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> PRESTAZIONI DI SISTEMI INTEGRATI TERMOVALORIZZATORI + CICLI COMBINATI IN CONDIZIONI DI FUORI PROGETTO

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RAPPORTO 2.1/5

> PRESTAZIONI DI SISTEMI INTEGRATI TERMOVALORIZZATORI + CICLI COMBINATI IN CONDIZIONI DI FUORI PROGETTO

In accordo con il piano di lavoro proposto, il rapporto 2.1/5 è costituito dall'articolo "OFF-DESIGN PERFORMANCE OF INTEGRATED WASTE-TO-ENERGY, COMBINED CYCLE PLANTS", autori S. Consonni e P. Silva, Atti del congresso ASME-ATI 2006, Meeting on Energy: production, distribution and conservation - Milan, May 14-17, 2006 Vol. 1, pp. 81- 92. Published by SGEditoriali, Padova, Italy. Il testo è allegato di seguito.

OFF-DESIGN PERFORMANCE OF INTEGRATED WASTE-TO-ENERGY, COMBINED CYCLE PLANTS

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Summary

This paper focuses on the off-design operation of plants where a Waste-To-Energy (WTE) system fed with Municipal Solid Waste (MSW) is integrated with a natural gas-fired Combined Cycle (CC). Integration is accomplished by sharing the steam cycle: saturated steam generated in a MSW grate combustor is exported to the Heat Recovery Steam Generator (HRSG) of the Combined Cycle, where it is superheated and then fed to a steam turbine serving both the CC and the WTE plant.

Most likely, the WTE section and the natural gas-fired CC section are subject to different operation and maintenance schedules, so that the integrated plant operates in conditions different from those giving full power output. In this paper we discuss and give performance estimates for the two situations that delimit the range of operating conditions: (a) WTE plant at full power and gas turbine down; (b) WTE plant down and gas turbine at full power. This is done for two integrated plants having the same WTE section, i.e. grate combustors with an overall MSW combustion power of 180 MW_{LHV}, coupled with Combined Cycles based on two different heavy-duty gas turbines: a medium-size, 70 MW class turbine and a large-size, 250 MW class turbine.

For each situation we discuss the control strategy and the actions that can help to achieve safe and reliable off-design operation. Heat and mass balances and performances at off-design conditions are estimated by accounting for the constraints imposed by the available heat transfer areas in boilers, heaters and condenser, as well as the characteristic curve of the steam turbine. When the gas turbine is down the net electric efficiency of the WTE section is very close to the one of the stand-alone WTE plant; instead, when the WTE section is down, the efficiency of the CC is much below the one of a stand alone CC. These performances appear most congenial to what is likely to be the operational strategy of these plants, i.e. paramount priority to waste treatment and CC dispatched according to the requirements of the national grid.

Nomenclature

A	Heat transfer area
CC	Combined Cycle
G	Reduced mass flow
GT	Gas Turbine
HP, IP, LP	High, Intermediate, Low Pressure
HRSG	Heat Recovery Steam Generator
LHV	Lower Heating Value
MSW	Municipal Solid Waste

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RH	Reheat
U	Overall heat transfer coefficient
WTE	Waste-To-Energy
ψ	Load coefficient

1. Background

The work presented in this paper originates from a research sponsored by CESI and supported by AMSA Milano on the integration between grate combustor WTE plants and natural gas-fired Combined Cycles [1]. Integration can be carried out by sharing just the steam cycle or by sharing also the flue gas paths. As discussed in [2], sharing the flue gas path by sending the gas turbine exhausts to the grate combustor is unattractive for a number of reasons. This is why in this paper we restrict our attention to the first option, i.e. configurations where the WTE plant and the CC share only the steam cycle, while the combustion products of waste and natural gas follow completely separate paths and undergo different treatment processes. The boiler of the WTE section generates saturated (or slightly superheated) steam that is exported to the HRSG of the Combined Cycle, where it mixes with the steam generated by the gas turbine exhausts and it is superheated up to typical CC temperatures. The potential advantages of this arrangement are illustrated in a number of papers [3-7] and can be summarized as follows.

1) Eliminating the superheater from the WTE boiler increases reliability and decreases costs. In fact, the corrosion of superheater tubes is typically a major source of forced outages and expensive maintenance operations.
2) By generating most of the steam in the WTE section, lower mean temperature differences can be achieved in the HSRG. This implies lower irreversibilities and thus higher efficiencies, although the corresponding increase of heat transfer areas gives somewhat higher capital costs.
3) The single, relatively large steam turbine serving both the WTE section and the CC can achieve efficiencies much higher than those typically achieved by the steam turbines of WTE plants, which rarely go beyond 50-60 MW.
4) Sharing a large fraction of the steam cycle (steam turbine, condenser, feedwater pumps, etc.), as well as controls, auxiliary systems, electric equipment and civil works allows significant reductions of capital and O&M costs.
5) Discharging the flue gases of the WTE plant and the CC through the same stack can significantly reduce local environmental impact.

The optimal configuration of the integrated plant varies with the operating parameters (evaporation pressure, maximum steam temperature, minimum flue gas temperature, etc.) and the ratio between the combustion power supplied by waste and that supplied by natural gas. In this paper we consider two situations relevant to the new WTE plant being considered for the city of Milano, which will comprise three air-cooled grate combustors fed with residual MSW[1] for a total LHV combustion power of 180 MW. For the CC section we've considered two different classes of heavy-duty gas turbines:

- medium-scale, 70 MW class;
- large scale, 250 MW class.

