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Improvement of part load efficiency of a combined cycle power plant provisioning ancillary services

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ABSTRACT

According to the type of ancillary service provisioned, operation mode of a power plant may change to part load operation. In this contribution, part load operation is understood as delivering a lower power output than possible at given ambient temperature because of gas turbine power output control. If it is economically justified, a power plant may operate in the part load mode for longer time. Part load performance of a newly built 80 MW combined cycle in Slovakia was studied in order to assess the possibilities for fuel savings. Based on online monitoring data three possibilities were identified: condensate preheating by activation of the currently idle hot water section; change in steam condensing pressure regulation strategy; and the most important gas turbine inlet air preheating. It may seem to be in contradiction with the well proven concept of gas turbine inlet air cooling, which has however been developed for boosting the gas turbine cycles in full load operation. On the contrary, in a combined cycle in the part load operation mode, elevated inlet air temperature does not affect the part load operation of gas turbines but it causes more high pressure steam to be raised in HRSG, which leads to higher steam turbine power output. As a result, less fuel needs to be combusted in gas turbines in order to achieve the requested combined cycle's power output. By simultaneous application of all three proposals, more than a 2% decrease in the power plant's natural gas consumption can be achieved with only minor capital expenses needed.

1. Introduction

After the electricity market deregulation, ancillary services provisioning quickly became a part of the energy policy of the largest electricity producers and consumers in Slovakia; with Slovenské elektrárne as the largest electricity producer in Slovakia [1], combined heat and power plants, such as the thermal power plant in Košice [2], or combined cycle power plant in Bratislava [3] being some of them. Generally, there are three most common types of ancillary services in Slovakia – primary and secondary frequency regulation and tertiary reserve. Tertiary reserve is further divided into several categories according to the time required to achieve and stabilize the requested electric output. A comparison of ancillary services markets in different countries can be found in literature [4,5]. Remuneration for ancillary services provisioning includes the payments for capacity and for operation. Maximal prices to be paid for capacity and for operation are yearly given by the Slovak Regulatory Office for Network Industries [6] decrees.

Based on the present situation, ancillary services provisioning is an attractive option for improving the economics of the companies involved, especially because of growing fuel costs which are not followed by an immediate increase of the electricity costs. According to the type and extent of ancillary services, various fuel savings can be achieved; however, especially the secondary frequency regulation, by its nature, is connected with a permanent part load operation of the given power plant. Worsened operation parameters in the part load operation cause smaller fuel savings than expected.

It has to be stressed that, according to the amount of scientific articles on this topic, still much more attention is paid to the optimization of full load operation [7–13] than to that of part load operation [14–17] of a combined cycle, as it is stated in [15], since the full load operation is assumed to be far more probable than the part load one. In [7], optimal solutions for gas turbine blade cooling were sought incorporating it into a complex thermodynamic model of the whole CCPP, which results in a set of plant efficiency and specific work values for different operation parameters; however, the possibility of part load operation is not even considered. A CCPP consisting of two gas turbines and one steam turbine is described in [8] which strive to find optimal operating modes of this CCPP at different ambient temperatures and load percentage. However, attention is paid to optimization of air cooled condenser operation; CCPP part load operation is mentioned only briefly with less attention paid to its optimization than to CCPP full load operation

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Nomenclature

CCPP	combined cycle power plant	P_e	electric output
GT	gas turbine	ST	steam turbine
HP	high pressure	T_a	ambient temperature
HRSG	heat recovery steam generator	T_c	condensing temperature
LHV	lower heating value	$\dot{V}$	volumetric flow rate
MP	middle pressure	η_e	electric efficiency
NG	natural gas		

optimization. The aim of [9] was to compare two gas turbine design solutions and to optimize their performance by varying design parameters such as gas turbine firing temperature, pinch point in HRSG, and steam condensing; however, the possibility to operate at lower outputs remained untouched. In [10], where the possibility of using liquefied natural gas as a source for gas turbine inlet air cooling is described, it is explicitly stated that the authors dealt only with full load operation at given ambient temperatures. Authors of [11] carried out a detailed analysis of the four most common types of combined heat and power plants in chemical industry by varying several decision variables, part load operation according to our definition is not considered. In [12], a Brazilian steel mill's steam power plant is considered to be supplanted by a CCPP fired by blast furnace gas and supplementary fired by coke oven gas; the CCPP is planned to run at full load with excess electricity being sold into the national grid. Design possibilities for efficiency increase of a CCPP were analyzed in [13], by means of its energy balances calculation and exergy losses minimization. They concluded that a multi-pressure HRSG adoption together with gas turbine inlet air cooling and natural gas preheating in full load operation are able to decrease CCPP's specific fuel consumption by several percents and to increase its output.

