CASE STUDY WITH COMPARISON AND EVALUATION OF DIFFERENT ALTERNATIVES TO A COMBINED CYCLE POWER PLANT UPRATE

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Abstract. *Nowadays the improvement of energy efficiency is the objective of many industries that adopt self generation and the use of waste energy to increases the plant efficiency is a measure of great potential. Following this idea the uprate of open cycle gas turbine plants with a bottom steam cycle can dratically increasing the plant efficiency. However, the coupling of different cycles one of them already implemented requires a detailed power plant analyis and a study about load and operation requirements. The levels of pressure and temperature at the steam turbine, the number and type of boilers and turbines are parameters that should be evaluated to determine the better scenario.The subject is more complex when the cycle is designed to generate low electrical power, due the lack of technological alternatives for small power systems, like two pressure HRSGs.The aim of this work is present a study of different possibilities of a combined cycle for a small gas turbine power plant, based in performance and cost. The system adopted has 4 Siemens PGT 100 gas turbines, which works at 75% of load, in open cycle and the alternatives are evaluated in different aspects like efficiency, maximum power, flexibility, cost and reliability. To achieve a better solution, the number of and type of heat recover steam generators, number and type of steam turbines, pressure and temperature levels, will be changed.*

Keywords*: turbine, steam, CAPEX, generation.*

1. INTRODUCTION

The gas turbine is the most cost effective power generation equipment, when natural gas is available and the power need is from medium to large. Compared to the most common options, the use of alternative engines or steam rankine cycles, it presents almost the same efficiency, smaller footprint and smaller CAPEX.

The most known way to improve the efficiency is install a bottoming Rankine cycle. But adapt such system at a existent plant, composed of gas turbines only and not designed to operate in combined cycle, brings some additional challenges and requires some extra care.

The efficiency of a modern gas turbine goes from 32% to 42%. The most part of the wasted energy is the hot exhaust gases leaving the gas turbine and discharge by the stack. The HRSG (Heat Recovery Steam Generator) captures approximately two thirds of the gas turbine exhaust energy with the remaining third being lost in the boiler stack. Finally, around 20% of the fuel input is converted into power via the steam turbine with 1% lost in the turbine and 21% of the fuel energy lost in the condenser.

The combined gross power of gas and steam turbines equates to 55% (LHV) of the fuel energy in very well designed plants. Plant auxiliary's account for ~2% of the fuel input finally leaving 53% as net output combined cycle efficiency. Therefore the main justification for utilizing HRSGs within utility power plants lies in the clear gain of overall thermal efficiency.

The case presented at this paper explains the study performed, in a conceptual design level, to convert a power plant with four Siemens PGT-100, operating in most of the time at 75% of the load, in a combined cycle plant. The goal is reduce the MW cost, keep the reliability and flexibility of the plant, with low CAPEX and few increase at the OPEX.

2. THE DESIGN PROCEDURE

The design of the plant follows a general procedure to determine and evaluate the several design alternatives.

The initial phase of the project is always the field visit and information gathering, like the operation practice, the data sheets of the turbines, the fuel characteristics, and environmental conditions, as well as the layout of the plant and the available space for new machinery.

The second is the concept. It starts with a sensitivity analysis of some basic cases, to evaluate the main variables (mainly the steam pressure, temperature and condenser pressure) direction of influence, over the evaluation parameters (power, efficiency, water consumption). Some basic designs are chosen according the common practice and standard equipments available at the market. This step was performed using the software Thermoflow GTPro®. From the sensitivity analysis four designs are selected to a detailed evaluation, equipment selection, and quotation. The results are compared between them, using the customer requirements as the ranking parameters. The existent gas turbines performance data is presented at the Table 1.

Table 1: Test results Siemens PGT 100 4.9 MW running natural gas

Performance Parameter	100%	75%	50%	25%
COMPRESSOR				
Air mass flow kg/s	19.17	19.19	19.28	19.26
Pressure Ratio	13.44	12.77	12.07	11.28
TURBINE TEMPERATURE				
Indicated Operating C	535	447	356	277
Turbine Exit C	549	456	372	290
Exhaust Stack C	541	454	374	298
OUTPUT				
Shaft Speed rpm	1580	1581	1583	1581
Exhaust Flow kg/s	19.47	19.43	19.45	19.37
Zero Loss Output MW	5.157	3.887	2.644	1.349
FUEL - Natural Gas				
Fuel Heat Input MJ/s (MW)	16.45	13.13	10.1	7.15
Thermal Efficiency %	31.35	29.6	26.18	18.86

The preliminary results are used to choose the equipments that best fit the simulation conditions. After the supplier's data receive, the model is tuned with the actual performance of each equipment to provide a realistic result of the expected performance and price. The gas and steam turbines performance data was provided by Siemens, for the HRSGs by Rentech Boilers and for the OTSG by IST.

The year average environmental conditions are 18.5ºC, 66% of relative humidity and 101.6 kPa of pressure. The other conditions evaluate are the average winter conditions and the average summer conditions.

The alternatives can be evaluated in main requirements of the customer, in order of relative importance: CAPEX, output, reliability, efficiency, flexibility.

The main variables to discuss in this project that are related to the achievement of the desired performance and premises are the following: number of and type of SGs (steam generators); Number and type of STs (steam turbines); steam pressure level; steam temperature level; safety and control aspects; the design point.

There are some decisions that have to be made at the beginning of the dimensioning and selection of a cycle. The first is the design point, and the second the maximum desired output. The options for design point at this project are:

- Maximum GT Power: HRSG designed to the exhaust gas at maximum GT output ant the steam turbine to the maximum steam production;
- Most frequent GT operation (75% load): The HRSG is designed to receive the exhaust gases of the GTs at 75% output, and the steam turbine and generator to this amount of produced steam.