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[1] "Residual" means that the waste fed to the grate combustor is what is left after selective garbage collection and material recycling.

As representative of the medium scale class we’ve referred to the General Electric 6FA; very similar results would be obtained for the Siemens V64.3a. For the large scale machine we’ve considered cycle parameters and performances close to those of the General Electric 9FA and the Siemens V94.3a2.

2. On-design conditions

The new WTE plant being considered for the city of Milano comprises a rather advanced steam cycle with evaporation pressure at 85 bar, dry flue gas clean-up with lime and active carbon and Selective Catalytic Reduction for NOx control. Tab. 1 summarizes the main on-design features of the stand-alone arrangement.

Thermal power input [MW_{LHV}]	180.8
LHV_{MSW} [MJ/kg]	10
Un-burnt fraction [% of thermal input]	1
Thermal losses [% of thermal input]	1
Air pre-heat final temperature [°C]	115
O_2 flue gases vol. content [%]	5
Exhaust gas recirculation [%]	15
Gross electric power output [MW]	58.91
Net electric power output [MW]	51.99
Net electric efficiency [%]	28.76

Tab. 1 *Basic on-design features of the stand-alone WTE plant.*

The main assumptions adopted to estimate the on-design heat and mass balances and the performances of the integrated plants are summarized in Tab. 2. The relative high subcooling ΔT of 25°C for the medium-size GT has been chosen to reduce the chances of steaming at off-design conditions. In all cases the condenser is cooled with water at 22°C made available by wet cooling towers.

In the plant based on the medium scale gas turbine, the thermal capacity of the gases in the HRSG is insufficient to heat the whole flow of water; in addition to the de-aerator, we have considered therefore a feedwater heater fed with steam bled from the steam turbine at 1.1 bar, placed in parallel to the economizer of the HRSG. A more sophisticated feedwater heating line with additional regenerators and more steam bleedings would increase electric power output by 1-1.5 MW; its convenience should be verified by weighing such added power output against added cost and complexity. In the plant based on the large scale gas turbine, a low-temperature regenerator has been considered only for the off-design condition where the gas turbine is down and no heat source is available to heat the water ahead of the de-aerator; at on-design conditions, the regenerator is excluded by shutting off the bleed that provides its steam supply.

Fig. 1 illustrates the configuration and the on-design operating conditions assumed for the integrated plant based on the medium scale gas turbine. Almost all the heat recoverable from the gas turbine exhausts is

	gas turbine	
	medium	large
Pressures [bar]		
evaporator(s)	65	85 / 23 / 4.3 (a)
bleed for air heater, de-aerator and district heating	2.4	2.4
deaerator	1.4	1.4
bleed for feedwater heater	1,1	1.1(b)
bleed for air heater	0.6	0.6
condenser	0.07	0.07
Temperatures and ΔT [°C]		
max steam temperature (SH / RH)	550	535 / 560
ΔT at pinch point	10	10 / 10 / 15 (a)
minimum approach ΔT	25	20 / 15 / 10 (a)
subcooling at economizer outlet	25	2 / 2 / 10 (a)

(a) Respectevely for HP / IP / LP level
(b) Only at off-design conditions, when the gas turbine is down

Tab. 2 *Main assumptions adopted for the on-design conditions of integrated WTE-CC plants.*

needed to superheat the steam generated in the WTE section and pre-heat the water. As a consequence, multiple evaporation pressures or reheat are useless and we have assumed a single level at 65 bar to optimize the temperature profile in the HRSG and maximize power output. At the on-design conditions reported in Fig. 1 the production of steam in the evaporator of the HRSG is nearly zero. Nonetheless, an evaporator tube bank with adequate heat transfer area is necessary to operate the plant properly when the WTE plant is down. As illustrated further, an acceptable off-design situation can be obtained by imposing an on-design pinch point ΔT of 10°C.

The configuration and the on-design operating conditions of the integrated plant based on the large scale turbine are reported in Fig. 2. In this case the heat recoverable from the gas turbine exhausts is much larger than that needed to superheat the steam generated in the WTE section and there is room for generating additional steam at multiple pressures in the HRSG. The configuration with three pressure level and reheat assumed for this case is similar to that adopted in utility-scale units.

3. Off-design operation

The two most severe conditions that delimit the operating range of an integrated WTE-CC plant are the ones where one section is down (either the WTE or the CC) and the other is at full power. In the liberalized electricity market established in Italy where natural gas fired CC are typically operated 5-6000 hours per year, the most probable occurrence is a shut-down of the CC. Given the utmost priority normally given to waste treatment, a complete shut down of the WTE section is much less likely; even more so with multiple parallel lines, which allow rotating maintenance operations without losing completely the waste treatment capability. In practice, the WTE section is down only due to forced outages or to the maintenance of subsystems serving all treatment lines (steam turbine, condenser, control system, etc.).

Given these considerations, the situation where the gas turbine is down has been analyzed either for the plant based on the medium scale gas turbine and for the one based on the large gas turbine. In order to assess the most critical conditions, the latter has been evaluated also for the extreme condition where not only the gas turbine is down, but also two of the three grate combustors. With just a single grate combustor in operation, the HP steam flow is a mere 16% of the on-design value and steam turbine output is 93% lower than its on-design value (14.4 MW vs 201.4 MW). To avoid sending saturated steam to the steam turbine, when the GT is down we have assumed that one of the convective tube banks of the evaporator works as superheater, a strategy already adopted by some manufacturer to adjust the operation of the boiler between "clean" and "dirty" conditions. The heat transfer area of the tube bank converted to superheater is such that the off-design superheating temperature is 420-430°C.

