THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

97-GT-39



# INTERCOOLED ADVANCED GAS TURBINES IN COAL GASIFICATION PLANTS, WITH COMBINED OR "HAT" POWER CYCLE

BREAK

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## ABSTRACT

Due to their high efficiency and flexibility, aeroderivative gas turbines were often considered as a development basis for intercooled engines, thus providing better efficiency and larger power output. Those machines, originally studied for natural gas, are here considered as the power section of gasification plants for coal and heavy fuels. This paper investigates the matching between intercooled gas turbine, in complex cycle configurations including combined and HAT cycles, and coal gasification processes based on entrained-bed gasifiers, with syngas cooling accomplished by steam production or by full water-quench. In this frame, a good level of integration can be found (i.e. re-use of intercooler heat, availability of cool, pressurized air for feeding air separation units, etc.) to enhance overall conversion efficiency and to reduce capital cost. Thermodynamic aspects of the proposed systems are investigated, to provide an efficiency assessment, in comparison with more conventional IGCC plants based on heavy-duty gas turbines. The results outline that elevated conversion efficiencies can be achieved by moderate-size intercooled gas turbines in combined cycle, while the HAT configuration presents critical development problems. On the basis of a preliminary cost assessment, cost of electricity produced is lower than the one obtained by heavy-duty machines of comparable size.

## NOMENCLATURE

| | |
|---|---|
| G | mass flow, kg/s |
| HHV | higher heating value, J/kg |
| LHV | lower heating value, J/kg |
| p | pressure, Pa |
| $p_{gas}$ | gasification pressure, bar |
| P | power output, MW |
| T | temperature, °C |
| TIT | total turbine rotor inlet temperature, °C |
| $\beta$ | pressure ratio |
| $\Delta i_{is}$ | isentropic enthalpy drop, J/kg |
| $\eta$ | net cycle LHV efficiency |
| $\eta_p$ | polytropic efficiency |

Acronyms

| | |
|---|---|
| AD | aero-derivative gas turbine |
| ASU | air separation unit |
| CC | combined cycle |
| HAT | "humid air turbine" cycle |
| HD | heavy-duty gas turbine |
| HRSG | heat recovery steam generator |
| IGCC | integrated gasification combined cycle |
| IGHAT | integrated gasification HAT |
| LP,IP,HP | low, intermediate, high pressure |
| SH,RH | steam superheating, reheating |

## 1. INTRODUCTION

Aero-derivative gas turbines are widely used in the power industry, representing a prime choice especially for independent power producers. One of their most attractive feature is the unsurpassed efficiency: recent models to be introduced into the market, like the GE LM6000 PC, the Rolls Royce Trent, the Pratt & Whitney PW4000, show simple cycle efficiency exceeding 40% in the 40 to 50 $MW_{el}$ power range.

Using those engines as a development basis, interesting projects were carried out to further increase their efficiency and output by the addition of intercooling. The aim of those studies is a reduction of specific plant and operating costs, made possible by the utilization of the same engine core, rendering aero-derivative gas turbines a viable alternative to heavy-duty machines in the utilities power range. This subject has been deeply investigated by the CAGT group (Cohn et al, 1993 and 1994), sponsored by various utilities, gas companies and R&D organizations.

The influence of intercooling has been discussed by various

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Orlando, Florida — June 2-June 5, 1997

authors (Horner, 1989, Rao et al.,1991, Day and Rao, 1992, Chiesa et al.,1995). Just to remind the basic features, we can summarize that intercooling makes possible: (i) a "supercharging" of an existing engine, blowing compressed and cooled air into a previously designed HP compressor section, allowing an higher pressure ratio;; (ii) a reduction of the compression work (iii) a reduction of the temperature of air used for blade cooling, thus allowing higher turbine inlet temperature by keeping the same blade metal temperature and cooling technology. With a proper combination of those effects, a large increase of power output and an higher efficiency are therefore possible. Recent CAGT studies quote a simple cycle efficiency up to 46% and a power output of about 120 $MW_{el}$, by using a 40 $MW_{el}$ machine as a development basis.

It must be noted that aero-derivative machines are candidate to be the core engine of complex and innovative cycle configuration, in addition to the well-known practice of the combined cycle with heat recovery steam plant. A variety of those cycle has been studied in a previous paper (Chiesa et al.,1995), recognizing that the HAT cycle (Humid Air Turbine, firstly proposed by Rao, 1989) is the only solution with an efficiency matching the one of combined cycles. The rationale behind the HAT will be recalled later: it is now sufficient to say that the possibility of its development, potentially leading to efficiencies beyond 55%, is strictly connected to the one of intercooled machines (Day and Rao, 1992). The solution proposed by Nakhamkin et al.,1995, consisting of a topping HP turbomachinery train added to an heavy-duty, is quoted for a lower efficiency.

However, all those studies were devoted to the use of natural gas. The aim of this paper is to investigate the potential of intercooled aero-derivative machines in conjunction with coal gasification processes and with advanced cycle configuration.

In this regard, IGCC technology is rapidly developing toward maturity: three important commercial plants are now entering into duty in the U.S. (Piñon Pine, Wabash River, Tampa Electric - refs: Demuth, 1996, Breton and Stultz, 1996, Black and McDaniel, 1996), the milestone plant of Buggenum in the Netherlands is operational (Zon, 1996) and several projects are worldwide in progress: for instance, Puertollano in Spain (Sendin, 1996) and three refinery residual gasification plants in Italy (Farina and Bressan, 1994). However, IGCC plants are invariably based on heavy-duty gas turbines with steam recovery cycle. The HAT cycle together with coal gasification has been studied by Rao (1991) and by Schipper (1993), but again based on heavy-duty gas turbines. Those machines are poorer candidates to the HAT configuration, due to the single-shaft arrangement (making more difficult the redesign of LP compressor) and to their moderate pressure-ratio, lower than optimum for HAT cycles with today's TIT levels (Chiesa et al., 1995). A study from Klara et al. (1996) refers to aero-derivative HAT, but a coal combustion system, rather than gasification, is used. Then, the question arises whether the aero-derivative technology may become a suitable solution for coal gasification plants.

The answer to this question is not trivial: IGCC processes are not a mere combination of a gasification plant and a power plant linked together by a fuel pipe, but integration between the two plants can be found at various levels. Therefore, integration with an aero-derivative is inherently different from integration to heavy-duties, and its optimization is essential to achieve better performance and lower plant cost. This paper addresses the configuration of some "deeply-integrated" plant schemes, making use of resources available from the intercooled cycle (i.e. recovery of intercooler heat, use of cool, pressurized air for feeding air separation units). The heat recovery from syngas cooling (the largest energy flow, apart from synthetic fuel, crossing the barrier between gasification and power islands) is addressed by considering three processes: (i) high pressure steam production by syngas coolers, (ii) low pressure steam production after raw gas quenching, (iii) hot water production for air saturation in HAT cycles.

The analyses here carried out are devoted to provide an understanding of the process thermodynamics, to predict the overall plant efficiency, to compare the various technical options and, in general, to assess the basic elements necessary to further engineering studies, including detailed cost estimation.

The answer to the basic question "*are intercooled aero-derivative a good choice for IGCCs?*" involves many further consideration abouts plant and operating costs, emission levels, reliability, maintainability, and so on. Those issues cannot be exhaustively discussed at the present status-of-the-art and are not the main concern of the paper. However, a tentative evaluation of the cost of electricity produced is presented, with the purpouse of outlining the reasons for which aero-derivative machines should be considered as a viable and cost-effective technology for gasification power plants.

## 2. CALCULATION METHOD

### 2.1 General description

The calculation method used for predicting plant energy-exergy flows and efficiency has been described in previous papers, with reference to the gas turbine model (Macchi et al.,1991, Consonni, 1992), the steam plant model (Lozza, 1990) and the system used to analyze gasification processes (Lozza et al., 1994). Its main features are: (i) capability of reproducing very complex plant schemes by assembling basic modules, such as turbine, compressor, combustor, steam section, chemical reactor, heat exchanger, etc., (ii) built-in correlations for efficiency prediction of turbomachines, as a function of their operating conditions, (iii) built-in correlations for predicting cooling flows of the gas turbine, (iv) calculation of gas composition at chemical equilibrium. A peculiarity of the present method is its ability to reproduce the whole IGCC process in a single computer run, without need of "matching" results coming from different computational tools: it enables the possibility of studying heavily integrated processes and of performing a complete second law analysis of the entire plant.