Published articles [14,15] on optimization of CCPP part load operation usually deal with strategies used to maintain the same firing temperature in part load as in full load operation by reducing the gas turbine inlet air mass flow by varying the inlet guide vane position. This concept is already being widely used by gas turbine manufacturers. Another operating mode can be applied in those CCPPs which comprise two or more gas turbines, [8] – one or more gas turbines can be shut down during part load operation. Yet another concept, indirectly described in [16] was proven and made use of in [17], and is based on measured CCPP performance data. This concept, as described below, is unique in that it does not change the GT design or operating strategy but uses gas turbine inlet air preheating which improves the CCPP part load operation efficiency. This may seem to contradict the widely adopted concept of inlet air cooling, but its applicability will be, similarly to [17], proven based on the measured data. Therefore, when optimizing CCPP part load operation, some surprising conclusions about what measures could be undertaken compared to what is known about optimization of full load operation can be drawn.

2. Case study – plant description

Object of our interest was a newly built 80 MW_e (ISO conditions) net power output CCPP in Central Slovakia fired by natural gas. Its scheme is depicted in Fig. 1. Besides producing electricity and providing ancillary services, this power plant was designed to deliver hot water for district heating of nearby town as well as to deliver MP steam to nearby industry. However, both district heating and MP steam supply have been under unsuccessful negotiations for longer time, thus the power plant operates in fully condensing mode with its cogeneration potential being not exploited. Design parameters of the CCPP in full load operation are shown in Table 1.

Because of secondary frequency regulation service provisioning, this power plant operates in part load whereby its performance parameters are worsened. In this article, part load operation is considered as delivering lower power output than the maximum possible at the given ambient temperature, which is a rather different definition than the usual one stating that a part load operation is every operation performed under non-ISO conditions.

In order to meet the conditions for secondary frequency regulation provisioning, a power plant in Slovakia must be able to offer at least ±3 MW automatic generation control to be controlled from the National Dispatch Centre of Slovak Republic, which is incorporated in the Slovak Electricity Transmission System, Plc. [18] and the power plant must be able to respond to change in load demand with a gradient of at least 1.5 MW min^{-1}. Fig. 2 elucidates the part load operation range of the studied power plant. As can be seen, gas turbines are the more flexible part of the whole set; their output is under automatic generation control. On the contrary, steam turbine's output is not controlled directly; it reacts on steam generation rate changes depending on gas turbines operation. The CCPP typically operates in the range of 28–82 MW (e.g. 30–100% output) with the base load of 55 MW. Usually, secondary frequency regulation covers the whole output range, offering ±27 MW for secondary frequency regulation. According to the CCPP energy policy, a decrease of the base load to 40 MW is not rare, which reduces secondary frequency regulation to ±12 MW and the output range of 52–82 MW is offered on the market as tertiary reserve. Considering the produced electricity selling price of approximately 1800 Sk[1]/MWh, growing natural gas cost and maximal payments for the capacity given by [6] as 2020 Sk/MW/h for secondary frequency regulation and 780 Sk/MW/h for tertiary reserve, it is understandable why part load operation combined with provisioning secondary frequency regulation is at the moment a more feasible option than the full load operation. As an example of gross generated output changes, Fig. 3 documents its typical course over a randomly chosen decade.

3. Methodology

After being introduced to the CCPP's technological scheme, an extensive set of measured data, averaged over one hour to suppress the possible influence of CCPP unsteady state operation on their accuracy was obtained. These data comprised values of temperatures, pressures and mass flows of process streams, and NG consumption and gross generated output. This data set described the CCPP operation during warm as well as cold days, in part load as well as in full load operation. Data describing operation during start-up or shutdown of GT or ST were discarded. Based on GT, HRSG, and ST operation principles, correlations among various measures were sought and quantified. Some of the obtained trends

[1] 1€ = 30.126 Sk.