With the same rationale of exemplifying the most critical conditions, the situation where the WTE section is down has been analyzed only for the plant based on the medium scale gas turbine. For this plant, the steam flow generated in the evaporator of the HRSG at on-design conditions is nearly zero, while when the WTE section is down the same evaporator must warrant the generation of all the steam that eventually goes to the steam turbine. As shown further, this drastic increase of steam production requires a large increase of the mean ΔT across the evaporator, which in turn calls for higher gas temperatures and lower evaporation pressure. The HRSG drum must be designed for the flow at this off-design condition to ensure good steam quality and adequate “hold-up” times.

The off-design conditions at which the plant will operate obviously depend on the control strategy. In our case, the most important degree of the freedom to be controlled is the evaporation pressure in the WTE section, which is also the pressure in the HP drum of the HRSG. To maintain efficiency (and thus power output) as high as possible, it is assumed that such pressure is controlled as follows.

- When the gas turbine is down, the evaporation pressure in the boiler of the WTE section is maintained near its on-design value by adjusting the opening of the first nozzle of the steam turbine. This is the same strategy typically adopted in fossil fuel-fired steam cycles where, given the high temperatures reached in the combustion chamber, keeping evaporation pressure constant maximizes the steam turbine enthalpy drop without penalizing the boiler efficiency.
- When the WTE section is down, the first nozzle of the steam turbine is assumed to be fully open and the evaporation pressure in the HP drum of the HRSG is let slide as determined by the available heat transfer areas and by the steam turbine characteristic curve (i.e. the relationship between steam turbine mass flow and expansion ratio). This is the same strategy typically adopted in Combined Cycles, where letting evaporation pressures slide gives the best compromise between heat recovery from the gas turbine exhausts (which increases when evaporation pressures go down) and efficiency of the steam recovery cycle (which increases when evaporation pressures go up).

4. Methodology

Once the heat and mass balances at on-design conditions have been evaluated, one can calculate the relevant non-dimensional parameters of the steam turbine (reduced mass flow and load coefficient), as well as the products ($U \cdot A$) of each tube bank and heat exchanger, where U is the overall heat transfer coefficient and A is the heat transfer area. This has been carried out by a computer program named GS developed at Politecnico di Milano to analyze complex energy systems [8, 9]

The off-design operating conditions and performances have been estimated by assuming that:

1) The steam turbine of the plant based on the medium-scale GT is modeled as a sequence of three sections: (i) HP, from admission to the control stage; (ii) control stage, which maintains the pressure of the first bleed at 2.4 bar and at design conditions discharges at 1.6 bar; (iii) LP section, from the discharge of the control stage to the condenser. The steam turbine of the plant based on the large scale GT comprises a fourth section, from the admission of reheat (RH) to the control stage.
2) The pressure at the inlet of the control stage is kept constant, assuming that its geometry is changed as needed (partial admission). This allows using the 2.4 bar bleed for district heating, an option that can significantly improve the environmental and economic outlook of WTE plants.
3) When the gas turbine is down, the pressure at the inlet of the HP section is kept constant by assuming that the opening of the first nozzle of the steam turbine is varied as required. When the WTE section is down, the first nozzle of the steam turbine is assumed fully open and the pressure at the inlet of the HP section slides as

required by its characteristic curve, i.e. by the relationship between reduced mass flow G and load coefficient ψ[2] suggested in [10].

4) The pressure at the inlet of the LP and IP (RH admission) section slides as required by its characteristic curve, i.e. by the same correlation G vs ψ of the HP section.
5) Also efficiency varies with ψ as assumed in [10] and such variation is the same for all sections (HP, control stage, LP). The non-dimensional correlations (G vs. ψ) and (efficiency vs ψ) are generic representations of the behaviour of axial-flow, multi-stage turbines.
6) Except for the bleed at 2.4 bar, which is maintained at constant pressure, the pressures of the other bleeds slide as required to maintain the same ratio between the iso-entropic enthalpy drop upstream and downstream of the bleed.
7) The overall heat transfer coefficient U and thus the product (U·A) of each tube bank and heat exchanger are constant, equal to their value at on-design conditions.
8) The maximum steam temperature is limited to 550°C by de-superheating with water taken from the feedwater pump.
9) The mass flow of refrigerated water stays constant and condensation pressure slides as allowed by the available (U·A) down to a minimum condensation pressure of 0.04 bar. Further reductions of condenser thermal power are accommodated by decreasing the refrigerant flow rate. This strategy gives the lowest condensation pressure and maximizes gross power output. Alternatively, one could reduce the flow of refrigerated water, thereby reducing auxiliary power consumption at the expense of higher condensation pressures. Which strategy gives the highest net power output depends on the off-design behaviour of the cooling towers, which however have been excluded from our analysis.
10) The auxiliary power consumption of the waste island is proportional to the number of grate combustors in operation. The one of the power island is the sum of four terms: (i) a constant for the GT (oil pumps, cabinet fans, control system, etc.); (ii) a constant for the steam cycle (lubrication systems, controls, etc.); (iii) feedwater pump consumption, calculated by assuming constant head and efficiency variations typical of multistage centrifugal pumps; (iv) consumption of the refrigeration system, assumed proportional to the power discharged by the condenser.