The first one leads to a plant with maximum power, cost and reliability, and reduced efficiency at daily operation. The GTs operation with part load causes a reduction of the steam production, what decreases the steam turbine generated power going also to an off-design condition. The second leads to a reduction in cost, efficiency, and maximum power, and some means have to be employed to ensure the safety of the system. For this case the problem is how the system will behave in an eventual operation of the GTs at 100% load, and if is possible to avoid the risk of the steam loop failure. The second option was the chosen one to provide a low investment option, by the customer requirement.

But, in the eventual full power GT operation, the steam pressure will be increased, as well as the steam temperature, but the ST, and the generator, being designed to a smaller power won't be able to withstand. The steam turbine inlet control valve will have to operate to reduce the flow at the ST entrance to protect the machine.

In a regular boiler, in such cases the burners load is reduced to match the new condition. In an HRSG it's usually not possible, at least, not without the restriction of the GT power. If the steam consumption at the steam turbine is reduced without a reduction in the HRSG thermal load, the pressure at the HRSG will continuously raise until the safety valves acts to protect the evaporator from an overpressure, followed by the trip of the boiler. Such thermal input regulation without the GT load reduction could be achieved only by means of a diverter damper.

With the GTs at full load the diverter damper redirects part of the flow to the by-pass stack limiting the thermal load at the HRSG, but wasting some energy. If the adoption of a by-pass stack with a diverter damper is required there are some checks that have to be made at the design, because some particular choices of the heat exchangers area, operating at higher temperature, can lead to water boiling at the economizer, what can cause vibration and fatigue of the piping. The regulation dampers are subject to failures that reduces the plant reliability, and are relatively costly.

There are some technological choices that can improve the operation reliability without drastic increase in cost. To avoid the need for the by-pass diverters can be used an OTSG. The OTSG (Once-through steam generator) is a particular kind of design where the need for a steam drum is eliminated, what brings several advantages that will be discussed later. Another hardware that can limit the maximum steam flowrate at the ST is the by-pass valve that is assembled close to the steam turbine main stop valve and is regulated to divert to the condenser any amount of steam that exceeds the maximum that is allowed by the turbine. These two technologies together can provide an alternative to the use of regulating by-pass stacks.

3. EQUIPMENTS AND CYCLE

The gas turbine HRSG is essentially a counterflow heat exchanger consisting of a series of superheater, evaporator, and economizer sections arranged from the gas inlet to the gas outlet in such a way that the heat recovery is optimized and the area reduced.

The most important design aspects, as in any heat exchanger, are the temperature differences that influence the amount of heat transfer surface area: the pinch point and the economizer and superheater approach temperatures.

A small pinch point and superheater approach temperatures correspond to a lower temperature difference between flue gas and the steam within the exchanger tubing. As a result of these smaller temperature differences the required surface area is much greater in order to produce the same heat transfer. The consequence is an increase in the capital cost and backpressure, while an increase in the pinch point reduces the heat recovery and the HRSG efficiency (typical values of pinch point and superheater approach point are between 15 to 30 K and 20 to 35 K).

It's interesting to extend the discussion about the significance of the pinch and approach points in gas turbine HRSG's design.

Unlike fired steam generators, where the inlet gas temperatures are very high, namely adiabatic combustion temperatures of the fuel fired (1750-1850 ºC), the gas turbine exhaust gas temperature at HRSG inlet is much lower, in this case of the order of 450ºC – 500ºC..

The HRSG stack gas temperature cannot arbitrarily select to determine the steam flow In fired steam generators for example, if the feed water temperature is say 110ºC, an exit gas temperature of 130 to 150 ºC is feasible. A lot of energy is transferred to the steam before the flue gases enter the economizer, while in an HRSG, it could be very small due to the low inlet gas temperature. This in turn affects the energy absorbed in economizer and hence the HRSG exit gas temperature.

At the Figure 1 can be observed the temperature profile of a typical single pressure HRSG. The exhaust gas temperature line can be drawn with 2 points: the inlet gas temperature and the evaporator temperature and the pinch temperature. The outlet gas temperature can be obtained selecting the expected inlet water temperature (from the deaerator). In the case presented, a recover of 100MW is achieved considering a 60ºC feedwater.

However, with a good choice of 2 pressure levels at the HRSG (two drums) with the same gas temperature can be obtained a better heat recover, and a much lower outlet gas temperature. The recover reaches almost 130MW, 30% of increase.

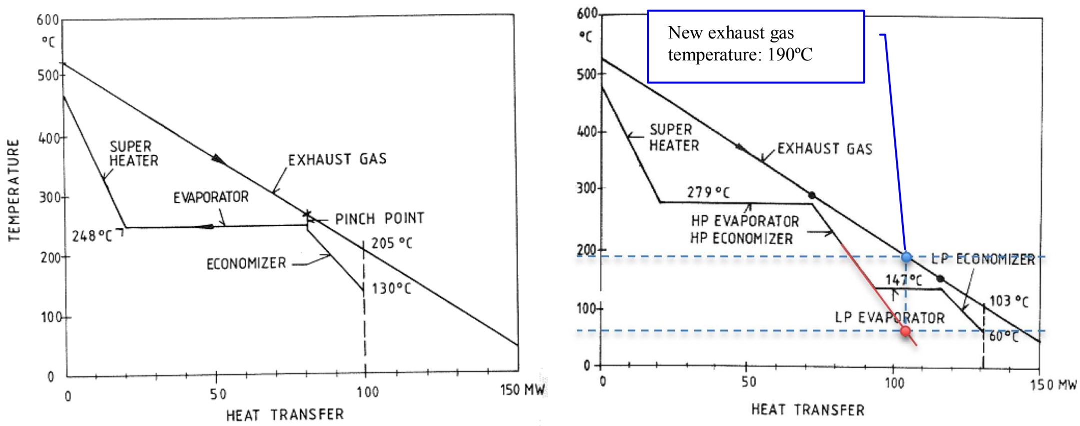

Figure 1: Gas-steam temperature relationship for single pressure (left) and dual pressure (right) (Kehlhofer,1997)

The Figure 2 shows are particular cases of two possible HRSGs tested for the GTs exhaust conditions of this project. The Figure 2 shows a 70 bar HRSG, and a 35 bar HRSG. Note the hot 250ºC stack gas temperature at the 70bar model while the 35bar HRSG shows a more acceptable 210ºC. This temperature difference means more heat recover with the 35bar boiler (a bigger steam production).

