

INTEGRATION OF SOLAR ENERGY IN HYBRID POWER PLANTS (COMBINED CYCLES & RANKINE CYCLES)

- Integration evaluation
- Integration proposals

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1. Nomenclature

A: Solar collector field area
C : Cost
E : Energy production
HRSG: Heat Recovery Steam Generator from a Combined cycle.
I : Irreversibility
 I_b : Beam solar irradiation
LHV: Fuel low heating value.
 m_f : Fuel mass flow.
 $m_{f,saved}$: Saved fuel mass/ mass flow by the solar integration.
 Q_s : Solar thermal energy input to power plant
 Q_f : Fuel thermal energy¹ input to power plant
 $Q_{f,saved}$: Saved fuel heat input by the solar integration.
Q : Total thermal energy input to power plant
SF : Solar Fraction
t: Time
T: Temperature
 W_R : Contractual plant electricity rated output.
 W_s : Electricity output from power plant due to Q_s
 W_f : Electricity output from power plant due to Q_f
 W_{GT} : Electricity output from gas turbine
 W_{ST} : Electricity output from steam turbine
 $W_{f,ref}$: Electricity output from an only fuel reference power plant
 W : Total electricity² output from hybrid power plant

Subindex

d: Design conditions
f: Fuel

Greek

ΔI_f : Avoided irreversibilities in the fuel use.
 η : Efficiency
 η_B : Boiler efficiency

¹ In general we will make a difference between the fuel thermal energy input and the fuel enthalpy input , in order to explicitly account for all boiler efficiencies in the integration evaluation parameters. That is

$$\dot{Q}_f = \eta_B \cdot \dot{m}_f \cdot LHV .$$

It is important to note that one very relevant aspect of solar energy integration in a power plant is how efficiently fuels are being used, and therefore the use of residual boiler thermal energy should be incorporated on evaluation parameters. A solar integration can both improve or worsen boiler efficiency. Performance parameters defined with ' Q_f ' are relevant to the thermodynamic power cycle, while performance parameters defined with ' $m_f \cdot LHV$ ' are relevant to the overall power plant. This is an important difference between hybrid (or only fuel) and only solar power plants. The first ones require burner efficiency specification, while in the latter the power cycle does not have any stack losses to account for.

² We will make no distinction between net and gross electricity output or efficiency. All parameters can be applied to both concepts.

η_{cycle} : Power cycle efficiency
 η_{cc} : Combined cycle efficiency
 $\eta_{\text{solar field}}$: Solar field efficiency
 η_{ST} : Steam turbine cycle efficiency (not steam turbine efficiency)
 η_{GT} : Gas turbine cycle efficiency
 η_{HRSG} : Heat recovery steam generator efficiency
 ζ : Integration evaluation parameter other than efficiency

2. Introduction

In the last years there have been many different proposals on how to integrate solar energy in hybrid power plants. In Figure-1 we show a schematic with different proposals to hybridise a Combined Cycle Power Plant. Indicated with an arrow coming from a sun, and pointing to the place where the solar energy would be introduced in the power block, are the names of the institutions that have proposed these integrations. In Figure-2 we find the same information shown on the thermodynamic cycle.

Due to the lack of a common basis to evaluate the quality of such integrations, it is difficult to compare them and to choose the most appropriate for a given application.

Proposals to Introduce Solar Energy on a Combined Cycle

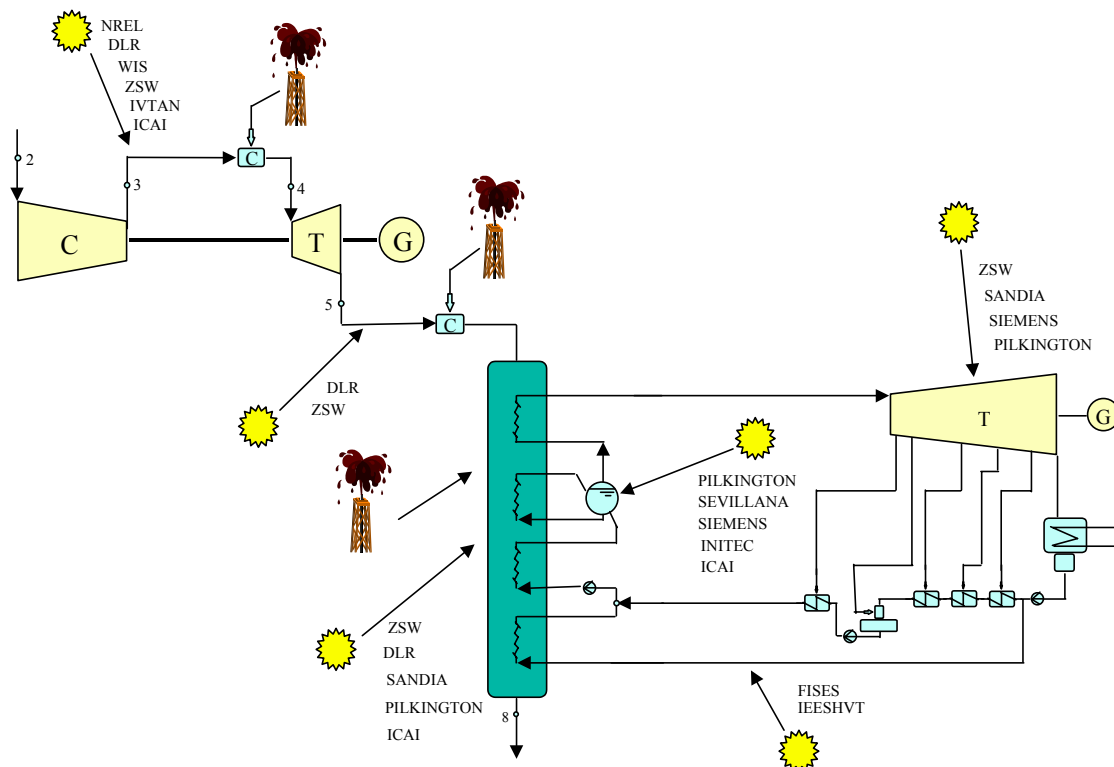


Figure-1: Proposals for integration of solar energy on a combined cycle power plant. Power block point of view.

Therefore, there is a need to have common and appropriate energy integration performance parameters that allow to compare the different proposed integration schemes, as well as to take on optimisation analysis that lead to the optimum integration conditions for a given integration scheme.

In the case of solar only thermal power plants, an adequate performance parameter is the total conversion efficiency, product of solar field efficiency and

power block efficiency, because it relates the useful output (generated electric energy) with the size from the solar field (needed investment). This was the parameter used to optimise the coupling between solar field and Rankine-type power block in design conditions for solar only plants in (García-Casals X., 2000), (García-Casals X., 1996). The power plants that can be operated in solar only mode in design conditions are Rankine cycles with parabolic through technology or central receiver technology, and Gas Turbine or Combined Cycles with central receiver technology. However, if these power plants are to produce electricity in hybrid mode or in fuel only mode for some hours a year, other integration parameters need to be used in order to judge how are solar and fuel energy sources being used.

Proposals to Introduce Solar Energy on a Combined Cycle

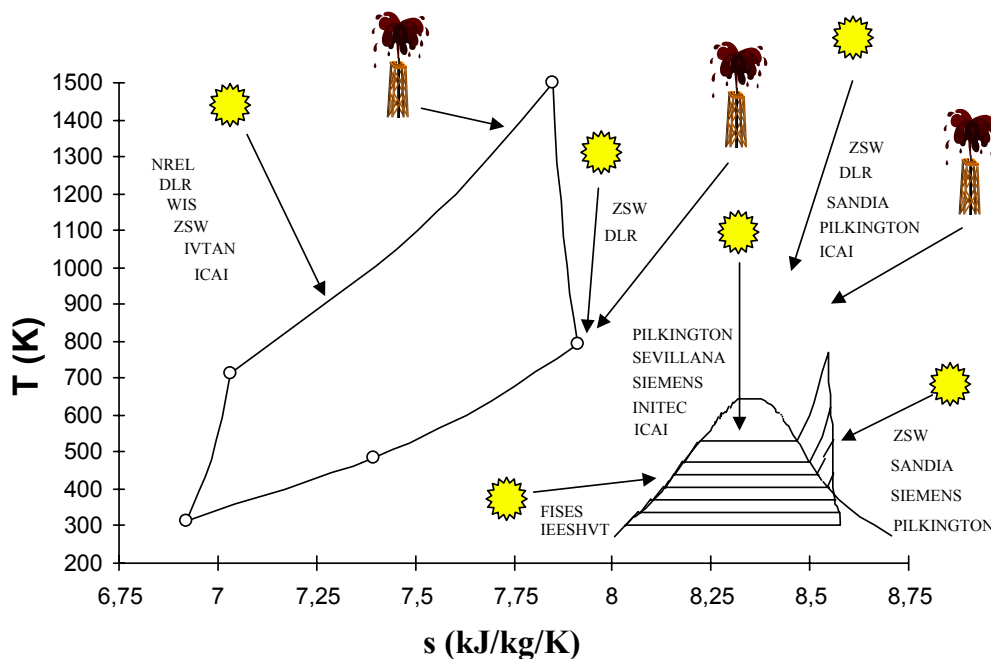


Figure-2: Proposals for integration of solar energy on a combined cycle power plant. Thermodynamic point of view.

In the case of Combined Cycles hybridised with parabolic through technology, on the one hand the fact that even under design conditions an hybrid operation is required, and on the other hand the more complex energy fluxes, have lead to a diversity of used parameters to characterise the behaviour and solar integration on these power plants. Some of the used performance parameters do not have a thermodynamic meaning when used without clear restrictions, and not only prevent the straight comparison between different integration schemes, but also often lead to misleading or wrong conclusions.

Therefore in this report we will first proceed to list together and discuss the different integration performance parameters proposed up till now, as well as the new ones proposed here, with the objective to identify the most appropriate ones. Latter we will review the main different integration possibilities, the ones

already proposed as well as new ones, with qualitative discussion on their pros and cons from the point of view of energy integration arguments. Hopefully this will lead to have some light on the appropriate future research work on optimisation of different integration schemes.

Almost all integration schemes to be considered, can be implemented both with parabolic trough technology (DSG and oil systems), and central receiver technology. From the thermodynamic point of view, the best solar technology would be the one that achieves the higher total conversion efficiency from solar radiation (I_b) to the required solar thermal input in power cycle (Q_s), which includes the solar collector efficiency as well as the effect from the effectiveness of required additional heat exchangers. However, the higher conversion efficiency from I_b to Q_s only means that the minimum solar collector area is needed to perform this solar thermal energy supply, and this becomes mixed with cost considerations when the optimum solar technology is to be chosen. Indeed, it could be that a solar technology with lower total conversion efficiency from I_b to Q_s would be the best choice because of a lower cost in solar equipment compensating the lower efficiency. Cost considerations depend on time and market development, so the more economic solar technology for a given integration scheme today can be different from tomorrow's best choice. In contrast, the thermodynamic arguments we present here will be constant in time. The choice of the appropriate solar technology for a given integration scheme is left to the reader, who should join to the thermodynamic argument the costs in a given moment in time.

3. Integration evaluation parameters

Ideally, the integration evaluation parameters should be annual average values, based on annual power plant simulations. However, in the actual state of these hybrid power plants, far from knowing the optimal options, and with a huge variety of conceptual proposals, it is not possible neither appropriate to go into enough calculation detail in all of them as for performing an annual simulation. Besides, such characterisation would be dependent on the weather data used (typical meteorological year), requiring an agreement about the TMY to be used in order to compare different schemes. Before investing all the resources needed to perform an annual simulation, we need tools to judge whether or not a given integration scheme is worthwhile.

For an hybrid power plant that will work only when the needed solar energy is available (straight or via thermal storage), it will be enough with evaluating the integration parameter in design conditions.

However, because of the low solar fractions achieved by the integration schemes on Combined Cycles (typically around 10 % when introducing solar energy in bottoming cycle), and the high investment on the power block, economic reasons will force the power plant to work with solar input under design conditions and even without solar input. In fact, for an intermediate or

base load power plant, most of the hours along the year the plant will be working under solar design conditions. This will be also the case for integration of solar on hybrid Rankine type power plants (although with higher solar fractions).

In all these cases, the solar input into the power plant often introduces penalties on the only fuel operation mode (or solar under design operation mode) because of having some plant components working in part load or out of design point, or because of having to use fuel on the low temperature part of the heat input (low exergy efficiency in fuel use). Therefore, to characterise these hybrid power plants often evaluation parameters will need to be evaluated at least in two working conditions: Design point, and only fuel operation mode. We think that this is the most general and appropriate way to characterise solar integration in hybrid power plants with optimisation or comparison aims. Average performance for a given operation strategy can be approximated using the value of evaluation parameter ζ in design condition (ζ_d) and in only fuel operation (ζ_f) by assuming the power plant to operate t_d hours a year in design conditions and t_f hours a year in only fuel operation by Eq.1.

$$\bar{\zeta} \approx \frac{t_d \cdot \zeta_d + t_f \cdot \zeta_f}{t_d + t_f} \quad (\text{Eq.1})$$

Of course, this approximation of average performance is not intended to substitute annual simulation in a more advanced design stage, but only as an efficient way to compare different plant proposals characterising each of them by two single parameters (ζ_d and ζ_f). Suggested values for t_d and t_f used in (Rheinländer J., Ratzesberger R., Hahne E., 1994) for an intermediate to base load plant, are $t_d = 1800$ hours/year and $t_f = 4200$ hours/year.

An intermediate approach between this two-parameter characterisation and annual simulation would be simulation along some typical days. However, this would present most of the drawbacks of an annual simulation.

3.1. ¿What should we ask from an integration evaluation parameter?

Lets write down some qualities we would ask from an integration evaluation parameter:

- Non dimensionality.
In order no to depend on the chosen units, and to allow a fast comparison between different schemes.
- $\zeta \in (0 , 100) \%$
In order to be able to fast situate a power plant on its appropriate range.

- A general and valid definition for different power cycle schemes that will allow a comparison between them.
- That gives a measure about one, or better both, of the following factors:
 - Effectiveness in the use of fossil fuels
 - Effectiveness in the use of the solar investment
- With physical fundament.
- Independent of time, social, economic or politic circumstances.
 For example this means no dependency on solar component costs or fuel prices, because this parameters fluctuate along time, and this would prevent comparison between different power plant integration schemes proposed on different times. Besides, the costs of solar components and fuel prices depend on many factors alien to conservation principles.
 In this way it could seem that this evaluation parameter would not allow to have a straight knowledge about which is the optimum integration scheme for a given time and place. However, if the integration evaluation parameter gives a good measure of energy and exergy conversion efficiencies, the absolute optimum plant scheme will be the same at any time, changing only the optimum solar fraction depending on relative costs. Site and time dependent boundary conditions can be incorporated to find the optimum plant for a given situation from the absolute ranking.

3.2. Listing of integration evaluation parameters

In this point we are going to list different integration evaluation parameters used up till now as well as some new evaluation parameters that we propose. In point 5 we will find more discussion about these parameters.

We will designate as η_i those parameters that may be interpreted as an efficiency, and as ζ_i those parameters that can not be interpreted as an efficiency.

- The parameter η_1 from Eq.2 is the thermal efficiency of the power cycle. It is important to note that this is the only cycle thermal efficiency with thermodynamic meaning (related to thermal energy input).

$$\eta_1 = \eta_{cycle} = \frac{\dot{W}}{\dot{Q}_s + \dot{Q}_f} \quad (\text{Eq.2})$$

- The parameter η_2 from Eq.3 is a thermal efficiency from the power plant based on fuel enthalpy input and solar thermal energy input. Although more appropriate than η_1 to evaluate integration schemes because of including

boiler efficiencies, somehow it does not give a fair treatment to both fuel and solar energy inputs, because ‘boiler’ efficiency from the solar contribution (Solar field efficiency) is not included. Therefore it does not reflect the relative cost of introducing Q_s at different temperature levels.

$$\eta_2 = \frac{\dot{W}}{\dot{Q}_s + \dot{m}_f.LHV} \quad (\text{Eq.3})$$

- The parameter η_3 from Eq.4 is a thermal efficiency from the power plant based on fuel thermal energy input and solar radiation energy input. Although more complete than η_1 because of including collector field efficiency, it does not give a fair treatment to both fuel and solar energy inputs, because boiler efficiency for the fuel is not included.

$$\eta_3 = \frac{\dot{W}}{I_b.A + \dot{Q}_f} \quad (\text{Eq.4})$$

- The parameter η_4 from Eq.5 is a thermal efficiency from the power plant based on fuel enthalpy input and solar radiation input. This efficiency is the one that incorporates all component efficiencies for conversion from solar radiation and fuel enthalpy to electrical energy.

$$\eta_4 = \frac{\dot{W}}{I_b.A + \dot{m}_f.LHV} \quad (\text{Eq.5})$$

In any parameter that involves solar and/or fuel energy inputs, we find all the above possibilities depending on weather we consider boiler and solar field efficiencies or not. From the cycle point of view η_1 is the relevant parameter, relating energy input and output from thermodynamic cycle. However, from a power plant point of view η_4 would be the most significant parameter because of relating energy inputs and outputs from the power plant, accounting for all the loss mechanisms in the power plant. In a solar only plant, the equivalent to η_4 (in this case the same as η_3), would be the total conversion efficiency, defined by Eq.6:

$$\eta_{tot} = \frac{W}{I_b.A} = \eta_{cycle} \cdot \eta_{solar \text{ field}} ; \quad \text{Solar only plant} \quad (\text{Eq.6})$$

This was the parameter we used to optimise solar only plants in (*Garcia-Casals X., 1996*) and (*Garcia-Casals X., 2000*).

As we will see later on, the solar fraction plays a very important role in most integration evaluation parameters. If we define the solar fraction in terms of the thermodynamic cycle by Eq.7 , the parameters η_2 , η_3 , and η_4 can be evaluated

in terms of the cycle efficiency (η_1), the solar fraction, and boiler or solar field efficiencies, as shown in Eq.8, Eq.9, and Eq.10.

$$SF = \frac{\dot{Q}_s}{\dot{Q}_s + \dot{Q}_f} = \zeta_1 \quad (\text{Eq.7})$$

$$\eta_2 = \frac{\eta_{\text{cycle}}}{\frac{1}{\eta_B} - SF \cdot \left(\frac{1}{\eta_B} - 1 \right)} \quad (\text{Eq.8})$$

$$\eta_3 = \frac{\eta_{\text{cycle}}}{1 + SF \cdot \left(\frac{1}{\eta_{\text{solar field}}} - 1 \right)} \quad (\text{Eq.9})$$

$$\eta_4 = \frac{\eta_{\text{cycle}}}{\frac{1}{\eta_B} - SF \cdot \left(\frac{1}{\eta_B} - \frac{1}{\eta_{\text{solar field}}} \right)} \quad (\text{Eq.10})$$

In this expressions we may begin to appreciate the important role that solar fraction plays on evaluation parameters, in such a way that generally it will not make sense to speak only about the value of a given integration evaluation parameter without relating it to a given solar fraction. For given values of cycle, boiler and solar field efficiencies, the values of η_2 , η_3 , and η_4 , that are evaluation parameters equivalent to total conversion efficiency on a solar only plant, change with solar fraction. Moreover, the change with solar fraction is different for each one of η_2 , η_3 , and η_4 . Indeed, Eq.8 shows that η_2 increases with SF, Eq.9 shows that η_3 decreases with SF, and Eq.10 shows that η_4 may increase or decrease with SF depending on the relative values of η_B and $\eta_{\text{solar field}}$ (usually $\eta_B > \eta_{\text{solar field}}$, and η_4 decreases with SF).

Therefore, the values of η_2 , η_3 , and η_4 may be changed as desired by modifying the solar fraction, giving a misleading information about the goodness of the integration scheme if they are not considered on relative basis to SF. On the other hand, η_1 does not have an explicit dependency on solar fraction, and therefore reflects how well thermal energy inputs are being used in the integration scheme. But η_1 gives no information at all about how are converted chemical fuel energy and solar radiation energy on thermal inputs to the power cycle.

- The parameter η_5 from Eq.11 is the efficiency at which fuel thermal energy input is converted to work. This kind of parameters that require to distinguish between the work output associated to fuel input from the work output

associated to solar input, are not always easy to calculate on a given power plant. Indeed, sometimes it is not possible to track independently the evolution of both thermal inputs on the power cycle.