These assumptions have been embodied in a computer code developed to estimate the off-design conditions of the steam cycle starting from the on-design features estimated by GS. Off-design calculations can be numerically involved because the solution is often very far from the on-design conditions.

5. Results

The results for the three off-design cases selected here are reported in Tab. 3 and Figs. 3-5. The table reports two marginal efficiencies that are helpful in appreciating the actual benefits brought about by integrating the WTE section and the CC:

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[2] The reduced mass flow G (a non-dimensional quantity) is defined as:

$$G = m_{in,T} \cdot \sqrt{R \cdot T_0} / (p_0 \cdot A)$$

where $m_{in,T}$ is the flow at the turbine inlet, R the gas constant, T_0 and p_0 the total temperature and pressure at turbine inlet, A an equivalent flow cross-section. The load coefficient ψ, also a non-dimensional quantity, is the ratio between iso-entropic enthalpy drop and $u^2/2$, where u is the blade peripheral velocity.

		Medium scale GT			***Large scale GT***		
		on-design	**GT off**	**WTE off**	**on-design**	**GT off**	**GT off**
Grate combustors in operation		3	3	0	3	3	1
W_{waste}	MW_{LHV}	180.80	180.80		180.80	180.8	60.27
$W_{nat\ gas}$	MW_{LHV}	216.76	-	216.76	680.93	-	-
$m_{steam,WTE}$	kg/s	92.17	61.75	-	98.71	61.37	21.59
$m_{steam,HRSG}$	kg/s	0.01	-	31.78	32.91, 6.46, 4.47	-	-
$m_{steam,tot}$	kg/s	92.18	61.75	31.78	131.62, 6.46, 4.47	61.37	21.59
$P_{steam,EVA}$	bar	65	65	25	85	85	65
$T_{steam,SH / RH}$	°C	550	433	550	535 / 560	430	429
$T_{exhaust,WTE}$	°C	130	114	-	130	-	-
$T_{exhaust,HRSG}$	°C	90	-	136	82	-	-
$W_{el,GT}$	MW_{el}	73.96	-	73.96	253.54	-	-
$W_{el,ST}$	MW_{el}	106.34	57.06	29.73	201.45	57.62	14.39
W_{aux}	MW_{el}	7.21	6.22	1.93	8.71	6.30	2.53
$\mathbf{W_{el,net}}$	$\mathbf{MW_{el}}$	**173.09**	**50.84**	**101.76**	**446.28**	**51.32**	**11.86**
$\eta_{\mathbf{nat\ gas}}$	**%**	**55.86**		**46.95**	**57.90**		
η_{MSW}	**%**	**29.80**	**28.12**		**39.69**	**28.39**	**19.68**

Tab. 3 *Performance characteristics of integrated plants based on medium and large scale GT, at on-design and off-design conditions.*

- natural gas efficiency $\eta_{nat\ gas}$, i.e. the ratio between (i) extra power generated by the integrated plant with respect to the power of the conventional WTE plant (51.99 MW, see Tab. 1) and (ii) combustion power of natural gas;
- MSW efficiency η_{MSW}, i.e. the ratio between (i) extra power generated by the integrated plant with respect to a CC fed by the same amount of natural gas and net efficiency 55% and (ii) combustion power of MSW.

To appreciate the variations occurring in the boilers and the steam turbine, Figs. 6-7 compare the on-design and off-design temperature profiles in the HRSG and the boiler of the WTE section, while Fig. 8 compares the steam turbine expansion lines.

When the gas turbine is down, the system behaves and reaches performances very close to the ones of the stand-alone WTE plant. In the extreme situation where either the large gas turbine and two grate combustors are down, net electric efficiency is close to 20%, about 8 percentage points below the one of the fully-fired, stand-alone WTE plant. The temperature of the flue gases at the outlet of the boiler is significantly lower than that on-design (~115°C vs 130°C). Should this be incompatible with the flue gas treatment system, the water temperature at the inlet of the economizer should be increased, e.g. by recycling warm water taken further downstream.

When the WTE is down, a desuperheater must be placed across superheater tube banks to limit the steam temperature to the maximum value of 550°C; as shown in Fig.7, the temperature drop across this superheater is very substantial. The significant decrease of evaporation pressures would cause steaming at the end of the economizers; to avoid this, the water pressure must be kept much above the drum pressure by valves[3] placed at the economizer outlet. Also, for the economizer ahead of the de-aerator a by-

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[3] Stellite valves with multiple flashes are often installed downstream of economizers and can accept some steaming. Economizer banks are designed for the shut-off pressure of the feedwater pump.

pass is needed to keep the water temperature low enough to insure proper de-aeration.

6. Conclusions

The integration between a WTE plant and a natural gas fired CC generates relevant technological, performance, environmental and economic advantages but poses significant off-design issues. The analysis presented in this paper illustrates suitable technical solutions and control strategies for plants where a 180 MW_{LHV} WTE section is integrated with either a medium scale or a large scale CC.

Results give a sense of the behaviour of the system under the most severe conditions, i.e. when either the gas turbine or the WTE section are down. When the gas turbine is down, the net electric efficiency of the WTE section is less than one percentage point lower than the efficiency of the stand-alone WTE plant. When the WTE section is down, the efficiency of the CC is much below the one of a stand alone CC: about 8 percentage points for the plant based on the medium-scale gas turbine. These performances appear most congenial to what is likely to be the operational strategy of these plants, i.e. paramount priority to waste treatment and CC dispatched according to the requirements of the national grid.

