Combined cycle power plants with post-combustion CO₂ capture: energy analysis at part load conditions for different HRSG configurations

Maura Vaccarelli*

Dept. of Industrial and Information
Engineering and Economics
University of L'Aquila
Via G. Gronchi, 18
67100- Italy
maura.vaccarelli@univaq.it

Majed Sammak

Dept. of Energy Sciences
Lund University
SE-22100 Lund
Sweden
majed.sammak@energy.lth.se

Klas Jonshagen

RPP
Siemens Industrial
Turbomachinery AB
SE-61283 Finspong
Sweden
klas.jonshagen@siemens.com

Roberto Carapellucci

Dept. of Industrial and Information
Engineering and Economics
University of L'Aquila
Via G. Gronchi, 18
67100- Italy
roberto.carapellucci@univaq.it

Magnus Genrup

Dept. of Energy Sciences
Lund University
SE-22100 Lund
Sweden
magnus.genrup@energy.lth.se

ABSTRACT

The part-load behaviour of combined cycle power plants (CCPP) equipped with post-combustion CO₂ capture is analyzed. Different CCPP configurations are compared, including single-, dual- or triple-pressure level steam generators. The gas turbine is a single shaft unit using the variable guide vanes and fuel flow to control the load. The 90% CO₂ capture is achieved using monoethanolamine (MEA 30% wt) as absorbent.

In all configurations, the reboiler duty to regenerate the absorbent is partly covered by the hot water of the economizer. This economizer-reboiler loop allows to increase the plant efficiency as the possibility to extract more energy from the economizer results in lowering the steam extraction from the turbine. The dual-pressure level CCPP with CO₂ capture gives the best efficiency at the design operation. As the load is reduced, the efficiency of the single-pressure plant becomes comparable to the efficiency of the dual-pressure plant, due to the effective thermal integration between the heat recovery steam generator and the absorbent regeneration process. Hence, when CCPPs with CO₂ capture operate at part-loads, a further benefit can be

*Corresponding author (M. Vaccarelli, tel +39 0862 434317, fax +39 0862 434403, e-mail maura.vaccarelli@univaq.it)

achieved with the single-pressure level plant, having it both design simplicity and low efficiency penalties for the CO₂ capture.

KEYWORDS

Post-Combustion CO₂ Capture; Off Design; Combined Cycle; Part-load operation.

NOMENCLATURE

Symbols

A	Area
C	Constant
c	Speed
h	Enthalpy
$\dot{m}$	Mass Flow
Nu	Nusselt Number
p	Pressure
Pr	Prandtl Number
Re	Reynold Number
t	Thickness
U	Heat Transfer Coefficient
u	Rotor Speed
X	Parson Number

Greek letters

α	Heat Transfer Coefficient
ξ	Blade Stagger Angle
Δ, δ	Difference
φ'	Flange Efficiency
η	Efficiency
λ	Heat Conductivity
μ	Dynamic Viscosity
ρ	Density
π	Pressure Ratio
φ	Capture Ratio

Subscripts

in	Inlet
is	Isentropic
out	Outlet
rel	Relative
des	Design
s	Static

Abbreviations and acronyms

CCPP	Combined Cycle Power Plant
CHIC	Compressor characteristic
CB	Combustor
CP	Compressor
EGR	Exhaust Gas Recirculation
EGT	Exhaust Gas Temperature
FG	Flue Gas
GT	Gas Turbine
HP	High-Pressure
HRSG	Heat Recovery Steam Generator
IP	Intermediate-Pressure
LG	Lean Gas
LP	Low-Pressure
MEA	Monoethanolamine
VG	Inlet Guide Vanes
1P	One pressure level
1PRH	One pressure level with reheat
2PRH	Dual pressure level with reheat
3PRH	Triple pressure level with reheat

1. INTRODUCTION

The use of fossil fuels covers more than 85% of the World's energy needs, thus providing a prominent contribution to the formation of greenhouse gases. It is expected that this situation remains unchanged in the next 25 years, since the World energy consumption is expected to increase by a factor of two [1]. Among the various energy uses, the electricity production is well suited to the adoption of strategies for CO₂ emission reduction. This is due to the large amount of CO₂ produced by power stations (ranging from

400 to 1200 kg CO₂ per MWh), to the widespread use of fossil fuels, to the high power concentration which makes possible the adoption of complex removal systems. Therefore, a drastic reduction of CO₂ emissions becomes possible only by integrating Carbon Capture and Sequestration (CCS) technologies in electricity generating plants [2].

Many methods have been depicted to obtain CO₂ capture and sequestration, basically resorting to three different technologies. The first, called *post-combustion technique*, is when the CO₂ separation from exhaust gases is produced by fired boilers or by gas turbines, using chemical or physical absorption processes. The second is the *oxy-fuel technique*, in which thermodynamic cycles have an highly enriched CO₂ mixture as the working fluid. This is obtained by using oxygen rather than air as the oxidizer, thus avoiding dilution with nitrogen. When fuel chemical treatments are used to obtain a highly enriched synthetic gas, through coal gasification or natural gas reforming, it is the case of the *pre-combustion technique*.