If possible to calculate, η_5 gives a convenient way to measure how good fuel thermal energy is being used in the power plant. Solar hybrid power plants lose a good deal of their meaning if they use fuel energy (exergy) worse than a conventional power plant, and therefore it is important to monitor how fuel energy is being used on an integration scheme.

$$\eta_5 = \frac{\dot{W}_f}{\dot{Q}_f} \quad (\text{Eq.11})$$

- The parameter η_6 from Eq.12 is the efficiency at which fuel energy input is converted to work. This parameter incorporates the information on boiler efficiency and therefore is more appropriate to evaluate an integration scheme than η_5 , because in some cases a solar integration can produce a reduction in η_B .

$$\eta_6 = \frac{\dot{W}_f}{\dot{m}_f \cdot \text{LHV}} = \eta_5 \cdot \eta_B \quad (\text{Eq.12})$$

- The parameter η_7 from Eq.13 is the efficiency at which solar thermal energy input is converted into work. Whenever it is possible to track the way solar thermal energy follows in the power cycle and distinguish the work output due to this source from the one associated to fuel input, this parameter is a good measure of how solar thermal input managed by the hybrid plant. However, η_7 includes no information about how solar radiation is converted into the thermal energy input in power cycle.

$$\eta_7 = \frac{\dot{W}_s}{\dot{Q}_s} \quad (\text{Eq.13})$$

- The parameter η_8 from Eq.14 is the efficiency at which solar radiation input is converted into work. With the same limitation as the former parameter associated to the possibility to decompose the total work output in the solar and fuel contribution ($W = W_s + W_f$), η_8 now incorporates the information about how solar radiation is converted on solar cycle thermal input, and therefore is more appropriate to relate required inversion to solar electricity production.

$$\eta_8 = \frac{\dot{W}_s}{I_b \cdot A} = \eta_7 \cdot \eta_{\text{solar field}} \quad (\text{Eq.14})$$

- The parameter ζ_1 from Eq.15 is the solar fraction associated to solar and fuel thermal inputs in power cycle.

$$\zeta_1 = \frac{\dot{Q}_s}{\dot{Q}_s + \dot{Q}_f} \quad (\text{Eq.15})$$

- The parameter ζ_2 from Eq.16 is the solar fraction associated to solar radiation and fuel enthalpy input into the power plant. Two other intermediate parameters between ζ_1 and ζ_2 are also possible incorporating only information about burner or solar field efficiencies. In Eq. 17 the relation between ζ_1 and ζ_2 is shown.

$$\zeta_2 = \frac{I_b \cdot A}{I_b \cdot A + \dot{m}_f \cdot LHV} \quad (\text{Eq.16})$$

$$\zeta_2 = \frac{1}{1 + \frac{\eta_{\text{solar field}}}{\eta_B} \cdot \left(\frac{1}{\zeta_1} - 1 \right)} \quad (\text{Eq.17})$$

- Sometimes, solar fraction³ is defined with electricity outputs instead than with energy inputs. In this case it would be given by parameter ζ_3 from Eq.18. The problem with this definition of solar fraction is that often it is much more complicated (or even impossible) to separate the total electricity output (W) in the solar and fuel contributions ($W = W_s + W_f$), while this distinction is always possible with plant energy inputs.

$$\zeta_3 = \frac{\dot{W}_s}{\dot{W}} \quad (\text{Eq.18})$$

It is important to notice that approximate annual solar fractions according any of the definitions given by Eq.15, 16 or 18 , are in general different from average values calculated by Eq.1. Indeed, only in the case that total power plant energy input in the case of Eq.15 and 16 , or total power plant output for Eq.18 are the same in design conditions than in fuel only operation will these values coincidence. In general, annual average solar fraction can be approximated by the knowledge of design conditions and only fuel operation by means of Eq.19 for the case of Eq.18 (or equivalent ones for Eq.15 and 16):

³ To distinguish between design solar fractions and annual solar fractions, the first ones are sometimes called solar ratios (Rheinländer J., et al, 1994).

$$(\zeta_3)_{average} = \frac{t_s \cdot (\dot{W}_s)_d}{t_s \cdot (\dot{W})_d + t_f \cdot (\dot{W})_f} \quad (\text{Eq.19})$$

- The parameter ζ_4 from Eq.20 has been very often used to magnify the advantages of a solar integration in a hybrid power plant. Moreover, often this parameter has been called cycle efficiency for the power plant. However, this parameter, by itself lacks thermodynamic meaning. Indeed, in the numerator we account for the fuel and solar contributions to work output, while in the denominator only the fuel thermal input is considered. Therefore this parameter always tends to infinity when solar fraction tends towards one. This can be very misleading and hide a very bad integration scheme behind a very high value of the integration evaluation parameter. Just by increasing the solar fraction we can get ζ_4 -values higher than an equivalent only fuel plant, and this even if in the integration process cycle efficiencies are reduced and coupling between cycles on a combined cycle power plant are worsened. Therefore, this hybrid power plant would be making a worse use of fuel energy than a conventional power plant, and still be claimed to have a higher efficiency than the fuel power plant.

$$\zeta_4 = \frac{\dot{W}}{\dot{Q}_f} \quad (\text{Eq.20})$$

- The parameter ζ_5 from Eq.21 is the levelized electricity cost. This parameter, although very important to justify the investment on a given power plant, on the one hand demands an annual simulation to know the annual electricity production (depends on site conditions, operation mode, ...), and on the other depends on component costs, subsidies and economic analysis assumptions. Therefore, from a technical point of view it is not possible to compare integration schemes by their ζ_5 -value evaluated by different people, in different places and moments in time.

The LEC value from a hybrid power plant should anyway be calculated in advanced design stages, but the decision on the more appropriate integration scheme should be taken in former analysis stages by using other integration evaluation parameters.

$$\zeta_5 = \frac{C}{E} = LEC \quad (\text{Eq.21})$$

- The parameter ζ_6 from Eq.22, relates the reduction of irreversibility in the use of fuel energy when integrating solar energy on the power plant, to the fuel enthalpy input. Therefore, its meaning is the equivalent increment of efficiency in the fuel only plant associated to the introduction of solar energy. A good integration scheme will have $\zeta_6 > 0$, while a bad integration scheme, that is, one that makes a worse use of fuels than a conventional fuel only plant, will have $\zeta_6 < 0$. The higher the value of ζ_6 for a given power plant, the

better is the integration scheme, because of allowing a better use of fuel energy on this power plant.

Caution has to be taken when comparing the ζ_6 -values from two different power plants. For example, consider integration of solar energy on a Rankine cycle and on a Combined cycle. It could be that the ζ_6 -value for the Rankine cycle integration would be higher than the one for the combined cycle, steaming from the fact that boiler irreversibilities on a Rankine cycle are much higher than irreversibilities on the HRSG of a Combined cycle. This would not mean that fuel energy is better used on an hybrid Rankine cycle than on an hybrid Combined cycle, because the starting values of the irreversibility in a Combined cycle are much lower than on a Rankine cycle. However, this would mean that from a fuel efficiency point of view it would be more effective to integrate solar energy on existing Rankine cycles than on existing combined cycles⁴. However if a new hybrid power plant is being considered, probably the hybrid Combined cycle will still have a lower specific irreversibility in fuel use than the Rankine cycle, and therefore the combined cycle should be chosen.

This parameter only looks to the fuel part of the hybrid power plant, and it says nothing about how efficiently is the solar investment being used in a given integration scheme.

Besides, some times it is not obvious how to make a difference between the irreversibility associated to the fuel input and the one associated to the solar input. For example, on the heat recovery steam generator from a combined cycle, total irreversibility after solar integration may be higher than before because of the increased steam mass flow due to solar input, but the irreversibility associated to fuel use is reduced if temperature differences in heat transfer are diminished.

$$\zeta_6 = \frac{\Delta \dot{I}_f}{\dot{m}_f \cdot LHV} \quad (\text{Eq.22})$$

- The parameter ζ_7 from Eq.23 is called the solar capacity factor. It relates the solar electricity production on a given period of time 't' (typically t = 1 year) to the solar electricity production if the plant would work always on design conditions (design solar insolation input). This parameter requires an annual simulation (dependent on location), and to be able to separate the solar electricity production from the fuel one, which is not possible in some plant schemes.

⁴ From the pollutant emission reduction point of view, because of having most Rankine cycles working on coal and Combined cycles working on gas, it would also be more effective to integrate solar energy on a Rankine cycle of existing power plants.

$$\zeta_7 = \frac{\int \dot{W}_S \cdot dt}{t \cdot \dot{W}_{S, design}} \quad (\text{Eq.23})$$

- The parameter ζ_8 from Eq.24 relates the incremental power output due to hybridisation to the solar thermal energy input. This is a way to overcome the indetermination associated to the knowledge of $\dot{W}_S$ in η_7 . However, to do so a reference only fuel plant has to be chosen. In principle, for this reference plant one can take the state-of-the-art fuel only plants being implemented at the time, the alternative only fuel plant to be erected at the site, or a fossil only plant as similar as possible to the proposed hybrid plant. The last choice would probably be the more adequate because of giving ζ_8 a meaning close to the efficiency at which solar thermal input is converted into work. However, the hybrid and only fuel plant will seldom be identical, and that is the reason why this parameter is not classified as an efficiency. Although always possible to evaluate, this parameter requires an agreement about the reference fuel plant, and even with this agreement, it may be difficult to compare two different hybrid plants by their ζ_8 -value. The parameter ζ_9 from Eq.25 is analogous to ζ_8 but including information about collector field efficiency.

$$\zeta_8 = \frac{\dot{W} - \dot{W}_{f, ref}}{\dot{Q}_S} \quad (\text{Eq.24})$$

$$\zeta_9 = \frac{\dot{W} - \dot{W}_{f, ref}}{I_b \cdot A} = \zeta_8 \cdot \eta_{solar field} \quad (\text{Eq.25})$$

- The parameters ζ_{10} and ζ_{11} from Eq.26 and Eq.27 have the same meaning as the parameters ζ_8 and ζ_9 , but now, the reference fuel plant is the hybrid power plant in only fuel operation. This has the advantage of avoiding the indetermination in the choice of the fuel reference plant. However, as in one of the two operation modes (only fuel and design) there will be components working in part load, or a different fuel usage will be done in both cases, this parameters are also not a conversion efficiency. Care should be taken when comparing ζ_{10} or ζ_{11} -values from different hybrid power plants because these can be done as high as wished by just spoiling the only fuel operation mode, or by reducing the fuel supply in only fuel mode. For a given solar hybrid power plant with a given fuel consumption in only fuel operation, the lower the only fuel mode efficiency, the higher the ζ_{10} or ζ_{11} -values (the worst the fuel usage, the better the integration evaluation parameter). Therefore, caution should be taken when judging a hybrid power plant by its ζ_{10} or ζ_{11} -values. The ζ_{10} -value was used by Siemens to characterise their combined cycle proposals for the STEM project.

$$\zeta_{10} = \frac{\dot{W}(\dot{Q}_S) - \dot{W}(\dot{Q}_S = 0)}{\dot{Q}_S} \quad (\text{Eq.26})$$

$$\zeta_{11} = \frac{\dot{W}(\dot{Q}_S) - \dot{W}(\dot{Q}_S = 0)}{I_b \cdot A} = \zeta_{10} \cdot \eta_{\text{solar field}} \quad (\text{Eq.27})$$

- The parameter ζ_{12} and ζ_{13} from Eq.28 and Eq.29 are the alternative way to look at partial conversion efficiencies. In this case emphasis is given to the efficiency at which fuel energy is converted in the hybrid power plant. As a reference power output, in these cases we take the power output the hybrid plant would produce if we would give no fuel input (and keeping solar input in design conditions), and we assume that the increase in work when going from the reference case to the design case is fully due to the fuel input. This assumption magnifies the goodness of the use of fuel energy that is being done in the hybrid plant, because on the one hand we take a part load operating mode as reference case, and on the other hand often we assume that the low exergy heat input zone is covered by solar, contributing fuel only to the high exergy heat input zone. In general, this parameter is always higher than cycle efficiency, and increases with solar fraction. The ζ_{12} parameter was used by (Shpilrain E.E., Popel O.S., 1994) and (Garcia-Casals X., 1998) to analyse hybrid gas turbine operation.

$$\zeta_{12} = \frac{\dot{W}_d - \dot{W}(\dot{Q}_f = 0; \dot{Q}_S = \dot{Q}_{S,d})}{\dot{Q}_{f,d}} \quad (\text{Eq.28})$$

$$\zeta_{13} = \frac{\dot{W}_d - \dot{W}(\dot{Q}_f = 0; \dot{Q}_S = \dot{Q}_{S,d})}{\dot{m}_{f,d} \cdot LHV} = \zeta_{12} \cdot \eta_B \quad (\text{Eq.29})$$

- The parameter ζ_{14} , ζ_{15} , and ζ_{16} from Eq.30, Eq.31, and Eq.32 represent the increment in cycle or plant efficiency between the hybrid solar power plant and a reference power plant. The parameter ζ_{14} gives a pure thermodynamic measure of the integration quality from the point of view of the power cycle. The parameter ζ_{15} incorporates the boiler efficiency effect, which could be modified by the integration, and the parameter ζ_{16} still incorporates solar field efficiency effects. These parameters, although can be used also with Rankine type power plants, become specially useful when dealing with combined cycle plants because of incorporating on a convenient manner the important effect of solar fraction.

In these parameters, the reference efficiency $(\eta_i)_{\text{ref}}$ is calculated by the expressions presented on chapter-4, as a function of plant components (gas turbine cycle efficiency, steam turbine cycle efficiency, HRSG efficiency

and boiler efficiency) and solar fraction. Therefore the functional relationship for $(\eta_i)_{ref}$ is the one shown by Eq.33.

As we will see, the efficiency from the different combined cycle schemes with solar integration can be evaluated as an analytical expression of component efficiencies and solar fraction.

The quality from a solar integration on a combined cycle may not be judged only by its η_1 to η_4 values, because this cycle and plant efficiencies present a strong dependency on solar fraction.

If we assume a reference for the component efficiencies on a combined cycle, with $(\eta_i)_{ref}$ we take into account the main effect of solar fraction, discounting it from η_i , in such a way that ζ_i is already useful to evaluate the quality from the solar integration for all the values of solar fraction.

On the integration of solar energy on a combined cycle using the parabolic trough technology, in most of the cases, the gas turbine cycle efficiency (η_{GT}) will not be strongly affected by solar fraction (the design changes on HRSG may alter gas outlet pressure loss, and slightly gas turbine cycle efficiency). But both steam turbine cycle efficiency, and specially HRSG efficiency may experiment changes with solar fraction.

$$\zeta_{14} = \eta_1 - (\eta_1)_{ref} \quad (\text{Eq.30})$$

$$\zeta_{15} = \eta_2 - (\eta_2)_{ref} \quad (\text{Eq.31})$$

$$\zeta_{16} = \eta_4 - (\eta_4)_{ref} \quad (\text{Eq.32})$$

$$(\eta_i)_{ref} = f(\eta_{GT,ref}, \eta_{ST,ref}, \eta_{HRSG,ref}, \eta_{B,ref}, SF) \quad (\text{Eq.33})$$

For the efficiencies ($\eta_{GT,ref}$, $\eta_{ST,ref}$, $\eta_{HRSG,ref}$, $\eta_{B,ref}$) there are three possible reference cases:

1. The hybrid power plant working in only fuel mode, normally at part load with $Q_s = 0$.
2. A conventional fuel power plant with the same cycle scheme.
3. Establish some fixed values of cycle component for all integration schemes.

The first one has the problem that a bad solar integration (high reduction in component performance without solar input) would lead to a higher ζ_i -value. Besides, it would be no possible to have a straight comparison between the ζ_i -values of different integration schemes because the difference between their qualities could be hidden by the different reference values.

The problem with the third option is that the ζ_i -value would no longer be a straight measure of integration quality when some component design parameters would change (for example, a change in gas turbine inlet

temperature or pressure ratio would lead to a change in η_{GT} , η_{HRSG} and η_{ST}), unless the established fixed cycle component efficiency values are also changed.

In principle, it looks as if the fact of using the same component efficiencies for all integration schemes would allow for a straight comparison between them. However, because cycle schemes would differ from one integration to another, we could have ζ_i -value < 0 for some integration schemes due not to the fact of a bad integration, but because of employing different component efficiencies.

Therefore, the most appropriate choice for the reference values seems to be the second option. However, this requires the knowledge and evaluation of a conventional fuel power plant with the same cycle scheme as the hybrid power plant under consideration.

Here we should point out that usually we have to know the only fuel power plant alternative to the proposed power plant in order to justify the solar investment in the power plant. However the alternative fuel reference plant may have a different power cycle scheme.

- The parameter ζ_{17} from Eq.34 proposed in (Steinmann W., 2000) relates the fuel mass saved by the solar integration scheme with the needed area of solar collectors. The fuel mass saved by the integration scheme is defined by the difference between the fuel mass consumed by the reference fuel only plant, and the fuel mass consumed by the hybrid plant. The reference fuel only plant is defined as one producing the same power output as the hybrid power plant. In order to evaluate fuel mass consumption from both power plants one needs to perform annual simulations. In (Steinmann W., 2000) it was proposed to evaluate this parameter by simulation over one characteristic day.

As commented before, the need to perform simulations over a certain period of time, on the one hand requires much computation effort to characterise the power plant, and on the other hand makes the evaluation dependent on climatic data used for the simulation. In order to be able to compare different power plants, an agreement should be reached about the evolution of solar radiation, ambient temperature and wind speed over the proposed typical day.

$$\zeta_{17} = \frac{m_{f, saved}}{A} \quad (\text{Eq.34})$$

- The parameter ζ_{18} in Eq.35 , based on the ζ_{17} above commented, on the one hand shows the information on ζ_{17} in a non-dimensional form , and on the other makes it independent from a time extended simulation by using fuel mass flows instead of consumed fuel mass.

Now this parameter looks like an efficiency, but however, we keep on assigning a ζ -value to it, because although generally it will be $\zeta_{18} < 1$ (because $\eta_B > \eta_{\text{solar field}}$), but a good solar integration will improve the use of fuel energy in the power plant, and therefore a good integration will be characterised by $Q_s < Q_{f, \text{saved}}$, opening the possibility of getting $\zeta_{18} > 1$. On the other hand, because now it is evaluated with fuel mass flows instead than with fuel mass, we can get $\zeta_{18} < 0$ (for example during night time operation of the hybrid plant).

In order to incorporate the information on the hybrid plant off-design performance when there is no solar energy available, the average ζ_{18} -value should be evaluated with Eq.1. It is important to note that if the solar integration worsens plant performance without sun, we will have $(\zeta_{18})_f < 0$ in Eq.1, in such a way that the average ζ_{18} -value can be above or below zero depending on the quality of solar integration, and on operation strategy (t_d and t_f values in Eq.1). To justify any integration, the average ζ_{18} -value should be above zero.

$$\zeta_{18} = \frac{\dot{m}_{f, \text{saved}} \cdot LHV}{I_b \cdot A} = \frac{\eta_{\text{solar field}}}{\eta_B} \cdot \frac{\dot{Q}_{f, \text{saved}}}{\dot{Q}_s} \quad (\text{Eq.35})$$

However, the fact of having an average ζ_{18} -value > 0 , still says nothing on how efficiently we are using the fuel. Indeed, the fuel used in the hybrid plant could still be used with a lower efficiency than on a conventional power plant, even for design conditions of the hybrid plant. Just by increasing solar fractions, a positive ζ_{18} -value can be reached even with a bad integration scheme.

The ζ_{18} parameter can be related to other parameters as follows:

We can use the reference fuel plant to define the solar electricity production from the hybrid plant as shown by Eq.36.

$$\dot{W}_s = \dot{m}_{f, \text{saved}} \cdot LHV \cdot \eta_{2, \text{ref}} = \dot{Q}_{f, \text{saved}} \cdot \eta_{1, \text{ref}} \quad (\text{Eq.36})$$

However it is important to note that this solar electricity production it is not the one really produced by the hybrid power plant, but only an artificial one defined from the reference fuel plant. This is easily noted by the fact that now it can be $W_s < 0$, while in the hybrid plant in spite of the bad fuel energy usage it makes, we will always have $W_s > 0$.

Therefore, we may find the relations from Eq.37 between ζ_{18} and other integration evaluation parameters.

$$\zeta_{18} = \frac{\eta_8}{\eta_{2, \text{ref}}} = \frac{\eta_7 \cdot \eta_{\text{solar field}}}{\eta_{2, \text{ref}}} = \frac{\eta_7 \cdot \eta_{\text{solar field}}}{\eta_B \cdot \eta_{1, \text{ref}}} \quad (\text{Eq.37})$$

The fact to introduce a fuel reference plant has the advantage of producing a measure of integration quality related to standard fuel plant technology, but presents two drawbacks:

- The ζ -value changes with the considered standard fuel technology.
- To evaluate the average ζ -value three power plant calculations and two power plant designs have to be done:
 - Standard optimised fuel plant in design conditions.
 - Hybrid plant in design and only fuel operation conditions.

- The parameter ζ_{19} in Eq.38, is called the entropic efficiency, and was introduced by (Pérez del Notario P., et al, 2000) as a way to standardise irreversibility evaluation in engineering adiabatic equipment by means of a second law based efficiency. For adiabatic systems, the argument from the exponential function in Eq.38 is related with total entropy increment through the second law, and because entropy is additive, the entropic efficiency from a complex system can be rather easily obtained as a product of the ζ_{19} -values from its elemental components. Because for most common working fluids the entropy is a logarithmic function of pressures and temperatures ratios, the exponential in Eq.38 takes away the logarithmic dependency.

Moreover, the ζ_{19} -value definition is common to all processes and components, verifies $0 < \zeta_{19} < 1$, and has a very straight relationship with process irreversibility, that is the ultimate thermodynamic variable that characterises an energy conversion process.

The ζ_{19} -value could also be used for non-adiabatic processes as long as the entropy flux associated to the heat fluxes can be evaluated.

It is a good approach to try to unify the efficiency definitions for all engineering processes, laying their foundations on basic physical principles. Following this line of thought, it would be good to evaluate integration schemes by their ζ_{19} -value, like with any other energy conversion equipment.

There are two ways to use the ζ_{19} -value to measure the quality from solar integrations:

1. Apply its definition to the overall hybrid power plant. This would produce a measure of the irreversibilities produced in the overall energy conversion process.
2. Apply its definition to those components that are critical to the solar integration quality (HRSG, bottoming cycle, ...). The application to the individual system components, gives the advantage of showing where the higher irreversibilities are present, allowing for a more efficient concentration of design efforts to improve plant quality. If the irreversibility in all components is determined, the overall plant irreversibility comes from adding them, and therefore the ζ_{19} -value for the overall plant can also be determined.

As well as with other integration parameters, there are 4 different possibilities when defining the overall plant ζ_{19} -value, depending on whether we take as starting energy inputs the different combinations m_f/Q_f and $I_b.A/Q_s$. The difference between them is whether we account for irreversibilities in combustion chamber and solar collectors or not. Probably, the most appropriate in order to judge integration quality it would be to take as plant inputs m_f and Q_s . That is because fuel exergy can be conserved, but solar radiation exergy may not. If anyway solar radiation exergy is to be included in the analysis, its value can be taken from (Winter C.J., et al, 1991).

There are two draw-backs associated to the ζ_{19} -value:

1. It could be difficult to separate the irreversibilities associated to the fuel usage in the power plant (the ones that mainly should be of interest in an integration evaluation), from the entropy fluxes associated to solar thermal energy inputs (we need to assign a temperature to these heat fluxes)
2. Irreversibility evaluations are less familiar to the engineering community than energy evaluations.

$$\zeta_{19} = \exp\left(-\frac{i}{T_o.(\dot{m}.c)_{min}}\right) \quad (\text{Eq.38})$$

4. Effects of solar fraction on combined cycle efficiency

As we have already mentioned, when introducing solar energy in the bottoming cycle, as in most of the designs with parabolic trough technology and some of the proposals with central receiver systems, solar fraction has an important effect on combined cycle efficiency. Much of this effect is not due to solar integration issues, but just to the fact to be introducing thermal energy in the bottoming cycle of the combined cycle power plant. Therefore, in order to be able to quantify the quality of a given solar integration, and to compare different solar integration schemes, it would be good to separate the effects of solar fraction associated to low bottoming cycle heat supply from those associated to integration issues. This is done by the integration evaluation parameters ζ_{14} , ζ_{15} , and ζ_{16} from Eq. 30, 31, and 32. However, to use this ζ -values, we need to evaluate the reference combined cycle efficiency as a function of component efficiencies and solar fraction, as indicated by Eq.33.