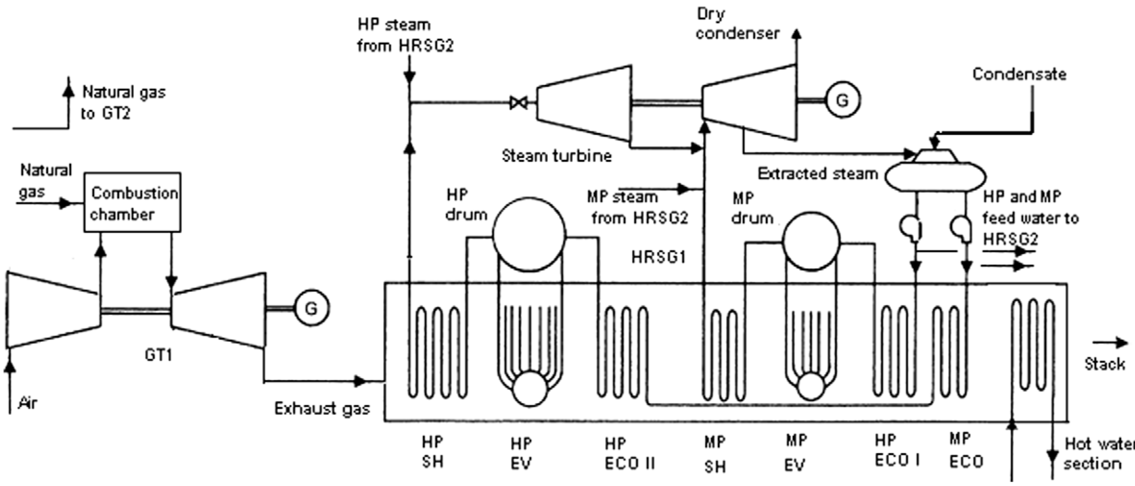

Fig. 1. Scheme of combined cycle power plant studied.

are shown in Figs. 4–10. These trends together with technological design of the studied CCPP helped us to identify areas of further interest with the aim to find technological devices whose operation strategy modification could, under present conditions, lead to overall CCPP efficiency improvement.

4. General remarks to CCPP operation

Based on the online monitoring of CCPP gross power output, natural gas consumption and under assumption of average LHV of natural gas – 34.26 MJ/m^3 [19] (15 °C, 101.325 kPa), overall electric efficiency is plotted as a function of heat input in natural gas in Fig. 4.

Overall electric efficiency is calculated via

$$\eta_e = \frac{P_{e,GT} + P_{e,ST}}{\overset{*}{V}_{NG} \cdot \mathrm{LHV}} \tag{1}$$

Based on Fig. 4 it can be seen how the electric efficiency of this power plant deteriorates in part load operation. Power output of the power plant as a function of heat input in NG can be approximated by a straight line [16] with a non-zero intersect and with the slope of 0.6364 MW/MW. At the same time, it gives the value of marginal electric efficiency of the CCPP (63.64%) which is higher than the overall electric efficiency even at full load (see Table 1). Or, inversely, marginal heat rate of 5657 kJ/kWh can be obtained by dividing 3600 by the marginal electric efficiency. This value characterizes the combined cycle heat input change when power output change is requested. Parasitic load of power plant currently ranges up to 1.2 MW, which is 1.5–3% of the gross power output. Because of its small influence on the power output, gross power output was used instead of the net one in source data for Fig. 4.

Table 1
Design parameters of combined cycle at full load at ISO conditions.

Parameter description	Value	Parameter description	Value
Overall electric efficiency in fully condensing mode (based on LHV of natural gas)	50%	HP steam pressure	6.25 MPa
Nominal net power output	80 MW	HP steam temperature	440 °C
Parasitic load	1.2 MW	MP steam pressure	1.3 MPa
Maximal district heating duty	32 MW	MP steam temperature	250 °C
Controlled steam extraction pressure	0.4 MPa	Deaeration temperature	102 °C

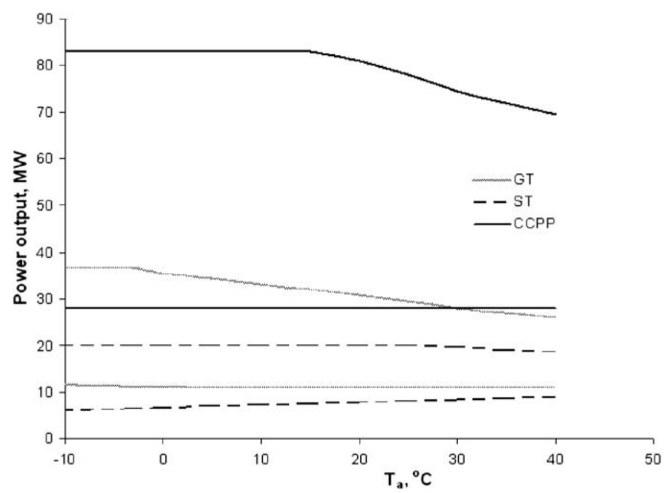

Fig. 2. Operation limits of gas turbines and steam turbine; typical lower and upper load limit of combined cycle.