### 2.2 Tune-up with aero-derivative gas turbines

The reliability of the efficiency prediction strictly depends on the accuracy of the assumptions made about the performance of the main plant components. Addressing the technology of advanced aero-derivative machines, some refinements of those assumptions were introduced to improve the accuracy of the results. In fact, the set of assumptions used in previous papers (Chiesa et al., 1995)

Table 1: *Assumptions adopted for gas turbine cycle calculations, with aero-derivative units. Numbers in brackets are relative to heavy-duty machines.*

Gas turbine compressor
Air inlet: 15°C, 101325 Pa, 60% RH
$\Delta i_{is}$=27 [23] kJ/kg for all stages
Leakage 0.8% of inlet flow, at HP exit
$\eta_p = 0.905 \cdot [1-0.07108 \cdot \log_{10}^2(SP)]$ for SP<1
$\eta_p = 0.905$ for SP≥1
Inlet Δp (filter)=1 kPa
Combustors
Oxidizer Δp/p=3%, fuel Δp/p=15%
Heat losses = 0.2% of combustion heat(LHV)
Gas turbine
$\Delta i_{is}$, kJ/kg: 300 (cooled stages), 100 [200] (uncooled stages)
$\eta_p = \eta_{p,\infty} \cdot [1-0.02688 \cdot \log_{10}^2(SP)]$ for SP<1
$\eta_p = \eta_{p,\infty}$ for SP≥1; $\eta_{p,nozzle}$=0.95
$\eta_{p,\infty}$ = 0.90 [0.89] (cooled stages), 0.925 (uncooled stages)
Diffuser recovery = 60% [50] of exit kinetic head
Maximum blade temperature: 860°C [830] (1st nozzle), 830°C [800] (cooled turbine)
First nozzle cooling: film [convective]
Others
Electric generators: see Lozza (1990)
Organic losses: 0.03% of turbomachine work

The size parameter SP used to evaluate turbomachinery efficiencies is defined as $V^{0.5}/\Delta i_{is}^{0.25}$, where V is the stage exit volumetric flow for turbines, and the average volumetric flow for the compressor.

provided inadequate results when applied to this class of machines. This is not surprising, since technological advancements often yield to better component efficiency and, consequently, assumptions must be frequently updated. In addition, aero-engine technology is traditionally more sophisticated than for industrial machines.

Therefore, an attempt was made to reproduce the published, expected performance of two state-of-the-art gas turbine engines, the GE LM6000PC and the Rolls-Royce Trent. Assumptions used are reported in Table 1. For the best fit of available data, it was necessary to improve the internal turbnmachine efficiency and cooling technology parameters, with respect to the values (quoted in brackets in Table 1) providing the best average results for modern heavy-duty machines. With those assumptions, we obtained the results shown in Table 2.

For the aero-derivative machines, the common set of assumptions slightly underestimates the performance of the first unit, but overestimates, of a comparable amount, the second one. For the heavy-duties, later used for comparison in IGCC cycles, the first unit is very well predicted, while the efficiency of the second one is underestimated. However, the V64.3A is a new machine whose data are not exactly defined in the open literature.
We can therefore state that the selected assumptions may represent a reasonable compromise to reproduce the state-of-the-art of modern gas turbine, with an accuracy adequate to the purpouses of this paper.

## 2.3 Other assumptions

Table 3 includes the most relevant hypotheses used for calculating the gasification island and the other equipment used in the power island. The reference gasification process includes an entrained-bed oxygen-blown single-stage gasifier, fed by coal-water slurry, depicting the Texaco technology. The chemical equilibrium model, used to predict the raw syngas composition, provides adequate results for this class of gasifiers. Coal used is Illinois #6 (bituminous) having LHV and HHV of 24.826 and 26.143 MJ/kg (as received); moisture, ash and sulphur contents respectively of 12%, 8.7%, 3.4% by weight. Other assumptions, especially concerning the heat transfer equipment, are retrieved from an analysis of the values referenced in the literature and in the plant practice. More details are given by Chiesa (1995).

The selected value for TIT in Table 3 (1450°C) requires some comment: this figure is much larger than values currently used both for aero-derivative units (about 1250°C) and for heavy-duties (about 1280°C) referenced in Table 2. The improvement in TIT is feasible, according to many researchers (e.g. Day and Rao,



Table 2: *Comparison between expected and calculated performance of two reference aero-derivative engines and two heavy-duty machines of the 70 MW class. β, TIT and exhaust flow as declared for each machine in open literature (Gas Turbine World, 1995 Performance Specs, Pequot Publishing).*

| | GE LM6000PC (60 Hz) | | Roll-Royce Trent | |
|---|---|---|---|---|
| **Aero-derivatives** | expected | calculated | expected | calculated |
| Power output, MW | 43.86 | 43.2 (-1.5%) | 51.19 | 51.9 (+1.3%) |
| Efficiency LHV, % | 41.87 | 41.2 (-1.6%) | 41.57 | 41.98 (+1%) |
| Exhaust temp., °C | 448 | 454 (+6°C) | 427 | 425 (-2°C) |
| | GE PG6101 (Fr.6A) | | Siemens V64.3A | |
| **Heavy-duties** | expected | calculated | expected | calculated |
| Power output, MW | 70.14 | 70.10 (=) | 70.00 | 69.1 (-1.3%) |
| Efficiency LHV, % | 35.2 | 35.1 (-0.3%) | 36.8 | 36.0 (-2.2%) |
| Exhaust temp., °C | 597 | 586 (-9°C) | 565 | 566 (-1°C) |

Table 3: *General assumptions used in the paper, in addition to the ones for the gas turbine (Table 1).*

| | |
|---|---|
| Gasification process (entrained bed) | |
| Coal input (300 $MW_t$ LHV - Illinois #6): | 12.085 kg/s |
| Auxiliary power consumption, gasif.island: | 3 $MW_e$ |
| Water/coal ratio in slurry: | 0.323 |
| $O_2$/coal ratio [(1)]: | 0.852 |
| Gasifier pressure: | 60/80 bar |
| Overall pressure loss[(2)]: | 15% |
| Heat loss (% of input LHV): | 0.5% |
| Other gasification island components | |
| ASU air feed pressure: | 4.8 bar |
| $O_2$ compressor polytropic efficiency: | 0.78 |
| No. of intercoolers in $O_2$ compressor: | 4 |
| Syngas cooler mininum $\Delta T$: | 25°C |
| Syngas cooler heat loss: | 1.5% |
| Syngas heater mininum $\Delta T$: | 25°C |
| Syngas heater heat loss: | 0.7% |
| Scrubbing/quench exit condition: | saturation |
| Min.$\Delta T$ in IP steam generator (quench): | 10°C |
| Min.$\Delta T$ in saturation water heater (HAT): | 10°C |
| Min.$\Delta T$ in syngas/water heat exchangers: | 10°C |
| Syngas expander: used only if available $\beta >$ | 1.5 |
| Intercooled Gas Turbine | |
| Intercoolers minimum $\Delta T$: | 10°C |
| Intercoolers $\Delta p/p$ (air side): | 1% |
| First rotor total inlet temperature (TIT): | 1450°C |
| HRSG and steam cycle | |
| Live steam pressure: | 110 bar |
| Max. live steam temperature (SH and RH): | 538°C |
| Minimum RH pressure: | 12 bar |
| Condensing pressure: | 0.05 bar |
| Approach $\Delta T$=25°C, Pinch point $\Delta T$=10°C | |
| Gas side $\Delta p$ 3 kPa, $\Delta p/p$ superheaters 8%, economizers 10%, heat losses 0.7% | |
| Minumum gas stack temperature: | 80°C |
| Steam turbine efficiency: see Lozza (1990) | |
| Auxiliary power consumption: 1% of condenser heat | |
| Pumps efficiency: 65% (incl. mech./el. losses) | |
| HAT cycle components | |
| Recuperator minimum $\Delta T$: | 25°C |
| Recuperator heat losses: | 0.7% |
| Recuperator $\Delta p/p$ (both sides): | 2% |
| Saturator $\Delta p/p$ (air side): | 0.7% |
| Maximum water temperature: | 240°C |

(1) Such to obtain a raw syngas temperature of 1310°C
(2) From gasifier to syngas expander inlet

1992), due to availability of cooling air at a much lower temperature than in the original engine, because of intercooling. In fact, in a previous paper (Chiesa et al., 1995) we recognized that a critical parameter to assess the maximum cooling duty, for a given cooling technology and blade temperature, is the ratio between the volumetric flow of coolant and the one of the main gas flow. The achievement of TIT=1450°C with coolant flows from an adequately intercooled compressor does not require an improvement of such parameter with respect to the original machine with lower TIT. Therefore, modification of the cooling technology is not necessary and assumptions of Table 1 can be considered valid for the intercooled machine. Higher values of TIT, even if respecting the above mentioned limit, were excluded, not to overstress problems related to combustor design and $NO_x$ formation.