In this point we are going to present the needed results to evaluate $(\eta_i)_{ref}$ from Eq.33 for the different combined cycle schemes we can find, using the opportunity to introduce and gain some insight on the different combined cycle integration scheme possibilities.

From a thermodynamic point of view, the main evaluation parameters to be concerned about are η_1 , η_2 , η_3 , and η_4 . To focus on the power plant we are going to leave aside collector field efficiency, and therefore concentrate on the η_1 and η_2 parameters from Eq. 2 and 3.

A combined cycle can be formed combining any two or more power cycles. The cycle's combination can take either the form of a mass or thermal coupling. Following (Horlock J.H., 1992) we can distinguish two main kinds of cycle couplings: External and internal couplings.

The external coupling is always a thermal coupling in which the bottoming cycle (lower temperature level) uses as heat input part of the residual thermal energy from the topping cycle. These cycles are also called serie combined cycles. There are several types of this kind of combined cycle, and they are by far the most extended.

The internal coupling can be a thermal or a mass coupling. In an internal mass coupling, two or more elemental power cycles share the same working fluid, like in the Field cycle and its variations (Horlock J.H., 1992), (Jericha H., Hoeller F., 1991). In an internal thermal coupling the heat source supplies heat to two or more elemental cycles in the same point, like in the Presurized Fluidized Bed Coal Combustion plants, or the STIG/Cheng cycle. The combined cycles with internal thermal coupling are also called parallel cycles.

The main combined cycle applications usually involve a gas turbine cycle and a Rankine cycle. That is why most of the times we will speak about combined cycles formed by a gas turbine cycle (GT) and a steam turbine cycle (ST).

4.1. Series cycles without heat loss between cycles

The ideal series coupling combined cycle uses as only thermal input to the bottoming cycle all the residual heat output from the topping cycle. In Figure-3 we show a schematic from this kind of cycles, as well as an evaluation of its η_1 -efficiency as function of component cycles efficiency. In Figure-4 we show the schematic associated to the η_2 -efficiency for these combined cycles, where the only boiler efficiency to incorporate is the gas turbine combustion chamber efficiency.

The η_1 and η_2 efficiencies from these combined cycles, as function of component efficiencies, are given by Eq.39 and 40.

$$\eta_1 = \eta_{GT} + \eta_{ST} - \eta_{GT} \cdot \eta_{ST} \quad (\text{Eq.39})$$

$$\eta_2 = \eta_1 \cdot \eta_{B_{GT}} \quad (\text{Eq.40})$$

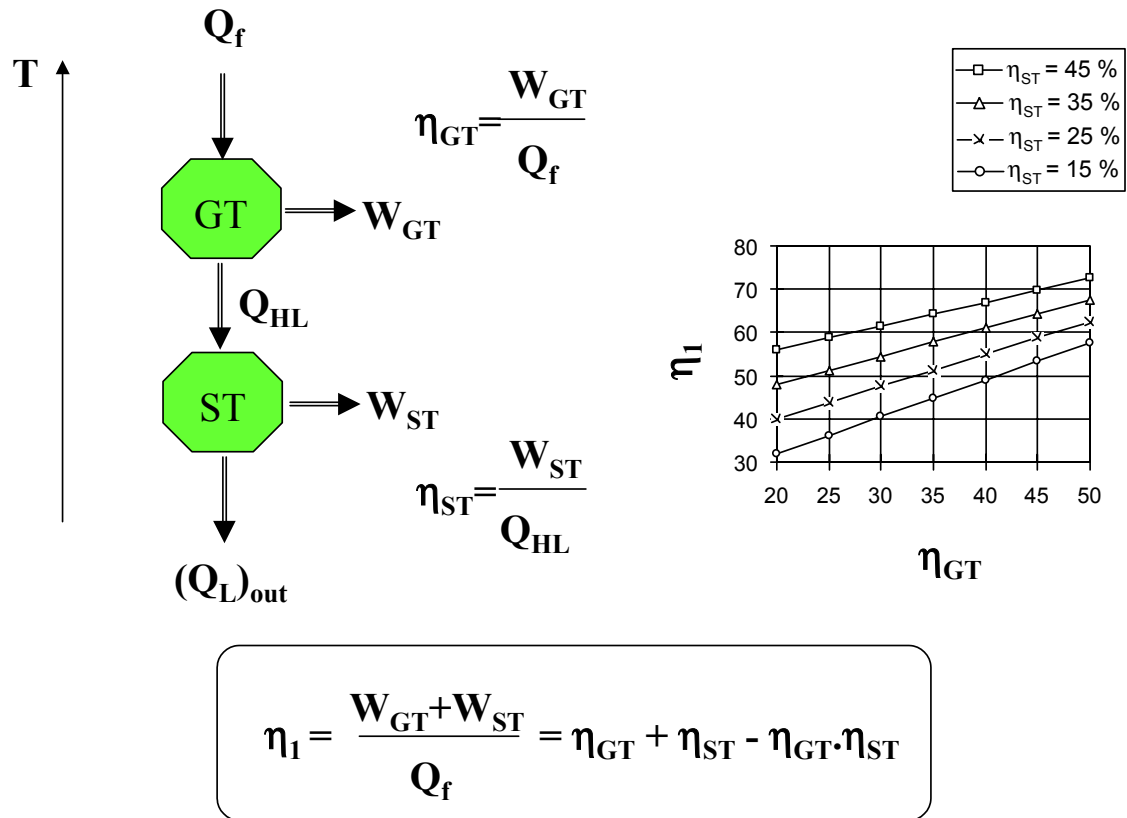


Figure-3: Series combined cycle without heat loss. η_1 -efficiency.

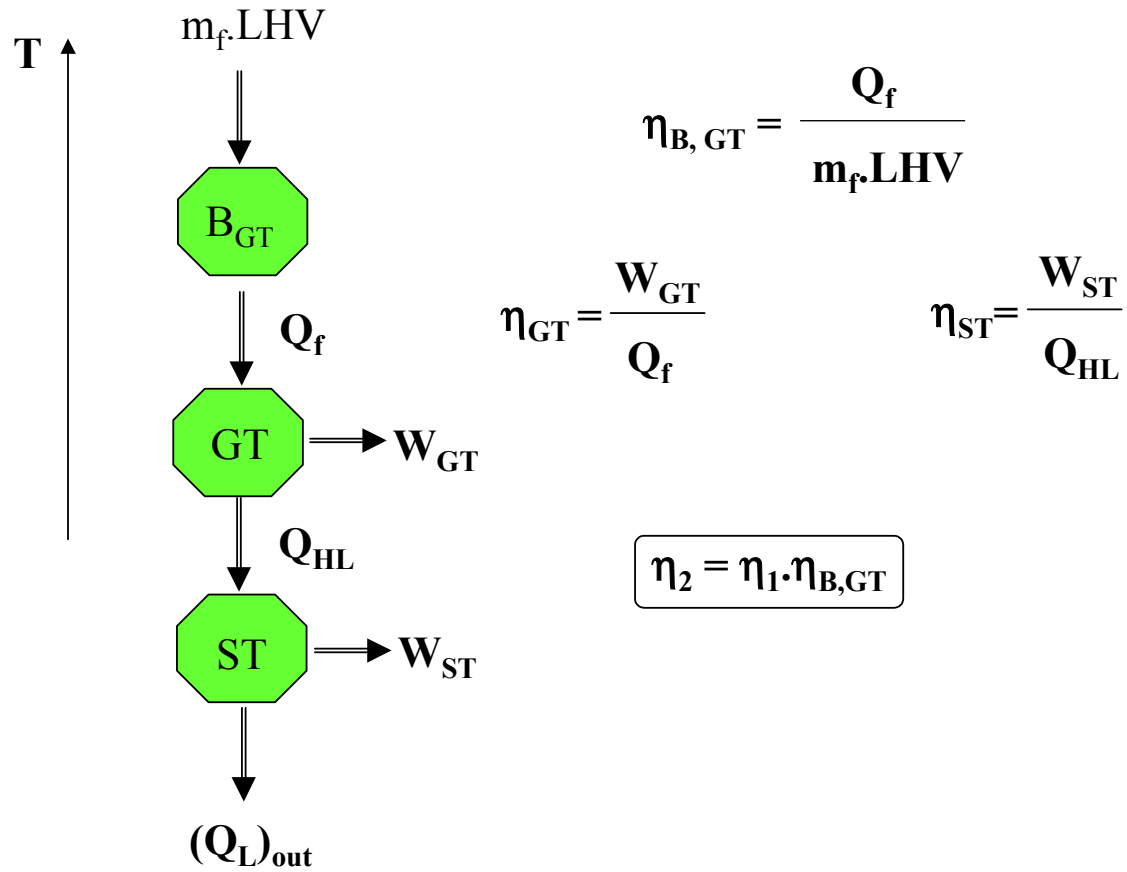


Figure-4: Series combined cycle without heat loss. η_2 -efficiency.

As we can see, the efficiency from this combined cycle is higher than any of the component cycles efficiency.

A practical application from this combined cycle scheme is the Field cycle. A Rankine reheated cycle can also be seen as a series coupling without heat loss plus a parallel coupling.

To further benefit from the exergy use from residual heats, series combined cycles with more than two cycles have been proposed. In Figure-5 we present the schematic from a series cycle combination with 3 component cycles, and an evaluation from its η_1 -value, given by Eq.41. An example from this cycle scheme would be three Rankine series cycles (Mercury – Water - Ammonia), or a closed Brayton cycle (Helium) with a bottoming steam turbine cycle.

$$\eta_1 = \eta_{GT} + \eta_I + \eta_L - \eta_{GT} \cdot \eta_I - \eta_{GT} \cdot \eta_L - \eta_I \cdot \eta_L \quad (\text{Eq.41})$$

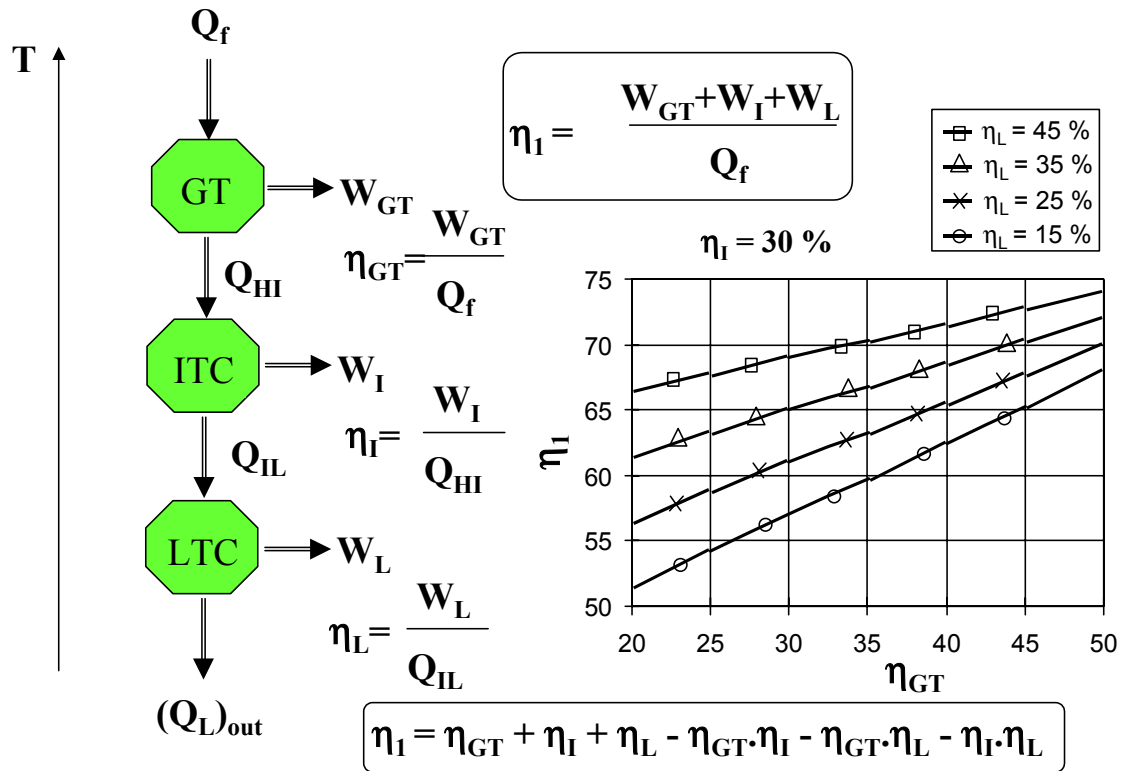


Fig.5: Three series combined cycle without heat losses. η_1 -value.

4.2. Series cycles with heat loss between cycles

In the case of external thermal coupling between series cycles, it is usually not possible to recover all the residual heat from the topping cycle as heat input for the bottoming cycle. In the most common combined cycle schemes (GT+ST), this heat transfer process is implemented by a heat recovery steam generator, and because of pinch-point and stack limitations a significant amount of the topping cycle residual heat is finally rejected to the atmosphere (typical HRSG efficiencies are around 55 %).

In Figure-6 we can find a schematic and η_1 -value evaluation for a series combined cycle with heat loss (Q_{UN}) between cycles. The η_1 -value from this combined cycle is the one given by Eq.42.

$$\eta_1 = \eta_{GT} + \eta_{ST} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST} \cdot \eta_{HRSG} \quad (\text{Eq.42})$$

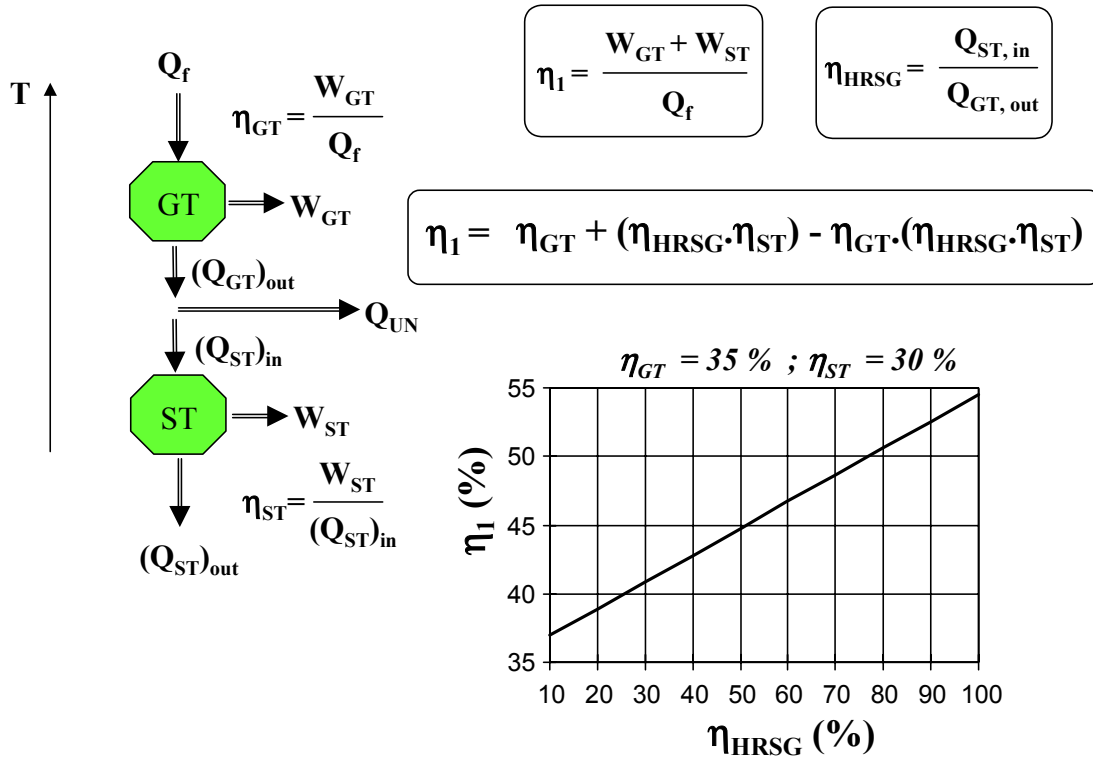


Figure-6: Series combined cycle with heat loss. η_1 -value.

4.3. Parallel cycles

The parallel heating of elemental cycles is a combination element that we find in several combined cycle power plants: Presurized Fluidized Bed Combustion (PFBC), STIG or Cheng cycle, conventional GT+ST series combined cycle with supplementary firing prior to HRSG, bottoming multiple pressure steam cycles, Rankine reheated cycle, ...

From the point of view of solar thermal power plants, integration schemes introducing solar energy in HRSG or in steam turbine are elements of parallel combination.

The combined cycle resulting from a parallel coupling of two power cycles, has a η_1 -value that lies somewhere in between of the η_1 -values from elementary cycles depending on the fractions of heat supply to each of the elementary cycles. Therefore, by itself, this combined cycle does not present any thermodynamic advantage with respect to the option of using all the thermal input in the elementary plant with higher efficiency. However, on the one hand parallel combination elements are usually introduced in combined cycle power plants that also have a series coupling in order to use residual heats from some of the component cycles. On the other hand parallel combination sometimes presents important technological advantages, as in the case of PFBC or some solar integration schemes. In the case of combined GT+ST cycles, the parallel coupling in bottoming cycle, although it reduces bottoming cycle efficiency, it improves HRSG efficiency in such a way that the effect on overall cycle η_1 -value is positive.

In Figure-7 we may find a schematic from this combined cycle, as well as its η_1 -value and associated nomenclature. The η_1 -value is also presented in Eq.43.

$$\eta_1 = \eta_{GT} - (\eta_{GT} - \eta_{ST}) \cdot v_{ST} = \eta_{ST} + (\eta_{GT} - \eta_{ST}) \cdot v_{GT} \quad (\text{Eq.43})$$

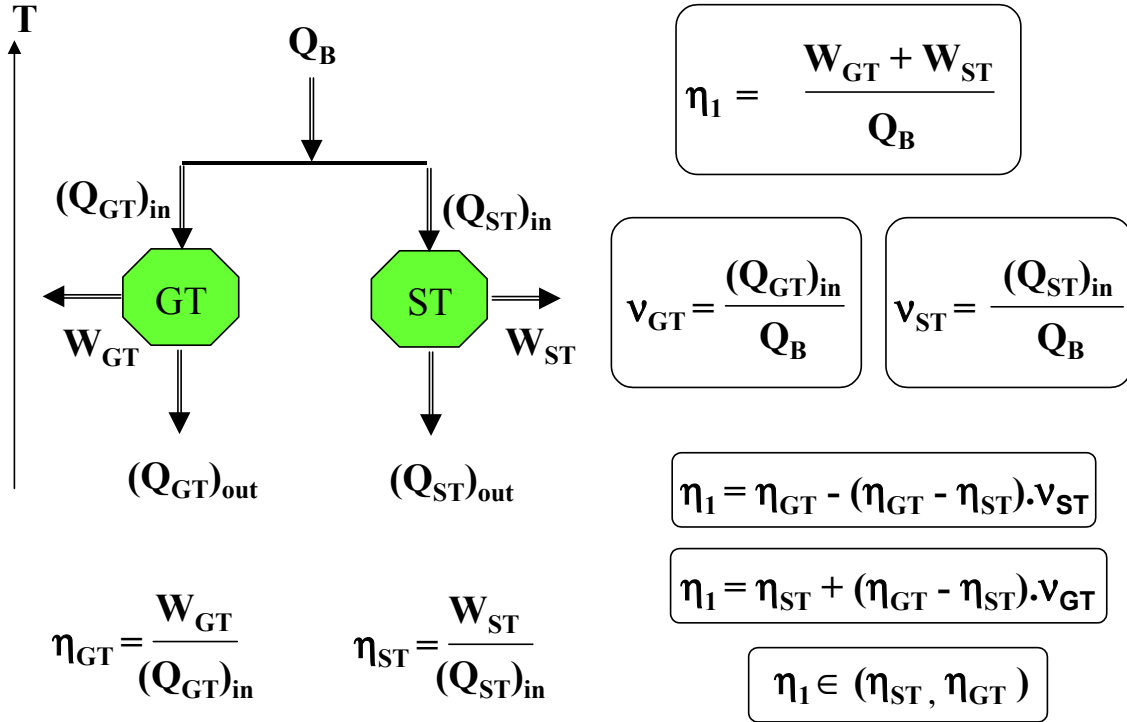


Figure-7: Parallel combined cycle. η_1 -value.

4.4. Series/parallel cycles without heat loss between cycles

In Figure-8 we show the schematic of the series/parallel combined cycle without heat loss between cycles, as well as an evaluation of its η_1 -value and ζ_4 -value. In Eq. 44, 45, and 46 we present the η_1 -value, η_2 -value, and ζ_4 -value from this combined cycle as function of component efficiencies and solar fraction (ζ_1 -value).

$$\eta_1 = \eta_{GT} + \eta_{ST} - \eta_{GT} \cdot \eta_{ST} - \zeta_1 \cdot \eta_{GT} \cdot (1 - \eta_{ST}) \quad (\text{Eq.44})$$

$$\eta_2 = \frac{\eta_1}{\zeta_1 + \frac{1 - \zeta_1}{\eta_{BGT}}} \quad (\text{Eq.45})$$

$$\zeta_4 = \eta_{GT} + \eta_{ST} - \eta_{GT} \cdot \eta_{ST} + \eta_{ST} \cdot \frac{\zeta_1}{1 - \zeta_1} \quad (\text{Eq.46})$$

A Rankine reheated cycle can be seen as a series/parallel combined cycle without heat losses between cycles. This helps to understand the best way to integrate solar energy in this cycle as we will discuss latter on.

This cycle is also, except by the heat loss between topping and bottoming cycles, an appropriate comparative scheme for many proposed solar integrations based on a series GT+ST combined cycle, where solar heat is supplied in the bottoming cycle. It would be the appropriate scheme in the case of implementing a closed Brayton as topping cycle, but when using an open Brayton cycle the appropriate comparative scheme has to include heat losses between cycles (see next point). However, as well for closed or open Brayton cycles the qualitative conclusions are the same.

