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AN ASSESSMENT OF THE THERMODYNAMIC PERFORMANCE OF MIXED GAS-STEAM CYCLES: PART A: INTERCOOLED AND STEAM-INJECTED CYCLES

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ABSTRACT

This paper discusses the thermodynamics of power cycles where steam or water are mixed with air (or combustion gases) to improve the performance of stationary gas turbine cycles fired on clean fuels. In particular, we consider cycles based on modified versions of modern, high-performance, high-efficiency aero-derivative engines.

The paper is divided into two parts. After a brief description of the calculation method, in Part A we review the implications of intercooling and analyze cycles with steam injection (STIG and ISTIG). In Part B we examine cycles with water injection (RWI and HAT).

Due to lower coolant temperatures, intercooling enables to reduce turbine cooling flows and/or to increase the turbine inlet temperature. Results show that this can provide significant power and efficiency improvements for both simple cycle and combined cycle systems based on aero-engines; systems based on heavy-duty machines also experience power output augmentation, but almost no efficiency improvement.

Mainly due to the irreversibilities of steam/air mixing, intercooled steam injected cycles cannot achieve efficiencies beyond the 52-53% range even at turbine inlet temperatures of 1500°C. On the other hand, by accomplishing more reversible water-air mixing, the cycles analyzed in Part B can reach efficiencies comparable (RWI cycles) or even superior (HAT cycles) to those of conventional "unmixed" combined cycles.

NOMENCLATURE

h	:	heat transfer coefficient	W/m^2K
i	:	enthalpy	J/kg
k	:	mass transfer coefficient	kg/m^2s
k_{bw}	:	blade wall thermal conductivity	W/mK
$\dot{m}$	:	mass flow	kg/s
p	:	pressure	Pa
P	:	electric power output	W
S	:	heat transfer surface	m^2
t	:	blade wall thickness	m
T	:	temperature	K or °C
x	:	molal fraction	
V	:	volumetric flow	m^3/s
Y	:	absolute humidity	kg_{water}/kg
β	:	overall cycle pressure ratio	
β_{LPC}	:	low pressure compressor pressure ratio	
Δi_{is}	:	stage isentropic enthalpy drop	J/kg
λ	:	vaporization heat	J/kg
φ	:	relative humidity	
η	:	net electric LHV efficiency	
η_p	:	polytropic efficiency	

Subscripts and superscripts

a	:	air
chrg	:	chargeable, i.e. for all turbine blade rows except first nozzle
cl	:	turbine coolant
ex	:	exit
g	:	main gas flow
i	:	liquid-gas interface in the saturator
in	:	inlet
nz	:	first nozzle of gas turbine
opt	:	optimum
w	:	water
1r	:	first rotor of gas turbine
'	:	referred to dry air
*	:	referred to saturated air mixtures

Acronyms

ICC	:	intercooled combined cycle
ICR	:	unmixed intercooled recuperated cycle
CC	:	combined cycle
HAT	:	humid air turbine cycle
HP,IP,LP	:	high, intermediate, low pressure
HRSG	:	heat recovery steam generator

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ISTIG	:	intercooled steam injected cycle
LHV	:	fuel lower heating value
LPC	:	low pressure compressor
RWI	:	recuperated water injected cycle
ST	:	steam turbine
STIG	:	steam injected cycle
TIT	:	first rotor total inlet temperature
TOT	:	turbine outlet temperature

1. INTRODUCTION

The steadily increasing performance of new "superfan" jet engines, in terms of both power and efficiency, is spurring increasing interest toward the use of aero-derived engines for large-scale, base-load electricity generation from natural gas (Cohn et al., 1993a; Stambler, 1993). With this regard, a first point to be emphasized is that, given the relevance of efficiency for base-load duty, no simple-cycle aero-engine alone, no matter how advanced, will ever compete successfully with combined gas/steam cycles based upon modern heavy-duty turbines. Evidence for this argument comes directly from the intrinsically poor thermodynamic "quality" of Brayton cycles: results presented in a previous paper (Chiesa et al, 1993) show that even pressure ratios above 60, turbine inlet temperatures above 1500°C and substantial advances in blade cooling techniques, materials and turbomachinery aerodynamics would be inadequate to reach simple cycle net electrical efficiencies of 50%, a value well below the potential of existing, commercial combined cycles.