In addition, it has to be noticed that the energy input of 300 $MW_t$ by coal LHV, stipulated in Table 3, establishes the size of the plant, and in particular the air flow to the gas turbine and the power output. Therefore, in this phase we will not make reference to a particular existing machine: however, this issue will be discussed later.

## 3. REFERENCE ADVANCED CYCLES WITH NATURAL GAS

This paper will basically consider two alternative power cycle concepts, the combined and the HAT cycles. The combined cycle is a mandatory reference: it is the most efficient and cost-effective proven technology for conversion of clean fuels (included the synthetic gas after all treatments) into electricity. About the HAT, let us recall briefly that it is an intercooled recuperative cycle, whose particular feature is that low-grade heat, recovered by intercooling, aftercooling and turbine exhausts (after recuperation), is used for heating water, which, in turn, flows into a saturator transferring sensible and latent heat to compressed combustion air (fig.1). Such a large heat content would be dissipated to the ambient in a simpler intercooled recuperative machine. Here, it is used to progressively evaporate water which enters the cycle and produces work in the turbine. Therefore, the outstanding performance of the HAT cycle derives from a very efficient handling (on the second-law point of view) of low temperature heat in the saturation process. A more detailed description of the HAT features and thermodynamic aspects is given by Chiesa et al., 1995.

Before addressing HAT and combined cycles in the frame of an integrated gasification plant, let us briefly discuss their performance with natural gas. This will be useful for testing the reliability of assumptions from Table 3 (by comparison with predictions from other researchers), as well as for establishing a reference case for IGCC applications. The results of our calculations are reported in fig.1:

- The intercooled gas turbine in simple cycle shows an efficiency of about 46.5%, for a selected pressure ratio of 46 (the same value used by the CAGT studies, providing very close-to-optimum thermodynamic performance). This prediction is in good agreement with values quoted by those researchers, not only for efficiency but also for exhausts temperature and specific work. The power output and mass flows quoted in fig.1 are obtained by maintaining the same volumetric flow in the HP compressor of a GE LM6000PC.
- The combined cycle, obtained by adding a three-pressure reheat steam cycle to the same gas turbine unit, shows a net efficiency of about 57%. This is slightly higher than predictions by CAGT (56%), probably due to a more sophisticated arrangement of the steam cycle.
- The HAT cycle provides an efficiency of 57.7%, at the same pressure ratio of 46 (again, very close to optimum). This efficiency is slightly higher than the one of the combined cycle,

confirming that HAT is a viable and interesting alternative. Power output, at the same HP compressor volumetric flow of previous cases, is as high as 287 MW, due to several factors: (i) the optimum intercooling pressure is larger (6.77 vs. 2.76 bar) giving a lower HP compressor outlet temperature and specific volume: this calls for an higher mass flow (310 vs. 230 kg/s) and a lower compression work; (ii) the large water addition (58 kg/s, i.e. 18.7% of air flow), expanding through the turbine without affecting compression work, greatly enhances the cycle specific work. This obviously calls for a more extensive re-design of the machine: for instance, the turbine nozzle area should be 2.25 times larger than the one of the original machine (vs. a 23% increase required by the previous cases).

## 4. CONCEPTUAL ASSESSMENT OF THE PLANT SCHEMES

Coming back to gasification plants, this paper will consider two alternatives for cooling the raw syngas (by syngas coolers producing HP steam and by water quenching) and two alternatives for the power section (combined cycle and HAT, both based on intercooled aero-derivatives). Let us first discuss the syngas cooling process: an entrained-bed gasifier produces raw syngas at temperatures exceeding the ash fusion point, say over 1200°C. Syngas must be promptly cooled for handling reasons and for separating the slag. Conceptually, this can be done by: (i) removing high temperature heat by production of high pressure steam (many processes are proposed by the various manufacturers - see for instance Mahagaokar and Doering, 1995), (ii) "quenching" the raw syngas by means of water addition. In the latter case, heat is not removed, but is used to evaporate a large amount of water, up to saturation. A large exergy destruction takes place, since temperature is abated without work extraction. Sensible and latent heat can be recovered from the water-syngas mixture afterwards, but only medium pressure steam can be produced, because of the moderate temperature of the mixture (about 250°C). Therefore quenching yields to poorer cycle performance, but it is seriously taken into consideration because (i) it is less expensive than syngas coolers, (ii) it is easy to maintain and operate, (iii) enhances plant reliability, because a syngas cooler, operating in hot and harsh environment, is a very sensitive equipment. We will consider this option in our analysis, being especially attractive in the case of relatively small and cost-effective plants like (promisingly) the ones based on aero-derivative turbines.

About the power cycle, let us underline the differences between natural gas and gasification applications. Apart from the change in fuel, the most relevant addition is that the power cycle must handle the heat recovered in syngas cooling to produce mechanical power. In a combined cycle, the large amount of HP steam (generally saturated) produced by syngas coolers is usually superheater and expanded into the steam turbine of the combined cycle (fig.2, upper left). The same happens for IP steam if syngas quenching is used (fig.2, lower left). More in general, all steam and preheated feedwater flows involved in the gasification process are connected to a single steam plant in an integrated power station. In the HAT cycle, the absence of a steam turbine makes questionable the use

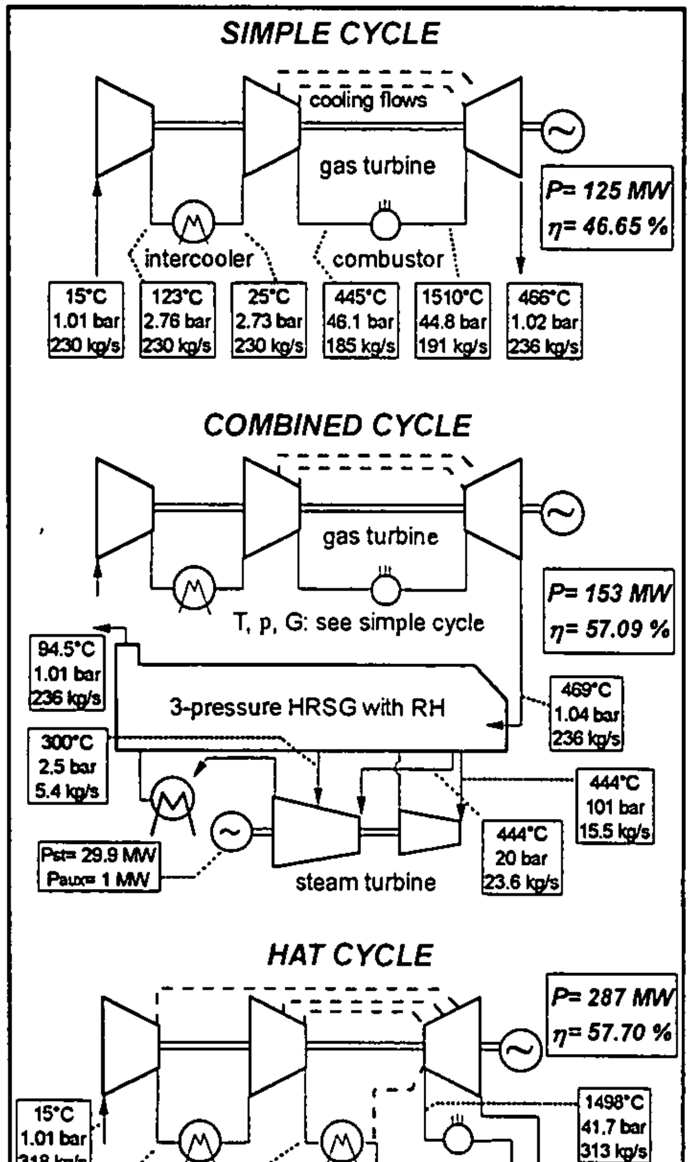


Fig.1: *Intercooled aero-derivative gas turbine cycles with natural gas. Mass flows and power output are calculated by assuming the same volume flow of an LM6000PC in the HP compressor.*