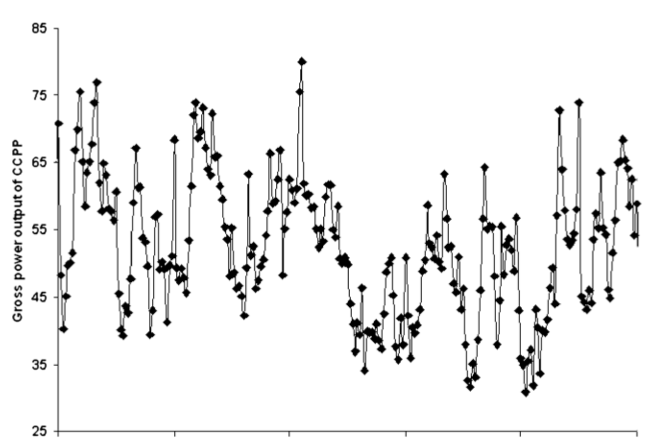

Fig. 3. Typical CCPP gross power output values over a decade.

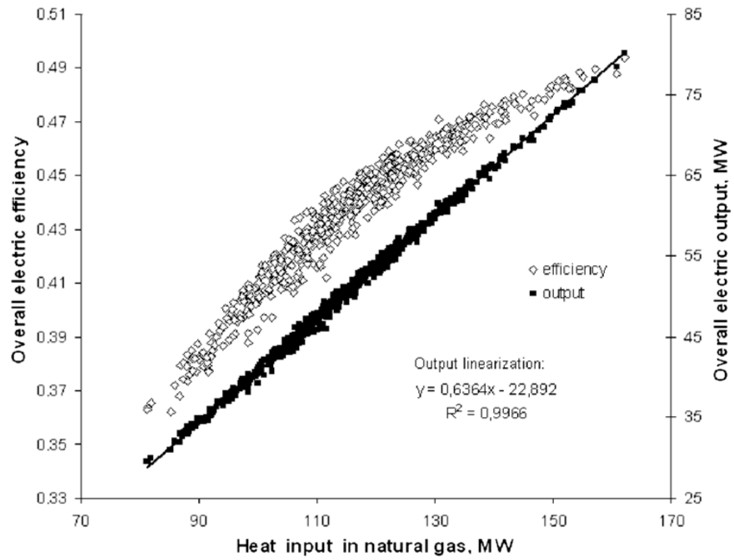

Fig. 4. Overall electric efficiency and overall electric output plotted as functions of heat input in natural gas.

In the fully condensing operation mode, overall power plant efficiency equals to its overall electric efficiency. However, the power plant was designed as a cogeneration device. For this purpose, HRSGs include a hot water section, where water should be heated from 70 °C to 110 °C and pumped in the heating network of a nearby city. Maximal heating duty of each section is 6 MW; peak heat demand is to be covered by a 20 MW duty heat exchanger, supplied by condensing 0.4 MPa steam extracted from steam turbine. Presently, flue gases leave the HRSGs at around 150 °C causing considerable stack losses; in case of the hot water section operation their temperature could be lowered to 100 °C.