7. Acknowledgements

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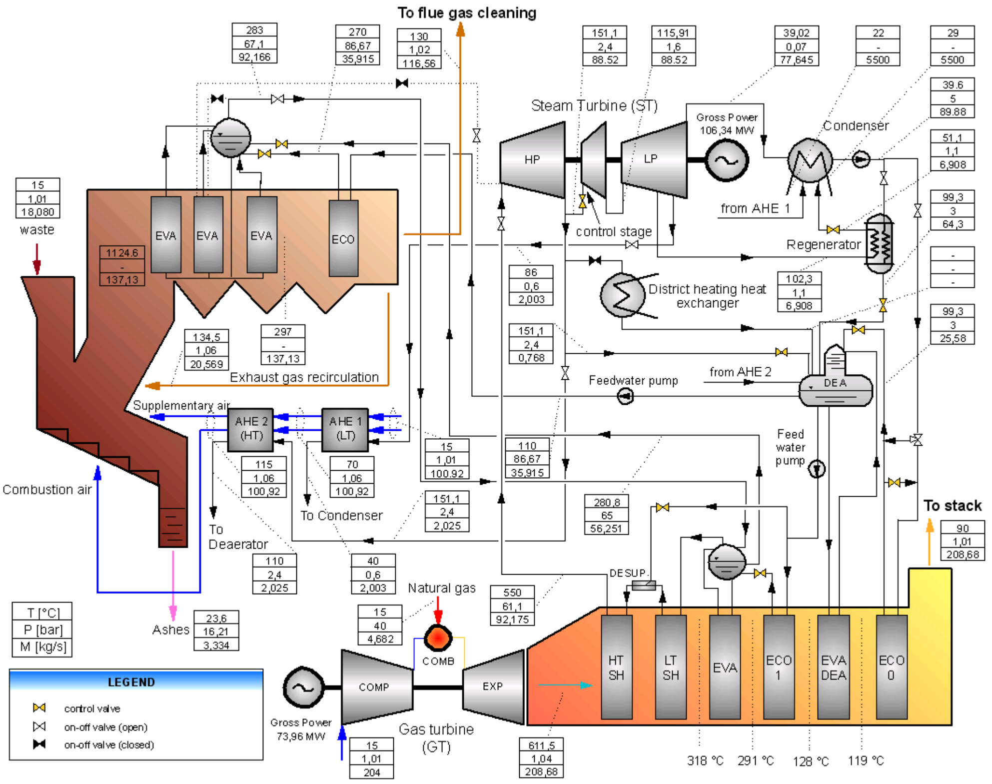


Fig. 1: *Configuration and on-design operating parameters for the plant based on the medium-scale GT.*

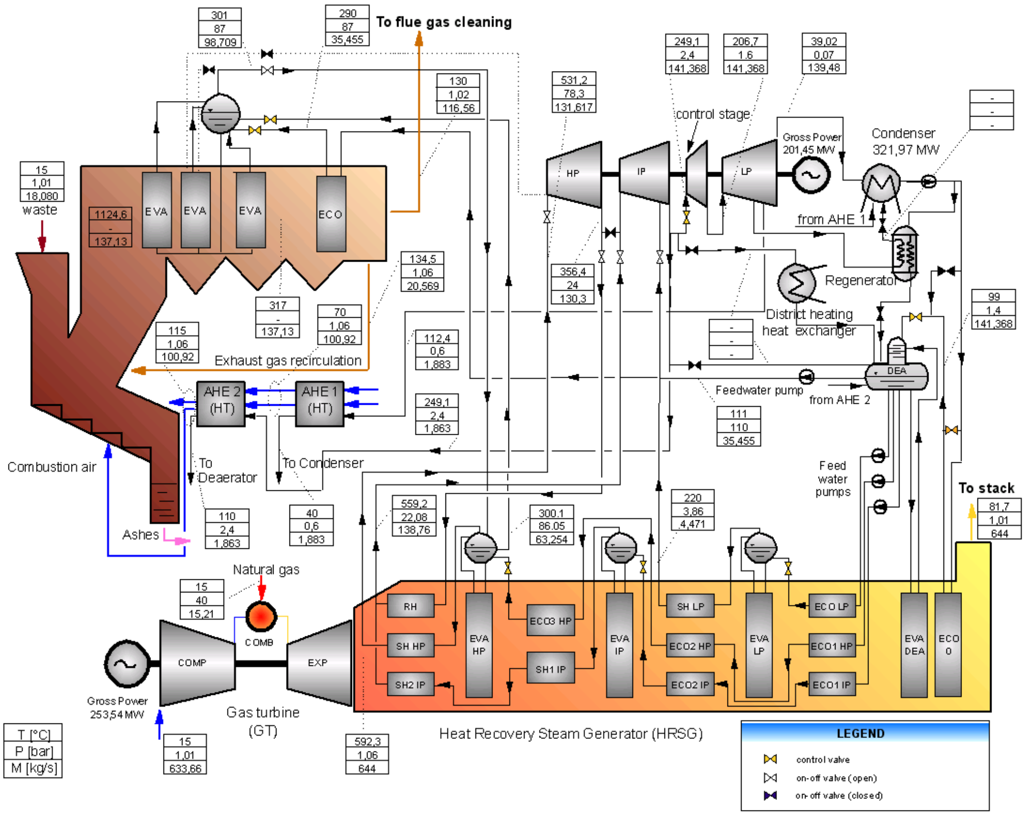


Fig. 2: *Configuration and on-design operating parameters for the plant based on the large-scale GT.*