of steam produced from syngas cooling: one solution is to employ a dedicated steam turbine (fig.2, upper right), as suggested in the Novem study (Schipper, 1993). However, this solution will affect the cost-effectiveness and the simplicity of the plant and will not be considered here: we believe that the addition of a complete steam plant to the already complex HAT machinery cannot be compatible with philosophy of compact, reduced-cost and relatively small aero-derivative gas turbines. Therefore, the HAT cycle will be considered here together with the quench cooling mode only (fig.2, lower right): low temperature heat available from syngas cooling after quenching will be mainly used to produce hot water to enhance the duty of the saturator. In this way, no substantial machinery is added to those already present in the HAT cycle. Plant arrangement does not include neither syngas coolers nor steam turbine, and the premises to get a less expensive, easy-to-operate plant become more serious.

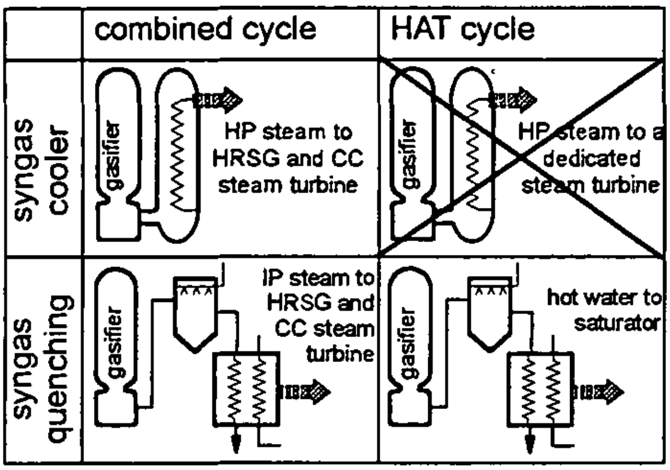


**Fig.2**: *Combinations between two options for syngas cooling and for power cycle configuration. HAT cycle and syngas cooling by steam production are not considered together in this paper.*

The above outlined aspects are not strictly connected to the use of an intercooled aero-derivative rather than an heavy-duty gas turbine, except for the fact that the HAT cycle calls for a pressure ratio and a multi-spool compressor which are typical of aero-derivative machines. The presence of intercooling introduces some interesting peculiarities in the frame of an IGCC, not encountered with heavy-duties: (i) the availability of cold, pressurized air at the intercooler exit for feeding the air separation unit (ASU), (ii) the possibility of recovering the intercooling heat, usually lost in the combined cycle arrangement.

The first issue is of particular help in reducing the cost impact of the oxygen production unit, being the air compressor in a conventional, non-integrated ASU (top of fig.3) a large fraction of its capital cost. In IGCC plants with heavy-duty gas turbines, the "integrated ASU" concept is often proposed (Rao et al., 1993, Koenders, 1995, Smith et al., 1996): air is bled from the gas turbine compressor outlet, at a much higher pressure and temperature (i.e. about 15 bar, 400°C) than required by common separation units. Therefore: (i) feed air must be efficiently cooled before

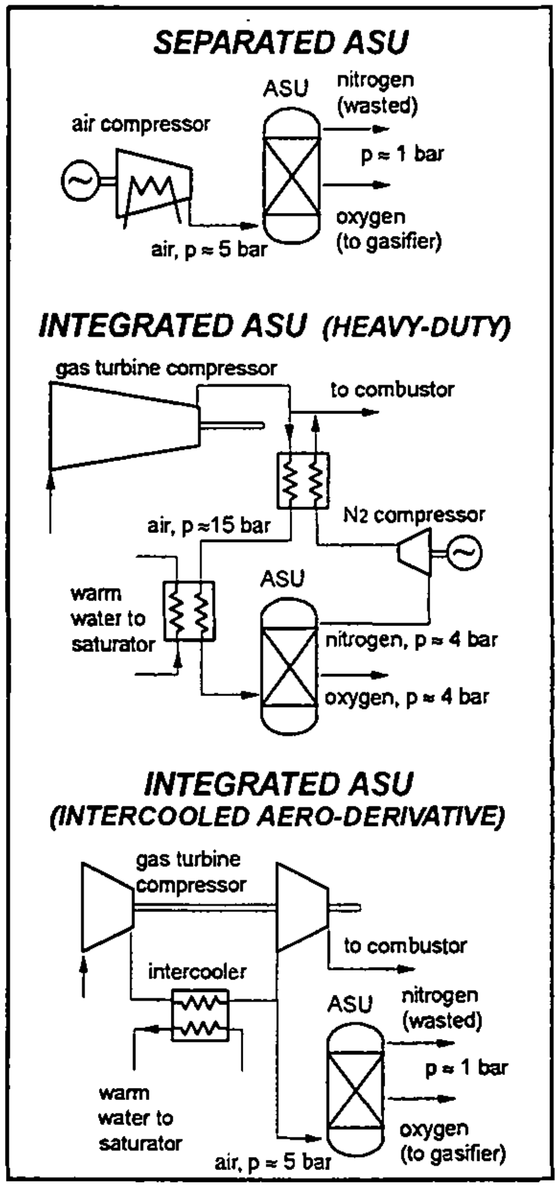


**Fig.3**: *Simplified schemes of three different methods for supplying compressed air to oxygen production units.*

coming to the ASU, (ii) separation columns must be pressurized, (iii) the pressure content of separated nitrogen should not be wasted. Fig.3 (middle section) shows a possible solution, re-injecting nitrogen into the gas turbine after compression and pre-heating, and recovering heat by warming-up water for syngas saturation (a similar concept is used in the Buggenum plant). This brings about many other components and costs, beyond some difficulties in operation and start-up. If an intercooled gas turbine is adopted, integration of ASU becomes much simpler, since there is ample availability of cold, pressurized air after intercooling (bottom of fig.3). It is sufficient to select an intercooling pressure matching the one required by the distillation columns (the value here selected is 4.8 bar - Table 3). This has shown to be compatible with the thermodynamic optimization of the power cycle, and takes advantage from the higher polytropic efficiency of gas turbine compressors compared to industrial compressors. Therefore, a standard air separation unit can be used without additional components: we can say that the advantages of "integration" remains without the drawbacks present in the heavy-duty case.

The other point is re-use of intercooling heat. This problem does not apply to the HAT cycle, in which recovery is inherently present. On the contrary, heat recovery from such a low temperature heat source is not practical in intercooled combined cycle with natural gas (see fig.1). In the frame of IGCC, a solution can be found by heating water for saturation of the synthetic gas, a classic arrangement often used for $NO_x$ suppression. In typical IGCC schemes, such hot water demand is partly satisfied by deriving feedwater from the HRSG: this is no longer necessary if use is done of intercooling heat, leading again to plant simplification and performance enhancement.

## 5. RESULTS ANALYSIS AND EFFICIENCY PREDICTION

### 5.1 The "syngas cooler" option

This configuration, producing high pressure steam in radiative and convective heat exchangers, is presented in fig.4, including arrangements conceptually discussed in the previous chapter: the same figure reports the main operating parameters in the case of $\beta$=36 and $p_{gas}$=60 bar.

Apart from the intercooled gas turbine cycle, the scheme is quite conventional for IGCC plants. A cold gas clean-up process is employed to remove acid elements from synthetic fuel, operating at near-ambient temperature: therefore, an effective heat recovery system is required. Beside HP saturated steam production (more than 90% of the HP steam at turbine inlet is evaporated outside the HRSG), heat from raw syngas is used to warm clean syngas in a regenerative exchanger and, after scrubbing, to produce LP steam and hot water for the slurry. Following the acid gas removal, syngas is pre-heated and saturated by means of water coming from the intercooler.