During the last years several studies on CO₂ capture have been done in different lines, from experimental studies at laboratory scale and pilot plants to the development and implementation of mathematical models [1]. A number of separation technologies could be employed with post-combustion capture, including adsorption, physical absorption, chemical absorption, cryogenics separation and membranes. The most near-term technology according to the IEA report is post-combustion capture from a natural-gas-fired power plant by chemical absorption using amine solutions [3].

This work treats post-combustion CO₂ separation on combined cycle power plants using monoethanolamine (MEA 30%wt) as absorbent. Compared with the other CO₂ capture techniques, the absorbent process does not require a new power plant concept while the absorption process is located downstream of the power plant. Hence conventional proven technology can be used, also for the retrofit of existing power plants [4]. This CO₂ removal technique is, however, still highly energy intensive due to heat requirements needed to regenerate the amine solution which drastically increases the operating cost. The thermal integration of the amine regeneration in the reboiler unit is a further objective of this work.

The energy required by the capture process is provided by the power plant in the form of heat and electricity. Flue gas containing CO₂ is flowing through an absorber where it is exposed to the MEA solvent flowing counter-currently. The reaction between MEA solvent and the CO₂ gives a water soluble salt. A CO₂-rich MEA stream is preheated in a heat exchanger and enters a stripper column to regenerate the solvent and reverse the reaction by means of heat supplied in a reboiler and release the CO₂ content as

a stream leaving the top of the column [5]. The lean MEA-stream is recycled back to the absorption column while the CO₂ stream goes to the compression section. Although it represents a well-established separation method, the costs and energy requirements of the CO₂ capture system are really remarkable, leading to higher cost of electricity produced and higher consumption of fossil fuels for a fixed power output [5].

Amine solvents, such as monoethanolamine (MEA), have been used for production of food-grade CO₂ long before post-combustion capture of CO₂ became a research and development topic as one of several possible means to combat global warming [6]. Post combustion capture of CO₂ from fossil power plants, using amine solvents, is therefore often regarded as a fairly mature technology in a new application. Nevertheless, in order to make the technology attractive compared to other CO₂ capture routes, several technical challenges remain, such as finding a suitable solvent that is non-toxic, configuring the capture unit (absorption and desorption of CO₂) to operate with a minimum energy penalty and finding the optimum integration of the capture unit with the power plant emitting the CO₂ [6]. In order to answer to these challenges, this work uses the principle of the eco-reboiler [11] to utilize low-grade energy to regenerate the absorbent.

A single-pressure cycle with and without reheat, a dual- and a triple-pressure combined cycle including reheat with a MEA CO₂ capture plant have been tested. Previous work has been focusing on the optimum design for such system while this work takes it one step further by investigating the part-load performance [4].

The purpose of the study is to evaluate the performance and limitations of CCPP's with carbon capture at part load.

2. CCPPs WITH CO₂ POST-COMBUSTION CAPTURE AT THE DESIGN POINT

The CCPPs analyzed are the single-pressure plant, with and without reheat (1PRH and 1P respectively), the dual- (2PRH) and the triple-pressure (3PRH) plants with reheat. Combined-cycle power plants have been modelled to accommodate post-combustion CO₂ capture unit with MEA as solvent. The thermodynamic cycle arrangements of these CCPPs are described by Jonshagen et al. in [4], where these configurations were also analyzed at design conditions. The study is here extended to investigate effects of the integration with the CO₂ removal system when the CCPPs operate at part-loads.

The study is based on a combined cycle using the General Electric 109FB gas turbine, a 300 MW single-shaft machine, as topping cycle. The machine is directly connected to a 50 Hz generator running at a fixed speed of 3000 rpm. The steam cycle comprises high, intermediate and double-flow low pressure turbines, a condensing system and a heat recovery steam generator (HRSG) with single or multiple pressure levels. In the configuration based on a three pressure and reheat HRSG, it includes a preheater, three economizers, evaporators and super-heaters, and a reheater. The separation unit model was developed by Möller [7], based on the concept described by Kohl and Nielsen [8]. The CO₂ capture unit is designed to have a capture ratio of 90% at the design point utilizing an absorbent solution of 30% MEA and 70% water. The absorbent regeneration temperature is set to 122°C, too high regeneration temperatures will lead to degradation of the amine. The separated CO₂ leaves the capture unit at 1.4 bar and is compressed up to 200 bar. Other calculation assumptions are reported in Table 1.