5. Improving performance of studied power plant

5.1. Condensate heating and deaeration

Condensate is heated from 35–60 °C to 102 °C and deaerated in a deaerator by direct contact with condensing low pressure steam (see Fig. 1). Accomplishing such temperature rise in one stage

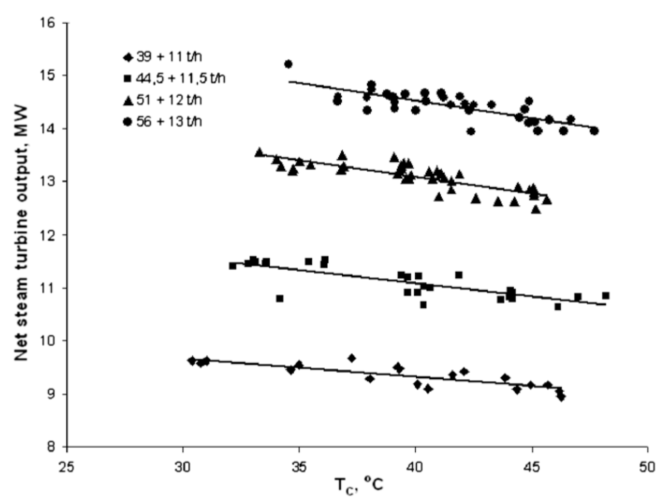

Fig. 6. Steam turbine output minus combined cycle's parasitic load as a function of steam condensing temperature for various HP + MP steam generation rates.

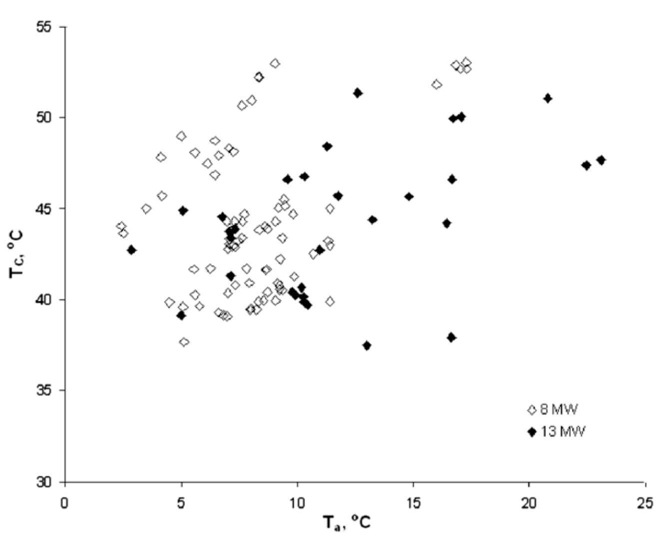

Fig. 5. Steam condensing temperature in dry condenser versus ambient temperature for two steam turbine outputs.

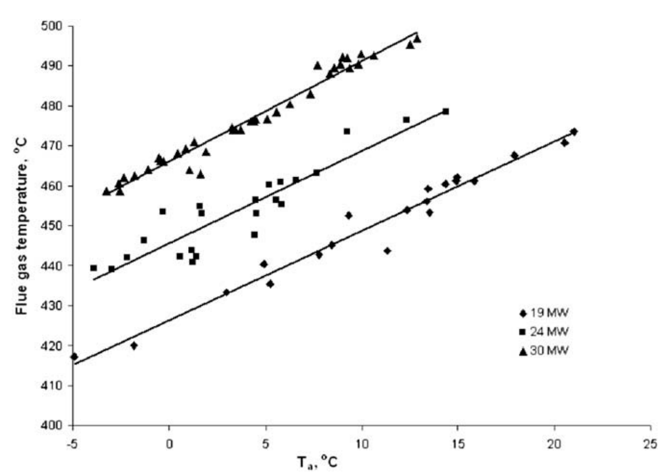

Fig. 7. Temperature of flue gases exiting gas turbine as a function of ambient temperature for three typical gas turbine power outputs.

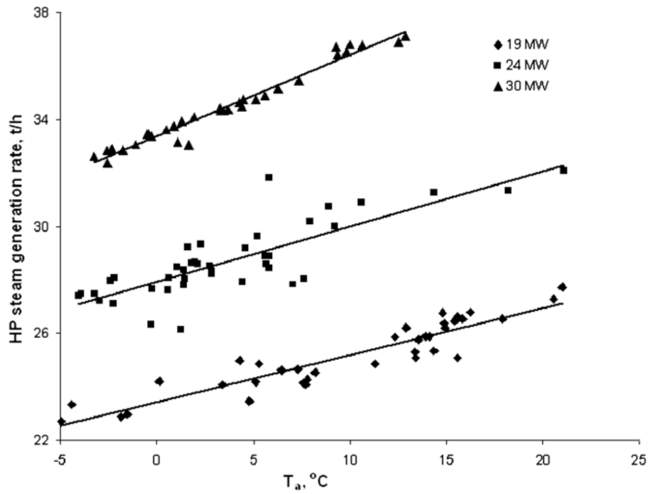

Fig. 8. HP steam generation rate in one HRSG as a function of ambient temperature for three typical gas turbine power outputs.