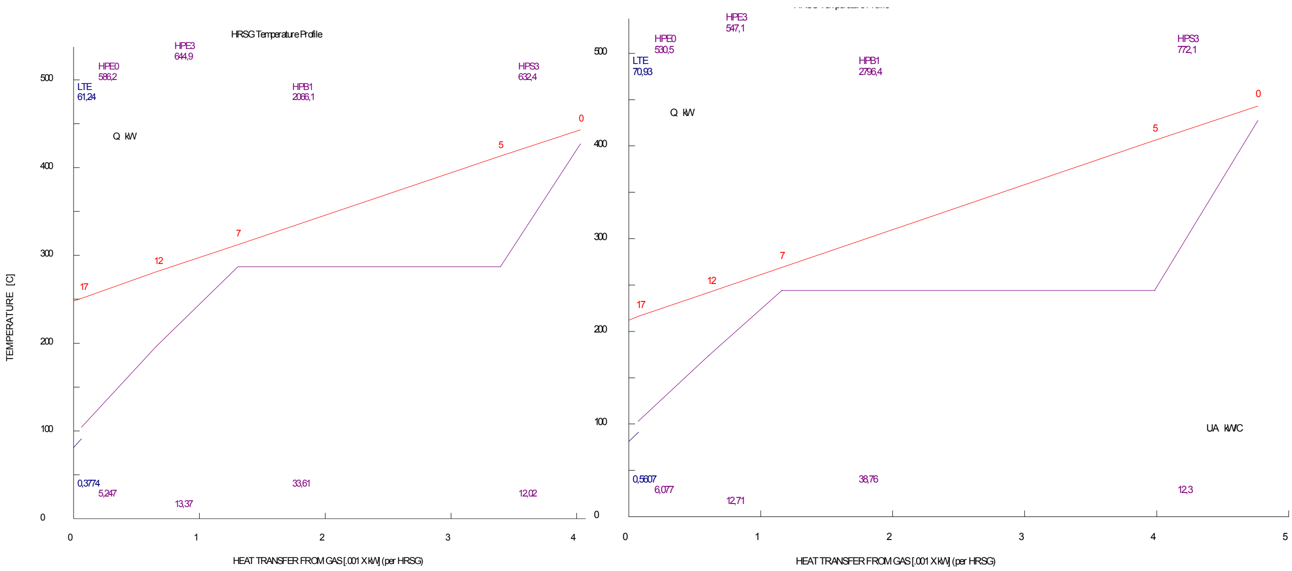

Figure 2: Gas-steam temperature relationship for a single pressure 70bar HRSG for this project, with a pinch of 25ºC and 35 bar, and the same pinch point.

If an exit gas temperature is arbitrarily assume, there is the risk to the designer of going into either an unfeasible design or a steaming economizer (water boiling at the economizer) at off design operation. Hence the right way is to define, as explained, the steam generation and gas/steam profiles using pinch and approach points.

Pinch and approach temperatures have to be selected in the unfired operation mode (even if the HRSG operates in fired mode) and preferably at the lowest inlet gas temperature to avoid steaming in the economizer. The pinch and approach points are dependent on the inlet gas temperature and whether the evaporator is constructed with bare tubes or finned.

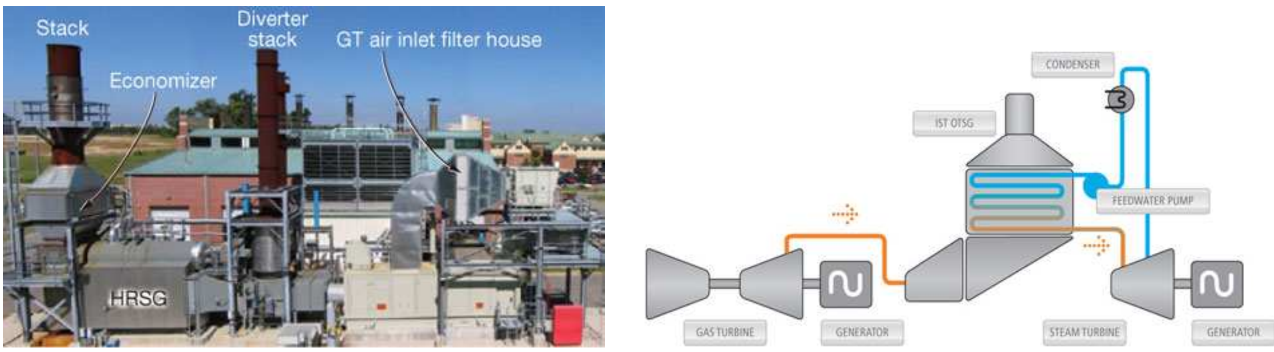

Figure 3: HRSG typical arrangement (Rentech Boilers) for a Solar Taurus 60 and the steam path diagram of an OTSG (IST)

In circulating boilers the steam drum is the beginning and the end of the evaporation process. Water is fed to the drum, and saturated steam is separated from the wet mixture returning from the boiler tubes to the drum. The steam production is constrained to occur in the boiler tubes.

One of the studied technological options is de adoption of OTSGs instead of the HRSGs. In the OTSG (Figure 3) there is no steam drum to separate the phases. No physical mechanism constrains the location of the boiling zone along the heat transfer path. The active boiling zone can move, upstream or downstream as operating conditions are varied.