The most straightforward method to increase the efficiency of an aero-engine for stationary applications is the addition of a bottoming steam cycle: indeed, a remarkable number of aero-engine-based combined cycles with net electric efficiencies close to 50% and power outputs below 50 MW_{el} are successfully operating throughout the world. At such low-medium power outputs aero-engine-based systems can outperform the ones based on heavy-duties by several efficiency percentage points; however, at larger power outputs (say over 100 MW_{el}), combined cycles based upon modern, high-temperature heavy-duties exhibit both superior efficiencies and remarkably lower specific costs.

There are several distinctive features of aero-engines which suggest to investigate cycles different from conventional combined cycles: the multi-shaft arrangement makes it simpler to insert intercoolers amidst the compression phase (or reheat in the expansion phase); the pressure ratio, already higher than optimum for combined cycles, can be further increased for optimum operation with unconventional cycles; the relatively low exhaust gas temperature and flow rate give poor bottoming steam cycle efficiencies. It is therefore not surprising that most of the innovative configurations alternative to the combined cycle have been proposed for aero-engines.

Several "complications" of the basic Brayton cycle have been proposed in recent years: injection of water and/or steam at various point along the gas cycle; insertion of heat exchangers (recuperators, compressor precoolers, intercoolers, aftercoolers), of reheat combustors, or of more complex components such as chemical recuperators and air/water saturators[1]. The effect of these complications is twofold: higher net electrical efficiency due to a more favourable "shape" of the thermodynamic cycle and larger unit power output due to an increase of both specific work and mass flow. Better cycles are realized by increasing the average combustor operating temperature and reducing the exhaust gas temperature, thus abating the two major losses of the simple cycle (introduction and release of heat to/from the cycle). Higher power outputs are accomplished by modifications which decrease compressor power (intercooling), increase turbine power (water/steam injection or higher average expansion temperature) or "supercharge" the existing turbomachinery by adding compression stages in front of the engine core.

Since none of these plants has been operated nor tested so far, the assessment of their performance potential can be based only on predictions of the authors proposing these cycles, who often claim very attractive efficiencies. However, due to inconsistencies among the hypotheses adopted by the various authors, such predictions do not warrant a comparison among the different schemes. The aim of this two-part paper is to investigate the thermodynamic performances of these plants and to compare them with those of combined cycles on the basis of the same, coherent set of assumptions. The analysis includes the optimization of the plant arrangement and the cycle parameters, as well as a detailed second-law analysis. The focus is on "mixed" cycles without reheat, i.e. cycles with substantial water and/or steam injection into air or gas and only one combustor. Reheat at constant TIT has been considered in a previous paper (Macchi et al., 1991); nonetheless, the variable-TIT results presented here and the recent commercial launch of a new heavy-duty reheat turbine (Anon., 1993c) make it worth of further future investigations.

2. CALCULATION MODEL

The calculation model used to generate the results described here has been specifically developed to predict the performance of complex gas-steam cycles, particularly "mixed" cycles (Consonni, 1992). Since the structure and the capabilities of the model have been extensively described in previous papers (Consonni et al., 1991; Lozza, 1990 and 1993; Chiesa, Consonni and Lozza, 1992; Chiesa et al., 1993) we recall here only the most significant features and the modifications

[1] All three major world aero-engine manufacturers are presently engaged in a research program called Collaborative Advanced Gas Turbine (CAGT) aimed at developing alternative cycle concepts for stationary high-efficiency power generation (Cohn et al., 1993a). Phase I of the program, originally organized by Pacific Gas & Electric Co. and now joined by a number of US, Canada and European utilities, includes research on the potential of intercooled combined cycles based on the General Electric *GE90*, intercooled regenerative cycles based on the Rolls Royce *Trent* and humid air cycles based on the Pratt&Whitney *FT4000* (Stambler, 1993).

The evaluation of innovative cycles is also among the projects selected for the Advanced Turbine Systems (ATS) Program sponsored by the U.S. Department of Energy (Anon, 1993a), as well as in other programs sponsored by the Electric Power Research Institute (Cohn et al., 1993b; Ghaly et al., 1993; Tittle et al., 1993).

introduced in the framework of this paper.