In all configurations, to increase CO₂ concentration in the flue gas, a fraction of exhaust gases is recirculated to the compressor inlet through EGR (exhaust gas recirculation). The EGR also reduces the volume of the flue gas to be treated in the separation plant. The limit on how much EGR can be utilized is determined by the level of O₂ at the combustor inlet and outlet, considering that the excessive decrease of the oxygen concentration can compromise the flame stability and combustion efficiency, as well as can affect energy performance [9]. As the recirculation fraction increases, also the main energy performances of the power plant change: in Figure 1, these effects are shown for a dual pressure reheat plant. In fact, the efficiency and the net power output of the gas turbine slightly decrease with EGR, due to the small increase of the temperature at the compressor inlet, which causes a growth of the compression work and a reduction of the mass flow rate at the compressor suction. However, EGR also gives a slight increase in gas temperature at the turbine exit and a higher power output of the steam section, allowing to keep almost unchanged the efficiency of the combined cycle. Therefore, the slight penalization in GT performance with EGR is compensated by the correspondent gain in power output of the steam section, also considering the reduction of power requirement in the carbon capture system related to the higher CO₂ concentration in flue gases to be treated.

Experimental studies showed that an excessive formation of unburned hydrocarbons may occur if the O₂ concentration at the combustor inlet drops below 16% [10], that is reached with EGR of about 40%. The limitation is generally caused by a lack of oxygen, which reduces the efficiency of the combustor, resulting in high CO level and unburnt fuel.

However, the extent of the gas recirculation is limited also by monitoring the O_2 level at the combustor outlet: as shown in Figure 2, the most restrictive situation occurs at about 70% of the load when the gas temperature at the turbine exit reaches its maximum value. Further load reductions require a reduction of the fuel flow rate and, consequently, of the gas temperature at the turbine inlet.

To keep the molar O_2 content at the combustor exit higher than 6.5%, the EGR value has been fixed at 40% in the whole load range explored, even if at the minimum load the EGR value could be increased at about 45% without having inadequate oxygen in the combustor.

In a previous work [11], a theoretical study was performed about the effects of EGR on the turbomachinery, in terms of Mach numbers, resulting from changes in the gas properties of the working fluid. It was found that the change in Mach number was very small, close to 40% EGR. The current model gives a CO_2 concentration of just below 8% vol with 40% EGR, which corresponds to the value suggested by Elkady et al. for the GE 109FB burner [12]. Hence, 40 % EGR is used throughout this study. EGR also requires exhaust gas cooling in order to reduce the compression power [13]. In this paper the EGR cooling, up to 20°C, is achieved with water at ambient temperature, which is sprayed through the exhaust in an exhaust gas condenser.

When CO_2 capture system based on amine scrubbing is integrated with the cycle, the heat recovery unit will be drastically affected. A significant amount of steam is extracted from the steam cycle to cover the heat requirement of the CO_2 capture plant. A fraction of the reboiler duty can be covered by introducing a water loop between the economizer and the reboiler, as shown in Figure 3. Consequently, the water mass flow through the economizer is increased and more energy can be recovered.

The new reboiler consists of two heating coils, one heated by pressurized water and one by condensing steam. The energy supplied by the economizer replaces partly the steam extracted from the turbine and, therefore, more power will pass through the low pressure steam turbine. When more heat is recovered from the flue gas, the requirement of exhaust gas cooling prior to the EGR and the carbon capture plant is reduced.

The control strategy for the economizer-reboiler loop allows to maintain the temperature of the water mass flow at the economizer outlet close to the saturated state at all loads. This will provide extra control, preventing evaporation and drying-out of the economizer.

When the CO_2 capture plant is integrated into the CCPP and the economizer-reboiler loop is used (Figure 3), some new aspects should be considered in the plant design. The further heat extracted from the economizer reduces the temperature difference at the exit of the HRSG. Thus, the LP pressure level of the

plant can be increased and the benefit of introducing multiple pressure levels is significantly reduced. The cycle configuration here presented uses the condensate leaving the reboiler for feedwater deaeration and fuel preheating.

Effects of the water loop on performance of a single-pressure CCPP with reheat, EGR and CO₂ capture system were evaluated by authors in [11]. The analysis highlighted the benefit of using highly pressurized water to cover a fraction of the reboiler duty: it allows a 20% reduction of the steam flow rate extracted by the turbine with respect to the conventional option of supplying the whole reboiler duty through LP steam. Consequently, the increased amount of steam expanded through the LP steam turbine increases its power output, reducing the efficiency penalty by 1.35 percentage points compared with the steam only option. Moreover this novel cycle concept produces a significantly lower thermal energy needed for cooling purposes in addition to power for the blowers, as the new economizer configuration with the added pressurized water stream results in a lower exhaust gas temperature at 93 °C compared with 144 °C in the case without water [11].