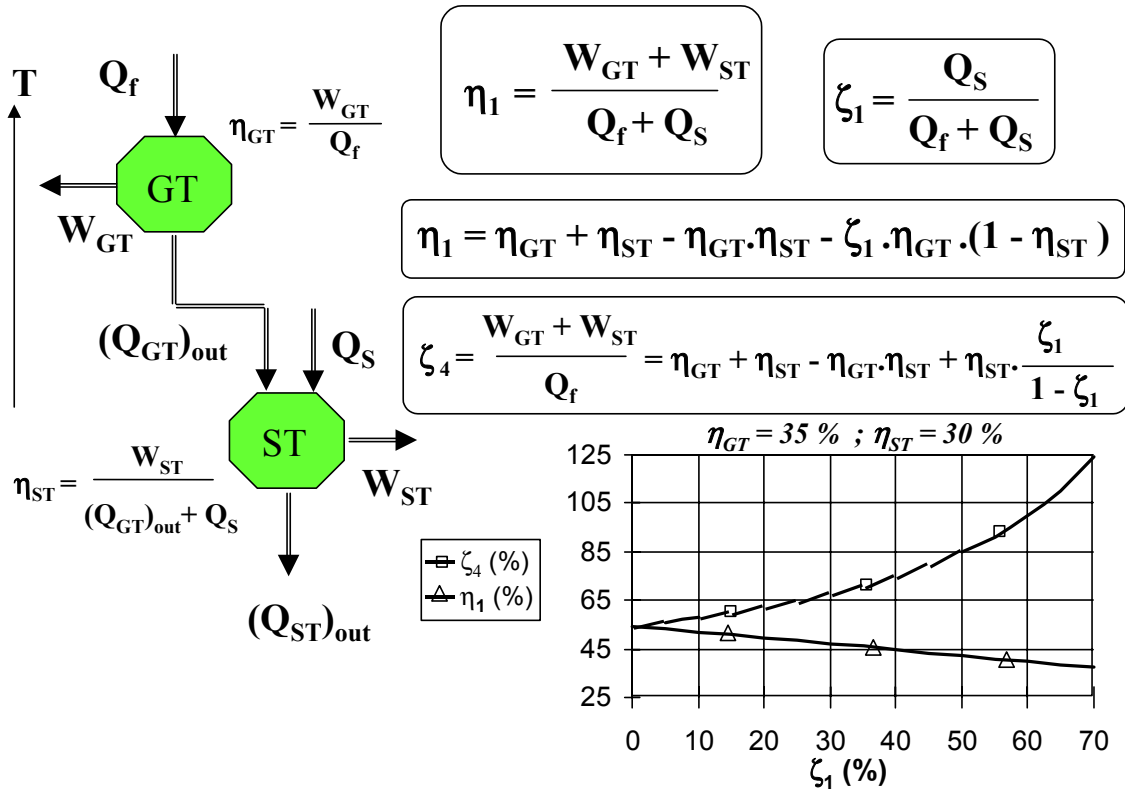


Figure-8: Series/parallel combined cycle without heat loss. η_1 and ζ_4 values.

The introduction of solar energy in the bottoming cycle introduces a parallel component in the combined cycle configuration (in this sense equivalent to a supplementary firing), and therefore, cycle efficiency is in principle reduced from the one of the original series combined cycle (unless component efficiencies

experiment a high improvement). This is shown by the η_1 -value representation as function of solar fraction in Figure-8.

However, a good solar integration can conduce to a better use of the fuel energy supply by an improvement in component efficiencies, reaching higher η_1 -values than the ones shown in Figure-8. The component efficiency that solar integration can improve in this combined cycle is η_{ST} , while in the case of the series/parallel with heat loss, both η_{ST} and η_{HRSG} may be improved by solar integration.

In Figure-8 we also show the evolution of the ζ_4 -value as function of solar fraction. The ζ_4 -value does not give by itself any useful information about the quality of solar integration. Just by increasing solar fraction it is possible to produce ζ_4 -values higher than the reference fuel only power plant, even if in the integration process η_{GT} , η_{ST} and η_{HRSG} are reduced, increasing therefore the irreversibilities in the use of fuel energy. In this, case when measuring plant performance by the ζ_4 -value, we would be hiding a bad use of fossil fuels.

4.5. Series/parallel cycles with heat loss between cycles

The most common combined cycle and therefore the most common solar integration proposals, use a series GT+ST combined cycle with heat loss in the HRSG.

In Figure-9 we show the schematics and η_1 -value evaluation in the case that solar heat input is mixed with gas turbine heat output prior to heat losses between cycles, that is, before HRSG. This is the integration scheme proposed using the PHOEBUS air central receiver technology by (Lippke F., et al, 1996), as well as one of the proposed schemes for the ISCCS in Egypt (Rheinländer J., 2000). For this situation, solar energy input is affected by HRSG efficiency, in contrast with the usual situation where only gas turbine heat output is affected by HRSG efficiency.

In Figure-10 we show the schematics and η_1 -value evaluation in the case that solar heat input is introduced in a heat supply region from the bottoming cycle such that it is converted at bottoming cycle efficiency (η_{ST}), as it would be the case if bottoming cycle would be a Rankine, single pressure, non reheated cycle. Therefore, solar heat input does not undergo losses due to the HRSG. This is the case in some hybrid plants where the solar heat is integrated in the HRSG by supplying thermal energy straight to the water-steam from the bottoming cycle.

In Figure-11 we show the schematics and η_1 -value evaluation in the case that solar heat input is introduced in a lower exergy zone of the bottoming cycle. This is the case of hybrid plants with solar integration in the intermediate or low pressure parts of the steam turbine, that is, after HRSG.

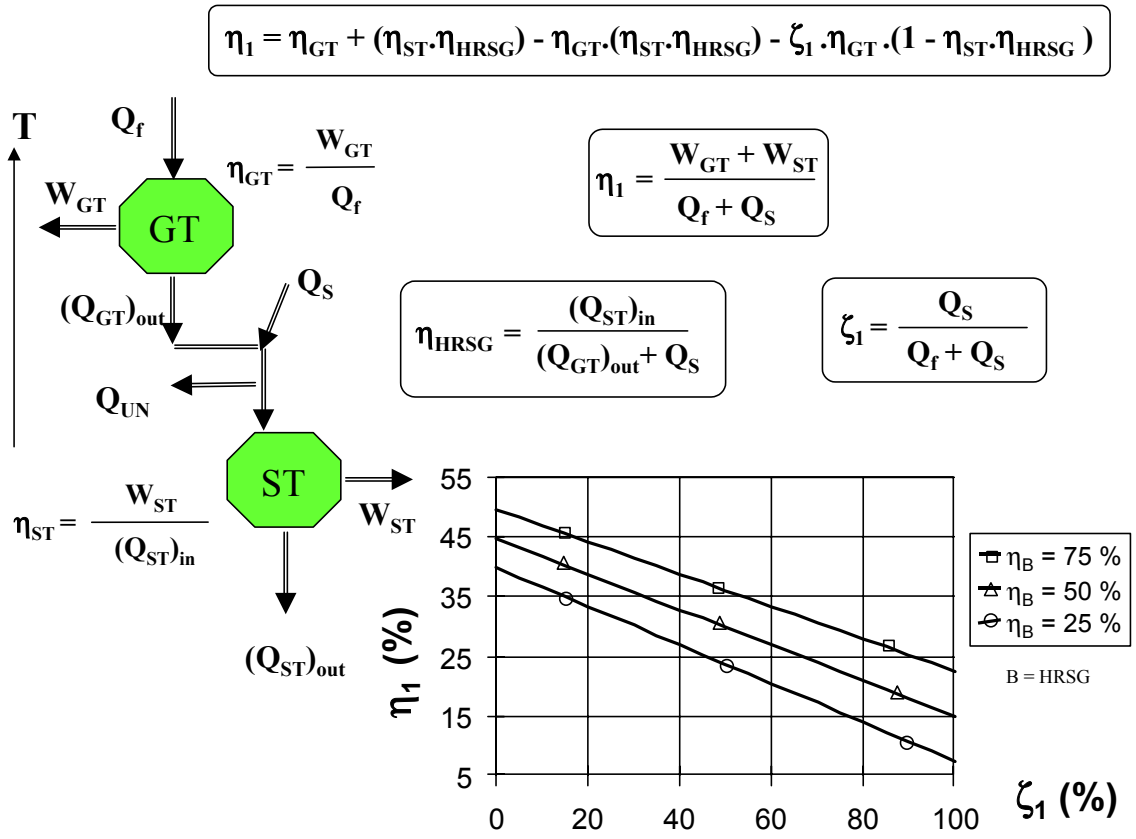


Figure-9: Series/parallel combined cycle with heat loss and solar integration before HRSG. η_1 -value.

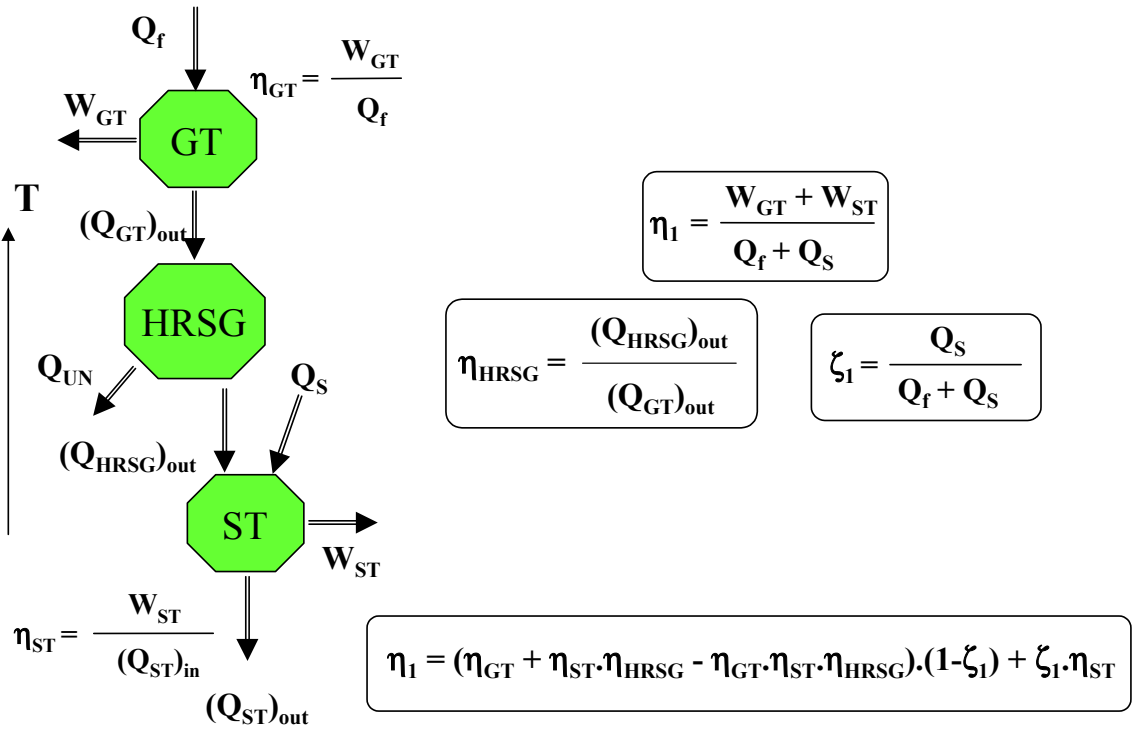


Figure-10: Series/parallel combined cycle with solar integration in HRSG. η_1 -value.

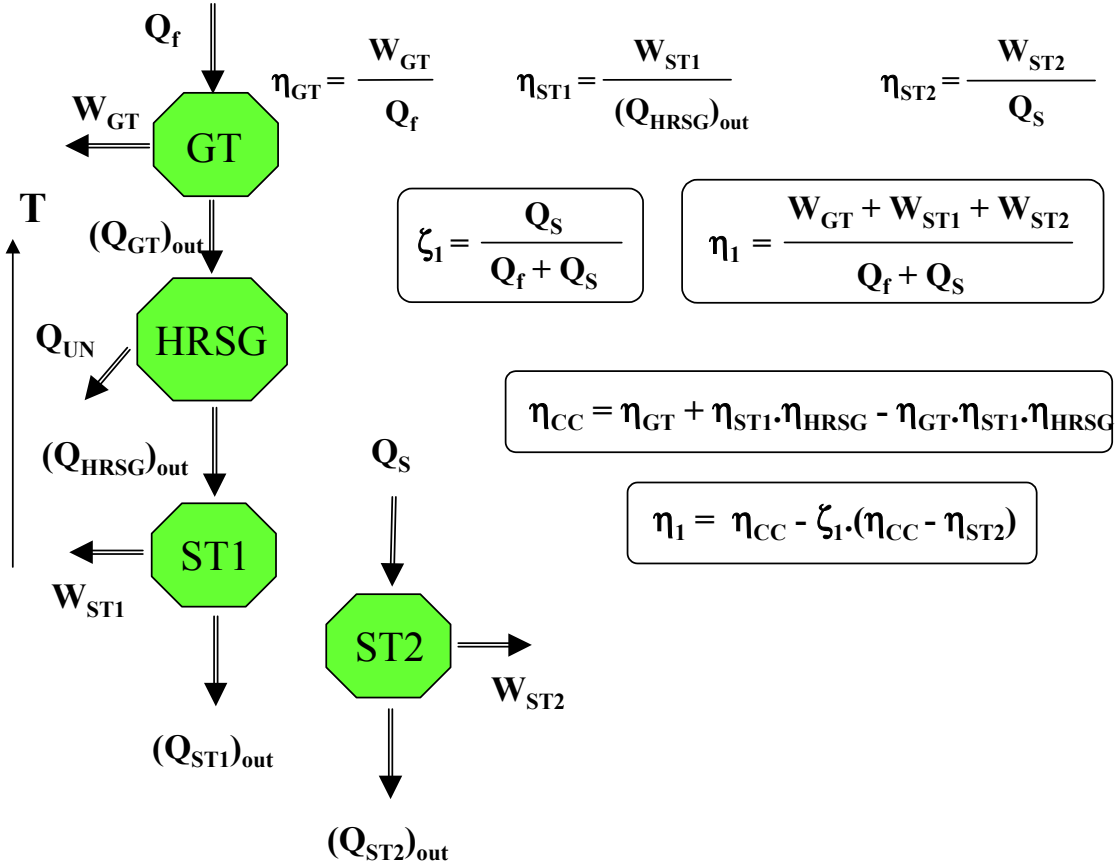


Figure-11: Series/parallel combined cycle with solar integration in steam turbine. η_1 -value.

In Eq.47 , Eq.48, and Eq.49, we present the η_1 -values for solar integration on a combines series/parallel cycle with heat loss, before the HRSG (Eq.47), in the HRSG with a single pressure no reheated bottoming cycle (Eq.48), and in the steam turbine or on lower pressure zones of bottoming cycle (Eq.49). The η_2 -value for integration in the HRSG is also given by Eq.45, but with the η_1 -value from Eq.48.

$$\eta_1 = \eta_{GT} + \eta_{ST} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST} \cdot \eta_{HRSG} - \zeta_1 \cdot \eta_{GT} \cdot (1 - \eta_{ST} \cdot \eta_{HRSG}) \quad (\text{Eq.47})$$

$$\eta_1 = (\eta_{GT} + \eta_{ST} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST} \cdot \eta_{HRSG}) \cdot (1 - \zeta_1) + \zeta_1 \cdot \eta_{ST} \quad (\text{Eq.48})$$

$$\eta_1 = \eta_{CC} - \zeta_1 \cdot (\eta_{CC} - \eta_{ST2}) \quad (\text{Eq.49})$$

$$\text{With : } \eta_{CC} = \eta_{GT} + \eta_{ST1} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST1} \cdot \eta_{HRSG} \quad (\text{Eq.50})$$

We should note that from these three ways to integrate solar heat into the bottoming cycle of a GT+ST series combined power plant with heat loss between cycles, that is, integration before HRSG, in HRSG (single pressure no reheated cycle) or after HRSG, the most appropriate from a thermodynamic point of view is integration into the HRSG. Indeed, when integrated before the HRSG, a part $((1-\eta_{\text{HRSG}}) \cdot Q_s)$ of the solar thermal input is lost by the HRSG. When integrated after HRSG solar energy is converted with an efficiency lower than the one of the bottoming cycle ($\eta_{\text{ST1}} > \eta_{\text{ST2}}$), which usually consists of a parallel combined cycle (multiple pressure steam cycle). Moreover, when integrating solar energy before or after HRSG, the efficiency from the HRSG is the same as for an only fuel equivalent combined cycle power plant, but when integrating solar energy in HRSG normally we get an increase in η_{HRSG} because of two reasons:

- Reduction in pinch point limitations if solar energy is used to evaporate water.
- Irreversibility reduction in superheater and preheater HRSG regions because of the increase in the ratio of heat capacity rates associated to the increase in steam flow.

When integrating the solar energy in a bottoming multiple pressure steam cycle somewhere in the HRSG, because the bottoming cycle consists itself on a parallel combined cycle, its efficiency (η_{ST}) will be an intermediate value between the different pressure cycles. If we call η_{HP} and η_{LP} the efficiencies from the high pressure and low pressure zones in the bottoming cycle, we will have $\eta_{\text{HP}} > \eta_{\text{ST}} > \eta_{\text{LP}}$. Therefore, depending on where in the cycle we introduce Q_s , it will be converted with one or another efficiency. The most favourable case is to introduce Q_s in the high pressure zone, in which case it will be converted with an efficiency even higher than bottoming cycle efficiency.

5. Comments about the integration evaluation parameters

Here we are going to give additional comments about different integration evaluation parameters listed in point 3.2, where some comments were already done.

5.1. Main evaluation parameters used up till now

First of all let's have a look to which parameters have been used up till now in the available literature about integration of solar energy on power plants.

In (Stein W., 1992) we find one of the first references analysing integration of solar thermal energy from parabolic trough technology into combined cycle power plants. An interesting analysis is developed comparing different integration alternatives. The main parameter used to compare different alternatives is the inverse of the ζ_4 -value, which is called the 'net heat rate'. As

already discussed, the ζ_4 -value is not an appropriate integration evaluation parameter, an moreover when trying to compare integration schemes with different solar fractions it leads to wrong and misleading arguments. Other integration evaluation parameters used in this reference to characterize some of the integration results are the solar capacity factor (ζ_7 -value), and what pretend to be the solar conversion efficiencies given by the η_7 -value and the η_8 -value. However, these two last values are not properly evaluated, and in fact the ζ_{10} -value and ζ_{11} -value are given. Both design values from these parameters for comparison between different integration schemes, and annual average values for the design cases analysed with more attention are presented.

In (Price H.W., 1997) we find a report written with a similar aim as this work, that is, to reach some common basis to measure the performance of solar power systems that will ease the comparison between them. However, (Price H.W., 1997) is more focused on reporting the performance of existing large parabolic trough power plants (extended response time to transients and constantly varying solar input due to one axis tracking), that is parameters to characterise the ongoing operation of a given power plant. Therefore it is recommended to use average annual values as appropriate evaluation parameters. On the other hand we are interested on characterise performance of non existing power plants on a design stage in order to find out the optimal designs. Anyway, we can find a reflection of the evaluation parameters proposed in (Price H.W., 1997) on some of the ones listed in point 3.2.

The main integration evaluation parameters proposed in (Price H.W., 1997) require the break down of the electricity generation (W) into its solar (W_s) and fuel (W_f) components. The way to perform this break down is not clear at all in (Price H.W., 1997) except for the trivial case that both Q_s and Q_f are converted with the same efficiency. On the one hand, fuel thermal input (Q_f) and chemical input ($m_f \cdot \text{LHV}$) are not clearly distinguished, and therefore boiler efficiencies are not clearly incorporated. Assuming that the Q_s and Q_f considered in (Price H.W., 1997) are given by $Q_s = I_b \cdot A \cdot \eta_{\text{solar field}}$ and $Q_f = m_f \cdot \text{LHV} \cdot \eta_B$, some indetermination remains in the evaluation of W_f and W_s when Q_f and Q_s are converted at different efficiencies ($\eta_{1,\text{sun}}$, $\eta_{1,\text{fuel}}$), because it is proposed that this efficiencies to be evaluated (Eq.51) as the ones from the power block producing the same W as the hybrid plant, but operating only with one thermal source input (solar or fuel). Therefore, the interactions between Q_s and Q_f with respect to their thermal efficiencies are neglected. For example, in SEGS VI-VII (Rankine reheated in solar, hybrid and fuel operation, but with higher turbine inlet temperature in hybrid or fuel operation), in hybrid operation Q_s is converted at overall plant efficiency (or even higher if reheating would be done solely by Q_f), which is much higher than solar only plant efficiency.

$$W_s = W \cdot \frac{Q_s \cdot \eta_{1,\text{sun}}(W)}{Q_s \cdot \eta_{1,\text{sun}}(W) + Q_f \cdot \eta_{1,\text{fuel}}(W)} \quad (\text{Eq.51})$$

The main evaluation parameters proposed by (Price H.W., 1997) are:

- Plant annual total capacity factor , given by Eq.52.

- Plant annual solar capacity factor, given by Eq.53. This parameter becomes the ζ_7 -value if we use design power output instead than contractual rated power output.
- Annual solar fraction, that is, the ζ_3 -value.
- Net system fossil efficiency, that is, the η_6 -value.
- Net system solar efficiency, that is, the η_8 -value.

$$CF_{annual} = \frac{W}{\dot{W}_R \cdot 8760} \quad (\text{Eq.52})$$

$$CF_{solar_{annual}} = \frac{W_s}{\dot{W}_R \cdot 8760} \quad (\text{Eq.53})$$

The integration evaluation parameters used in (Rheinländer J., et al, 1994), for integration schemes into low temperature zone of combined GT+ST cycles with a bottoming two pressure steam cycle, are average solar fraction values based in Eq.19. The solar fraction used is the ζ_3 -value. However, as an approximate approach to determine W_s , they subtract from the total hybrid power plant output the output from the plant working in only fuel operation mode at part load with $Q_f = Q - Q_s$. In this paper, reference is also made to a total conversion efficiency, which is not defined, but could probably be the η_4 -value.

The SOLGAS project (Blanco M.J., et al, 1994), (Blanco M., 1996), a combined cycle (ST+GT) plus cogeneration power plant was analysed. Solar energy was used to assist the water evaporation in the HRSG using central receiver technology, but this integration scheme could be perfectly implemented with parabolic trough technology. The quality of the integration scheme was evaluated by its ζ_4 -value in design conditions, which as we have already commented gives no useful information at all about how solar energy has been integrated. In out of design conditions (no sun), fuel energy is supplied by burning in the HRSG, introducing a parallel combination element that lowers fuel conversion efficiency. Besides, in this project it is assumed that Q_s is converted at bottoming cycle conversion efficiency, which would be true only in the case of implementing a single pressure no reheated bottoming cycle.

The Colon-Solar Project (Blanco M., et al, 1996), (Silva M., Blanco M., Ruiz V., 1998) follows a similar scheme from that of the SOLGAS project but is dedicated to the analysis of GT+ST combined cycles with integration in the HRSG to evaporate water. Integration evaluation is again done with the ζ_4 -value in design (solar) conditions. When no solar energy is available, fuel is burned in HRSG.