On the power cycle side, ambient air is taken by LP compressor to 4.8 bar (a value imposed by the oxygen plant requirements) and then cooled to 127°C in an exchanger by recovering heat for fuel saturation. The reduction in air temperature (72°C) is directly depending on the operating conditions of the above-described fuel saturator. Further air cooling, by releasing heat to the ambient, is detrimental to the plant efficiency. Air supply to the ASU is withdrawn before the HP compressor. Following combustion and

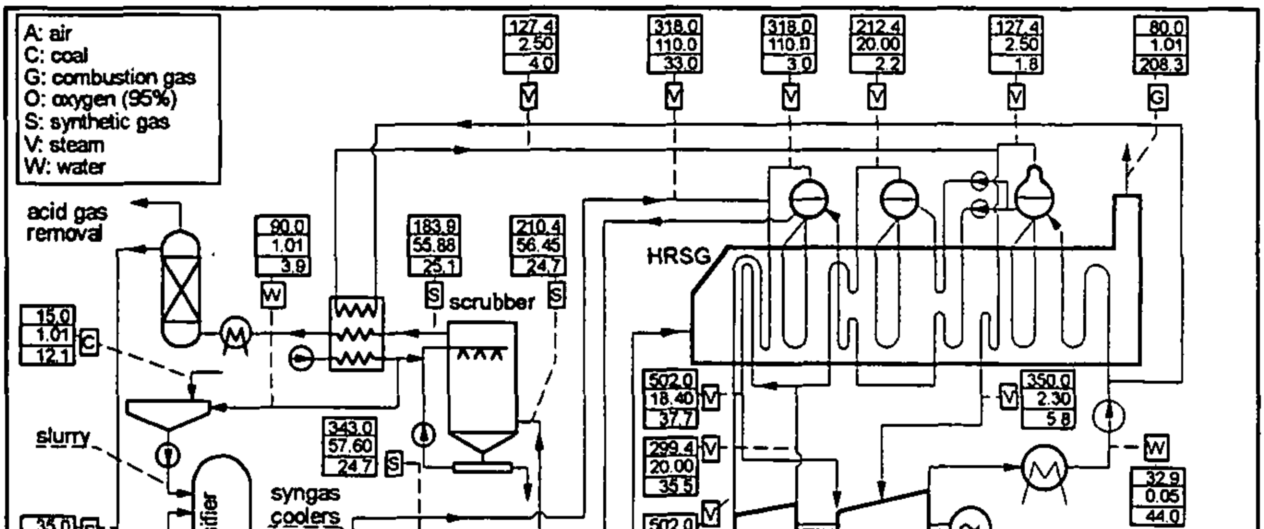


**Fig.4**: *IGCC plant based on aero-derivative intercooled gas turbine, including syngas cooling by steam production and combined cycle, with a pressure ratio of 36 and a gasification pressure of 60 bar.*

expansion, exhausts from the gas turbine enter the HRSG: a three-pressure reheat configuration has been selected, offering the best performance in the realm of an acceptable plant complexity.

Overall plant net efficiency exceeds 47% (based on coal LHV input). Data summarized in fig.5 show that the effects of the overall pressure ratio and gasification pressure upon the plant performance are very weak. The increase of the latter parameter from 60 to 80 bar only induces a 0.1 percentage point rise in efficiency. Table 4 reports the second-law loss breakdown for the considered plants. In the present case (first column), gasification accounts for about 13.6 points of efficiency loss but, since the crude fuel introduced in the plant undergoes a partial oxidation in the gasifier, the loss due to the gas turbine combustion is drastically reduced compared to a natural gas fired combined cycle. The sum of these two terms, which are responsible for more than the half of the overall loss, is only slightly higher than the combustion loss of the intercooled cycle depicted in the upper part of fig.1 (26.3%). Other major losses are located into: (i) syngas coolers: even if producing HP steam, the temperature difference between steam (saturated) and raw syngas remains large; (ii) acid gas removal: it accounts for the loss of heating value of sulphurated compounds, here supposed to be partly used to generate steam, in the Claus furnace, to feed strippers and tail-gas treatment; (iii) steam cycle, mainly accounting for steam turbine inefficiency and condensation.

## 5.2 The "quench" option

This configuration (fig.6) includes full water quenching of the raw syngas at the gasifier outlet. The substantial reduction of the syngas temperature (from 1310 to about 250°C) is obtained by means of the evaporation of a mass flow of water almost equalling the one of the raw syngas. Water condensation in the following processes makes available a large, low temperature heat source. For an effective heat recovery, a complex network of heat exchangers produces LP and IP saturated steam, supplies hot water for make-up and slurry, provides heat to the clean fuel before the combustion. The rest of the plant is very similar to the previous scheme: heat released in the intercooling provides hot water for fuel saturation and the steam bottoming cycle is three-pressure reheat, although it now receives LP and IP instead of HP steam. In order to enhance heat recovery from syngas cooling, the intermediate level pressure decreases from 20 to 12 bar.

Comparing the performance of the two IGCC configurations it is clear that cost savings afforded by water quench are paid in term of conversion efficiency. Water quench causes a 6 $MW_{el}$ cut of the steam cycle power output, with a decay of the overall plant efficiency greater than 2 percentage point in comparison with the previous scheme. Referring to the second law analysis in Table 4 ($\beta$=36, $p_{gas}$=60 bar), syngas cooling causes a 2 points higher loss, almost completely accounting for the efficiency decay: this occurs both in the quenching process, due to the highly irreversible mixing, and in the low temperature heat recovery, because of the large condensation heat to be handled. The overall loss due to the steam cycle is virtually the same: the lower losses related to the reduced power output are offset by the larger term due to heat transfer (HP steam generation is now entirely located in the HRSG, resulting in an higher temperature difference between

Table 4: *Breakdown of second-law losses for the following IGCC plants: (i) aero-derivative (AD) with syngas cooling (SC) and combined cycle (fig.4); (ii) aero-derivative with syngas quench (SQ) and combined cycle (fig.6); (iii) aero-derivative with quench and HAT cycle (fig.7); (iv) combined cycle with syngas cooling based on a commercial heavy-duty (HD) of comparable size (see section 6). Notes: (*) losses located in syngas cooler or in quenching; (**) includes all other losses in heat recovery, pre-heating and saturation of syngas; (***) thermal, mechanical, electric, auxiliary losses of the power cycle (auxiliary losses of gasification section are included in "Coal handling, aux.").*

| Plant type / Second law losses | AD SC CC | AD SQ CC | AD SQ HAT | HD SC CC |
|---|---|---|---|---|
| Gasification | 13.57 | 13.57 | 13.57 | 13.57 |
| Combustion | 14.92 | 15.24 | 13.56 | 16.83 |
| Air separation | 2.52 | 2.52 | 2.29 | 3.22 |
| Coal handling, aux. | 1.00 | 1.00 | 1.00 | 1.00 |
| HT cooling(*) | 3.04 | 4.31 | 4.47 | 3.04 |
| LT cooling(**) | 0.95 | 1.64 | 0.92 | 0.72 |
| Acid gas removal | 2.99 | 2.95 | 2.96 | 2.98 |
| GT compression | 2.13 | 2.10 | 1.39 | 1.54 |
| GT expansion | 3.97 | 3.92 | 5.08 | 3.69 |
| Exhausts discharge | 0.92 | 1.18 | 6.99 | 1.10 |
| HRSG heat transfer | 1.18 | 1.54 | - | 1.76 |
| Steam cycle | 4.43 | 4.08 | - | 4.66 |
| Intercooling | 0.14 | 0.14 | 0.48 | - |
| Aftercooling+eco. | - | - | 0.80 | - |
| Recuperator | - | - | 1.53 | - |
| Air saturation | - | - | 0.51 | - |
| Other losses(***) | 2.12 | 2.08 | 1.09 | 2.61 |
| Total 2nd law losses | 53.88 | 56.27 | 56.64 | 56.72 |
| 2nd law efficiency | 46.12 | 43.73 | 43.36 | 43.28 |
| LHV net efficiency | 47.35 | 44.89 | 44.51 | 44.43 |

exhausts and steam). Eventually, fig.5 shows that, differently from "syngas option", the gasification pressure has some influence upon the plant performance. Temperature at quench outlet grows from 242 to 259 °C as a result of an increase of the gasification pressure from 60 to 80 bar. Consequently a larger amount of higher-quality heat can be recovered, allowing an average 1.2 MW increase of the steam cycle power output which means a 0.4 point gain in the overall efficiency.