Figure 4 shows the temperature versus heat flux (TQ) diagram for a single-pressure CCPP comparing the regular steam reboiler with the new configuration in which the proposed water-reboiler loop, described in Figure 3, has been integrated. As shown, there is a huge potential for extracting more energy from the economizer of the HRSG and, consequently, covering a considerable fraction of the reboiler duty without interfering with the rest of the steam production. Hence, the efficiency penalty associated with solvent-based CO₂ capture could be reduced. By utilizing the excess energy in the economizer for the reboiler, more energy can be recovered from the exhaust gas. The temperature difference in the second economizer is very small, being the process close to optimal.

The further losses related to the introduction of other pressure levels reduces the gain in increased steam production, resulting in that the triple-pressure plant has lower or equal efficiency as the dual-pressure plant. To provide a comprehensive overview of the potential of the proposed method, the efficiencies of a single-pressure and a triple-pressure plant with a conventional steam-heated reboiler are calculated. The efficiency values at design conditions are highlighted in Figure 5, showing the best behaviour of the dual-pressure level plant. The off-design analysis refers to these power plant configurations, whose thermodynamic cycles, parameters and performances at full load are described in [4].

3. OFF-DESIGN THERMODYNAMIC MODELLING OF CCPP PLANTS WITH CO₂ CAPTURE

Different ambient conditions and part-load strategies and their influence on the cycle performance are the base for a study in off-design mode. The off-design modelling of a combined cycle power plant with CO₂ capture comprises the study of four units: the gas turbine, the heat recovery steam generator, the steam cycle and the CO₂ capture plant.

A 300 MW natural gas combined cycle has been considered, using the General Electric 109FB gas turbine as topping cycle. The gas turbine model is as described by Walsh and Fletcher [14] and uses the fully dimensionless parameter groups. The compressor map is based on a publication by General Electric [15]. The map is actually for a Frame 7 unit, which is the 60 Hz version of the Frame 9 type. It is assumed that the Frame 9 compressor is a direct scale-up of the smaller Frame 7 unit. The turbine uses a standard turbine map, choking occurs in the stator. The wet efficiency is modelled according to Baumann and the design exhaust loss in the low-pressure turbine is set to 30 kJ/kg. Parasitic losses and transformed step-up losses are not included in the presented efficiencies [4].

The used platform is the commercially available heat- and mass balance software IPSEpro developed by Sim-Tech [16].

3.1 GAS TURBINE

The load of a single-shaft unit is usually controlled in a large range by the firing level and the engine mass flow rate. As known, in a CCPP plant the load control through the firing level reduces the efficiency of both the Brayton and the Rankine cycles. In fact, they suffer for the reduced maximum temperature; however, lowering the maximum temperature in the GT will extend the lifetime [17], being it crucial for the lifing of hot parts.

The part-load operation of the CCPP investigated in this study is obtained reducing the mass flow rate at the compressor suction, controlling the variable inlet guide vanes (VGV). Their setting controls the axial velocity into the first stage of the compressor, decreasing the mass flow rate through it and reducing the pressure ratio of the turbine. By setting the VGV angle, the exhaust gas temperature will increase due to reduced expansion ratio. This produces a loss in the stand-alone GT, but it can be recovered in the steam cycle of CCPPs. The effects of EGR are taken into account in the off-design model of the gas turbine group by considering its influence on the volumetric flow rate at the compressor inlet, varying with temperature and gas composition.

The off-design model consists of modified standard components and specifically designed components developed at Lund University, the Division of Thermal Power Engineering. The gas turbine is modelled with the referred parameter groups on the rigorous form. This gives the possibility to calculate component performance under a wide range of operational conditions, i.e. changing media compositions. In this study it is assumed that the flow at the inlet of the gas turbine expander is always choked, a reasonable assumption for a wide range of GT operation conditions.

The basis for the compressor model is either a generic- or a certain non-dimensional map, which can be parameterized for the specific engine. The compressor map is based on a GE publication [15] and should be more accurate than a generic map. Providing design levels the map can be adimensionalized with values of unity at design conditions for normalized flow, efficiency, pressure ratio and aerodynamic speed [7]. The off-design calculation is highly recursive since all components have to be matched together to form a gas turbine unit. On a more detailed level, each point in the maps represents unique velocity triangles, based on Mach number. Hence, the aero-thermal behaviour of the compressor and turbine is set if two dimensionless parameters are known for each of them. In order to avoid separate maps for each VGV setting, influence coefficients are introduced, according to:

$$\dot{m}_{rel}^* = \dot{m}_{rel,CHIC}^* \left(1 + \frac{a_{VIGV}}{100} \Delta\xi \right), \quad a_{VIGV} = \frac{\delta(\dot{m}_{rel}^*)}{\delta(\Delta\xi)} \quad (1)$$

$$\pi_{rel} = \pi_{rel,CHIC} \left(1 + \frac{b_{VIGV}}{100} \Delta\xi \right), \quad b_{VIGV} = \frac{\delta(\pi_{rel})}{\delta(\Delta\xi)} \quad (2)$$

$$\eta_{rel} = \eta_{rel,CHIC} \left(1 - \frac{c_{VIGV}}{100} [\Delta\xi]^2 \right), \quad c_{VIGV} = \frac{\delta(\eta_{rel})}{\{\delta(\Delta\xi)\}^2} \quad (3)$$

The preceding equations show a linear behaviour for pressure ratio and mass flow, whilst the efficiency follows a quadratic behaviour [18].