2.1 Basic Outline

The system to be calculated is defined modularly as an ensemble of interconnected components, which can be of ten basic types: compressor, gas turbine expander, splitter, mixer, heat exchanger, combustor, pump, saturator, steam cycle (including all its components) and shaft (accounts for turbomachine spool interconnections, as well as electric losses). Operating characteristics and mass and energy balances of each component are calculated sequentially until the conditions (pressure, temperature, mass flow, etc.) at all interconnections converge toward a stable value. Aside from the algorithm handling the component network (it is virtually possible to analyze any cycle configuration), the most distinctive features of the model lie in the calculation of the key cycle components: turbomachines, heat recovery steam generator and saturator.

The cooled gas turbine expansion is calculated as a sequence of small steps, each consisting of an expansion followed by gas-coolant mixing. At each step, the coolant flow required to maintain the blade temperature within an assigned value is found by the heat flux balance across the blade wall. The coolant is bled at the minimum pressure required to overcome coolant circuit pressure drops, and then discharged into the main flow; the optimistic implications of this idealized "continuous" compressor bleed (one for each expansion step) is compensated by imposing a high (40%) coolant-side pressure drop. The polytropic efficiency of both the cooled expansion steps and the uncooled turbine varies with a similarity size parameter to account for scale effects; exit kinetic energy is partly recovered in the diffuser.

The calculation of the steam bottoming cycle and the evaluation of the steam turbine expansion have been addressed in previous papers by Lozza (1990 and 1993), while the evaluation of the saturator is extensively discussed in Part B.

2.2 New Features

Since all cycles analyzed here feature intercooling (STIG is the only exception), it is important to assess whether – and how much – lower coolant temperatures allow increasing the TIT of a given engine. This was accomplished by defining "critical" values for the ratios $V_{cl,nz}/V_g$ and $V_{cl,1r}/V_g$ between the volumetric cooling flow of the turbine nozzle ($V_{cl,nz}$) or of the 1st rotor ($V_{cl,1r}$) and the volumetric gas flow at the nozzle exit (V_g, see Ch.3). Both ratios are now calculated at each iteration, based on the coolant mass flow rate and the density at the conditions of injection of the step placed halfway the cascade[2]; if their "critical" values are exceeded it means that, for the stipulated technology, turbine cooling is unfeasible (see Par.3.1 for further comments).

[2] Along the step-by-step expansion, the conditions of the spent coolant injected into the mainstream vary continuously, thus preventing from defining an actual volumetric flow for the whole cascade. By referring to the density at the step located halfway along the cascade, the definition used here gives a value of V_{cl} corresponding to "average" (for the cascade) injection conditions.

Beside this addition, two adjustments were introduced to represent more closely the situation encountered in actual engines:

1) Rather than being a fixed input datum, the blade wall Biot number $Bi = h_g \cdot t_{bw}/k_{bw}$ is now calculated on the base of the blade material thermal conductivity k_{bw} assigned in input, the calculated gas-side heat transfer coefficient h_g and the blade wall thickness t_{bw}, which is a constant fraction of the blade chord (2.5%). This allows suitable variations of blade wall thermal resistance due to different operating conditions (h_g increases with β) or size (in aero-derivatives the blade chord and thus t_{bw} are smaller).
2) The polytropic expansion efficiency of the cooled and uncooled turbine sections can be different. This allows accounting for the poorer performance of the cooled section (larger trailing edge thickness and flow disturbances due to coolant ejection).

2.3 State-of-the-art Performances

Detailed information on the characteristics and the operating parameters of commercial gas turbines (turbomachinery efficiencies, metal temperatures, cooling flows, etc.) are considered strictly proprietary by all gas turbine manufacturers. Thus, several of the parameters needed to run the calculation model are not known, but can only be estimated based on experience, theoretical analyses, pieces of information collected from manufacturers or in the literature. Despite this handicap, Consonni and Macchi (1988) and Consonni (1992) have shown that a proper scrutiny of the data publicly available (power output, efficiency, TOT, etc.) allows calibrating the most crucial model parameters (turbomachinery efficiencies and parameters describing the cooling technology) to reproduce satisfactorily the engines belonging to the same technological "generation".