In the STEM project (UNION FENOSA et al., 1997), the solar energy was integrated on a bottoming dual pressure no reheated steam cycle, and the

integration evaluation parameters used were ζ_4 and ζ_{10} . See the next point for specific comments about this integration evaluation.

To evaluate the integration for the ISCCS in Egypt (Rheinländer J., 2000) the parameters η_8 and ζ_3 are used, as well as ζ_5 (LEC). The η_8 -value can be used when solar heat is converted at known efficiency. In this reference they assume that Q_s is converted at η_{ST} . This happens when mixing solar input with gas turbine exhaust gases (PHOEBUS option), or when integrating solar energy from a parabolic trough solar field in the HRSG with a bottoming cycle consisting of a single pressure non reheated steam cycle. When a multiple pressure or reheated bottoming cycle is used, generally Q_s it is not converted at η_{ST} .

In this study, the performance parameters evaluation was done by computation of hourly energy balances for one typical day of each month of the year, including 28 non-solar days as well as 20 days of scheduled outage for maintenance and repair actions, and 11 days of not-foreseen outages due to technical failures, per year.

In (Kribus A., et al, 1998) an integration scheme for combined cycles using central volumetric receiver technology is analysed. The integration evaluation parameters used are the annual average η_8 -value, the ζ_7 -value (solar capacity factor), and the ζ_5 -value (LEC).

In (Allani Y., 1992) an integration scheme for combined cycle using parabolic trough technology is preliminarily analysed. The integration evaluation parameter used is the ζ_4 -value (wrongly called cycle efficiency), although results for the η_1 -value are also presented. However, no interpretation at all is given about the different results showed by the ζ_4 and η_1 values. Indeed, while the ζ_4 -value shows (as should have been expected) a much better behaviour of the solar plant compared with the only fuel plant, with an increasing difference as solar fraction is augmented, the η_1 -value shows a better behaviour of the only fuel plant.

5.2. Specific comments about some evaluation parameters

Sometimes, the use of the wrong integration evaluation parameters, together with the misinterpretation of their meaning, and of the thermodynamic concept of how is solar energy being introduced, can conduce to surprising results. As an example, we will analyse the conclusions from the STEM project (UNION FENOSA et al, 1997), where integration was analysed in terms of ζ_4 and ζ_{10} -values. In this project, two main integration schemes were analysed for a GT+ST combined cycle with a two pressure no reheated bottoming cycle. In both integration schemes solar energy was used in the bottoming cycle. In one of them it was used to evaporate high pressure water, and in the other to generate intermediate pressure steam to be injected in the steam turbine. Solar field was working at higher average temperature in the second case. The

integration evaluation parameters used were the ζ_4 and ζ_{10} -values. In the report conclusions we can read:

‘ The most remarkable point is, injection of solar steam into the HRSG yields the highest electrical efficiency for the collected solar energy ...The highest gain for the solar field is achieved by operating the field at lower temperatures, that means at lower exergy of the collected solar energy. This result is very surprising and has to be considered by a future research and development program ...’

Besides, the integration scheme with steam injection in HRSG is the one more detrimental for only fuel operation, because the imbalance in HRSG when solar energy is not available forces to spray cold water before the superheater, with the consequent irreversibility. So it looks as if the best integration option (higher ζ_{10} -value), is the one that more affects cycle efficiency in only fuel operation.

To better understand this case, in the Table-1 we present results from the analysed integration schemes, complemented with additional evaluation parameters not reported in the (UNION FENOSA et al, 1997) study : η_1 -value, $(\eta_1)_{av}$ -value, ζ_1 -value.

$(\eta_1)_{av}$ is the average η_1 -value according to Eq.1 with $t_f = 4200$ h/year and $t_d = 1800$ h/year. The different cases considered are as follows:

- C_o : Reference fuel only combined cycle.
- $C_1(m_s=0)$: Solar integration in intermediate pressure turbine. Only fuel operation.
- $C_1(m_s=15)$: Solar integration in intermediate pressure turbine. Design point operation with maximum solar fraction.
- $C_2(m_s=0)$: Solar integration in HRSG. Solar preheating and evaporation. Only fuel operation.
- $C_2(m_s=15)$: Solar integration in HRSG. Solar preheating and evaporation. Design point operation with maximum solar fraction.
- $C_3(m_s=0)$: Solar integration in HRSG. Solar evaporation. Only fuel operation.
- $C_3(m_s=15)$: Solar integration in HRSG. Solar evaporation. Design point operation with maximum solar fraction.

	η_1 (%)	$(\eta_1)_{av}$ (%)	ζ_1 (%)	ζ_4 (%)	ζ_{10} (%)
C_o	54,4	-	-	54,4	-
$C_1(m_s=0)$	54,5		-	54,5	-
$C_1(m_s=15)$	49,7	53,06	19,44	61,7	29,96
$C_2(m_s=0)$	54,04		-	54,04	-
$C_2(m_s=15)$	50,55	52,99	17,42	61,21	34,01
$C_3(m_s=0)$	53,40		-	53,40	-
$C_3(m_s=15)$	53,35	53,39	10,68	59,72	52,89

Table-1: Integration evaluation parameters for STEM project.

First of all, we should indicate that the different analysed options have different solar fractions (ζ_1 -value), and as shown in point 4, this makes it impossible to have a straight comparison between some integration parameters as pretended in the STEM project. For example, ζ_4 increases with ζ_1 in such a way that $\zeta_4 \rightarrow \infty$ when $\zeta_1 \rightarrow 1$. The ζ_4 -value, by itself gives no useful information about the integration quality.

On the other hand, because of integrating solar energy in the bottoming cycle, as we saw in point 4, the η_1 -value tends to decrease with the ζ_1 -value. In all the integration schemes proposed by the STEM project, the η_1 -value in solar operation is lower than the corresponding η_1 -value with only fuel operation. And this in spite of the fact that when introducing the solar energy some plant components (HRSG, ST) may be working closer to their design conditions.

In the STEM report the ζ_{10} -value is used as a way to measure how good the solar energy input is used. In this integration scheme where Q_f is constant with or without solar input, it can be seen that the ζ_{10} -value is related to solar fraction and η_1 -values by Eq.54. Therefore we can see in Eq.54 how with a very bad solar integration, that is, very low $\eta_1(Q_s=0)$, and the same ζ_1 and $\eta_1(Q_{s,d})$, we get a higher ζ_{10} -value than with a good solar integration ($\eta_1(Q_s=0) \approx \eta_1(Q_{s,d})$).

$$\zeta_{10} = \frac{\eta_1(\dot{Q}_{s,d}) - \eta_1(\dot{Q}_s = 0) \cdot (1 - \zeta_1)}{\zeta_1} \quad (\text{Eq.54})$$

If we consider an integration in which η_{HRSG} and η_{ST} are not modified when operating with or without sun (which could be a good design objective in order not to penalise operation without sun), and assuming that the introduced solar energy is converted at η_{ST} , from Eq.48 we can get Eq.55:

$$\eta_1(Q_{s,d}) = \eta_1(Q_s = 0) - \zeta_1 \cdot (\eta_1(Q_s = 0) - \eta_{\text{ST}}) \quad (\text{Eq.55})$$

Which when introduced in Eq.54 yields $\zeta_{10} = \eta_{\text{ST}}$, for all ζ_1 -values.

But in general, a good integration in the HRSG can improve η_{HRSG} and η_{ST} when working in design conditions (although η_{HRSG} in only fuel operation usually is lower than for the reference only fuel plant). In a hypothetical ideal solar integration giving $\eta_1(Q_{s,d}) = \eta_1(Q_s=0)$, from Eq.54 we would get $\zeta_{10} = \eta_1(Q_s=0)$ for all ζ_1 -values, which would be an upper bound for ζ_{10} -value.

But in general, an acceptable integration scheme will have $\partial \eta_{\text{HRSG}} / \partial \zeta_1 > 0$ because of reducing the pinch point limitation, and although $\partial \eta_{\text{ST}} / \partial \zeta_1$ can in principle be positive or negative depending on the steam turbine design point, its variations will be small, and therefore we will have $\partial \eta_{\text{CC}} / \partial \zeta_1 > 0$. The ζ_{10} -value, will be given by Eq.56, 57, 58 and 59.

$$\zeta_{10} = \frac{\eta_{CC1} - \zeta_1 \cdot \eta_{CC2} - \eta_{CC0} \cdot (1 - \zeta_1)}{\zeta_1} \quad \text{Eq.56}$$

$$\eta_{CC0} = \eta_{GT0} + \eta_{ST0} \cdot \eta_{HRSG0} - \eta_{GT0} \cdot \eta_{ST0} \cdot \eta_{HRSG0} \quad \text{Eq.57}$$

$$\eta_{CC1} = \eta_{GT} + \eta_{ST} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST} \cdot \eta_{HRSG} \quad \text{Eq.58}$$

$$\eta_{CC2} = \eta_{GT} + \eta_{ST} \cdot \eta_{HRSG} - \eta_{GT} \cdot \eta_{ST} \cdot \eta_{HRSG} - \eta_{ST} \quad \text{Eq.59}$$

The limit of ζ_{10} given by Eq.56 when ζ_1 tends towards zero or unity can be evaluated to give the following results (Eq.60-61):

$$\lim_{\zeta_1 \rightarrow 1} \zeta_{10} = \eta_{ST} \quad \text{Eq.60}$$

$$\lim_{\zeta_1 \rightarrow 0} \zeta_{10} = \eta_{ST0} + \left(\frac{\partial \eta_{CC1}}{\partial \zeta_1} \right)_{\zeta_1=0} \quad \text{Eq.61}$$

And therefore we will have $\zeta_{10}(\zeta_1 = 0) > \zeta_{10}(\zeta_1=1)$, and assuming a monoton behaviour of ζ_{10} (ζ_1), we will have a decreasing ζ_{10} -value with ζ_1 , in such a way that it is not possible to compare integration schemes with different ζ_1 -value in terms of their ζ_{10} -value as done in (UNION FENOSA et al, 1997).

Another curious aspect from the (UNION FENOSA et al, 1997) project, which explanation was left for future research, was the fact that from the point of view of solar conversion (ζ_{10} -value), it was more favourable to introduce solar energy at lower temperature in the power block (integration in HRSG), than to introduce it at higher temperature in the intermediate pressure turbine (integration in IPT)⁵.

Even if cycle exergy supply would be higher with the IPT integration scheme, the explanation from this apparent paradox lies in the fact that in both integration schemes the solar heat is supplied to a different thermodynamic cycle.

In the case of integrating solar energy in HRSG for water evaporation (300 °C, 75 bar), the solar thermal energy supply undergoes afterwards a further superheating in HRSG with flue gases until 540 °C, in such a way that solar

⁵ On the one hand, we have to say that the exergy interpretation given in (UNION FENOSA et al, 1997) is not adequate, since it could be that cycle exergy supply would be higher in the HRSG integration than in the IPT integration as long as average heat supply temperature would be higher in the first case.

heat input is finally converted at an efficiency that is even higher than overall bottoming cycle efficiency (parallel combined cycle). Indeed, solar input is converted at the efficiency from a Rankine cycle with 540 °C and 75 bar turbine inlet conditions.

On the other hand, when solar energy is integrated in IPT (400 °C, 35 bar), because solar input does not undergo any further superheating in this integration scheme, solar heat will be converted at an efficiency equal to the one of a Rankine cycle with 400 °C and 35 bar turbine inlet conditions, which is much lower than the efficiency at which solar heat is being converted in the HRSG integration scheme.

The important conclusion from this analysis is that independently of how much exergy the solar field introduces in the power block, the solar thermal energy is converted into work with the η_1 -value of the local thermodynamic cycle where it is introduced. Therefore, superheating the solar thermal input with fuel thermal energy a higher conversion efficiency for the solar energy input will be achieved, as far as component efficiencies are not spoiled. Therefore, taking into account that solar field efficiency diminishes with average solar heat supply temperature, keeping other factors unchanged, the solar energy will be more efficiently used in the low cycle temperature regions, because the same ζ_{10} -value will be achieved with a smaller collector field area.

However, care should be taken when implementing this conclusion to the way HRSG efficiency is affected by the solar input. Indeed, the very low temperature preheating zone has to be left to flue gas cooling in order to avoid HRSG or boiler efficiency fall. In general, the most favourable place to introduce solar input is in the evaporation zone, because the avoidance of the pinch point problem allows for increases in HRSG or boilers efficiency that are pinch-point controlled. However, in stack controlled boilers, the use of solar energy in the evaporation zone will be more neutral to boiler efficiency in design conditions. In both cases (pinch point controlled or stack controlled), the only fuel operation will be generally affected by an efficiency decrease associated to the compensation of heat exchanger zones imbalance.

To conclude with this case study, in the (UNION FENOSA et al, 1997) project, the improvement in ζ_{10} -value from the IPT integration scheme to the HRSG integration scheme with water preheating and evaporation is associated, on one hand to the reduction in the ζ_1 -value, and on the other hand to the increase in the thermodynamic cycle efficiency at which both solar inputs are converted into work. The improvement in ζ_{10} -value from the HRSG integration scheme with water preheating and evaporation, to the HRSG integration only for evaporation is associated on the one hand to the huge reduction in the ζ_1 -value, and on the other hand to the probable improvement in HRSG efficiency, and to the possibility that on the first case solar heat is input both to the high and low pressure bottoming cycles (parallel combined cycle). Indeed, in the case that solar energy input wants to be used for preheating purposes in a two pressures bottoming cycle as in the (UNION FENOSA et al, 1997) project, preheating should be restricted to the high temperature cycle, in which case the ζ_{10} -value would be similar to the case when only solar evaporation is performed, but solar field efficiency would increase.

6. Conclusions and proposal of integration evaluation parameters to be used

Herewith we are going to resume the main conclusions about the integration evaluation parameters and to recommend some guidelines to chose the appropriate set of parameters to characterise a given hybrid power plant.

Because of the important effect that solar energy integration can have on power plant only fuel operation mode, and because usually many power block operation hours will be only fuel or part load solar, there is the need to give values for integration evaluation parameters both in design conditions and in only fuel operation mode, in spite of the fact that this usually demands component part load performance evaluations. Simple average values like the one from Ec.1 can then be used to characterize overall power plant performance.

In general, to characterise a given solar hybrid power plant, several performance parameters should be reported, being not adequate to use only one parameter.

We should distinguish two situations where we are interested in measuring the quality of an hybrid power plant: The need to compare between themselves different hybrid power plants, and the need to optimise the solar integration on a given power plant scheme. For the first situation we need performance parameters with an absolute comparative basis. In general, performance parameters good to measure integration quality are relative evaluation parameters to the given power plant scheme, and are therefore not very useful to compare different power plants with each other. However, these integration evaluation parameters will be good to optimise solar energy introduction in a given power plant scheme.

In general it is not possible to compare integration schemes with different solar fractions, because many η -values and ζ -values have an important explicit dependence on solar fraction, in such a way that their relative numeric value does not reflect a good or bad integration scheme. Therefore, solar fraction (ζ_1 and ζ_2 values) should always be reported.

Some integration evaluation parameters should be avoided because of their misleading information about the goodness of a given integration scheme. This is for example the case of the ζ_4 -value so often used up till now. The risk to camouflage a bad integration scheme making a worse use of fuel energy than a conventional power plant with a high value of an integration evaluation parameter should be always avoided.

A convenient way to measure, in the field of energy efficiencies, the improvement due to solar input, that is, the quality of solar integration, is the one given by the ζ_{14} , ζ_{15} or ζ_{16} values. Therefore, these parameters are

recommended as integration evaluation parameters. However, these ζ_i -values do not allow to have an absolute comparison between different power plant schemes because of taking a different reference value.

In integration schemes, attention should be paid to the improvement in fuel energy use by solar integration. This could be done in the field of exergy by parameters like the ζ_6 -value and the ζ_{19} -value. Exergy based integration evaluation parameters, although are the ones thermodynamically more closely related to the ultimate expression of integration quality (that is, irreversibility), do not look a very appropriate choice because of the following reasons:

1. Irreversibility evaluations are less familiar to the engineering community than energy evaluations.
2. In some situations, the required irreversibility evaluations may be difficult to perform and interpret.
3. Supplying enough more familiar energy field evaluation parameters, all the information contained in the exergy field evaluation parameters is also supplied.

The η_1 and η_4 values, although do not give straight information about the integration quality, should always be reported because they contain a unique useful information about overall cycle and power plant energy conversion efficiencies.

Whenever it is possible to distinguish the electricity production originating from the fuel and solar energy inputs, the η_5 and η_6 values should be reported to characterise the way fuel energy is being used, while the η_7 and η_8 values should be reported to characterise the way solar energy input and investment are being used. These values, although by themselves say almost nothing about the integration quality, but give an absolute basis to compare different power plants from the point of view of how efficiently fuel and solar energy are being used.

Solar conversion efficiency is an important evaluation parameter. When it is not possible to distinguish the electricity production that originates from the solar input, the η_7 and η_8 values have to be replaced by ζ_8 and ζ_9 values or the ζ_{10} and ζ_{11} values. However, an agreement should be reached about the chosen reference values.

7. Integration on Rankine Cycles

The integration concepts developed up till now for combined cycles can also be applied to the analysis of solar energy integration in hybrid power plants working with a Rankine reheated cycle, the nowadays more extended power plant scheme. Indeed, this power plant can conceptually be viewed as a combined cycle with a series thermal coupling without energy loss, and a parallel thermal coupling. From this point of view the analysis of solar integration processes in Rankine or conventional combined cycles is completely equivalent.

It is important to note the important differences between integration of solar energy in hybrid Rankine cycles, or only solar power plants implementing Rankine cycles, as the ones evaluated in (Garcia-Casals X., 2000). When speaking about hybrid power plants the same issues affecting the quality of solar integration in combined cycles have to be taken into account for the analysis of Rankine cycles, and the same parameters have to be used in order to evaluate solar integration quality.

Integration into hybrid Rankine cycles deserves at least as much attention as integration into combined cycles for several reasons:

1. Is the more extended thermal power plant currently in operation.
2. In many cases this power plants use a lower environmental quality fuel (coal , fuel-oil) than combined cycles (gas), and therefore the environmental benefits associated to hybridise Rankine cycles is higher than in combined cycles.
3. Many old coal burning thermal power plants will find restrictions into their operation hours because of limitations in CO₂ emissions. Retrofitting these power plants with a solar field could be a good strategy for solar energy introduction by using the already existing investment in the power block.

In Figure-12 we find a reheated Rankine cycle decomposed in such a way that it may be interpreted as a combined series-parallel combined cycle.

The higher temperature cycle is a Rankine working between high pressure turbine inlet pressure and reheating pressure, with a η_1 -value given by η_H . The lower temperature cycle in a Rankine working between low pressure turbine inlet pressure and condensing pressure, with a η_1 -value given by η_L .

The upper cycle transfers all its condensing energy (Q_{HL}) to the bottoming cycle (without energy loss), and besides there is a parallel thermal coupling because heat is introduced into both cycles (Q_H , Q_L). The total cycle conversion efficiency (η_1 -value) is given by Eq.44, which in terms of Figure-12 may be evaluated as Eq.56.

$$\eta_1 = \eta_H + \eta_L - \eta_H \cdot \eta_L - \frac{Q_L}{Q_H + Q_L} \cdot \eta_H \cdot (1 - \eta_L) \quad (\text{Eq.56})$$

In Figure-12 we called Q_H to all the thermal energy input into the topping cycle, which can happen in preheating, evaporation or superheating. We called Q_L to all the external thermal energy input into the bottoming cycle, which can happen in preheating or superheating zones. Physically, both Q_H and Q_L are thermal energy inputs that take place in the boiler. In Figure-12 we called Q_{LPT} to a heat supply associated to an injection of solar steam, raised in parallel with the main boiler, straight into the low pressure turbine. This heat supply would be converted at the same η_1 -value as the Q_L , but collector field efficiency and solar fractions would be different.

The η_1 -value for the heat supplied in the topping cycle (Q_H) is the one of a series combined cycle without heat loss between cycles, and in terms of Figure-12 is given by Eq.57.

$$\eta_{1H} = \eta_H + \eta_L - \eta_H \cdot \eta_L \quad (\text{Eq.57})$$

On the other hand, the heat supplied to the bottoming cycle (Q_L or Q_{LPT}) will be converted into work with the η_1 -value of the bottoming cycle:

$$\eta_{1L} = \eta_L \quad (\text{Eq.58})$$

The solar thermal energy input can be performed anywhere in the cycle heat supply zone, but the conversion efficiency for this thermal input will depend on where it is introduced. The relation from Eq.59 holds for the different η_1 -values introduced:

$$\eta_{1H} > \eta_1 > \eta_{1L} \quad (\text{Eq.59})$$

Therefore, from the point of view of solar energy conversion efficiency, the solar thermal input should be done in any of the locations indicated by Q_H , in which case solar energy would be converted into work at an efficiency (η_{1H}) higher than the Rankine cycle efficiency (η_1). From the point of view of thermal solar energy conversion it does not matter the temperature level at which we introduce it (preheating, evaporation, or superheating), that is, the η_7 -value would be the same for any solar thermal energy input in these regions. However, collector field efficiency and boiler efficiency will be affected by the temperature level where solar energy is introduced. From the point of view of collector field efficiency, the lower the temperature level the better use we will do from the solar investment, that is, the higher the η_8 -value. But care should be taken in the low temperature zone for not spoiling boiler efficiency (increasing flue gas outlet temperature). A good integration will be a balance between these two factors.

Any solar thermal energy input in the Q_L zone will be converted at a very low η_1 -value, and therefore should in principle be avoided. The solar heat input in Q_{LPT} will also be converted with a low η_1 -value, but since in this case solar energy will be used to preheat, evaporate and superheat, solar fractions will be high, and collector field operating temperatures will be lower than those from the superheating region for Q_L .