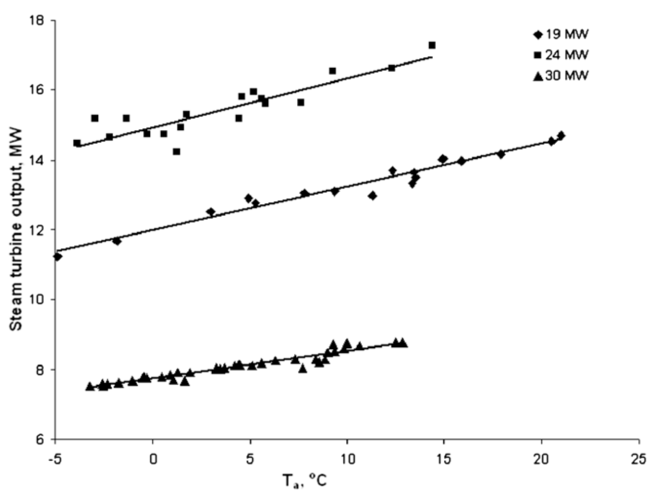

Fig. 9. Steam turbine output as a function of ambient temperature for three typical gas turbine outputs. Squares and diamonds: both gas turbines are in operation. Triangles: only one gas turbine operates.

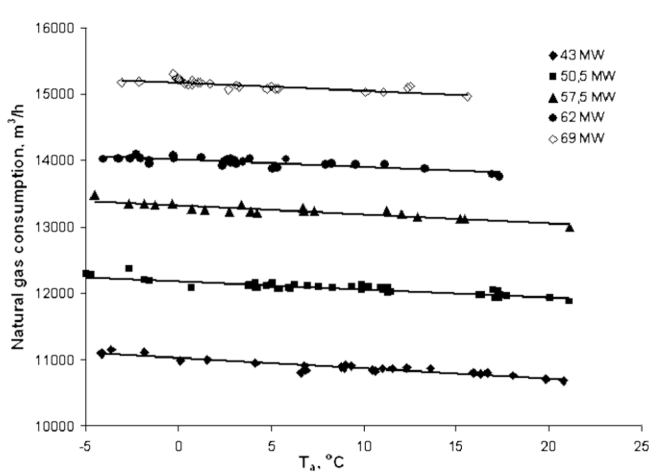

Fig. 10. Natural gas consumption needed to achieve specific combined cycle's outputs (see legend) as a function of ambient temperature.

requires extracting around 0.1 kg of steam per 1 kg of water to be heated, which subsequently leads to a considerable decrease of steam mass flow in the low pressure part of steam turbine thus decreasing its power output. According to [10,13] and to common practice, it is usual to preheat the condensate before deaeration by flue gases prior to their exit from HRSG. In our case study, hot water section, which has been designed to recover additional heat from flue gases, is idle and therefore may be used to preheat condensate. Two options are possible: (a) Allow the condensate to warm up directly in the hot water section; in order to ensure satisfactory deaerator performance the upper preheat temperature limit was set to 90 °C. (b) If the first option is not possible, hot water section should be used as designed; water heated to 110 °C is used to preheat the condensate to 90 °C in a plate heat exchanger. Nominal heat duty of 4 MW is enough to preheat all condensate to 90 °C even in peak load operation.

Even if the combined cycle starts to deliver heat to the nearby city, the expected heat demand of 4 MW during summer is readily covered by one hot water section duty (see Table 1) while the other one can still be used according to one of the two options described above. This measure can be thereby applied for nearly 5 months in a year, which still allows it to be economically attractive. Savings resulting from applying this measure are summed up in Table 2.

5.2. Steam condensing temperature regulation

Wet steam leaves the low pressure part of steam turbine and condenses in dry condenser at 8–20 kPa. Condensing pressure is much more dependent on ambient temperature than in the case of condenser cooling with cooling water, thus, the resulting steam turbine output depends on condensing steam mass flow, ambient temperature as well as cooling fans operation [8]. Analysis of condenser performance is shown in Fig. 5, which clearly indicates that condensing pressure regulation by cooling fans operation does not work properly. Condensing temperature differences by more than 10 °C at the same ambient temperature and the same steam turbine output mean that the condensing pressure control is by far not optimal. In future, regulation of cooling fans operation is planned to ensure stable condensing pressure of 12 kPa (e.g. condensing temperature of 49 °C).