The main supplier of the OTSGs has been Innovative Steam Technologies (IST), in Canada, who has around 70 units installed worldwide. Also, Siemens have developed some applications of this technology (sometimes called Benson Boiler) for large sized power plants. The main advantages of IST's OTSG technology are perceived as being: the absence of a by-pass stack, because the unit can run dry; smaller weight, that reduces the thermal stresses and allow a fast start-up after a shut-down and the absence of a deaerator that saves some steam consumption.

A reduction in the steam generation equipment price could be achieved by the use of one HRSG for several GTs, but the references in the market shows some difficulties related to operation in such arrangement. As presented by Bell and Nitzke (2003) the efficiency and maintenance of a HRSG is dependent of a homogeneous gas distribution over the tubing. It is possible to design a system that provides good gas distribution with all gas turbines operating at the same load. But the opposite is also true, being very difficult to achieve such flow smoothness if one of the GTs fails. In this case, the most of the times the practice is the HRSG shut-down.

The feedwater loop source also impacts the cycle efficiency. The key is to bring the feedwater in contact with the gas stream at a temperature as low as possible to get the most heat out of the gas stream, before it flows away by the stack. But there are two reasons for using some feedwater heating, that increases the water temperature before enters the HRSG. The first is to deaerate the steam. The second is to prevent corrosion due to condensation on the exterior of cold feedwater tubes in the HRSG.

Air inleakage into steam cycles is almost impossible to prevent. The major problem area is the condenser, which operates substantially below atmospheric pressure. Air also enters the steam cycle with makeup water. The main problem with air is that the O2, when present in the superheated steam (i.e., under high temperatures), becomes corrosive. To remove this air, a small amount of steam is sometimes extracted partway through the turbine. This is added to the feedwater to just bring it to the boiling point (i.e. a saturated liquid) at the deaerator. But the simpler arrangement is deliver to the HRSG only deaerated water. The cold condensate will be heated, into a separated loop, at a water-water heat exchanger, by the deaerated feedwater coming from the deaerator, before it enters the HRSG. It will reduce the temperature of the deaerated water and heat the cold condensate closer to the deaerator temperature (what reduces the consumption of saturated water or steam). An external compact plate and frame heat exchanger can be made from corrosion resistant materials at modest cost. Thermodynamically the energy to heat the feedwater still comes from the HRSG gases, because the deaerated water is cooled before it enters the high temperature economizer.

The deaerator heat source can be low pressure steam extracted from the ST, saturated water form the high pressure economizer or. The one that most reduces piping is the second, saturated water, and will be considered the standard choice.

3.1 The steam power extraction

The steam turbines (ST) can be divided as action or reaction, depending of how the blades extract the energy from the steam, and can be further divided as single or multistage turbines. The STs applied in power plants that have no integration with chemical process plants shall to be of condensing type without extraction.

There are some features that can improve the thermodynamic efficiency of the cycle and other that raise the "internal efficiency" of the turbine. The internal efficiency is evaluated considering the internal losses at the nozzles, buckets, sealing system, and all other losses related with some constructive aspect of the ST. The main source of energy loss is without doubt the latent heat at the condensing of low pressure outlet steam, to saturated water, that accounts for an average of 50% of the steam heat at the ST inlet.

Other losses like the exhaust hood, last stage buckets aerodynamic losses and leakages at the shaft ends sealing are related to the turbine design of each manufacturer, and reflected at the well now isentropic efficiency. This losses usually represents 9% to 15% of the ST heat input. The other aspects and in this case the most important is the influence of the steam inlet and outlet pressure and temperature onto the turbine performance.

The steam inlet condition are one of the key parameters consider at the steam turbine choice. The steam density and the enthalpy are proportional with the pressure and temperature. The specific volume influences the cross sectional area of the stages, with direct consequence on the length of the buckets and nozzles. The increase in the machine size impacts the efficiency. Since the stages are all designed for similar leakage control and minimum clearance area (the space between the nozzles and the shaft or the rotating blades and the shell), the leakages around larger stages will have less of an effect, as compared with smaller stages.

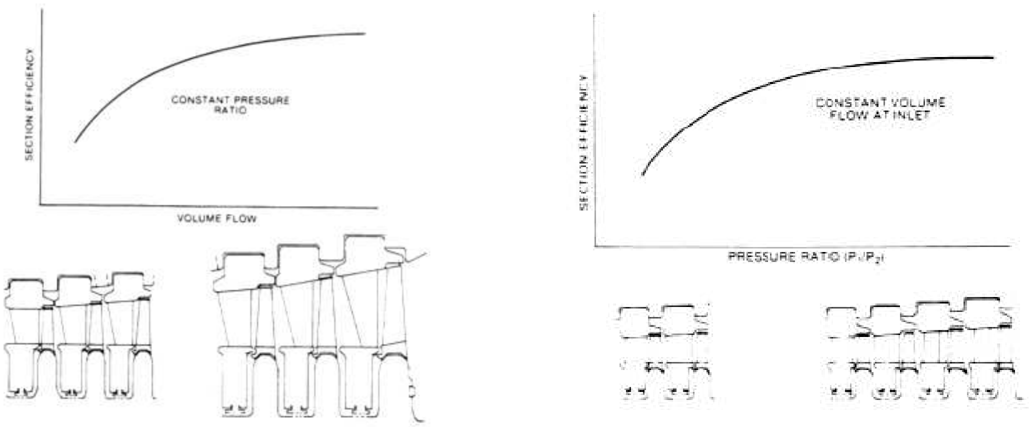

Figure 4: Effect of the volumetric flow rate on the stage efficiency. At the left, the increase in the volume flow to a fixed temperature, and at the right: the expansion ratio increase to a defined volumetric flow. Consider P2 as the condenser pressure (fixed) and P1 the inlet pressure steam.