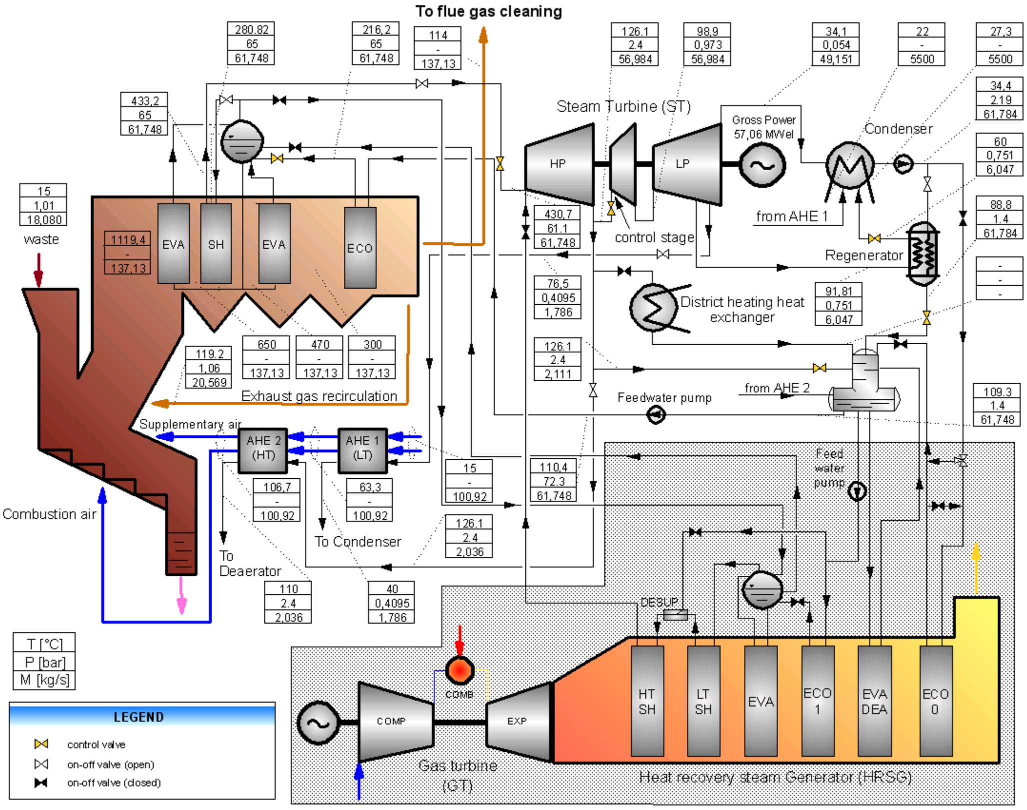


Fig. 3: *Configuration and operating parameters of the medium-scale GT plant when the GT is down.*

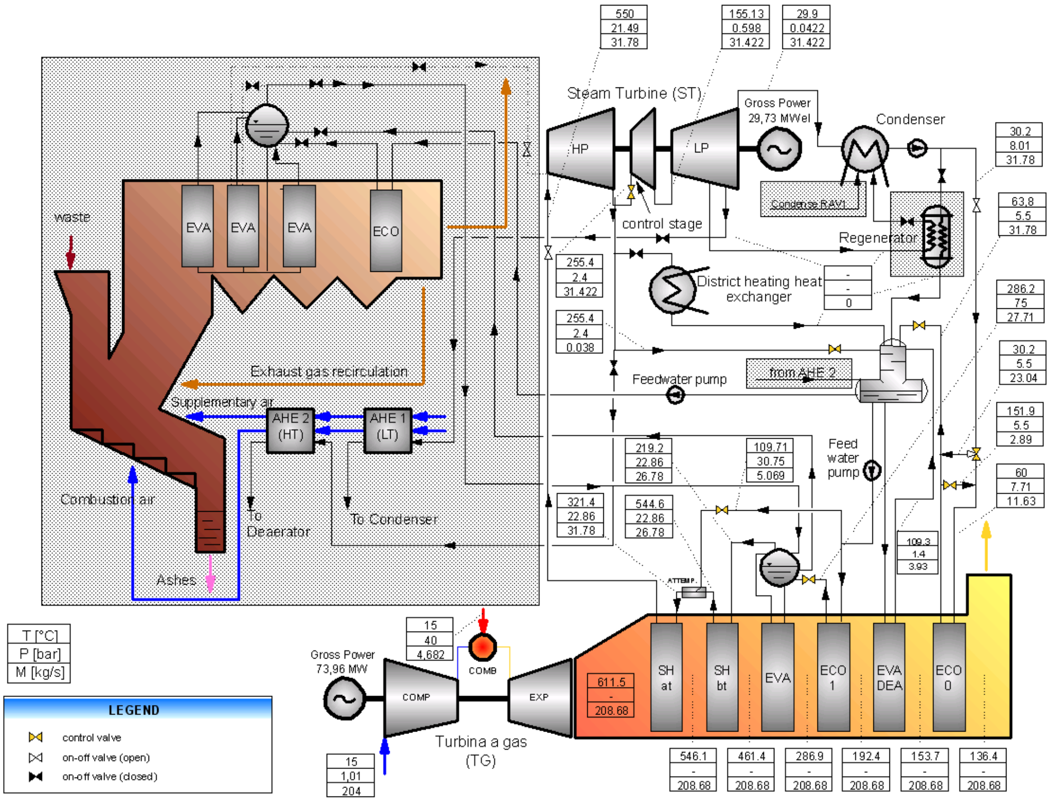


Fig. 4: *Configuration and operating parameters of the medium-scale GT plant when the WTE is down.*