However, this strategy seems also false since lower condensing pressure can be achieved without a significant increase of parasitic load during colder months, as was described in [8]. This can be proved by Fig. 6, which, based on online monitoring data gives steam turbine power output minus parasitic load over a broad range of condensing temperatures at constant HP and MP steam generation rate. Since the net power output increases with decreasing temperature difference it seems reasonable to change the regulation strategy of condensing pressure. Considering the cooling fan system, temperature difference $T_c - T_a$ as low as 20–30 °C can be achieved, which gives the operation system the possibility to keep the condensing pressure at lower values than 12 kPa. Because, according to its technical data, the condenser can be safely operated under pressures down to 8 kPa and a range of ±1 kPa must be allowed for reliable cooling fan operation, condensing pressure can be set to 9 kPa, which ensures both safe condenser operation as well as decreasing fuel consumption in part load operation of the combined cycle.

Savings resulting from changing the condensing pressure control strategy can be identified via the net steam turbine output increase (see Fig. 6). The average value taken into consideration was 50 kW °C^{-1}. Due to climatic conditions on the site, a decrease of the condensing pressure from 12 kPa to 9 kPa is possible during six colder months, as is indicated by yearly average ambient temperature of 11 °C as well. This corresponds to the decrease in condensing temperature from 49 °C to 44 °C. Table 3 gives an overview of annual achievable savings. This measure requires no investments and can be applied immediately.

Table 2
Overview of natural gas savings calculation by condensate preheating.

Factor description	Value
Average extraction steam savings by condensate preheating	5 t h^{-1}
Steam turbine power output increase	700 kW
Natural gas savings (based on CCPP marginal heat rate and average LHV of natural gas)	116 m^3 h^{-1}
Annual natural gas savings (5 month operation per year)	420,000 m^3

Table 3
Overview of natural gas savings calculation by changing regulation strategy of steam condensing pressure.

Factor description	Value
Net steam turbine power output increase	50 kW °C^{-1}
Condensing temperature decrease	5 °C
Net steam turbine power output increase	250 kW
Natural gas savings (based on marginal heat rate and average LHV of natural gas)	40 m^3 h^{-1}
Annual natural gas savings (6 month operation per year)	180,000 m^3

Table 4
Overview of natural gas savings calculation by gas turbine inlet air preheating.

Factor description	Value
Natural gas specific consumption decrease	13 m^3 h^{-1} °C^{-1}
Natural gas consumption decrease	312 m^3 h^{-1}
Parasitic load increase	300 kW
Natural gas consumption increase to cover parasitic load increase (based on marginal heat rate and average LHV of natural gas)	50 m^3 h^{-1}
Resulting natural gas consumption decrease	262 m^3 h^{-1}
Annual operating hours	8000 h
Annual natural gas savings (incorporates factor 0.8; see corresponding text)	1,680,000 m^3

5.3. Gas turbine inlet air conditioning

Gas turbine inlet air conditioning is a well known operation which includes air moisturizing [20,21] or indirect cooling via compression or absorption cooling [13,22], or via other methods [10]. It is applied to existing as well as to newly built gas turbines, especially in areas with hot climate. Its aim is to prevent gas turbines from performance deteriorating in hot climate if they have to operate at full load or if they have to provide peak power to the grid; it enables the gas turbines to provide higher peak load than without inlet air cooling. However, the yearly average ambient temperature in the area where the power plant is built is around 11 °C, which means that for the colder half of the year, inlet air cooling is not needed at all, since the inlet air is cold enough. After considering the investments and maintenance costs of an air cooling device, its pay back period was not attractive enough and the CCPP refused this investment possibility.

In order to investigate the ambient temperature influence on the part load operation, firstly, flue gas temperature at the exit of gas turbines and HP steam generation rate in HRSGs were studied. They are shown in Figs. 7 and 8. As can be seen, both flue gas temperature and HP steam generation rate are strongly affected by ambient temperature changes. Higher ambient temperature causes the flue gas temperature to increase as well and the same holds true for the HP steam generation rate. Moreover, higher flue gas temperature enables the HP steam to reach its design temperature and pressure whereas on cold days with lower gas turbine output, HP steam temperature derates to less than 400 °C and its pressure lowers to less than 5.3 MPa.