The leakage of steam outside the steam path is a major concern of the turbine design. Any steam that flow out of the steam path is not going to produce power. The pressure has a similar effect, once it has the strongest impact on the steam density. As big the pressure, smaller will be the steam volumetric flowrate. But, as can be expected, between two machines designed to the same steam production, the one with the higher pressure will be probably the smaller.

Also leakage at the shaft ends sealing packages, in the modern large sized STs, can be used in the stages with compatible pressure, but in small STs where the extra piping and complexity is not justified by the small amount of recovered energy, it is directed to the condenser or vented. The choice of a separated gland steam condenser is the most adequate for this case.

There are losses that the steam incurs while flowing through the exhaust hood and into the condenser which the stage efficiency dos not account for. These losses are called exhaust losses and they are essentially a function of the velocity of the steam leaving the last stage.

An exhaust curve loss is provided for each last stage bucket and is composed of some components: the leaving loss; the annulus restriction loss; the gross hood loss; and the turn-up loss. The leaving loss is the kinetic energy of the steam as it leaves the last stage. The annulus restriction occurs when the pressure ratio of the last stage is large enough to cause the axial steam velocity to reach the sonic. When it happens any increase in the pressure ratio (by means of the upstream pressure increase or condenser pressure decrease) will not change the effective stage pressure ratio, and as a consequence, there will be no increase of the produced work. The gross head loss is that pressure drop across the condenser tubing, less the pressure recover at the turbine exhaust diffuser

If the pressure ratio across the last stage is reduced (like due the increase at the condenser pressure) the axial velocity will be lower. The ratio of the bucket to stream velocity (the so called stage velocity) can become very high, resulting in efficiency loss. At some point, the low axial velocity will cause the blades to no longer produce power, but absorb power, being driven by the upstream stages. This is the called turn-up loss. The last stage efficiency have a strong impact over the turbine efficiency and power, and it is in great instance defined by the condenser vacuum, that in turn is dependent of the cooling water temperature and vacuum steam ejectors.

3.2 The variables to choose the operating point

As is very well discussed and explained with the aid of the first and second law of thermodynamics, the increase in the steam temperature raises the cycle efficiency. At a thermodynamic point of look this can be explained simply by the increase of the average temperature of the hot energy reservoir. Speaking about a cycle with real world equipment, once the input heat at the boiler is constant, the increase of the steam temperature reduces the steam production. But, the ratio of the absorbed heat and rejected heat increases, thus, increasing the cycle efficiency. The steam pressure raise causes a similar effect at the cycle efficiency increasing the efficiency.

The condenser pressure is the third operational variable that can be managed to improve the cycle efficiency. A lower pressure allows a reduced temperature at the outlet steam. Thermodynamically it can be considered a reduction of the cold reservoir temperature. The reduction of the low temperature energy sink widens the temperature interval raising the efficiency.

Note that the steam outlet lies in the multiphase region, what means that both saturated steam and some saturated water are leaving the last stage of the steam turbine. This amount of condensate has to be limited to not more than 10%. Values larger than this can cause damage of the buckets as well as the reduction of the internal efficiency of the turbine.

The outlet pressure defines the length of the last stage (related to the volumetric flowrate). To avoid a larger turbine frame the vacuum must be defined at a reasonable value, to be defined to each project. In practice, the increase of the ST frame size raises the cost. This raise in cost must be justified by the increase in the output.

The most used method to improve a Rankine cycle performance is the steam reheat at intermediate pressure levels. This measure increases both the HRSG/boiler efficiency (as already explained) as well as the ST efficiency but requires a two pressure level HRSG, unavailable for machines of this small with a maximum steam production up to 10 t/h of steam. Must be highlighted that the ST also will be a double pressure model, more expensive and of complex control.

The selection of the operating conditions depends also of the constraints imposed by the environment and the existent gas turbines. The exhaust temperature of the GTs at the defined operating range by the customer, from 75% to 80% of the rated maximum power, may vary from 435ºC to 477ºC, that is considered low for combined cycle applications. Due this restriction and considering that this power plant main goal is the reliability, and is subjected to cycling operation, the design of a combined cycle in not a simple task.

As a characteristic of single shaft gas turbines for power generation, the air mass flow rate is constant despite the load. The increase of the generated power is a function only of the fuel flow rate. Related to the power there is only the exhaust gas temperature change. Considering this exhaust temperature range the maximum attainable steam temperature, adopting typical approach temperatures at the superheater is 430ºC.

As already explained the operation with the maximum steam temperature is a clear benefit to the efficiency of the steam cycle. Also, as big the temperature smaller is the steam flowrate, what benefits also the cost, once the gas turbine and condenser have to be designed to a smaller volumetric flow.

The operating temperature would be set as a free function of the steam produced at the HRSG, but it would impose a metal temperature variation that is not beneficial to the ST from the mechanical point of look. For this reason is considered the use of an atemperator to keep the temperature constant over the upper part of the operating range (from 75% load, at the winter, to 80% load at the summer). A maximum temperature of 420ºC at the steam side and 477ºC at the gas side, in regular operating conditions keeps the tubing and piping material requirements at the standard of the market, and helps to keep the price of the HRSG and piping at a reasonable value.

The steam pressure also has an influence both in cost and efficiency. At this particular project, this is the most important choice to be made.

At the HRSG section, the means to improve a steam generator efficiency that operates with the relatively low temperature of this GTs exhaust gas is include another pressure level (already discarded) or reducing the steam pressure. The feasible option is the reduction in the steam pressure. The preliminary simulations attested that there is a raise in the net power output with the reduction of the pressure until 15 bar. If one considers only the energy recovered, that would be the best choice, but the reduction of the steam pressure also impacts the cost.