3.2 HEAT RECOVERY STEAM GENERATOR

The purpose of the heat exchangers in the HRSG of a CCPP is to transfer heat from the gas turbine exhaust to the water or steam of the bottoming Rankine cycle. The driving force of the heat transfer process is the temperature difference between the hot and cold sides [19]. The total amount of heat transferred is given by:

$$\dot{Q} = U \cdot A \cdot LMTD \quad (4)$$

where U is the overall heat transfer coefficient, LMTD the logarithmic mean temperature difference, and A the heat exchange surface.

The heat transfer coefficient U is dependent on the convective heat transfer coefficients (inside, outside) and material characteristics:

$$\frac{1}{U \cdot A} = \frac{1}{A_i \cdot \alpha_i} + \frac{t_w}{\lambda_w \cdot A_i} + \frac{1}{\phi' \cdot A_o \cdot \alpha_o} \quad (5)$$

This equation is difficult to rearrange and requires numerous input parameters, several of which are not easily estimated. It is therefore desirable to simplify the equation thus limiting the number of parameters needed in the model. In a steam boiler or a HRSG, where the medium is water/steam or evaporating water on one side and flue-gas on the other, Eq. (5) can be simplified as:

$$U \cdot A = \phi' \cdot A_o \cdot \alpha_o \quad (6)$$

Equation (6) represents a good approximation of Eq. (5) since the heat conductivity through the heat exchanger material is very high and the convective heat transfer coefficient on the gas side is much worse than on the other. Caution should be exercised when employing this simplification as the flow on the steam side in some off-design scenarios is significantly reduced, causing the heat transfer to decrease.

In Eq. (6), the heat transfer coefficient α is dependent on the fluid properties, the Reynolds number and the geometry of the unit. In a steam boiler or a HRSG the heat exchangers are normally of shell-and-tube type, and the heat transfer coefficient is given by:

$$\alpha \cdot d = C \cdot Nu \cdot \lambda_f \quad (7)$$

in which the Nusselt number

$$\alpha \cdot d = C \cdot Re^n \cdot Pr^{\frac{1}{3}} \quad (8)$$

is the product of the Prandtl and Reynold numbers. The constant n is dependent on the diameter and configuration of the tubing; in the general case it can be set at 0.6. The dimensionless Prandtl number (Pr) depends only on the fluid properties and state of the fluid. It does not vary greatly for different gases, and the temperature dependency is very small. However there is a pressure dependency and, under some conditions, it should be included.

The Reynolds number should be calculated for the highest fluid velocity.

$$Re = \frac{\rho \cdot c_{max} \cdot d}{\mu_f} \quad (9)$$

As when modelling a pressure loss, it is now desirable to reduce the number of variables. By combining Equations (6), (7) and (8), and dividing by the design value, the following equation is obtained:

$$\frac{UA}{U_{des}A} = \frac{\mu_{f,des}(\rho c_{max})^{0.6}}{\mu_f(\rho c_{max})_{des}^{0.6}} \quad (10)$$

The flange efficiency is assumed to be constant throughout the load range. The dynamic viscosity (μ) can be omitted in most cases, but if the temperature varies considerably over the operational range of the unit it can be replaced by:

$$\left(\frac{\mu_{f,des}}{\mu_f} \right)^{0.6} \approx 1 - (\bar{t}_{des} - \bar{t}) \cdot 5 \cdot 10^{-4} \quad (11)$$

The continuity equation can be applied and, as the flow area is fixed, Eq. (10) can be simplified:

$$\frac{UA}{UA_{des}} = \frac{\dot{m}^{0.6}}{\dot{m}_{des}^{0.6}} \quad (12)$$

so that it is only a function of the ratio of the actual mass flow and the design mass flow.

The heat exchanger model can now be constructed from Equations of the LMTD, (4) and (12), together with an energy balance, a mass balance and the pressure drop as:

$$\frac{\Delta p}{\Delta p_{des}} = \left(\frac{\dot{m}}{\dot{m}_{des}} \right)^2 \quad (13)$$

If a heat balance through the heat exchanger is available, the boundary conditions can be used to evaluate and set all design values ($U_{des} \cdot A$, $\dot{m}_{des,i}$, $\dot{m}_{des,o}$, $\Delta p_{des,i}$ and $\Delta p_{des,o}$).