Increased steam generation rate causes the steam turbine output to increase, as is demonstrated in Fig. 9. The increase in steam turbine power output is quite significant – mostly over 100 kW per 1 °C increase in ambient temperature. Summed up, in part load operation, elevated ambient temperature has positive effect on the overall electric efficiency of this power plant, which is clearly proven by the online measured data depicted in Fig. 10. As can be seen, in order to achieve the same electric output of the whole power plant, less fuel needs to be combusted at elevated ambient temperatures than at lower temperatures. This finding is quite surprising but it can be used to boost the combined cycle electric efficiency in part load operation. Supporting evidence of the elevated ambient air temperature beneficial effect on combined cycle's part load performance can be found in [16] and especially in [17].

Therefore, our proposal was to use some waste heat source for controlled inlet air heating. The largest waste heat source by far is the dry condenser; a part of the air heated in it could directly be supplied to gas turbines. In winter, air could be further heated either by flue gases exiting the HRSGs in a gas/gas heat exchanger, or by condensing low pressure steam extracted from steam turbine in a gas/steam heat exchanger. Thus our proposal is somewhat different from that in [17], since we have adopted dry condenser as the prime source of heated air, whereas in [17] it is a gas/water heat exchanger with water being heated by condensing steam. Only if the peak load operation is required, heated air flow will be diverted elsewhere and gas turbines will take up fresh cold air.

Savings estimation is based on the fact that it is a new concept; to ensure its safe start, target temperature of heated air was initially set to 35 °C, which is a temperature that is measured each year a few times in Slovakia. Compared with the average temperature of 11 °C, there is an average difference of 24 °C which can be made use of. Further limits of the temperature increase are set by gas turbine operation which in the case of peak load demand require fresh air intake. Considering 70 MW overall power plant output (see Fig. 6) as the upper limit for air preheating to 35 °C, power plant operates above the 70 MW output approximately 20% of the time; therefore annual savings have to be multiplied by factor 0.8. Parasitic load increase by approximately 300 kW must be taken into account; it is caused by more intense cooling fans operation due to higher steam turbine output as well as by a new fan delivering preheated air to gas turbines. Based on the source data in Fig. 10, natural gas consumption is expected to decrease by 13 m^3 h^{-1} for each 1 °C of gas turbine inlet air temperature increase. Table 4 gives an overview of savings calculated.

Table 5
Overview of total natural gas savings and its impact on average electric efficiency and CO_2 emissions.

Factor description	Value
Average power output of power plant	55 MW
Average natural gas consumption	13,250 m^3 h^{-1}
Average annual natural gas consumption (8000 annual operating hours)	106,000,000 m^3
Average electric efficiency (condensing mode)	0.44
Total natural gas savings	2,280,000 m^3
Average electric efficiency after implementation of proposed measures (condensing mode)	0.45
Annual avoided CO_2 emissions	3400 t

In Table 5, achievable savings are summed up and compared with the average yearly natural gas consumption of combined cycle power plant. Total natural gas saving can be calculated as a sum of savings resulting from each proposal since the beneficial and malefic effects of simultaneous application of all three proposals are likely to cancel each other out or the beneficial ones may slightly prevail. In summary, NG savings lead to more than a 2% decrease in the average NG consumption. As a result, average electric efficiency of the given power plant in condensing mode increases from 44% to 45%. Moreover, CO_2 emissions decrease by nearly 3400 t per year.

6. Conclusions

Part load operation of a power plant deserves more attention since it becomes a more usual mode in a deregulated electricity market with ancillary services provisioning being an established part of it. In order to discover and quantify possible savings accompanied by efficiency improvement in the part load operation mode, an 80 MW$_e$ combined cycle power plant was studied. Based on on-line monitoring data, it was possible to propose three ways of achieving fuel savings, with gas turbine inlet air preheating being a quite unusual one. Benefits obtained from gas turbine inlet air preheating were clearly demonstrated, thus from theoretical point of view, it can be applied to every CCPP operating mostly in the part load mode. However, as with each new concept, CCPP designers need proofs of its successful operating capability. That means it has to be tested in several CCPPs willing to undergo this procedure before technical issues are clarified and this concept becomes a more widespread technology. In future, we will continue to monitor the CCPP in question, especially after it starts to work as a real cogeneration unit to fully assess its impact on proposed NG saving measures.

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