The most part of the standard small STs at the market are designed to the maximum pressure of 45 bar and temperature of 450C. It means that an operating pressure of 15 bar do not means a reduction in the ST price. The shell will be designed to 45 bar anyway. In fact, is possible that it means an increase in price. This can happens due the fact that the turbine will need a bigger cross sectional area at the first stages, and inlet nozzle due the larger volumetric flowrate (due the steam less density). Between two standard types STs rated to the same power, the one operating with a larger steam flowrate with the lower pressure may have to be of a superior shell frame size.

The best economical point depends of several information's that are proprietary of the manufactures, and for this reason, it is not safe to go to far from the market common practice. The Figure 5 shows the results of an extensive research performed by Fridh to study the tendency of the nowadays and future STs operating conditions.

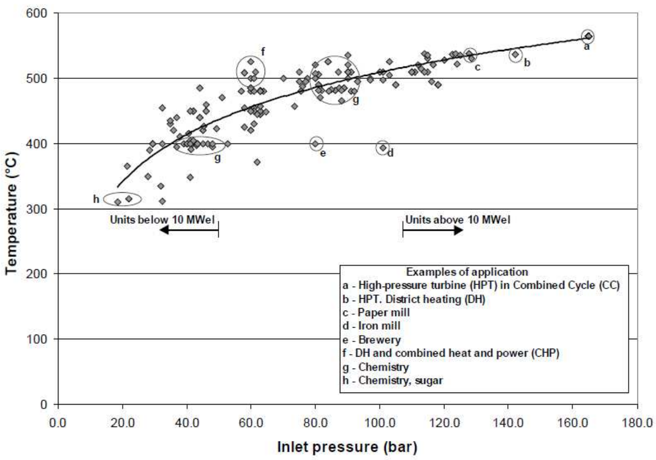

Figure 5: references from 6 manufacturers, globally spread, on ordered turbines between the years 1990 and 2001 (~600 units) in the range of approximately 1 to 25 MWel (Fridh, 2001)

Clearly there is a strong concentration of the machines below 10 MW at the range of 40 bar to 50 bar with temperatures from 400ºC to 450ºC. There are few machines of 15 bar and the few ones operating with lower steam temperatures.

The first machines operating with a steam temperature at the desired value starts at 25 bar. To follow the market practice the boundaries of this study are restricted to this 2 borders, the minimum at 25 bar, and the most common at 45 bar. Must be highlighted that the Figure 5 shows machines used at CHP (combined heat and power) plants where the steam generated by regular boilers, which operates without the low temperature inlet gas that is imposed to an HRSGs.

Once is defined the operating pressures and temperature, the next definition is the condensing pressure. Typical values of condenser pressure are in the range of 0.05 to 0.1 bar, with condensing temperatures of 32ºC and 44ºC respectively.

The size of the steam condenser is function of the difference between, the cooling water temperature, and the steam temperature. The cooling water temperature is defined by the cooling tower. The first dimensioning, using wet cooling towers, revealed a need for a make-up water of 450 m3/day, much bigger than the available at the site (located in dry climate). The choice, then, is the use of air coolers (also called dry cooling tower). The minimum temperature of the cooling water becomes a function of the air temperature for the different seasons.

The average summer temperature of almost 18ºC allows a minimum cooling water temperature of 28ºC with an approach temperature of 10ºC. Considering 28ºC of cooling water, the steam maximum temperature at the condenser falls in the range of 38ºC. Condensing of the steam with lower temperature difference to the cooling water could be achieved with the increase in the condenser area. Considering that the operation at the summer occurs at a limited amount of time, and the amount of power that can be reached with the reduction of the steam temperature from 38ºC to, lets say, 32ºC would cause, is hardly justifiable the oversizing of the air coolers and condenser. So, the condenser pressure will be kept at the somewhat conservative value of 0.1 bar. In practice the system is designed to average annual conditions, and not to the most restringing condition, the summer. In this period the steam turbine will operate with some reduction of output.

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3.4 Results

The first part of the design has begun with the simulation of a standard machine operating with several pressure levels. It was not considered the equipment specific performance data of the manufactures, but simulations with a typical accuracy of 5%. For the cost estimates, the absolute values need some adjustments, but the relative difference between each design is trustable.

The Figure 6 shows an efficiency increase of 1% using 25 bar instead of 45bar. The increase of the efficiency means a raise in the net generated power, of around 400 kW, or more less 7.5%. It was observed that the efficiency increases with the decline of the pressure until 15 bar due the best energy recover at the HRSG.

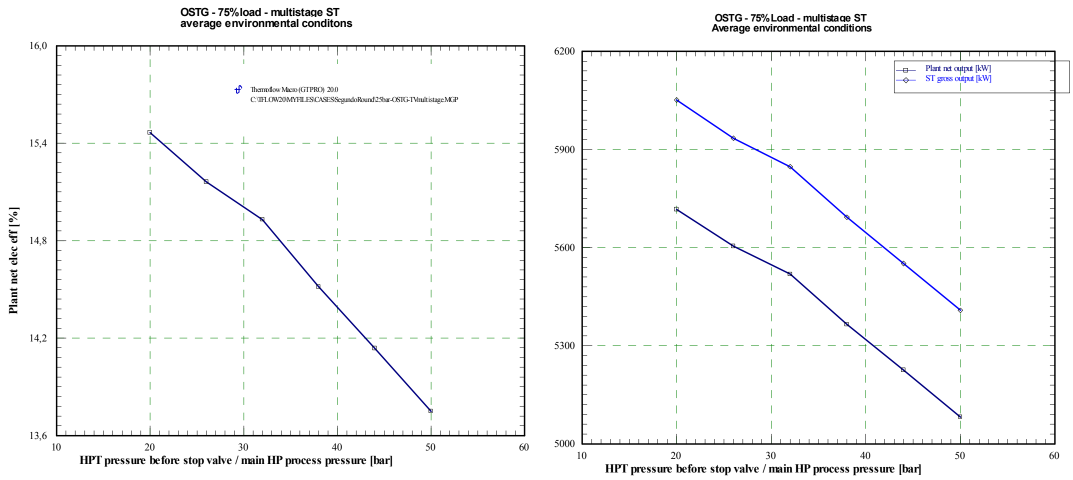

Figure 6: ST electrical efficiency and output change with steam pressure.