## 5.3 The "HAT" option

The integrated gasification HAT cycle plant is shown in fig.7, reporting the main operating parameters for $\beta$=36 and $p_{gas}$=60 bar. It is important to outline that the power cycle is quite different from the natural gas-fired plant depicted in fig.1. The main difference concerns the saturation water circuit, receiving heat from syngas cooling after quench, in addition to the one

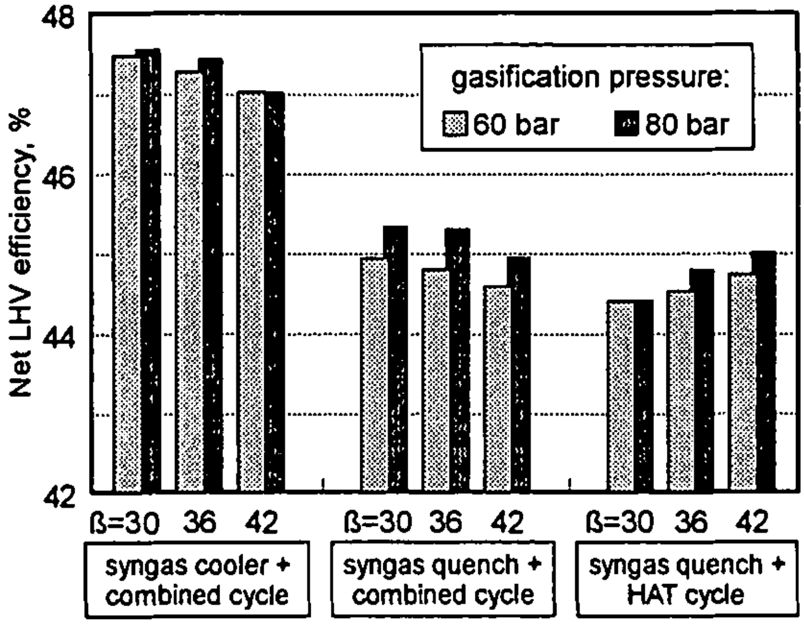


**Fig.5:** *Net LHV efficiency of the three plant configurations studied in this paper, with different overall gas turbine pressure ratios and gasification pressures.*

retrieved from intercooler, aftercooler and economizers. A much larger hot water flow is therefore available for air saturation: a liquid withdrawal in an intermediate section of the saturator was added to optimize heat transfer temperature profile of the syngas coolers. This is clearly shown in fig.8: the heat capacity of saturated syngas decreases during cooling, due to progressive condensation, thus a reduced water flow at the cold end helps in accomplishing a more complete heat recovery; on the saturator side, the larger water flow in the upper section corresponds to the increasing heat capacity of air during saturation. Thanks to the additional heat provided by syngas cooling (more than 40% of the total figure), the water/air mass flow ratio in the saturator jumps from 0.99 of the natural gas case to 3.3 of IGHAT and the water molal fraction in the air entering the combustor from 27 to 48%. Because of the massive water evaporation occurring in the saturator, the make-up flow heat capacity is larger than that of the compressed air flowing in the intercooler, thus accomplishing a complete air cooling without additional heat exchangers. With regard to the gasification section, some differences can be found with respect to the previous IGCC plants: fuel saturator has been removed

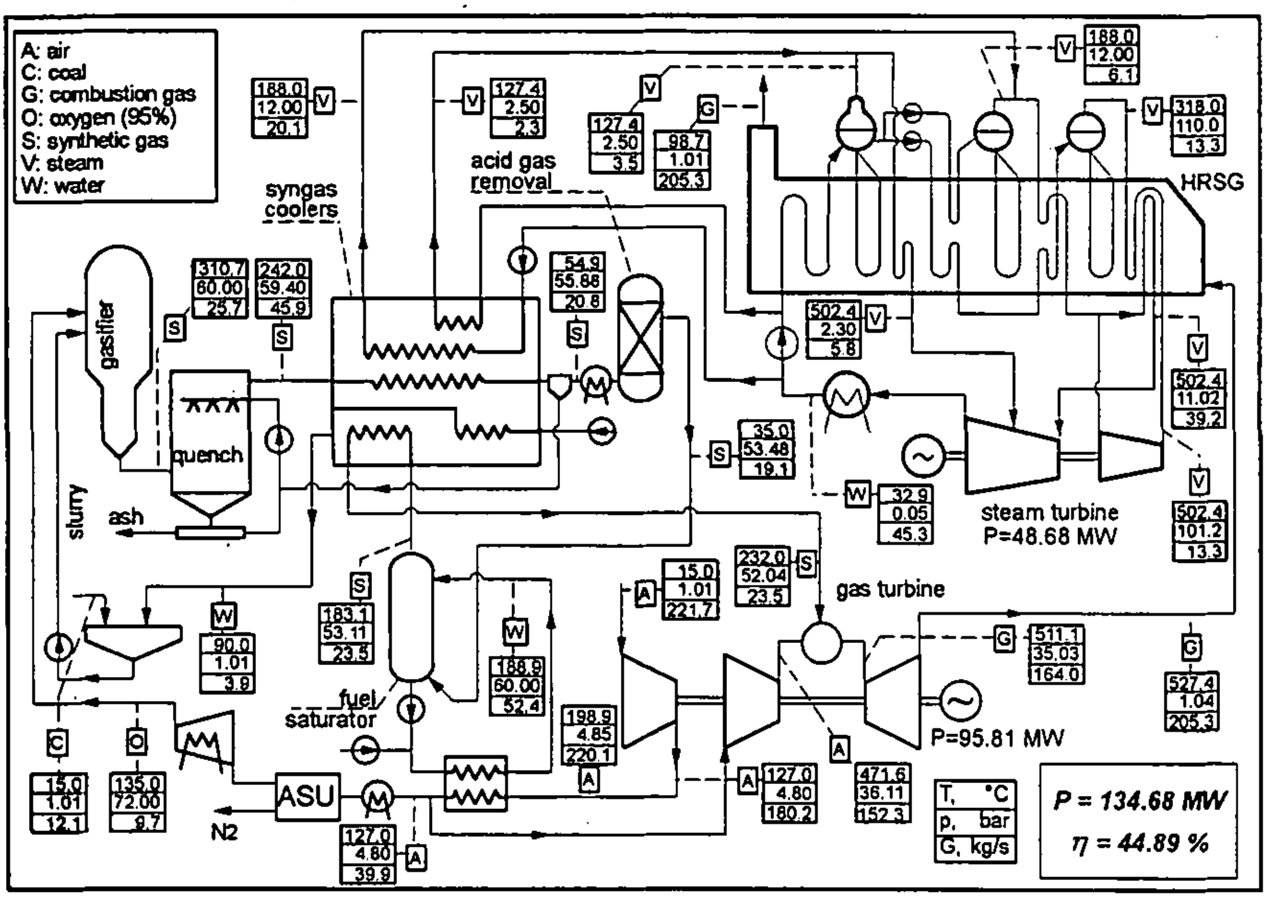


**Fig.6:** *IGCC plant based on aero-derivative intercooled gas turbine, including syngas cooling by water-quench and combined cycle, with pressure ratio of 36 and gasification pressure of 60 bar.*

since it is useless both for heat recovery, wholly accomplished by water recirculating in saturator, and for $NO_x$ prevention assured by the high water content in the combustion air.

Plant efficiency almost equals that of the quench option with combined cycle and slightly increases while increasing the overall pressure ratio and the gasification pressure (fig.5). Comparing these two options on the basis of the entropy analysis (Table 4), IGHAT mainly takes advantage in air compression (due to the lower mass flow rate, -30%) and combustion (because of recuperation which provides warmer oxidant to the combustor); in addition, recuperator and economizer losses are smaller than the ones present in the steam cycle. However, these gains are more than offset by the exhaust loss related to the huge water amount evaporated in the saturator and then released to the ambient, bringing about condensation heat as large as 138 $MW_{th}$ (46% of coal input). Its temperature level is too low for a realistic recovery system able to produce work (condensation begins at 78°C), but it is high enough to reduce the cycle efficiency.