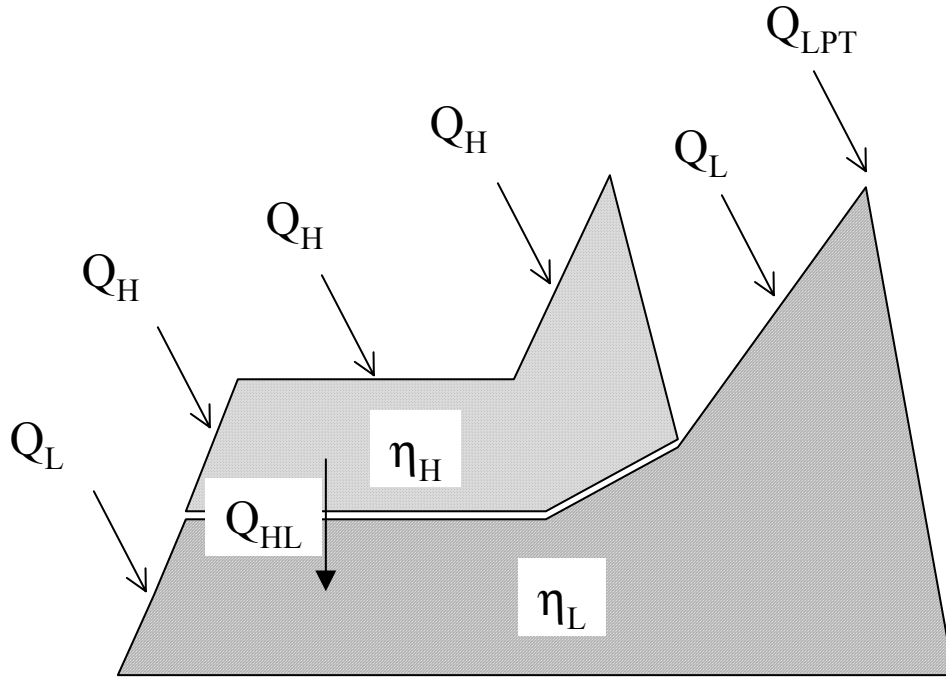


Fig-12: A Rankine reheated cycle viewed as a series/parallel combined cycle.

From the point of view of part load operation, solar thermal energy supplies in the Q_L and Q_H zones could introduce imbalances in boiler heat exchangers that could force to introduce irreversibilities by mixture of cold water with the generated steam. As an example, a solar thermal input in the preheating or evaporating zone of Q_H could mean the need to introduce cold water before the superheater and reheater in only fuel operation (if total fuel supply is kept constant and water mass flow reduced) for not going beyond the maximum allowed operation temperatures in boiler and turbine inlet. In contrast, integration schemes in Q_{LPT} will not introduce any heat exchanger imbalance since solar steam is generated completely outside the boiler.

Integration schemes in Q_{LPT} are better suited for 'boosting' integration strategies, forcing the low pressure turbine to work with a higher flow mass that on design conditions. However they could also be used in 'fuel-saver' integration strategies, although HPT would probably have to work in part load.

Integration schemes in Q_H are better suited for operation in fuel saver mode. However, if boiler heat exchangers are designed to perform the required functions in the zones without solar heat input with less fuel input, in only fuel operation these heat exchangers will be oversized and could force to some

mixing irreversibility. Integration schemes in Q_H , can only operate in booster mode if all the regions from the Q_H heat supply region may be covered by the solar field in parallel with main boiler.

Integration schemes in Q_L can only operate in fuel saver mode. However, if boiler heat exchangers are designed to perform the required functions in the zones without solar heat input with less fuel input, in only fuel operation these heat exchangers will be oversized and could force to some mixing irreversibility.

Comparing the introduction of solar thermal energy on a Rankine cycle with that on a bottoming multiple pressure of a steam cycle, from the point of view of solar conversion efficiency, we can have the following conclusions:

1. Higher solar fractions are achievable with integrations on Rankine cycle, where the upper limit value for the ζ_1 -value is 100%, that is, a only solar power plant in design conditions.
2. Conversion efficiencies for the solar thermal energy input will be higher in Rankine cycles than in bottoming steam cycles. Indeed, turbine inlet pressures and temperatures are usually higher on conventional Rankine cycles than on bottoming steam cycles, and as we have already indicated the high pressure zone is the one that allows for the higher solar thermal energy conversion efficiencies. Therefore, higher η_7 -values have to be expected from Rankine cycles.

Due to the lack of regeneration and the lower saturation pressures from bottoming cycles, higher collector field efficiencies are to be expected for integration into bottoming steam cycles. However, as we saw in (Garcia-Casals X., January 2000), cycle efficiency usually weights more than collector field efficiency with turbine inlet parameters under those of conventional power plants, and therefore higher η_8 -values are to be expected from integration schemes in Rankine cycles than on combined cycles.

3. The multiple pressures available in bottoming cycles do not represent any advantage because of the lower solar thermal energy conversion efficiencies associated to these cycle regions.
4. Conventional Rankine cycles are usually Stack limited, and therefore, the fact of using solar energy to evaporate the water will not bring about an increase of the efficiency how fuel is being used. Indeed, in Rankine cycles this improvement in fuel energy use could come only from an increase in boiler efficiency, and if the boiler is stack limited, the Pinch point elimination will not improve boiler efficiency.

On the other hand, in bottoming cycles the HRSG usually is pinch point limited. Therefore, when using solar energy to evaporate the water, two improvements in fuel conversion efficiency can be achieved: An improvement in HRSG efficiency (η_{HRSG}), and an improvement in bottoming cycle efficiency (η_{ST}) by upgrading the characteristics of the steam at the high pressure turbine inlet. Of course, this has to be balanced with the probable decrease on the

use of fuel energy efficiency in only fuel operation to see which is the net effect on plant performance.

8. Integration on Combined Cycles

8.1. Introduction

There are many different ways to integrate solar energy on a combined cycle scheme. Below we are going to comment the main ones that could be implemented with parabolic trough technology. We will emphasise the thermodynamic point of view from such integrations. There are other factors (commercial equipment availability, economic considerations, operational flexibility, development strategies, ...) that also play an important role on selecting the appropriate integration scheme for a given application.

When considering an integration scheme with parabolic trough technology in a combined GT+ST cycle, if no solar energy storage is considered, the additional solar thermal energy input will force some equipment to be working out of design conditions in solar or in only fuel operation. The main issue in solar integration is how to avoid that this variable solar thermal input spoils power block performance either in solar operation or in only fuel operation. Speaking about integration in HRSG, there are different ways to cope with this variable solar thermal energy input:

1. To have the gas turbine operating at part load when solar energy is available and at full load when no solar energy can be used, in such a way that only fuel operation is performed with a combined cycle in design conditions.
2. To overload the steam turbine when solar energy is available, and recover a more or less conventional combined in only fuel operation.
3. To use an upgraded bottoming cycle to manage the solar thermal energy input in design conditions, and in only fuel operation to burn fuel in HRSG to make up for the missing solar thermal energy input.

In all of them HRSG changes its operating conditions when solar energy is not available, which often means increased irreversibilities to compensate for heat exchanger imbalances.

The first approach has the drawback of the bad part load performance from the gas turbine, but has the advantage of not imposing such severe limitations on solar fraction as the second approach.

It is important to note that the bad part load behaviour from the gas turbine, together with the fact that steam turbines that can be fed with a single gas turbine usually have small power and therefore small efficiency, have led conventional combined cycle power plants to be implemented with several gas

turbines feeding a single steam turbine. An example is the 2000 MW_e combined cycle power plant recently implemented in Korea with 8 GT24 gas turbines from ABB feeding in parallel a single steam turbine. This, on the one hand allows for a power plant part load operation without reductions in efficiency by changing the number of active gas turbines. On the other hand, the high power steam turbine has better efficiency than what could be reached in a combined power plant with a single gas turbine.

This approach could also be very advantageous to solar thermal power plants with integration in bottoming cycle. Indeed, in design solar operation, only some of the parallel gas turbines would be operated at full load, and when solar thermal energy input would be under design conditions, other gas turbines would go into operation. In this way, both gas turbine and steam turbine would be operating in almost design conditions as well with or without solar energy input. Only the HRSG, like in almost all the integration schemes, would suffer from heat exchanger imbalances.

The second approach imposes limitations on the solar fraction due to the capability of the steam turbine to manage the increased steam mass flows when solar energy is available, and demands some design modifications in the last steam turbine stages. The reduced steam mass flow when solar input is under design conditions demands a steam turbine operation in sliding pressure mode, which together with heat exchanger imbalances in HRSG produce a reduced power block efficiency in only fuel operation.

The third approach has the drawback of having to burn fuel in the bottoming part of the combined cycle when solar thermal energy input is under design conditions. This is in part compensated by the fact of using an improved bottoming cycle (high pressure and temperature turbine inlet conditions + reheating) compared with conventional combined cycles, but still means a reduction in fuel usage efficiency.

8.2. Considerations about integration in bottoming steam cycles.

A useful way to understand the quality from different integration schemes in the bottoming cycle of a combined cycle is to view such bottoming steam cycle as a combined cycle itself. Therefore, the arguments and formulation presented in point 4 can be applied to determine conversion efficiencies when solar energy is supplied in different cycle zones.

If the bottoming cycle is a single pressure no reheated Rankine cycle, wherever solar energy is introduced it will always be converted at bottoming cycle efficiency.

If the bottoming cycle is a single pressure reheated Rankine cycle, it can be viewed as a combined cycle formed by a series combination without heat loss between cycles plus a parallel thermal combination. In order to do so, one should consider the topping cycle as a Rankine that goes from turbine inlet pressure to reheating pressure, and the bottoming cycle as a Rankine that goes from reheating pressure to condenser pressure. In this way, conversion efficiencies for solar thermal inputs in the different heat supply zones of bottoming cycle are as discussed in point 7.

If the bottoming cycle is a multiple pressure no reheated cycle, it can be viewed as a parallel combined cycle, where any of the component cycles are Rankine cycles operating between the corresponding HRSG pressure and condensing pressure. In this case, the higher the pressure level where we introduce the solar thermal input, the higher the efficiency at which it will be converted into work, independently of the temperature at which solar energy is supplied.

If reheating is introduced in a multiple pressure bottoming cycle, it can be viewed as an additional parallel thermal coupling as in a Rankine reheated cycle.

Nowadays bottoming steam cycles from commercial combined cycles tend to have several pressure levels, looking for a better use of the gas turbine exhaust gases on the pinch point controlled HRSG. For combined cycle power plants above the 100 MW_e power range, the three pressures reheated cycle seems to be becoming a standard. Even a 4 pressure level bottoming cycle may be worthwhile (Pérez del Notario P., et al, 2000) when the compresor is intercooled with water from the steam cycle.

However, when considering solar integration issues, a high number of pressure levels in the bottoming cycle it is not necessary the most appropriate configuration.

Indeed, on the one hand, the integration of solar energy plays a bit the same role as the several pressure levels, that is, to reduce the pinch point limitations and to allow a better use of the gas turbine flue gas thermal energy. In fact, solar integration in design conditions is a more powerful tool to eliminate pinch point limitations than the multiple pressure strategy, with the potential to completely get rid of this limitation. However, if no solar storage is available, solar integration schemes have the drawback the HRSG pinch point problem completely unattended when solar input is no available.

On the other hand, the heat exchanger imbalances associated to the presence/absence of solar thermal energy input can be more severe on a multiple pressure bottoming cycle than on a single bottoming cycle because of the high number of heat exchangers interacting in a HRSG with multiple pressures.

As we have discussed, from the point of view of solar thermal energy conversion efficiency the high pressure level of a multiple pressure bottoming cycle is the most appropriate place to introduce solar energy. Besides, the low pressure levels have much smaller mass flows. All of this, together with the convenience on DSG fields to work at only one pressure level, will probably recommend to introduce solar thermal energy only on the high pressure level.

Therefore, the expected advantages, from the solar integration point of view, of using a multiple pressure bottoming cycle instead of a single pressure are :

1. A higher turbine inlet pressure will be available, increasing the conversion efficiency of the solar thermal energy input.
2. In only fuel operation a higher fuel conversion efficiency will be achieved.

However, there are also some expected disadvantages when using a multiple pressure bottoming cycle instead of a single pressure:

1. Lower solar fractions can be achieved.
2. The available preheating zone is very reduced because of being occupied by lower temperature levels.
3. Higher collector field working temperatures (lower efficiencies).
4. More heat exchanger imbalances to make up for due to the discontinuity of solar thermal energy input.

An intermediate approach it would be to dispose different sets of heat exchangers in the HRSG and have the bottoming cycle working in one or several pressure levels depending on whether solar thermal input is available or not. However, this would demand a flexible steam turbine design.

Therefore, a careful comparative analysis based on integration performance parameters is needed to decide whether a multiple pressure bottoming cycle is the best option for a new solar thermal hybrid power plant.

8.3. Possible integration schemes in GT+ST combined cycles

In general, when integrating solar energy without thermal storage in the bottoming cycle from a conventional GT+ST cycle, some plant components will have to undergo part load performance if no fuel is to be burnt in the low temperature steam cycle (low exergy efficiency), with the consequent efficiency reduction. In some integration schemes the gas turbine is operated at part load when solar energy is available. Other integration schemes operate the steam turbine at part load when solar energy is not available, or in boosting mode when solar energy is available. In many integration schemes the different heat exchangers in the HRSG go out of balance when solar energy is not available, demanding a mixing process with cold water (irreversibility) in order to avoid excess temperatures. The key to a good integration process is to minimize this situations without a high increase in investment, and to choose the best place to introduce solar energy in the sense of reaching the higher total conversion efficiency for the solar input or synergies with other plant components that improve their component efficiencies.

Integration schemes where solar energy is mixed with gas turbine exhaust gases prior to inlet in HRSG, like the ones proposed in (Lippke F., et al, 1996), can be applied to any of GT+ST combined cycle schemes below commented. The thermodynamic draw back from this integration scheme is the fact that the fraction $(1-\eta_{\text{HRSG}})$ of the collected solar energy is thrown to the atmosphere with the stack gases. The fraction $\eta_{\text{HRSG}} \cdot Q_s$ is converted at bottoming cycle efficiency. There are other integration schemes into the HRSG where a higher total conversion efficiency from solar to electricity can be achieved. Although it is also true that in some ways to integrate solar energy into or after HRSG a lower total conversion efficiency from solar to electricity can be produced.

We will not pay much attention to integration schemes where solar energy is introduced in the high temperature zone from the combined cycle (gas turbine) , because in general they are out of reach from parabolic trough technology. However, from the thermodynamic point of view this is the best way to use a given Q_s , because it will be converted at overall combined cycle efficiency. However, one should note that this does not mean that these integration schemes are the ones that make the better use of purchased collector aperture area. Indeed, the efficiencies from collector field and needed additional heat exchangers are in between and deserve careful consideration.

There have also been proposals of integration schemes where solar generated steam is introduced into the combustion chamber of a gas turbine belonging to a combined cycle or forming a Cheng cycle. These integration schemes, although favourable from the points of view of conversion efficiency and operability, have the problem of a high water consumption. Indeed, all the water that goes through the solar field is thrown to the atmosphere with the gas turbine exhaust gases, and therefore, water supply and treatment must be constantly available. Taking into account that the candidate regions for this kind of power plants have also potable water supply problems, it does not look likely that such integration schemes may become appropriate. In fact, one of the factors for the Cheng type cycles not having prospered in Europe for conventional applications is the water consumption issue.

In (Kolb G., 1997) an integration scheme valid for any steam bottoming cycle is presented. In this integration scheme solar thermal energy is supplied to the bottoming cycle in parallel with HRSG. The objective is not to affect the HRSG working conditions with or without solar input, because the HRSG mass flow and thermal functions will not be altered by solar thermal input. However, depending on steam turbine operation mode, HRSG pressure levels would be modified by the presence of solar energy input into steam turbine, which in turn would affect the thermal performance from the HRSG heat exchangers. Another advantage from this integration scheme is that solar thermal energy input is converted at an efficiency equal than bottoming cycle efficiency, for a single pressure steam cycle, or higher than bottoming cycle efficiency for a multiple pressure steam cycle, while collector field efficiency is benefited from the fact of having cold water as input.

8.3.1. Combined GT+ST cycle with a bottoming one pressure steam cycle.

One of the first proposals to integrate solar energy in a combined cycle was due to FLAGSOL . On their proposal, solar energy is integrated on the HRSG from a combined GT+ST cycle, increasing the capabilities of the HRSG by the use of solar thermal energy for water evaporation. In this way the bottoming steam cycle can be upgraded to a conventional Rankine cycle, with high pressure and temperature turbine inlet conditions, and reheating. In design conditions this approach gives very good performances, because of having a high efficiency bottoming cycle and reduced irreversibilities in HRSG. However,

in only fuel operation, the upgraded bottoming cycle may no longer be operated only with gas turbine exhaust gases, and therefore there is need to burn fuel in the HRSG. This is not so bad as it would be on a conventional combined cycle with a lower efficiency bottoming cycle, but reduces fuel use efficiency compared to a conventional combined cycle. In only fuel operation steam turbine inlet temperature and pressure are reduced (from 130 bar and 492 °C to 103 bar and 428 °C), and flue gas temperature at HRSG outlet is increased (from 113 °C to 184 °C).

In (Allani Y., 1992) an integration scheme for combined cycle using parabolic trough technology is preliminarily analysed. The 22,3 MW_e power block consists of a 12 MW_e gas turbine and a bottoming steam cycle (probably one pressure although not clearly stated). Solar energy is introduced in series between HRSG preheater and superheater to evaporate the steam. Steam turbine is operated at constant load, while gas turbine operates at part load depending on available solar irradiation. Although results are presented for ζ_4 and η_1 values as function of solar input, it seems like if no considerations are incorporated at all about components part load operation. In this integration scheme, special consideration should be given to the variable functions that the HRSG has to perform, with an increasing evaporation taking place in HRSG heat exchangers as available solar energy is reduced.

In the SOLGAS project (Blanco M., 1995), solar energy is used to evaporate water in a bottoming single pressure, no reheated Rankine cycle. Additional flue gas energy is recovered at lower pressure for cogeneration purposes. When no solar energy would be available, fuel would be burned in HRSG.

In the first approach to the Colon-Solar project (Blanco M., et al, 1996) a conventional repowering (topping) was considered. The bottoming cycle was a Rankine (single pressure) reheated cycle, and solar integration was used to evaporate water.

The integration schemes with the PHOEBUS central receiver technology in combined cycles (Lippke F., et al., 1996), as well as the proposed integration schemes for the ISCCS plants in Egypt (Rheinländer J., 2000) also fall into this category.

The other plant alternative analysed by ZSW for the ISCCS plants in Egypt (Rheinländer J., 2000) uses parabolic trough technology with oil as heat transfer fluid to preheat and evaporate water in parallel with the HRSG, introducing the solar thermal energy before the superheater. Because of using a single pressure steam bottoming cycle, solar thermal input is converted at bottoming cycle efficiency. To complement the solar input in off design operation, fuel energy is introduced in two locations of the bottoming cycle: In parallel with the solar field to complement short time deficits of solar input, and into the HRSG, to operate at full load without solar input (used from 18-22 h to cover the pick in demand). Besides, if no bottom firing wants to be introduced, in operation at part load without solar thermal energy input, the superheater will be oversized and need to compensate for this imbalance.

In (Kribus A., et al, 1998) an integration scheme for combined cycle using central volumetric receiver technology is analysed. Solar thermal energy input is used for gas turbine air preheating from compressor outlet. The 34 MW_e combined cycle implements a Pratt & Whitney FT8 gas turbine and a bottoming (probably one pressure, although not stated) steam cycle, giving a power block gross conversion efficiency of 47 %.

In Figure-13 we find a possible plant scheme for an integration in the bottoming steam one pressure steam cycle.

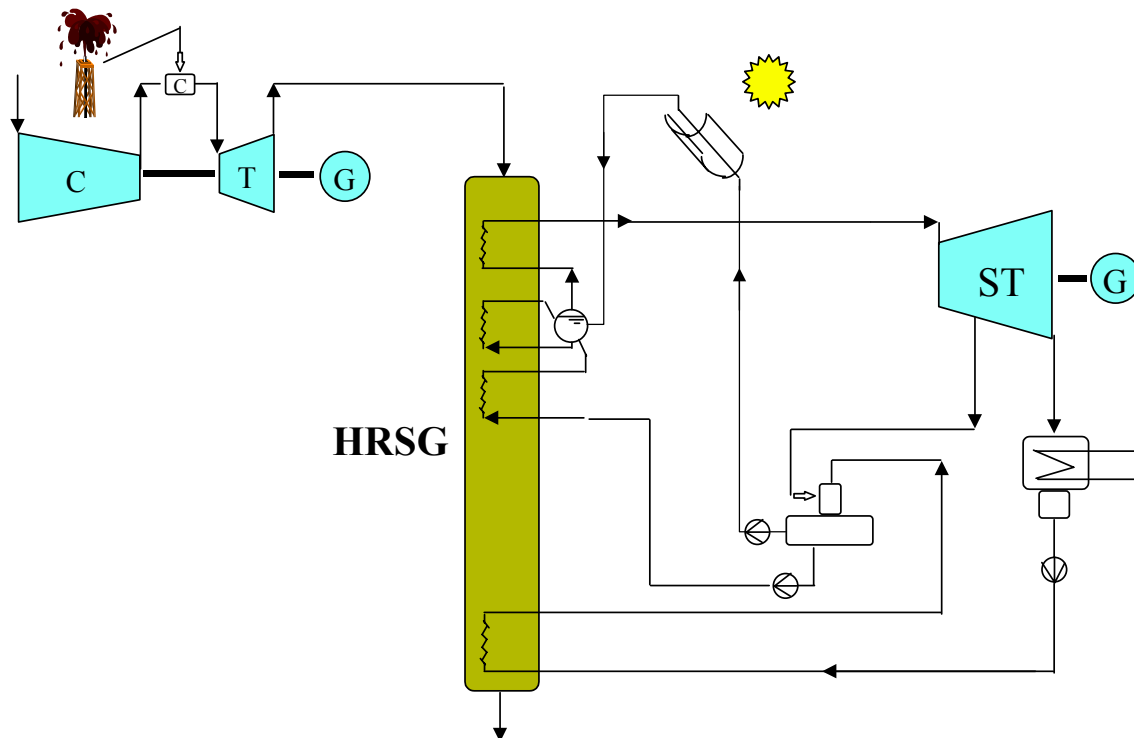


Figure-13: Integration in one pressure bottoming steam cycle.

8.3.2. Combined GT+ST cycle with a bottoming two pressures steam cycle.

In (Stein W., 1992) several integration schemes into bottoming cycle of combined power plants are analysed, using parabolic trough technology. The combined cycle is in the 200 MW_e power range, formed by two KWU 64.3 gas turbines feeding in parallel their exhaust gases into a HRSG that produces steam for a single steam turbine working with a two pressures (90 bar and 7 bar) reheated steam cycle. The gas turbine's working conditions are slightly changed in order to increase their exhaust gas outlet temperature until 564 °C, in order to be able to achieve high and low pressure steam turbine inlet temperatures of 538 °C. The different integration schemes analysed are the following ones:

1. Direct injection into steam turbine.
2. Solar booster heater (complementing solar with steam turbine extractions when no enough irradiation is available).
3. Solar in series into steam drum.
4. Solar in parallel into steam drum.