Similar simulations were performed considering the equipment cost and the specific energy cost (USD/kW). As expected the equipment cost is higher with the reduction of the pressure, due the raise in the steam volumetric flowrate. The difference between the two options is around 4%.

At the balance, the difference of the net generated power is bigger than the cost difference, in opposite directions. The final specific Investment cost (USD/kW) of the 25 bar option is 10% lower than the 45 bar.

The achieved exhaust gas temperature at the HRSG stack shows the main reason of the difference of output when the pressure is changed. Comparing the options, the one with 25 bar, reaches an average production of 26 t/h of 435 C steam with and HRSG exhaust temperature of 186 ºC, while when using a steam generation of 45 bar, the stack temperature rises to 207 ºC and the steam production is reduced.

As an example of one of the evaluations performed, using a 2 pressure levels boiler (10 bar and 45 bar) is possible to achieve until 6.4 MW, with a stack gas temperature of 135 ºC. The price to pay for this 400kW of extra power is the increased CAPEX and operational complexity.

After this preliminary comparison, four cases were selected to be studied in details, including the equipment selection and quotation with the main suppliers. The Table 2 summarizes the results.

The CASE 1 provides the lowest CAPEX, but uses a single stage steam Siemens turbine type SST-060. This kind of turbine is unable to generate vacuum at the condenser that have to operate at a pressure close to the atmospheric. The hotter outlet steam (around 100ºC) also provides an economy due the less area requirement at the condenser, but due the low Rankine cycle efficiency the specific cost is the worst between all the options. The main advantage is the low thermal inertia, what allows the machine to be started and shut-down fast without major concerns.

The CASE 2 uses an Siemens machine model SST-110. This is a two stage condensing machine. In fact, consists of 2 single stage machines in series arrangement (looking at the steam path). Both machines are coupled to the same gearbox, attached to the generator. This machine generates vacuum at the condenser (0.1bar) providing a good efficiency and keep the advantage of a low thermal inertia of the machine adopted at the CASE 1.

At the CASE is employed a standard 25bar Siemens SST-300 turbine together with the 4 OTSGs. The choice for an OTSG is related to the better efficiency and to reach a condition of better reliability. There is no diverter damper and stack, and the safety of the ST is provided by a steam by-pass valve that diverts the any excess of steam production to the condenser, in case of the GTs operation at full load. Due the reduction in the ST outlet pressure the condenser and cooling system have the larger dimensions and capacity. Also, the condensing and cooling systems have to be designed to the worst situation, when there is some HP steam being diverted to it. The OTSG have no deaerator, reducing some pipework. Also, this kind of steam generator needs no blowdown what improves the overall efficiency. The extra power generation can reach 1 MW compared to the CASE 2, being the most efficient of the alternatives. Due the bigger volumetric flow of steam both the OTSG and the ST are more expansive than the 45 bar equipment, as well as the pipework that need a larger diameter.

