally pressure drop at recovery heat exchanger and more expensive and complex system would make the benefits negligibly.

Influence of compressor internal efficiency onto cycle energetic efficiency was investigated [21]. For proposed model decrease of compressor efficiency of 1 pp. leads to decrease in energy efficiency of gas cycle of about 0.8 p.p. Decrease of working fluid mass flow rate cause also decrease of gas turbine internal efficiency with two times higher influence on the cycle efficiency. That means 1 pp. of gas turbine internal efficiency drop cause 1.6 pp. decrease of the cycle efficiency. For comparison, decrease of internal efficiency of high-pressure (HP) and low-pressure (LP) steam turbine cause respectively 0.33 and 0.08 decrease of steam cycle energy efficiency for parameters proposed by [21]. Similar analysis has been conducted in [34].

Changes in steam turbine internal efficiency have not so significant influent onto cycle energetic efficiency than gas turbine does. Moreover, steam turbine internal efficiency is also more stable during partial loads [11,32,33]. Additionally, output power of a steam cycle can be adjusting in few ways.

Three main methods of steam turbine governing, the most common in power engineering, are: throttle governing, nozzle governing and boiler fallow mode. In general throttle governing is a live steam stream throttling in control valve. In this mode steam mass flowrate is proportional to pressure drop, temperature and enthalpy is held at the design point [11, 20, 32, 33]. Advantage of this regulation is good flow conditions through the turbine first stage, and high internal efficiency of HP turbine. Main exergy loss of control process, according to entropy production, is generating during throttling. According to the Eq. (22) mean thermodynamic inlet temperature $\overline{T}_{ST\,in}$ is decreasing. To reduce those losses, distribution of live steam can be divided into few groups. Regulation process consists on throttling only part of the stream to reduce mass flow rate and not the total inlet pressure, this calls nozzle governing. This type of regulation causes less degradation of $\overline{T}_{ST\,in}$ but also leads to uneven load of the turbine first stage, pressure bifurcation and finally deteriorate of turbine internal efficiency. In third governing system, the boiler fallow mode, turbine inlet pressure in adjusting by feed water pump adjustment. Decrease of centrifugal pump rotation speed cause decrease of mass flow rate and pressure ratio at the same time. It reduce amount of energy consuming by pump, as well as it increases entropy production during the water evaporation under lower pressure than design level. Mentioned above governing methods often occur in hybrid solutions and each of them can be applied in gas – steam cycle.

In opposite to gas turbine, steam turbine outlet temperature does not change in wide range of power adjusting process, generally it depend on ambient temperature, but the turbine outlet mean thermodynamic temperature $\overline{T}_{ST\,in}$ does change. Nonetheless, in exergy analysis, those changes are negligible because heat exchanging in condenser is useless.

5 Conclusions

Exergetic analysis of hierarchical cycles highlights the processes during which the significant amount of entropy is produced. During those processes the largest stream of exergy is losing, which is equal to degradation of thermodynamic potential.

According to conducted investigation, partial loads did not cause exergy losses in ideal cycle, as well in hierarchical cycles as in single thermodynamic cycles. Though in real power plants increase of exergy losses under partial loads is nonnegligible. To reduce those losses, exergetic optimization is advisable. To undertake optimization of real facilities detailed analysis of cycle parameters, it is necessary to estimate exergy losses. However parameters like pressures, temperatures, mass flow rates, chemical composition of fuel, dryness fraction of turbine out flow steam, etc., should be gathered from real machinery load characteristics or from computational fluid dynamics (CFD) modeling. Only accurate input data can provide reliable results which eventuate from fact that hierarchical cycles are complex, machinery are linked together and changes of one parameter influence onto others. The complexity of the hierarchical cycles justifies usage of the exergy analysis to optimization processes, particularly the partial load states.

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