From all of these, the parallel integration was chosen as the most appropriate. Integration schemes with different design solar fractions were generated by keeping both gas turbines working in design conditions and increasing HRSG area and steam turbine capacities to accommodate the differing amounts of solar steam as the solar field size was changed.

Once a design was selected, annual performance was analysed assuming that gas turbines always worked at full load, while steam turbine and HRSG worked at part load following solar insolation.

In this integration scheme, the solar field takes cold water from the deaerator, and produces steam in parallel with the HRSG until the steam drum, where solar steam is mixed with HRSG steam and further superheated in the HRSG.

The introduction of the solar steam into the steam drum of the evaporator allows to accommodate solar steam with qualities varying over the entire subcooled liquid to superheated steam range. However, if subcooled liquid or superheated steam are introduced, an additional irreversibility is produced.

A degree of freedom in this integration scheme worthwhile to analyse with an appropriate integration evaluation parameter is solar steam quality when introduced into the evaporator drum. Indeed, a balance exists between irreversibilities in the solar field and in the HRSG (gas turbine flue gas path). If solar mass flow is introduced in subcooled conditions, there will be a mixing irreversibility for the steam risen in the HRSG, and the solar mass flow will have to be evaporated in HRSG worsening the pinch point. However, in this case solar field efficiency will be high. If solar mass flow is introduced in saturated conditions ($0 < x < 1$), there will be less mixing irreversibility, but a part of the solar mass flow will still have to be evaporated in HRSG. If solar steam is introduced in superheated conditions, there will be a mixing irreversibility for solar steam (paid for in terms of a lower solar field efficiency), but the excess energy in the solar steam will alleviate the HRSG evaporator, reducing the pinch point irreversibility.

By the analysis performed in (Stein W., 1992) it was concluded that the optimal operation strategy involved varying mass flow through solar field in such a way that the solar quality would be kept above saturated vapor when introduced into the steam drum. However, caution should be taken with these conclusions since the integration evaluation parameter used to judge optimal configurations was the ζ_4 -value, comparing schemes with different solar fractions, and as we have already said, this is not an appropriate measure of integration quality.

Indeed, in (Stein W., 1992), when evaluating the convenience to superheat the solar steam injected in the HRSG evaporator drum, they come to the conclusion that the higher the superheating, the better, in spite of mixing 400 °C solar steam with 304 °C steam in the drum. They support this conclusion by the fact of the increasing ζ_4 -value with superheating. However, as solar steam mass flow is kept constant, solar fraction increases with superheating, producing a ζ_4 -

value that increases with superheating in spite of a possible worse integration scheme.

Regarding the pressure level where solar steam should be introduced, in (Stein W., 1992), modelling demonstrated what should be obvious from the thermodynamic considerations we introduced in points 4.3 and 8.2, that is : There are significant performance improvements when injecting the solar steam into the high pressure drum rather than into the intermediate or low pressure drums.

Regarding the part load operation in only fuel mode, the following conclusions can be taken from (Stein W., 1992):

1. With a design having 30 % of mass flow into steam turbine from the solar field, in only fuel operation, cycle efficiency was the same as in the reference only fuel plant. The larger HRSG surface (and probably condenser surface, although not commented) compensated for the steam turbine part load operation.
2. With a design having 80% of mass flow into steam turbine from solar field, in only fuel operation mode cycle η_1 -value is reduced from 51,25 % to 51,17 % , indicating that the steam turbine part load operation is starting to take effect, though still not with great significance.

However, it has to be taken into account that the higher HRSG and condenser areas that compensate for the steam turbine part load performance, mean an investment increase relative to the reference fuel power plant.

In (Rheinländer J., et al., 1994) two integration schemes for a combined GT+ST cycle with a bottoming two pressures no reheated steam cycle are analysed. They are called additive and substituting solar integration. In both integration schemes solar energy is supplied to the low temperature zone in the HRSG.

In the additive integration scheme, the solar field acts in parallel with the HRSG, using as input steam from the deareator and producing high pressure (100 bar) steam for the HP turbine. Therefore, in this integration scheme solar energy is converted at a higher efficiency than overall bottoming cycle efficiency (parallel combined cycle of the two pressure levels). In this integration scheme, the maximum solar fraction is limited by the steam turbine capacity, allowing higher solar fractions than the substituting integration, where maximum solar fraction is limited by the needed energy input to evaporate the water.

In the substituting solar integration, solar energy is used to evaporate high pressure (105 bar) and low pressure (5,5 bar) steam, in parallel with the two steam drums in the HRSG, using two DSG solar fields working at different pressures. In this scheme, the high pressure steam is converted at an efficiency higher than the overall bottoming cycle efficiency, but the low pressure steam is converted at a lower efficiency than bottoming cycle efficiency. The advantages from this scheme are the reduction in irreversibilities in the HRSG because of eliminating the pinch point, and an easy and efficient solar field operation because of producing saturated steam.

In this two integration schemes, the gas turbine is operated in part load when solar energy is available (for not having to burn fuel in the low temperature zone when no solar energy is available) , and in order to increase solar fraction, two steam turbines are used in parallel (only one would be working in fuel only operation). A total plant output of 87 MW_e is achieved for all solar fractions. Gas turbines are rather inefficient at part load operation.

This integration scheme keeps a constant power output in all operation conditions (with and without sun). In order to do so, in design conditions the gas turbine is operated at part load (50 %), while in only fuel operation the gas turbine works at 100 % load. To further increase achievable solar fractions two steam turbines are disposed in parallel, in such a way that both are operative in design conditions, but only one of them will be working in only fuel operation mode.

With one steam turbine, the additive integration scheme reaches a $\zeta_3 = 0,20$ in design conditions, while the substitutive integration scheme gives $\zeta_3 = 0,16$. With two steam turbines, the additive integration scheme reaches a $\zeta_3 = 0,62$ in design conditions, while the substitutive integration scheme gives $\zeta_3 = 0,33$.

This ZSW integration scheme does not force any boosting operation mode in power block equipment. The price to pay for it is to have the gas turbine operating at part load in design conditions, and one steam turbine stopped in fuel operation mode.

In the (UNION FENOSA et al, 1997) project, the integration schemes from SIEMENS consist on using solar thermal energy as a supplement to a conventional GT+ST combined cycle plant with a two pressures no reheated bottoming cycle. This implies having the steam turbine working above 100 % load, and demands some adjustment in turbine last stages design. With this integration scheme maximum gas turbine overload determines maximum design solar fraction, and without solar input the power block does not experience a very severe efficiency reduction (some penalty remains due to HRSG heat exchangers imbalance). A 15 % power output increase in design conditions is achieved by solarization.

SIEMENS proposes two options to integrate solar thermal input:

1. Integration in HRSG. More easy integration, with the only drawback of having to compensate HRSG heat exchangers imbalance by cold water injection in order to avoid turbine inlet temperatures above design conditions (540 °C) in no solar operation.
2. Solar integration in intermediate pressure steam turbine. With this integration scheme, HRSG efficiency is not modified in no solar operation, but high pressure turbine inlet pressure must also be reduced. In design solar operation, an additional irreversibility can be produced due to the mixing of solar steam with outlet high pressure turbine steam. A higher η_1 -value in no solar operation than the reference combined cycle is achieved thanks to the no shaded effect of increased condenser area. As discussed in point 5.2 , this integration scheme leads to lower thermal solar energy conversion efficiencies.

In the second approach to the Colon-Solar project (Silva M., et al, 1998), according to this reference, changes in the Spanish legal framework made the initial (topping) scheme economically unfeasible in the new electricity generation scenario. Therefore, the plant configuration evolved into a hybrid conventional GT+ST combined cycle, using as bottoming cycle a two pressures steam cycle. When no solar energy would be available, fuel would be burned in HRSG.

In Figure-14 we present a possible plant configuration for integration in a bottoming two pressures steam cycle.

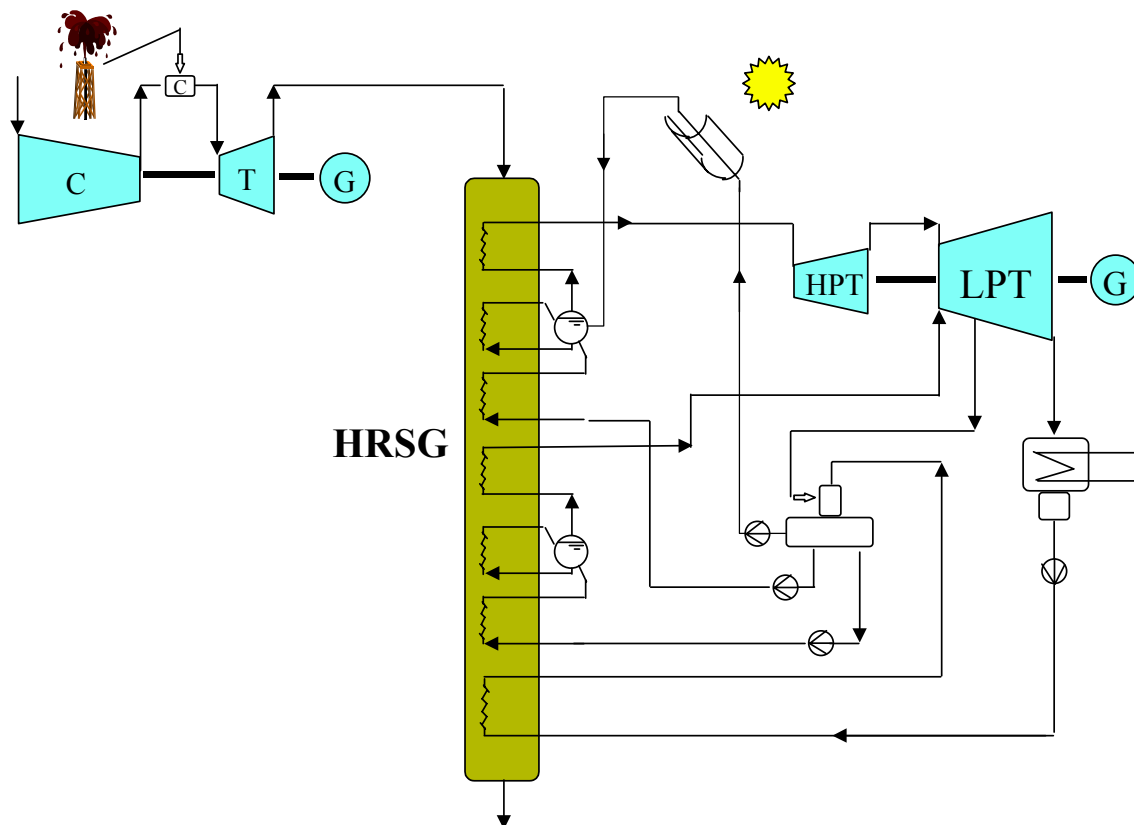


Figure-14: Integration in two pressures bottoming steam cycle.

8.3.3. Combined GT+ST cycles with a bottoming three and four pressures steam cycle.

Conventional combined cycle technology on the high power range is implementing a three pressure reheated bottoming cycle. We already commented in point 8.2 about the relative advantages and disadvantages of having different pressure levels in bottoming cycle.

Recently, in (Pérez del Notario P., et al, 2000) a fuel combined power plant has been proposed with 4 pressure levels in bottoming cycle, bringing about a

significant increase in cycle efficiency. The fourth pressure level finds its place because of intercooling the gas turbine compressor using water from the bottoming cycle. Besides the intercooling, the gas turbine from this combined cycle also implements a reheating. Another peculiarity of this combined cycle is that the steam turbine mechanical power is used straight for the gas turbine compressor, in such a way that only one generator for the supercharged gas turbine is required. From the solar integration point of view this mechanical coupling should be avoided to have more freedom for accommodating the variable solar thermal input.

8.4. Repowering

Conventional Rankine thermal power plants can be upgraded by adding a gas turbine to form a special configuration of GT+ST combined cycle called repowering.

As environmental limitations on low quality fuels such as coal increase, several Rankine type coal or oil burning thermal power plants with a significant life time left, can be upgraded by a repowering. This opens the way for solar thermal technologies to be considered in the upgrading process if they can be well integrated with the repowering strategy. The first version of the Colon Solar project (Blanco M., et al, 1996) dealt with such an integration scheme.

Since in all repowering schemes fuel is also burned in the bottoming cycle, solar thermal integration looks interesting because of its potential to reduce bottoming cycle fuel usage.

8.4.1. Series repowering (topping)

In the conventional repowering scheme (topping), a gas turbine is added on the top of an existing Rankine power plant in order to increase its power and efficiency, and reduce its specific pollutant's emissions. The gas turbine exhaust gases are sent into the steam cycle boiler and used to burn the Rankine cycle fuel.

The gas turbine from a topping produces around 10 % to 30 % of total power. Therefore, the gas turbine is relatively much smaller than on a conventional series GT+ST combined cycle, where it produces around the 65 % of total power. Because with parabolic trough technology only the bottoming cycle can be accessed, the repowering offers the potential of higher solar fractions than a conventional series combined cycle.

In contrast with the conventional GT+ST series combined cycle, the Rankine cycle from a repowering is a very regenerative cycle with high pressure and temperature turbine inlet conditions, and often a reheating, since the power plant comes from a pure Rankine cycle. In a repowering most of the bottoming cycle thermal energy input still comes from the boilers combustion process, and the boiler's flue gas thermal energy is used to preheat combustion air⁶. This

⁶ Since when the repowering is implemented part of the combustion air is substituted by gas turbine exhaust gases, there is no so much need for combustion air preheating. That is why in most topping

means that bottoming cycle efficiencies are higher than on conventional GT+ST series combined cycles, and therefore solar energy conversion efficiencies (η_7 -value) will be higher.

Another advantage from the repowering is that it has a much better part load performance than conventional GT+ST combined cycle. This facilitates the solar integration and permits an under base load operation strategy, increasing achievable solar fractions.

Conventional repowering schemes are encountering increasing difficulties because of gas turbine technology and environmental legislation evolution. However, solar thermal integration can benefit from some of these drawbacks:

1. Increasing gas turbine outlet flue gas temperatures require the use of Cr-Mo alloys in the boiler and gas ducts. This makes it difficult to keep on using the boiler from the old Rankine power plant. On the one hand this situation increases the repowering investment costs, but from the solar point of view, this situation makes it more easy to introduce the required boiler modifications for the solar integration.
2. With the higher turbine inlet temperatures in modern gas turbines, the available oxygen in gas turbine flue gases is smaller than before, and often not enough to complete combustion requirements in the Rankine cycle boiler. Therefore there is need to add about 25 % of the comburent flow as ambient air, or even it may be impossible to use all the gas turbine flue gases if the gas turbine power is in the upper range. If solar energy is integrated, less fuel has to be burned in the bottoming boiler, and therefore the need of ambient air is reduced, and maybe all gas turbine flue gas can be accepted.

In Figure-15 we present a plant scheme of a series repowering.

schemes water flow is split from condenser outlet into two parts: One goes through the steam preheaters, and the other through water preheaters feed with boiler flue gas.

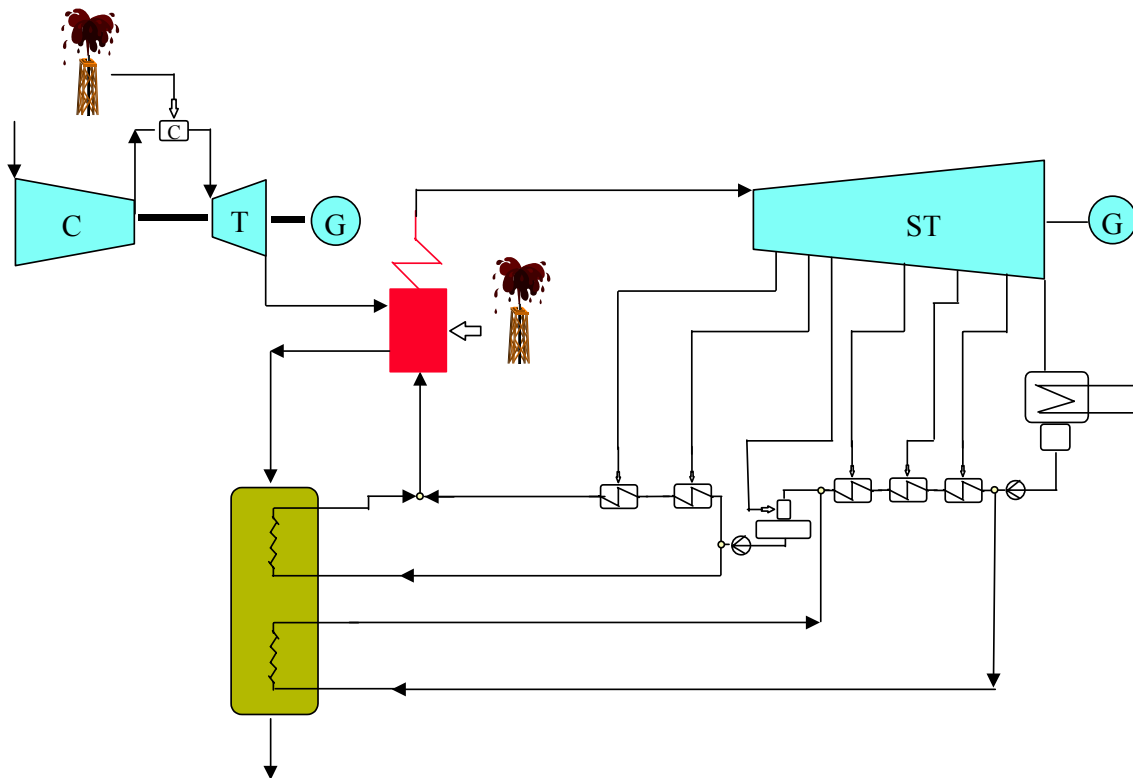


Figure-15: Series repowering power plant.

8.4.2. Parallel repowering

This repowering scheme originated to give an answer to the problems arisen with the conventional repowering. The gas turbine power in this scheme is between 10 % and 40 % of total power, and part load performance is very good. Like in a conventional repowering these are positive points for solar integration.

In this scheme, the condensed water flow from the Rankine cycle is preheated, evaporated and superheated in parallel by a HRSG using the gas turbine flue gases, and by the boiler from the Rankine cycle. The boiler uses as comburent ambient air preheated with boiler flue gases. The steam risen in the HRSG is fed into the intermediate pressure steam turbine, while only the steam produced in the boiler expands through the high pressure turbine. Normally this means that high pressure turbine inlet pressure can be reduced, which although increasing boiler life time, reduces bottoming cycle efficiency. Solar thermal cycle energy could be introduced to increase high pressure turbine mass flow, although maybe steam tyurbine generator maximum power could be reached, demanding a generator upgrade. But in fact, a parallel repowering offers many other possibilities to integrate solar energy (HRSG, boiler, HPT, LPT, ...).

In Figure-16 we present a plant scheme for a parallel repowering.

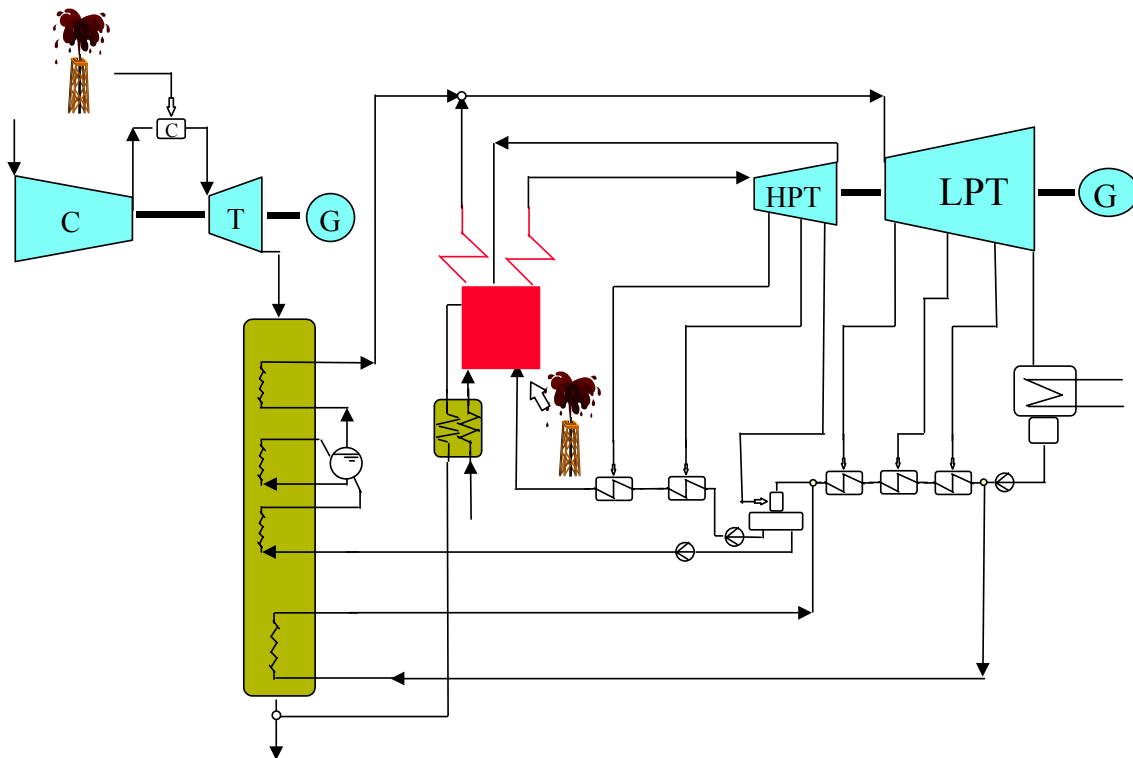


Figure-16: Parallel repowering power plant.

8.5. Combined GT+ST cycle with a reheated and regenerative gas turbine.

In (Garcia-Casals X., 1997) a combined cycle implementing a reheated and regenerative gas turbine was proposed for integration of solar energy in bottoming cycle. The concept pretended to avoid most of the problems associated with power block performances degradation in only fuel operation mode due to solar integration in bottoming cycle. To do so, an alternate use was made of gas turbine flue gases thermal energy depending on whether solar energy was available.