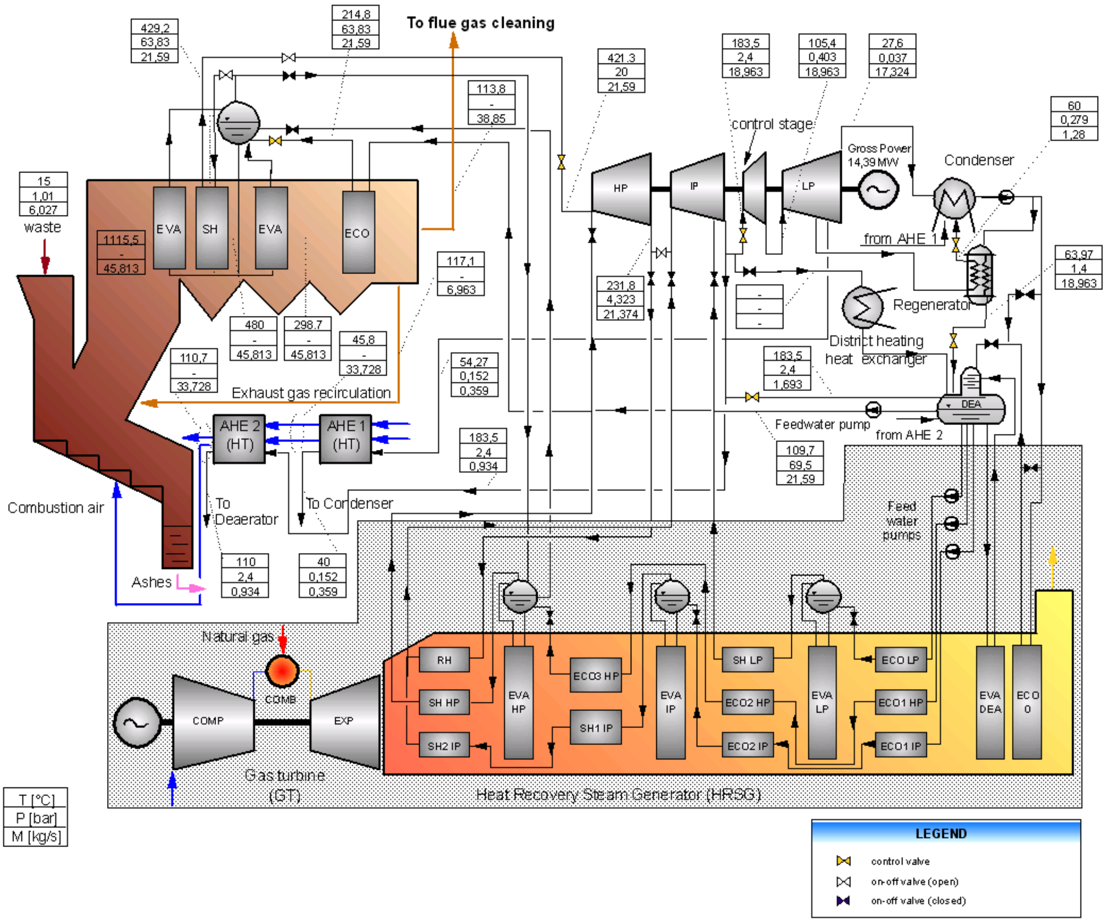
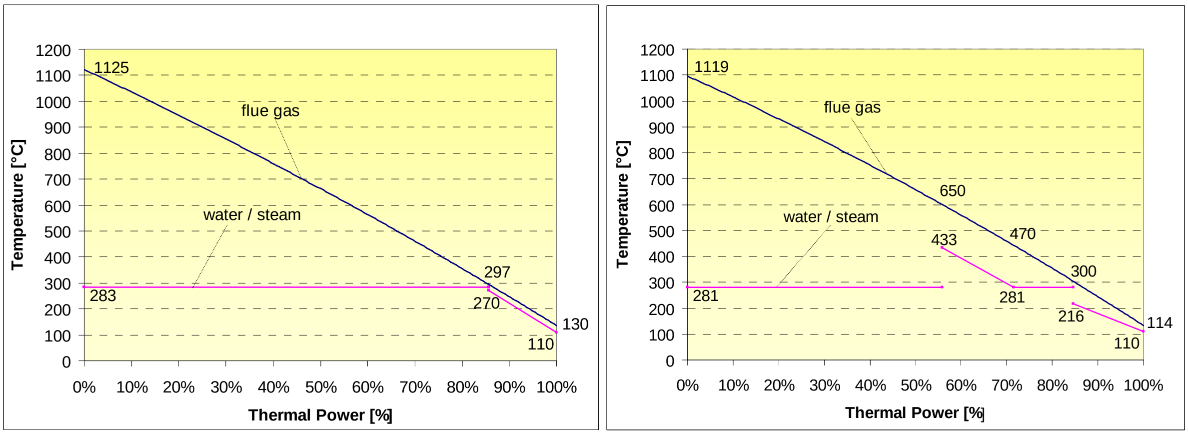
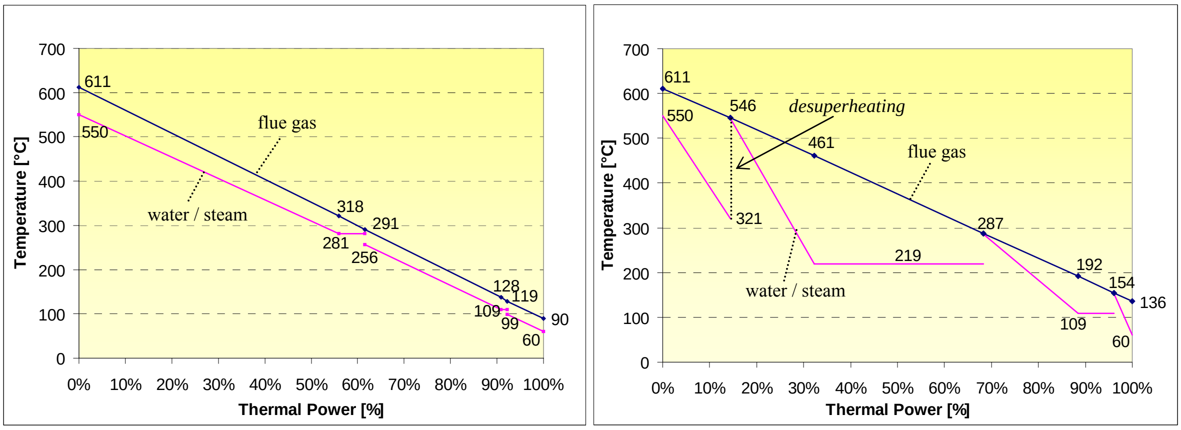