3.3 STEAM TURBINE

The steam turbine model is based on a part turbine methodology that is the same for all the plants analyzed. In this approach, a number of turbine stages are gathered to a group or part turbine. Each of these turbines is mainly described by two parameters, the efficiency and the swallowing capacity. The efficiency of a single turbine stage is a function of the stage loading, that affects the velocities and their diagrams. The stage loading can be described by the relationship between the fluid speed and the speed of the rotating blades. To shorten the iteration process, the speed obtained by the isentropic enthalpy drop is used, rather than the actual enthalpy drop which depends on the efficiency itself:

$$\eta = f\left(\frac{u}{c_s}\right) \quad (14)$$

The function in Equation (14) is often called the v number (Nu number). For heat- and mass-balance calculations the speed could be more conveniently expressed in terms of the enthalpy drop:

$$\eta = f\left(\frac{u}{\sqrt{2\Delta h_{is}}}\right) \quad (15)$$

Equations (14) and (15) are only valid for a single stage, being the efficiency normally correlated with the Parson number for a group of stages. In fact, with part turbines, the Parson number normally replaces the isentropic velocity ratio used for a single stage. The Parson number uses the sum of the mean rotor speed for a turbine section:

$$X = \frac{\sum u^2}{\Delta h_{is}} \quad (16)$$

For a fixed-speed steam turbine the rotor speed u is constant for all loads. Normalizing Equation (16) to the design point, it gives the isentropic efficiency:

$$\frac{\eta}{\eta_{des}} = f\left(\frac{\Delta h_{is,des}}{\Delta h_{is}}\right) \quad (17)$$

that can be determined using operational data. In this study a standard function is considered with a quadratic dependence of adimensional efficiency on isentropic enthalpy drop ($a x^2 + b x + c$, with $a=-0.156$, $b=0.438$ and $c=0.719$).

The turbine capacity is crucial for off-design modelling of turbomachinery. The capacity of a turbine or swallowing capacity describes the relation between inlet and outlet pressure, inlet temperature and mass flow rate. At a certain pressure ratio the turbine will choke and the mass flow rate will become independent on the outlet pressure. For modelling the choked turbine, in this study we use the model thermodynamically derived by Beckmann and used by Jonshagen in [19], more accurate than the classical steam-cone law empirically derived by Stodola in the 1920s. The swallowing capacity or flow relation with pressure is therefore used to calculate the node pressures, setting the enthalpy drop distribution through the turbine.

3.4 CO₂ CAPTURE PLANT

The CO₂ capture plant is composed by two main sections, the absorber and the stripping columns. The flue gases from the power plant, after cooling, enter the bottom of the absorption column, in which the CO₂ is absorbed by a MEA solution flowing in countercurrent. The treated gas is discharged into the atmosphere, while the CO₂-rich solution is collected at the bottom and sent to the desorber. Providing thermal power in the reboiler, the CO₂ absorbed is stripped from the CO₂-rich solution and the solvent regenerated. The nearly

pure CO₂ stream is then sent to the compression system in order to be dispatched to the storage site. The design and modelling of this carbon capture system is based on mass and energy balances referred to both the columns, in order to have 90% of capture ratio with a lean sorbent loading set at 0.25 kmol_{CO2}/kmol_{MEA}.

The heat balance equation over the absorber

$$\dot{Q}_{reaction} = (\dot{Q}_{sensible,LG} - \dot{Q}_{sensible,FG}) + (\dot{Q}_{sensible,rich-MEA} - \dot{Q}_{sensible,lean-MEA})$$

refers to the adiabatic process, relating the heat of reaction, due to CO₂ capture ratio, to the sensible heat of the flue gases (from base composition to lean gas) and to the MEA-water solution (from lean to rich).

The total heat required in the reboiler to regenerate the solvent is given by three contributions

$$\dot{Q}_{reb} = \dot{Q}_{des} + \dot{Q}_{sensible} + \dot{Q}_{latent}$$

Where $\dot{Q}_{des}$ is the heat for desorption of the CO₂ from the solvent, $\dot{Q}_{sensible}$ the heat to raise the solvent temperature from inlet stripper to reboiler conditions, and $\dot{Q}_{latent}$ the heat required to evaporate the water in the stripper.

In this work the part-load model of the CO₂ capture plant is limited to pressure drop correlations scaling the pressure drop to changes in massflow and velocity squared through the absorber and stripper, with a constant geometry. The regeneration temperature and the MEA heat demand are proportional to the mole-flow of CO₂ captured and kept constant, while the amine over-circulation rate changes [20]. As the load decreases also the capture ratio changes varying from 90% to about 86%.