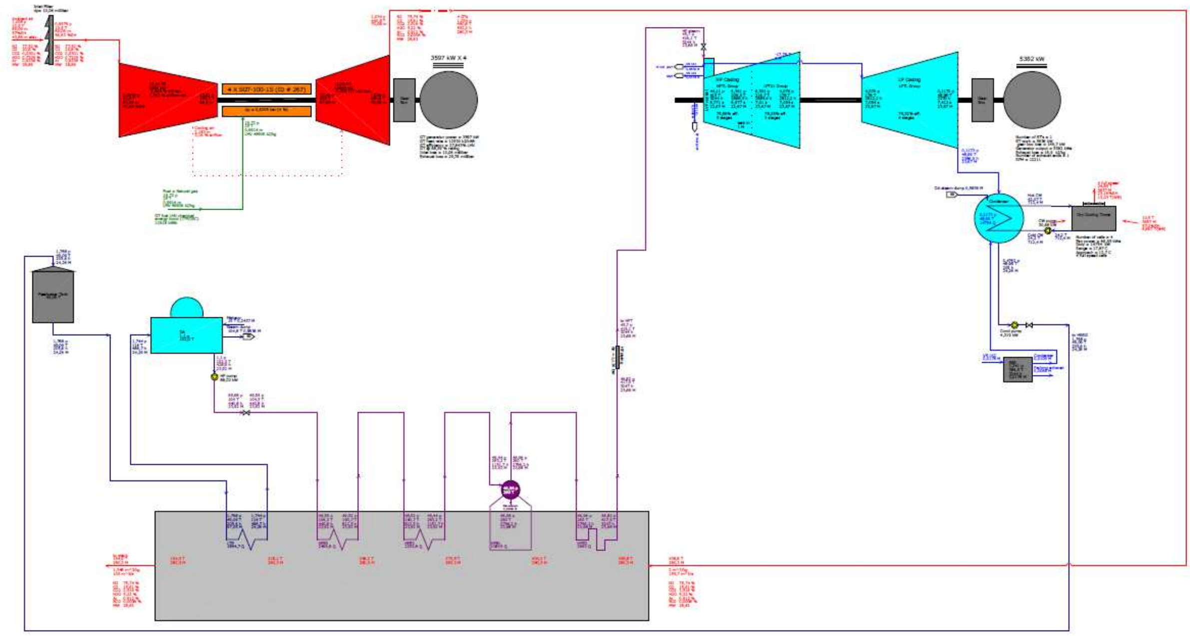

Figure 7: The CASE 2 mass and energy balance

Table 2. . Summary of CASES characteristics

	CASE 1	CASE 2	CASE 3	CASE 4	
Steam Generator	ton/h HRSGs with by-pass stack and diverter damper. The HRSG is composed of HPE, HPB and HPS.	ton/h HRSGs with by-pass stack and diverter damper. The HRSG is composed of LTE; HPE, HPB and HPS.	· 4x 25bar, 6.5 ton/h OTSG with**out** by-pass stack. The OTSG is composed of LTE; HPE, HPB and HPS.	ton/h HRSGs with by-pass stack and diverter damper. The HRSG is composed of LTE; HPE, HPB and HPS.	
Steam Turbine	· 1 x 3.5 MW single stage steam turbine with arc admission valves, 15000 rpm;	double casing, twin arrangement steam turbine with one set arc admission valves for casing, 15000 rpm;	· 1 x 6.6 MW multistage condensing, steam turbine with arc admission valves, 8300 rpm;	· 1 x 6.5 MW multistage condensing, steam turbine with arc admission valves, 12000 rpm;	
Accessories	· Gearbox 10:1 rpm, and a 6.3kV 4 pole air cooled generator.	· 1 Gearbox with two reduction rates and a 6.3kV 4 pole air cooled generator.	with 5.5:1 reduction rates and a 10 kV 4 pole air cooled generator.	with two reduction rates and a 6.3kV 4 pole air cooled generator.	
Condenser	· 1 x 540m2 shell and tube water condenser;	shell and tube water condenser with steam ejector vacuum system;	shell and tube water condenser with steam ejector vacuum system;	shell and tube water condenser with steam ejector vacuum system;	
Deaerator	· 1 deaerator;	· 1 deaerator with HPE water as heating source;	with HP steam extraction as heating source;	· 1 deaerator with HPE water as heating source;	
Air cooler	cooler;	cooler;	cooler;	cooler;	
PERFORMANCE CARACTHERISTICS					
GT Power	75%	75%	75%	75%	
Net Power ST	3002	4896	5882	5356	kW
Rankine cycle Eff.	15,8	24,1	27,9	26,4	%
Combined Cycle eff	39%	43%	44%	44,5&	%
FINANCIAL INFORMATION					
Total - Owner's Cost	100%	112%	135%	125%	
Net Plant Output	3,5	5,7	6,6	6,2	
Cost per kW - Owner's	100%	69%	71%	71%	

HPE: High pressure economizer; HPB: High pressure boiler; HPS:high pressure superheater; LET: Low temperature economizer

The CASE 4 is designed to use a solution is also proposed by Siemens, the model SST-200. This is a multistage condensing machine of classic design. In some loads, this turbine can accept more vacuum in the condenser, benefiting the part load performance. The condenser, by these reasons is a little more expensive due the smaller temperature difference between the steam and the cooling water.

In common, the solutions proposed at the CASES 1, 2 and 4, have a diverter damper that have to be kept partially closed if the GTs operates for some reason in part load.

The data for the full range of operation was evaluated and the same relative differences of performance could be seen to GT load. The environmental conditions can change the order of preference if one evaluate only the efficiency.

4. CONCLUSIONS

From a technical point of look, and considering only the investment cost the alternatives can be classified as follows:

- To provide the lowest investment and minimum output: CASE 1
- To provide the best specific investment and acceptable output: CASE 2
- To provide more output: CASE 3
- An alternative of larger output and better cost: CASE 4

In terms of reliability and simplicity of operation, the cases can be graded as follows: CASE 3, CASE 2, CASE 1; CASE 4.

The ratio of price per kW is higher that the typical that can be achieved in new power plants (Figure 8). But, this plant does not increase the fuel compsumption. It’s an investment entirely based in energy recover.

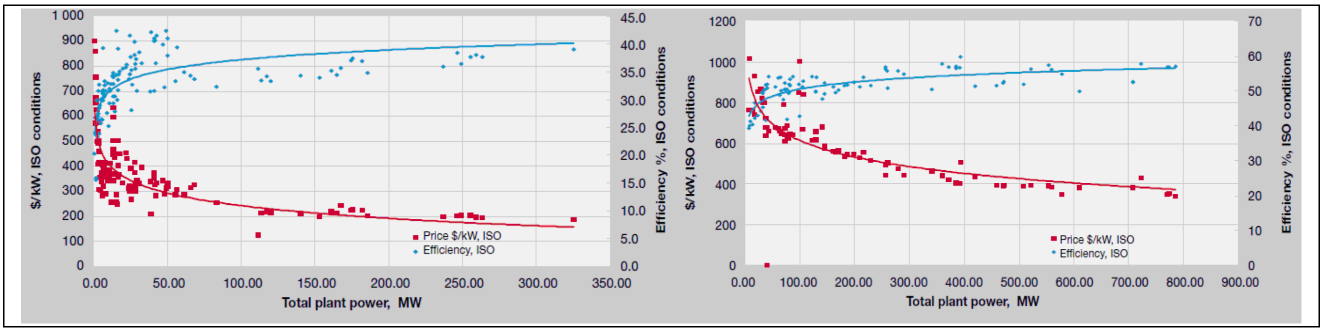

Figure 8. Charts of US$/kW for several power plant outputs, simple cycle (left) plants and combined cycle plants (right), always at seal level and 15ºC (Gas Turbine World 2001-2002 Handbook)

The overall efficiency of the plant, as well, is below the regular achievable values, but have to considered that this is due the low temperature gases of the GT exhaust, caused by the preference for operation in partial load, and the difficulty to implement the widely adopted means to increase the efficiency at the large sized Combined Cycle power plants (several pressure levels with reheat and control of the GT exhaust temperature).

Due the increase in the backpressure, there will be an increase between 0.5 and 1% at the GTs fuel consumption to keep the same output with only the GTs in operation.

The differences in the CAPEX evaluation obtained from the simulation software (GTPro) and the obtained by the selection and quotation of the main equipments is in the range of 10%, with the software providing higher price values.

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