Therefore, the HAT cycle, in the frame of a gasification system including quench, shows a slightly lower efficiency than a combined cycle (an opposite situation was found for natural gas - fig.1): differences are small, but it is clear that the efficiency decay resulting from syngas quenching is severe and cannot be compensated by a different cycle configuration.

## 6. COMPARISONS AND DEVELOPMENT CONSIDERATIONS

Results shown in the previous chapter clearly underline the thermodynamic merits and drawbacks of the three considered solutions. In this section, we will address some of the technical issues brought about by a possible development of those machines and we will compare the obtained performance to the ones of more conventional IGCC plants.

At first, let us comment the HAT solution. Its development for gasification applications suggests serious technical problems. Some of them are common to the ones for natural gas utilization: (i) heat exchanger design, especially for the recuperator and the saturator, a novel component in power plants; (ii) turbomachinery design: the turbine must accommodate a much larger gas flow than the compressor, with an increased enthalpy drop, and, consequently, power output. Therefore, an extensive aerodynamic and mechanic re-design is required and the possibility of adapting existing aero-derivative units is more remote than in the case of natural gas, due to the larger water addition. Other problems, mainly related to the combustor, are peculiar to the gasification application: in fact, the water content of the oxidizer is as high as 49% (molal), and the oxygen content at the combustor outlet is as low as 2%. In such conditions, the combustor stability and flammability must be carefully analyzed: preliminary investigations by Nakhamkin et al. (1994) suggest flame stability at 90% water/air mass ratio, with very low $NO_x$ emissions, but show an unacceptable CO presence.

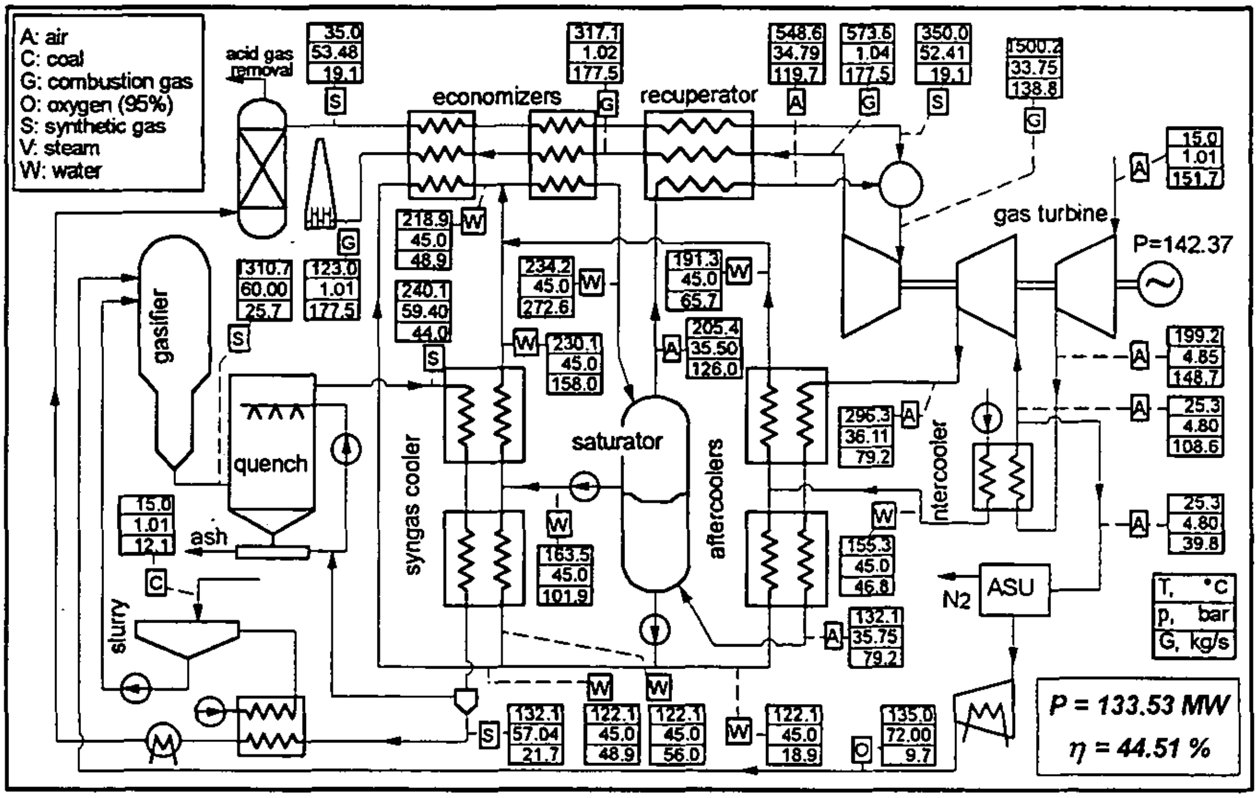


**Fig.7:** *IGCC plant based on aero-derivative intercooled gas turbine, including syngas cooling by water-quench and HAT cycle, with pressure ratio of 36 and gasification pressure of 60 bar.*

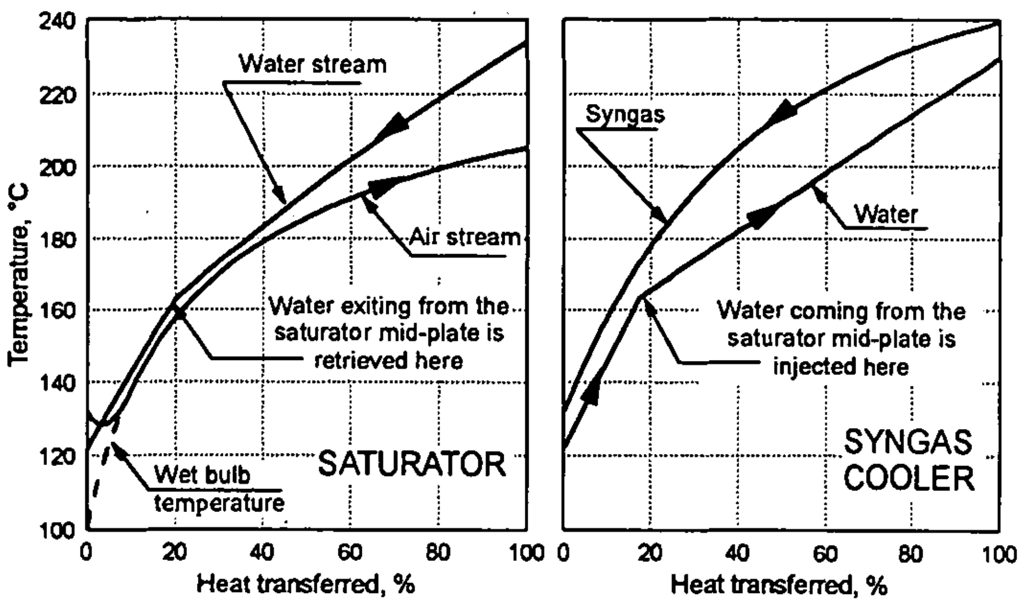


**Fig.8**: *Heat transfer diagrams of saturator and syngas cooler, from the plant scheme of fig.7 (IGHAT cycle). Water extraction from the saturator mid-plate enhances heat recovery in both heat exchangers.*

The actual condition calls for a quasi-stoichiometric combustor, in which oxidizer is not available neither for cooling nor for dilution. Also assuming that $NO_x$ formation is acceptable with a diffusive combustion (the adiabatic flame temperature is only 1700°C), the achievement of high combustion efficiency (i.e. limited CO emission) and of reliable operations is a matter requiring major R&D efforts. To increase the amount of oxygen (i.e., air) available to the combustor, steam cooling for bladings and liners has been recently proposed for advanced heavy-duties (Corman, 1996, Diakunchak et al.,1996): this is not feasible here, simply due to the absence of steam in the HAT. Another possibility to cool the engine without air is using nitrogen, recompressed after the air separation: this will increase plant complexity and operating risks, and should be avoided.