4. PART-LOAD PERFORMANCE OF CCPP PLANTS WITH CO₂ CAPTURE

Four types of combined-cycle power plants with different HRSG configurations have been modelled in part-load conditions with a CO₂ capture system based on amine scrubbing using hot water from the economizer to partly cover the reboiler duty. The typical method for reducing the load of a single shaft gas turbine in combined cycles is to decrease the mass flow rate with the variable inlet guide vanes (VGV) and, eventually, reduce the firing level in order to avoid too high exhaust gas temperatures. In the present work the load is reduced by 70% compared with the design point. Excellent heat recovery capabilities at all loads is achieved using the eco-reboiler couplings ability; in order to control the economizer mass flow rate, the flow in the reboiler circuit is increased when more heat is available.

4.1 EFFECT OF PART LOAD CONDITION ON CCPPs EFFICIENCY

The results obtained in terms of efficiency are presented in Figure 6. It shows that all the HRSG configurations answer in the same way to the load decreasing. It also shows that the plants utilizing the eco-reboiler concept increase their benefit towards the standard configured plant at low loads. This is due to the high fraction of heat supplied by the water to the absorbent regeneration process in the reboiler. Infact, the efficiency difference between the single-pressure level plants and the dual-pressure plant will be reduced (Figure7): at half load the efficiency difference is equal to 0.23%. When the load is reduced by 50%, as shown in Figure 8, the 2PRH still remains the plant with the highest efficiency, but the 1PRH recovers points overcoming the 3PRH which uses a conventional reboiler. Even though the advantage is less, the dual-pressure level plant with reheat has the best efficiency during the entire load range.

The single-pressure level non reheat plant, which has a much lower steam pressure to avoid moist problems in the expansion, is still the unit with the lowest performance.

Moreover, the efficiency difference between the dual- and single-pressure reheat plants decreases up to 0.16 percentage points. Figure 7 shows that passing from one to two pressure levels is no more justified in terms of plant efficiency when it operates at part-loads. With a single steam drum, the plant is quicker to start and responds more rapidly to load variations. Furthermore, the specific reboiler remains quite constant during the decrease of the load, passing from 4.41 to 4.54 MJ/kg for the 1PRH and from 4.39 to 4.50 MJ/kg for the 2PRH.

The advantages associated with an additional pressure level consume their gain in increased steam production, resulting in that the single-pressure plant with reheat has almost equal efficiency as the dual-pressure reheat plant and higher than the triple-pressure reheat plant with conventional reboiler (Table 3). For example, at half-load the efficiency of the dual-pressure reheat plant is 43.20% compared with 42.94% of the single-pressure reheat plant. This can be explained also because the single-pressure reheat plant is the one that utilizes more hot water from the economizer and less steam from the steam turbines (30% and 70% respectively) penalizing to a lesser extent the total efficiency of the integrated plant.

So, if the combined cycle power plant operates largely at part-loads during the year, a more simple configuration could be preferred, having it higher flexibility, lower operational costs, and performance competitive with those of a dual-pressure reheat plant. During load variations, the single-pressure reheat plant achieves performance higher than the triple-pressure reheat plant that utilizes a conventional reboiler. It is also interesting to see that the single-pressure reheat plant performs better than the triple-pressure almost immediately when the load is being reduced.

4.2 EFFECT OF PART LOAD CONDITION ON THE ECO-REBOILER LOOP

Thanks to low regeneration temperatures in the reboiler section of the CO₂ capture unit, that is fixed at 122°C to prevent the degradation of the MEA solvent, a significant fraction of the heat requested for the stripping process can be supplied by the hot water coming from the economizer, reducing the number and the mass flow rate of the steam extractions. In this way efficiency penalties of the integrated plant will be limited because the regeneration does not affect markedly the steam production.

Besides, in an operational mode deviating from the design point and with an eco-reboiler loop for the CO₂ capture, the analysis has highlighted that it is no much convenient to increase the number of pressure levels in the HRSG.

With the economizer-reboiler loop included in all the plants (except in the triple-pressure level plant that operates with a conventional reboiler), the reboiler duty comes partly from the economizer and partly from the steam turbine. At full load conditions the heat duty is provided by hot water and steam in different proportion, as shown in Table 2. The single-pressure plant with reheat uses for the most part heat from the economizer hot water for the stripping process (30% of the total heat duty) and the remaining heat is supplied by the extractions from the steam turbine, compared with the other two plants.

When the plants are operated in part-load conditions, the heat provided from the economizer to regenerate the amine in the reboiler is quite constant during the decreasing of the load. Not surprisingly the single-pressure level plant with reheat is still the one which uses the most heat from the economizer, as it can be seen from Figure 9. In this Figure, the variation of the reboiler heat fraction supplied via water, during the load decreasing, is shown. The point at which the maximum allowed exhaust gas temperature is reached can be seen in that Figure as a change of derivative. From this load point the firing temperature is reduced as VGV continues to close.