The input data used for the calculations performed here (see next paragraph) produce performances in good agreement with those of the latest, most advanced engines. Tab.1 shows the performance predicted for the two "reference" engines with operating conditions (TIT, β, $\dot{m}_a$) representative of state-of-the-art large-size aero-derivatives and heavy-duties recently introduced by major world manufacturers.

Tab.1: *Performances of "reference" state-of-the-art aero-derivative and heavy-duty engines (ISO conditions), as predicted by the calculation method used in the paper.*

Engine Type		Aero-derivative	Heavy-duty
TIT	°C	1250	1280
β		30	15
$\dot{m}_a$	kg/s	125	600
η	%	39.9	35.8
W	kJ/kg$_a$	327.2	373.1
P	MW	40.9	223.9
TOT	°C	450.7	595.8

2.4 Basic Assumptions

The assumptions adopted to obtain all results obtained here are summarized in Tab.2. Most values equal the ones adopted in previous analyses (Macchi et al., 1991; Chiesa et al., 1993). The higher value assumed here for the film cooling parameter (r_{fc}) of aero-derivatives is meant to account for improvements in film cooling technology incorporated in latest engines; for heavy-duties, film cooling is supposed to be used only in the first nozzle. The cooling technology parameters Z and r_{fc} have been held constant throughout all calculations: therefore, the results shown here are meant to represent the potential of current technology, even when the assumed TIT is above the state-of-the-art 1250-1280°C range. Pressure drops and temperature differences of unconventional components like the saturator, the recuperator or the aftercooler conform to the assumptions of Day and Rao (1992), which in turn are based on calculations for reasonably sized piping and equipment as quoted by vendors (Day, 1994).

3. TURBINE INLET TEMPERATURE: TRADE-OFFS AND LIMITS

TITs adopted in commercial engines presumably represent the best compromise among a number of requirements and constraints: high efficiency, low cost, high reliability, long life, blade heat transfer and temperature distribution, coolant temperature, available coolant-side pressure drop, etc. If any of the "boundary conditions" affecting this best compromise is changed, also the optimum TIT will change. For a given engine, the prediction of this change can be effectively performed only by the manufacturer, who can master all technical and economic details of his machines.

General-purpose thermodynamic analyses must rely on a simpler approach: given the intricacies of cost assessment and the need for criteria with the widest applicability, it is appropriate to evaluate TIT trade-offs within the realm of thermo-fluid-dynamics. Then, given the cooling technology, the pressure ratio and the cycle configuration, the constraints to be considered are related to:

- Thermodynamic optimization. In general, there will be an optimum TIT which maximizes efficiency; for TIT > TIT_{opt} the penalties due to larger cooling flows more that offset the advantages of better cycle thermodynamics (Chiesa et al., 1993).
- Fluid dynamics. Since the coolant cross-section is limited by the size (and shape) of the blade and of the channels driving the coolant to the turbine, there will be a limit on the flow rate which can be forced through the coolant circuit. In general, larger cooling flows can be obtained by increasing the bleed pressure; however this (i) definitely hurts efficiency and (ii) may not be possible without adding a compressor for the coolant.
- Emissions. The abatement of NO_x emissions calls for adequate amounts of dilution or secondary air in the combustor hot section. Consequently, the flow available for turbine cooling is much smaller than the one left after stoichiometric combustion.

Since the prediction of NO_x formation is much beyond the scope of this work, we've neglected the third issue, thus implicitly assuming that all configurations discussed in the paper are not emission-constrained. This simplifying assumption has been mitigated by introducing a ceiling of 1500°C on TIT (corresponding to combustor outlet temperatures below 1600°C); this limit should insure that problems like the exponential increase of thermal NO_x with temperature,

Tab.2: *Assumptions adopted for the calculations presented in the paper. The size parameter SP used to evaluate turbomachinery efficiencies is defined as* $V^{0.5}/\Delta i_{is}^{0.25}$.