Let us now focus the attention to the combined cycle options, not affected by the above outlined problems. Reasonably, once intercooled machines are available for natural gas, their basic technology can be easily transferred to IGCC applications. However, comparing the optimized combined cycle solution of fig.1 with the gas turbines quoted in figs.4 and 6, pressure ratios are different for both compressors (2.75/16.7 vs. 4.8/7.5, for LP/HP compressors) due to a variation of the overall pressure ratio and of the intermediate pressure, selected for coal application to provide air to the ASU. This requires a different stage allocation on the LP/HP shafts of turbines and compressors: in the reality, this issue can be rediscussed once prototype machines are better characterized than today. Scaling the size of plants showed in figs.4 and 6 in order to obtain the same volumetric flow in the HP compressor of an LM6000PC (the same hypothesis used in Chpt.3), a slight reduction in output can be predicted (to 137.7 and 132.7 $MW_{el}$ respectively for plants of fig.4 and 6), at the same efficiency. In those conditions, a 33% increase of the turbine nozzle area should be anticipated, with respect to the original engine, vs. a 23% required by the natural gas case shown in fig.1.

Apart from those preliminary remarks, a very important consideration, emerging from previous performance predictions, is that the coal conversion efficiency is very high compared to the values obtainable by heavy-duty solutions of comparable size. To better discuss this issue, performance were calculated for an IGCC power plant based on a modern heavy-duty machine matching the design data of a GE Fr.6A or a Siemens V64.3A, having a TIT of 1280°C and a pressure ratio of 15/16. This case represents the status-of-the-art of commercially available gas turbines in the power class here considered. The solutions used for the gasification process match the ones of fig.4: gasification pressure of 60 bar, slurry feed, syngas coolers producing HP steam, cold gas clean-up, fuel saturation. Due to a lower pressure ratio, a syngas expander was adopted before the combustor. Air feed to the ASU is provided by a separated, intercooled compressor (upper scheme of fig.3). The efficiency of this plant is 44.4% (see Table 4, last column), three percentage point less than what we achieved with the aero-derivative intercooled engine with syngas cooler, and something less of the "quench" solution too. Reasons for this poorer performance are shown in Table 4. Essentially, a larger combustion loss takes place, due to (i) lower pressure ratio, providing cooler combustion air (400 vs. 470°C), (ii) lower TIT. Losses in air separation increase by 1 point, since they now include air compression; lower compression losses are offset by larger losses in the steam cycle, now operating with warmer exhausts (606 vs. 527°C) and providing a larger output (64.5 vs. 54.8 $MW_{el}$, compensated by a smaller gas turbine output of 81.9 vs. 97.3 $MW_{el}$).

Higher efficiencies are however predicted for larger heavy-duties of the novel generation, providing about 400 $MW_{el}$ with efficiency of about 45/46% with "F" machines, going to 48/50% with "G" or "H" machines (Koenders, 1995, Newby et al., 1996). This is the consequence of a larger size, improving the performance of turbomachines (especially of the steam turbine), of technology advancements (like steam-cooling, higher TIT and pressure ratio) and, in the quoted papers, of the adoption of a more efficient dry-feed gasification system. Even in comparison with those more sophisticated systems, the solutions presented here are able to achieve competitive energetic efficiencies.

## 7. COST ASSESSMENT

A preliminary estimation of the cost of electricity produced by IGCC plants based on aero-derivative vs. heavy-duty gas turbines is presented in Table 5. The assessment of investment cost is based on data available from open literature (sources partially used are: Maude, 1993, Todd and Joiner, 1994, Klara et al.,1996). The absolute value of some items in the plant cost estimation is affected by large uncertainties, but it should be considered that: (i) investment cost of IGCC plants is a matter of discussion and estimations cover a very broad range (from 1200 to over 2000 $\$/kW_{el}$); (ii) the size of plants considered here is rather small, making cost extrapolation more difficult. Therefore, the scope of the present analysis is simply to outline the differences between the two technologies and to stress the reasons for which aero-derivatives may emerge as a winning solution. The rationale behind the results of Table 5 is that costs of gasification system, auxiliary equipments, civil works, engineering, etc. are the same for the two plants, having the same coal input. Differences arise from (i) different specific cost of gas turbine package, for which a 20% increase has been stipulated to account for a new and more sophisticated technology (notice that the ICAD group has a goal of only 10% increase); (ii) different output from gas and steam turbines; (iii) a 12 M$ saving in the ASU due to the absence of the air compressor. Following those assumptions, Table 5 shows a little difference in the plant cost, but the higher output of the aero-derivative solution (due to the higher efficiency) yields to a 9.3% reduction of the specific capital cost. Due to the better efficiency again, fuel cost is smaller allowing for a 7% reduction of the final cost of electricity.

Stating whether such a cost reduction is worth the adoption of a novel technology is beyond the scope of this paper. However, it is authors'opinion that, once intercooled aero-derivative are an available option in the power generation market, their use in IGCC plants is a matter to be pursued.

## 8. CONCLUSIONS

Aero-derivative intercooled gas turbines, with combined cycle, are an interesting option for gasification applications, for the following reasons:

- they can achieve a very high coal conversion efficiency: with syngas coolers, it is comparable to the one of very large and sophisticated plants with advanced heavy-duties not yet available on the market; with quench it can be compared to the one of modern heavy-duties of the same size with syngas cooling system;
- a very efficient plant can be operated at a moderate size (about 130 MW), reducing the risks connected to a large capital investment in a technology (IGCC) which cannot be considered, at present, fully proven;
- capital cost of such plants is to be investigated with more detail; however, preliminary considerations developed in Table 5 help in predicting a favorable situation, mostly due to the better conversion efficiency, allowing a larger power output for the same coal consumptions: this brings about a reduction of the plant specific cost, in addition to a lower fuel cost per kWh produced.

Contrarily to the proposals from other researchers quoted in the paper (but they make reference to different plant configurations), the IGHAT solution does not seem particularly attractive, if a steam plant is not included for syngas cooling: efficiency is lower than other solutions and serious technical problems (mostly due to the huge amount of water injected into the cycle) may impose critical obstacles to its development.

Table 5: *Cost analysis and cost of electricity obtained by medium-size IGCC plants having the same gasification islands and different gas turbine technology: (i) HD uses a commercial heavy-duty with TIT=1280°C and $\beta$=15; (ii) AD uses an intercooled aero-derivative with TIT=1450°C and $\beta$=36. Both plants include syngas cooling and combined cyle. Notes: (1) including syngas treatment, acid gas removal, sulphur plant; (2) based on 300 \$/kW for heavy-duty and 360 \$/kW for aero-derivative; (3) based on 700 \$/kW: includes HRSG, steam turbine, cooling system and water treatment plant; (4) efficiencies quoted in Table 4, with a 5% penalty due to ageing, fouling, part-load behaviour, etc.*

| Type of gas turbine | **HD** | **AD** |
|---|---|---|
| **Performance:** | | |
| Coal input LHV, MW | 300 | 300 |
| Gas turbine output, MW | 81.9 | 97.3 |
| Steam turbine output, MW | 64.5 | 54.8 |
| Net power output, MW | 133.3 | 142.0 |
| **Costs (M$):** | | |
| Coal handling & treatment | 12.1 | 12.1 |
| Gasification island (1) | 66.1 | 66.1 |
| Gas turbine (2) | 24.6 | 35.0 |
| Steam plant (3) | 45.2 | 38.4 |
| Air separation unit | 32.0 | 20.0 |
| Electric, instr. & site | 20.0 | 20.0 |
| Mounting, fees & contingencies | 50.0 | 50.0 |
| Total plant cost, M$ | 250.0 | 241.6 |
| Specific plant cost, $\$/kW_{el}$ | 1875 | 1701 |
| **Common assumptions** | | |
| Plant utilization, kWh/kW·year | 7000 | |
| Interest rate, % | 8 | |
| Plant life, years | 20 | |
| Interest during construction, % | 15 | |
| Cost of coal, $/GJ | 1.5 | |
| **Cost of electricity, mills/kWh** | | |
| Annualized capital cost | 31.37 | 28.46 |
| Fuel cost (4) | 12.76 | 11.97 |
| O&M cost | 8.00 | 8.00 |
| Total cost | 52.13 | 48.43 |

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