5. CONCLUSIONS

This study has shown the behaviour of four types of combined cycle power plants with MEA post-combustion CO₂ capture at part-load conditions. Configurations with single-, double- and triple-pressure levels HRSG have been investigated; the 1P configuration was investigated with and without reheat, the others just with reheat. To limit the mass flow rate to the CO₂ removal system and increase its concentration, a 40% fraction of exhaust gases was always recirculated to the compressor suction. The part-load operation

was obtained by varying the inlet guide vanes angle up to 40° with respect to the design condition. The basic off-design modelling of a combined cycle included four main models for simulating the behaviour of the gas turbine group, the heat recovery steam generator, the steam turbines and the CO_2 capture system.

A study was conducted in order to find the best configuration, in terms of efficiency, when the CCPP plant with the carbon dioxide capture was requested to operate at part-loads. The results obtained showed that the gap between the efficiency of the single- and dual-pressure level plants, both with reheat, diminishes when load decreases up to a difference of a few thousandths. Hence, it may be preferred to build a plant with single steam pressure level, also because it requires less capital, operational and maintenance costs. Furthermore, there were no reasons to choose a triple-pressure level plant with reheat and with conventional reboiler because its efficiency, when CO_2 is captured, was always smaller than the efficiency of the correspondent dual-pressure level reheat plant. This was also valid when triple-pressure was compared to the single-pressure reheat plant, except at full load operation.

Moreover, combined-cycle power plants have low investment costs and short construction times compared to large coal fired stations and even more compared with nuclear plants. This is an important factor in today's deregulated electric power market, where high investment costs are connected to a larger risk. In this scenario the results obtained in this work are important, also considering that combined cycle power plants are commonly used at partial loads to follow variable energy requests during the day. Infact, they could not be covered through other electricity generating plants due to their impossibility to be planned, if fed by renewables sources, or due to the long start-up time, like for the conventional steam power plants.

In conclusion, if the CCPP plant operates largely at part loads during the year, a single-pressure reheat plant could be preferred because it is cheaper to construct, offers better flexibility in terms of rapid response to load changes, and gives competitive performance with CO_2 capture.

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Figure 9. Heat fraction in the reboiler supplied via hot water in part-load conditions.

		1P	1PRH	2PRH	3PRH
Admittance pressure	[bar]	100	160	160	160
η_{LP} (wet)	[%]	83.4	84.8	84.7	85.0
η_{IP}	[%]	90	90	90	90
η_{GT} (at 40% EGR)	[%]	37.7	37.7	37.7	37.7
Pressure loss in HRSG	[mbar]	40	40	40	40
Evaporator pinch point	[°C]	8	8	8	8
Ambient temperature	[°C]	15	15	15	15

Table 1. Main calculation assumptions.

		<i>1P</i>		<i>1PRH</i>		<i>2PRH</i>	
		<i>design point</i>	<i>30% load</i>	<i>design point</i>	<i>30% load</i>	<i>design point</i>	<i>30% load</i>
Capture ratio ϕ	[%]	90	86.76	90	86.53	90	86.21
CO ₂ compression power	[MW]	11.35	4.76	11.24	4.95	11.59	4.92
Specific reboiler duty	[MJ/kg _{CO2}]	3.58	3.63	4.41	4.54	4.39	4.50
Heat fraction from hot water to reboiler	[MW]	31.86	14.16	52.58	24.84	31.07	12.97
	[%]	18.79	19.74	30.90	33.17	18.32	17.56
Heat fraction from steam extraction to reboiler	[MW]	137.68	57.55	117.58	50.04	138.50	60.88
	[%]	81.21	80.26	69.10	66.83	81.68	82.44

Table 2. Key parameters at the design point of the reboiler duty in the CO₂ capture unit for single-pressure level and single- and dual-pressure level plants with reheat

<i>1P</i>		<i>1PRH</i>		<i>2PRH</i>		<i>3PRH</i>		
<i>Load</i> [%]	η [%]	<i>P</i> [MW]	η [%]	<i>P</i> [MW]	η [%]	<i>P</i> [MW]	η [%]	<i>P</i> [MW]
100	50.55	395.56	51.27	402.50	51.65	403.88	51.31	401.29
90	49.78	355.92	50.75	362.07	51.02	362.84	50.53	361.27
80	48.40	316.49	49.44	322.18	49.68	322.36	49.20	321.18
70	46.75	277.72	47.78	281.05	48.05	282.42	47.51	280.87
60	44.48	236.98	45.67	241.84	45.89	242.42	45.31	240.49
50	41.79	198.09	42.94	201.24	43.20	202.40	42.55	200.10
40	38.23	158.03	39.48	161.35	39.62	161.26	39.01	160.13
30	33.66	118.99	34.81	120.55	34.95	120.88	34.47	121.05

Table 3. Efficiency and power of various plant configurations at different loads.