In design conditions the topping cycle is a gas turbine cycle with reheating and full regeneration. The exhaust gases from the regenerator, together with the solar thermal energy, are used to supply heat for the bottoming Rankine cycle.

In only fuel operation the topping cycle is a gas turbine cycle with reheating but without regeneration. The excess thermal energy content of non regenerative gas turbine is used to cover the missing solar thermal energy input.

With solar input under design conditions, the gas turbine exhaust gases are partly bypassed along the regenerator to cover the reduced solar energy supply.

This concept presents several advantages:

1. Avoids burning fuel in bottoming cycle when solar energy is not available.
2. It has the potential to keep a constant combined cycle thermal efficiency (η_1 -value), both with and without solar energy.

3. The main power block components are always working in design conditions.
4. Simple regulation for different solar energy inputs.
5. Design flexibility for different combinations of design solar fraction (ζ_1 -value) and combined cycle efficiency (η_1 -value).
6. Solar energy can be used only to evaporate steam or to preheat, evaporate and superheat.
7. Allows to implement a high performance Rankine cycle (100 bar, 550 °C), increasing solar energy conversion efficiencies (η_7 -value) in comparison with conventional steam bottoming cycles.

As an example, using a gas turbine with 1500 K turbine inlet temperature, the following designs may be achieved changing gas turbine compressor ratio (r_c):

1. $r_c = 6$; $\eta_1 = 55,6 \%$; $\zeta_1 = 46 \%$
2. $r_c = 20$; $\eta_1 = 59,9 \%$; $\zeta_1 = 19,4 \%$
3. $r_c = 23$; $\eta_1 = 55,0 \%$; $\zeta_1 = 19,5 \%$ (Solar energy only to evaporate)

In Figure-17 we present a plant scheme for this combined cycle. In Figure-18 we find the thermodynamic cycle and main thermal energy flows for design (solar) operating conditions, while in Figure-19 we can see this information in only fuel operation.

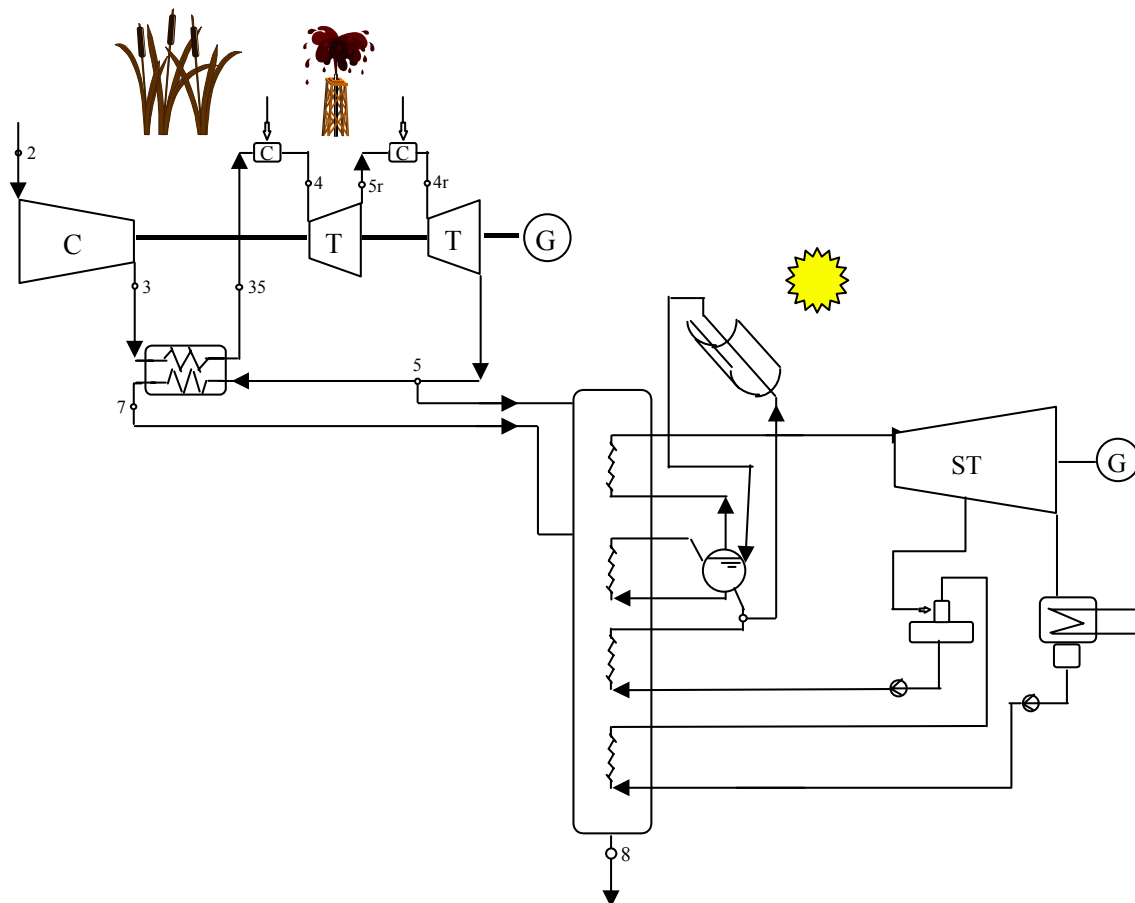


Figure-17: Combined cycle with reheated regenerative gas turbine with solar integration in bottoming steam cycle.

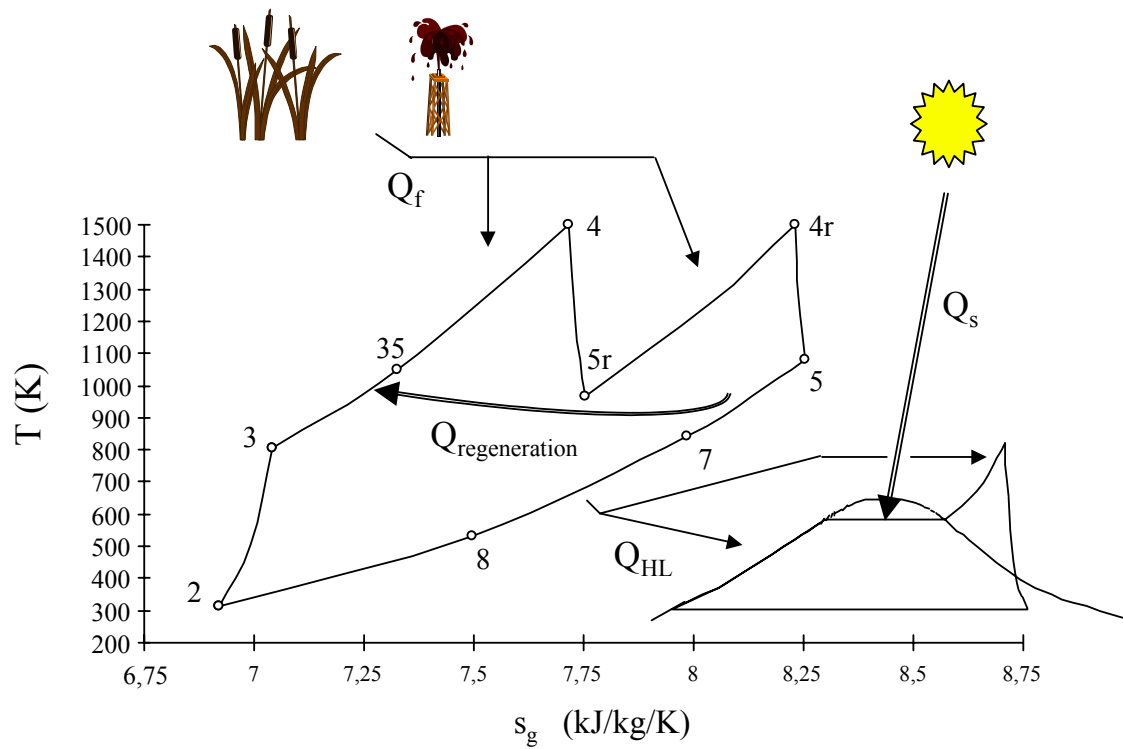


Figure-18: Thermodynamic cycle and main thermal energy flows in design conditions.

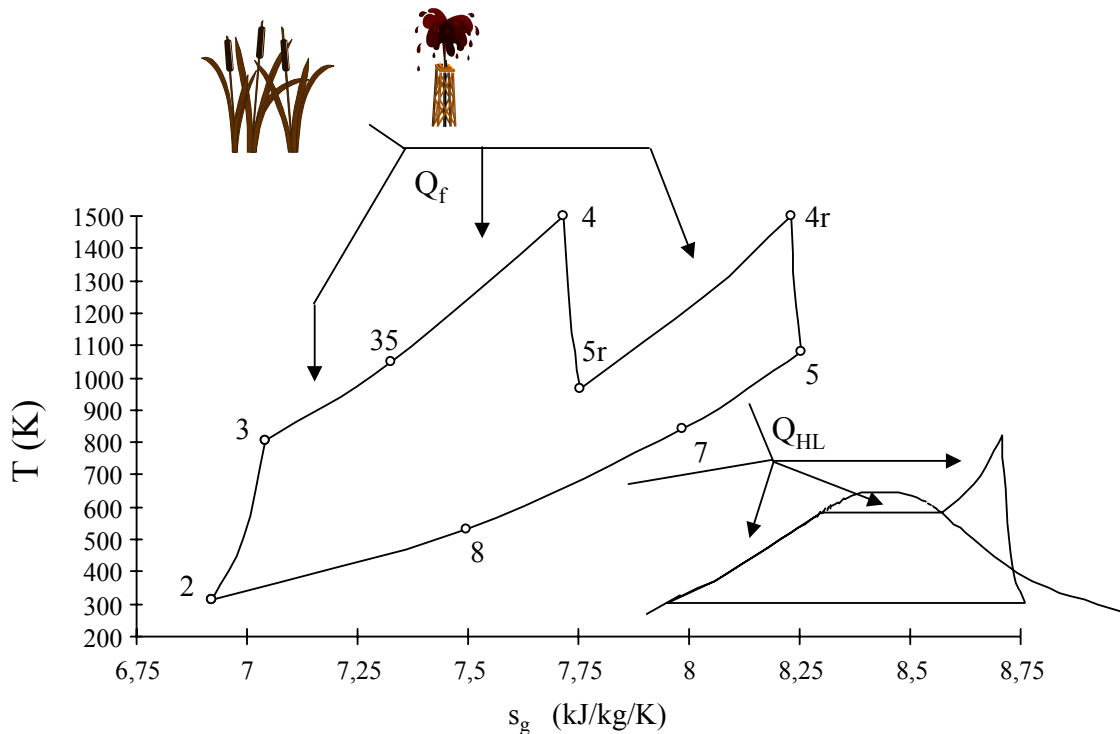


Figure-19: Thermodynamic cycle and main thermal energy flows in only fuel operation.

8.6. Combined GT+ST cycle with an intercooled gas turbine and solar integration in gas turbine.

In this integration scheme, as well as if parabolic trough technology with thermal oil or DSG are used, a heat exchanger oil-air or water-air needs to be incorporated in the scheme. The heat exchanger effect on integration evaluation parameters is that it will produce an increase in collector working temperature, and therefore a decrease in collector field efficiency.

The Kokhala-type integration scheme on combined cycles has been proposed (Bohn M.S., et al, 1995), (Price H.W., 1996) for use of central receiver molten salt technology. This solar technology allows to produce solar thermal energy at 565 °C, which is a rather low temperature when one thinks to integrate solar energy in the high temperature zone of the combined cycle (gas turbine). Using parabolic trough technology with DSG, it should be possible to reach working temperatures in the same order of magnitude, moreover when for this application there is no need at all to work at high pressures. Indeed, the DSG solar field could be working at pressures of around 5 bar (just in the order of the pressure loss), since collected solar energy will be transferred via a heat exchanger to the air at the outlet from the gas turbine compressor. This has the additional advantage to decrease water saturation temperature, and therefore increase collector field efficiency, which could more than compensate the

increase in collector field working temperature due to the heat exchanger. At this low working pressures, the only limitation to reach DSG outlet temperatures in the range of 550 °C would be the selective coating stability of current parabolic trough technologies (cermets). However, it would be always possible, to use another selective coating in the last collectors from the solar field, like the ones actually implemented in central receiver technology, which would decrease the efficiency from these collectors (only a small fraction of total aperture area), until new selective coatings currently under R+D programs are proved.

Another possibility would be to implement this integration scheme with the actual selective coating limits (450 °C), with the only effect to diminish solar fraction. Taking in account that other integration schemes introducing solar energy in HRSG allow for relatively small solar fractions, the resulting Kokhala-type parabolic trough plant could give even a higher solar fraction, with all the advantages associated to solar energy input in gas turbine, like the fact that cycle efficiency would not be reduced in no solar operation mode because all components would work in design conditions. Indeed, in this integration schemes all the main power block components work in design conditions with and without solar input.

The way to overcome the limitations of low solar temperatures and still reaching relatively high solar fractions, is to use an intercooled gas turbine. Indeed, because of intercooling, high pressure compressor air outlet temperature is much lower than on a conventional gas turbine, which allows to heat gas turbine air from the compressor outlet temperature till the limit achieved with the available solar irradiation and technology. Afterwards, the air is fuel-heated until turbine inlet temperature.

The introduction of solar thermal energy in the gas turbine from a combined cycle has important advantages over other integration schemes:

1. The fact of introducing or not the solar thermal energy does not modify the power block, neither demands operation at part load or boosting mode from any of the power block components. The only exception in the gas turbine burner, whose operation at part load is rather efficient. Therefore, power block efficiency is almost the same with solar and only fuel operation modes.
2. The solar thermal energy input is converted with a 50-60 % η_1 -value , much higher than the 20-30 % achievable when solar energy is introduced in the bottoming steam cycle from a conventional combined cycle.

The full potential for solar thermal energy introduction in gas turbine can only be realised with central receiver high temperature technology (volumetric receivers), but even with parabolic trough technology, solar fractions higher than with other integration schemes can be achieved. Indeed, in the original Kokhala scheme with central receiver molten salt technology design solar fractions (ζ_1 -value) around 30 % are achieved.

Besides, an alternative integration scheme able to reach even higher solar fractions would be to introduce solar energy both in the gas turbine after compressor outlet, and in the HRSG to preheat and evaporate water for the

bottoming Rankine cycle as in conventional integration schemes. This has also been proposed as an integration scheme using molten salt central receiver technology (Bechtel, 1995).

In Figure-20 we find a plant scheme, thermodynamic cycle and solar fraction evaluations for a Kokhala-type combined cycle power plant. The solar thermal technology indicated in this figure is a central receiver because this was the original proposal for this plant, but it could perfectly be substituted by parabolic trough technology.

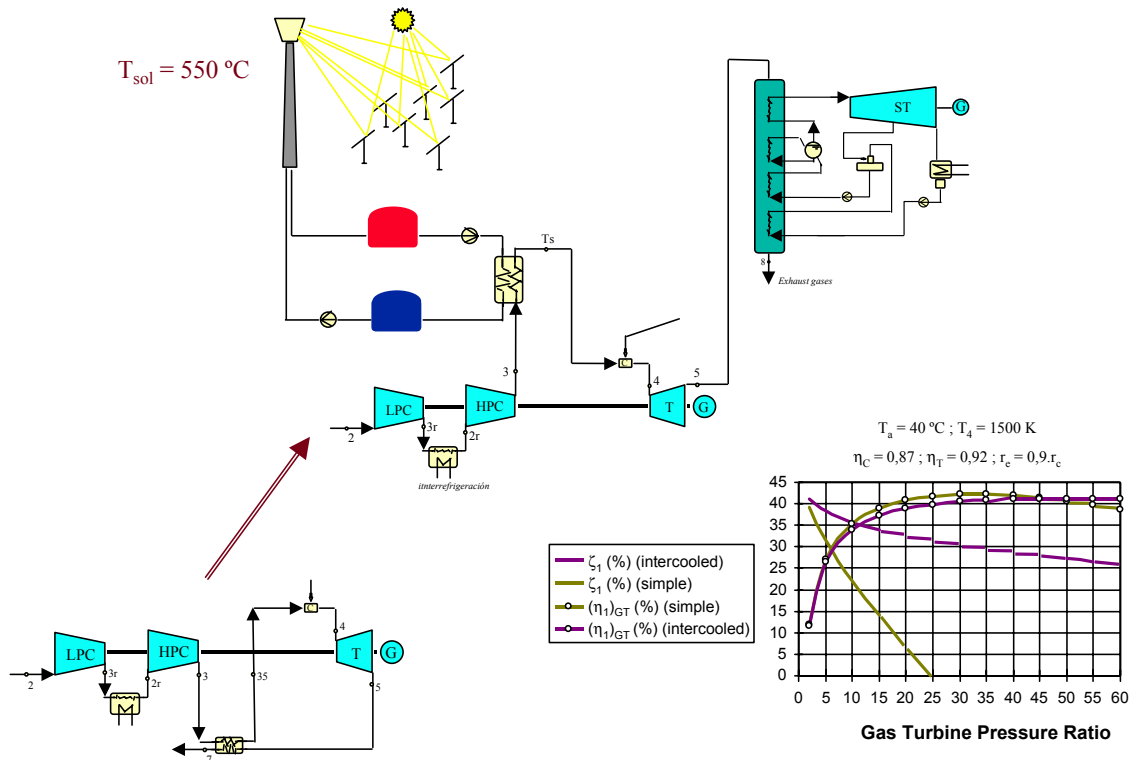


Figure-20: Kokhala-type combined cycle power plant implementing an intercooled gas turbine.

8.7. Brayton-Rankine steam cycle

In (Garcia-Casals X., 1997), an integration scheme using a high temperature steam combined cycle (Field cycle) was proposed. The topping cycle is a Brayton cycle using steam as working fluid with 25 bar and 850 °C turbine inlet conditions. The bottoming cycle is a Rankine regenerative cycle with 75 bar and 520 °C turbine inlet conditions. Solar energy is introduced in topping cycle, and therefore at low pressures. Fuel energy is used in this zone to complement solar energy input. After expansion in the high temperature turbine from the Brayton cycle, the steam is mainly used to evaporate and superheat steam for the bottoming Rankine cycle, achieving a high degree of internal regeneration.

Solar thermal energy can be introduced from 350 °C until 850 °C to preheat the steam in the Brayton cycle. Since solar thermal energy is introduced in the high

In Figure-21 we can find a plant scheme and thermodynamic cycle representation for this power plant.



technologies in order to have access to more R+D resources, and make itself a place in the future energy scenario.

In PFBC technology, a combined cycle power block with the gas turbine and the steam turbine working in parallel is implemented. The external thermal parallel coupling takes place in the pressurized fluidized bed combustor, where the fuel (coal or biomass) is burned to generate the thermal energy input to both cycles.

In this combined cycle, the gas turbine is relatively small (about 20 % of total installed power), and the needed gas turbine is of lower technology (turbine inlet temperature around 900 °C), compared with a conventional series GT+ST combined cycle. On the other hand, the Rankine cycle has less limitations than on a conventional series combined cycle, and therefore higher steam cycle efficiencies can be achieved. All of this can offer some advantages to solar thermal energy integration schemes:

1. Because of having both cycles in parallel, it is more easy to integrate solar energy without worsening the power block performances in only fuel operation mode.
When solar energy is available, less fuel is burned in the pressurized fluidized bed, and when no solar is available, more fuel is burned. The exergy efficiency in the use of fuel is to be kept rather constant with and without solar input.
2. Because of the higher contribution of steam turbine to total power, much higher solar fractions can be achieved than in conventional series GT+ST combined cycles. Moreover, since steam cycle efficiency can be higher in a PFBC, higher solar conversion efficiencies (η_7 and η_8 values) can be achieved.
3. The low turbine inlet temperature needed in the gas turbine (which besides needs no combustion chamber), makes it easier to incorporate heat exchangers for regeneration, intercooling and reheating, producing an advanced gas turbine cycle, since all the needed components are commercially available. That is, the gas turbine cycle can be very easily tailored to optimise solar energy integration keeping a good fuel conversion efficiency: Compressor intercooling can be performed with steam cycle condensed water, and the use of heat exchangers can ease the management of the variable solar input into the power plant.
4. The integration scheme can be very modular, allowing to implement different solar field sizes on a given power plant. This allows to ease the inclusion of a relatively small solar field on demonstration projects actually being implemented, or even to go increasing the size of the solar field along the life of the solar plant, reducing financing requirements.

9. Conclusions about interesting integration alternatives to be analysed

For conventional Rankine and GT+ST series combined cycle power plants, several guidelines have been proposed to produce good integration schemes. However, due to the different available alternatives, a comparative evaluation study where each concept is optimised using the appropriate integration evaluation parameter needs to be undertaken.

A same amount of solar thermal energy is converted into work at a higher efficiency (higher η_7 -value) when introduced into a higher η_1 -value cycle, independently from the temperature level at which solar energy is introduced. This conclusion, although seems obvious, can successfully be used to distinguish many bad integration schemes. Contrary to what it is often thought, in an integration scheme, the temperature at which solar energy is introduced has no direct effect on the η_7 -value, and high temperatures are always detrimental to the η_8 -value. Therefore it is better to integrate solar energy into the high pressure zone of the HRSG or boiler at low temperature and to superheat it with fuel energy, than to introduce solar energy straight into the intermediate pressure turbine, even if at higher solar temperature.

Besides the more common gas turbine / steam turbine series combined cycle plant or the Rankine power plants, there are other combined cycle plant schemes that present good perspectives for solar integration and therefore deserve a more detailed evaluation. The huge amount of interesting alternatives recommends to undertake an exhaustive comparative assessment using the same integration evaluation parameters in order to be able to have a strict comparison among the final results.

Among these other alternatives there are some new coal and biomass power plants that render themselves to good solar integration possibilities, opening the way to take advantage from the huge economic, financial, and technological resources that are being devoted to these technologies. This is the case of the PFBC. The repowering schemes, both the conventional series and the parallel repowering, also offer interesting possibilities for solar thermal energy integration.

The Kokhala-type combined GT+ST cycle with integration into the intercooled gas turbine, as well as the scheme based on integration both into the intercooled gas turbine and into HRSG, look interesting alternatives to be considered and compared with conventional ISCCS in terms of their respective η -values and ζ -values.

The integration scheme based on the combined GT+ST cycle with a regenerative and reheated gas turbine also offers interesting possibilities for an efficient use of both solar and fuel energy sources.

A comparative assessment from the most interesting integration schemes taking as a comparative basis the appropriate integration evaluation parameters would be a useful tool to better direct solar integration strategies, R+D resources and investments.

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