Fig. 5: *Configuration and operating parameters for the large-scale GT plant when the GT is down and only one (out of three) grate combustor is operated.*



(a) (b)

Fig. 6: *Temperature profiles of water/steam and flue gas in the WTE boiler for on-design (a) and off-design (b) operation of the plant based on the medium-scale GT.*



(a) (b)

Fig. 7: *Temperature profiles of water/steam and flue gas in the HRSG for on-design (a) and off-design (b) operation of the plant based on the medium-scale GT.*

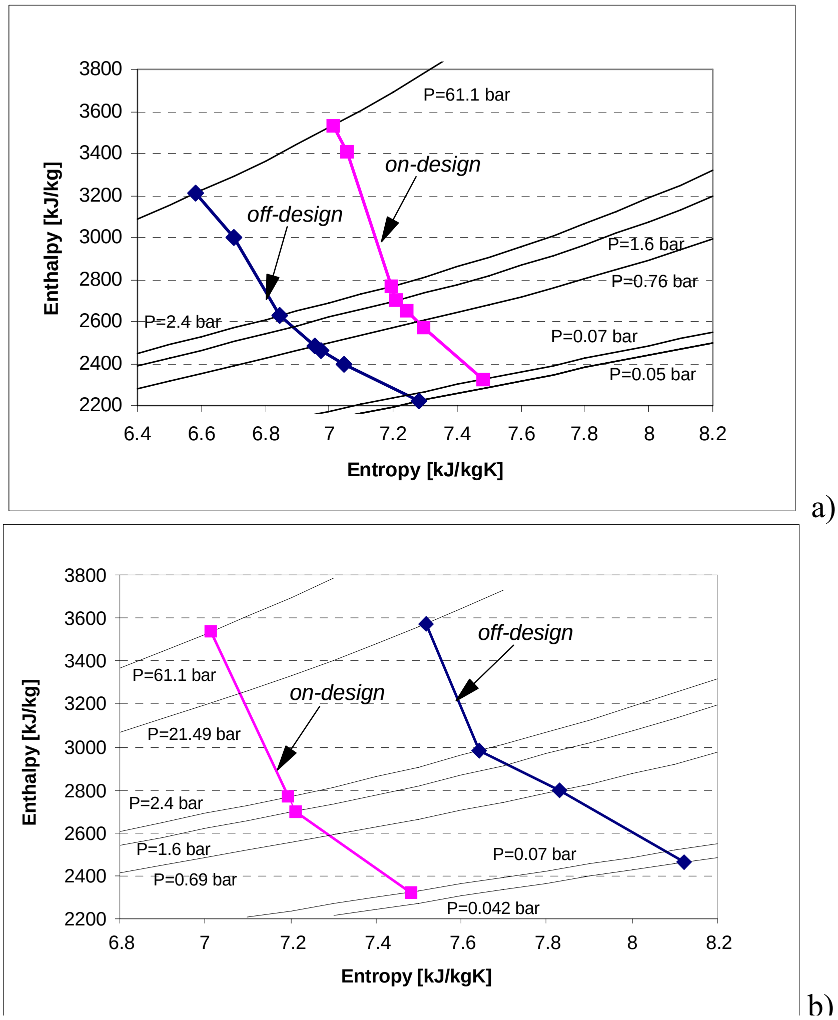


Fig. 8: *Comparison between the design and off-design expansion lines of the steam turbine when the GT is down (a) and when the WTE section is down (b). Both figures refer to the plant based on the medium-scale GT.*