Compressors	Δi_{is}=27 kJ/kg for all stages. Leakage 0.8% of inlet $\dot{m}$, at HP exit
	$\eta_p=\eta_{p,\infty}\cdot[1-0.07108\cdot\log_{10}^2(SP)]$ for SP<1; $\eta_p=\eta_{p,\infty}$ for SP≥1; $\eta_{p,\infty}$: 0.905 (AD), 0.895 (HD)
	Inlet Δp (filter)=1 kPa
Combustors	Δp/p=3%, heat losses=0.4%·$\dot{m}_f$·LHV
	Fuel compressor: isothermal with η=0.55, followed by fuel preheat (except CC, ICR)
Turbines	Δi_{is}: 300 kJ/kg (cooled stages) and 100 kJ/kg (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; $\eta_{p,\infty}$: 0.89 (cooled stages) and 0.925 (uncooled stages); $\eta_{p,nz}$=0.95; diffuser recovery =50% of exit kinetic head
	Cooling parameters: Z=100, r_{fc}=0.4 (AD) or 0.25 (HD). Maximum blade temperature: 830°C (1st nozzle), 800°C (cooled turbine)
Water-air heat exchangers	Air-side Δp/p=1%, minimum ΔT=10°C for surface HE; for evaporative intercoolers exit φ=90%
Recuperators	Δp/p=2% (both sides), minimum ΔT=25°C Heat losses 0.7% of the heat transferred
Saturators	Air-side Δp/p=0.7%
HRSG and steam cycle	Approach ΔT=25°C, Pinch point ΔT=10°C
	Gas side Δp 3 kPa, Δp/p superheaters 8%, economizers 10%, heat losses 0.7%
	Steam turbine: η=0.7 (includes el./mech. losses) for ISTIG; for CC see Lozza (1990)
Pumps	η=0.65 (includes el./mech. losses)
Other	Ambient air (φ=60%) and water: T=15°C, p=101325 Pa; Fuel: methane at T=15°C, p=4 MPa, LHV=50.01 MJ/kg
	Electric generators: see Lozza, 1990; Organic losses 0.03% of turbomachine work

combustor cooling, corrosion, fatigue or thermal stress can actually be solved within the realm of current technology.

Whether emission control technology can really compensate for the changes in operating parameters stipulated here will have to be verified, although two notable circumstances give credit to our simplifying assumption: (i) the dramatic improvements recently achieved by dry-low-NO_x technology; (ii) the favourable situation of mixed cycles, where the lower adiabatic flame temperature resulting from the high moisture content in the oxidizer allows achieving low NO_x emissions even with diffusion burners and fuel pre-heat.

3.1 Setting the Turbine Inlet Temperature

Based on the criteria set forth above, the gains achievable by increasing TIT without changing the cooling technology have been investigated by raising TIT until:

- efficiency reaches a maximum, or
- cooling flows reach the maximum value allowed by the cooling circuit characteristics and operating conditions, or
- TIT reaches the ceiling of 1500°C.

The maximum cooling flow allowed by the cooling circuit depends on a number of factors: available pressure drop, Mach number inside blade cooling channels, Mach number of spent coolant ejected from film cooling holes, etc. In this work we have assumed that:

a) The "critical" condition corresponding to the maximum flow allowable through the coolant circuit is identified by a limiting value of the ratio V_{cl}/V_g between the spent coolant volume flow rate and the volumetric flow at the first nozzle exit.

b) The limiting value of V_{cl}/V_g for both the nozzle and the first rotor is the one calculated for the two reference state-of-the-art engines of Tab.1. These "critical" values are listed in Tab.3.

The choice of the volumetric rather than the mass flow ratio is closer to the physical basis of the problem, because the quoted limits on pressure drops, Mach numbers and flow cross-section are mainly related to the coolant volume flow.

It is worth noting that, by defining state-of-the-art simple cycle engines as "critical", assumption b) implies that their TIT can be increased only by (i) improving the cooling technology or (ii) improving materials or (iii) reducing the coolant temperature. The first two options have been analyzed in a previous paper (Chiesa et al., 1993); in this paper we discuss the potential of the last option.

4. INTERCOOLING IN SIMPLE AND COMBINED CYCLES

Thermodynamic textbooks show that for an ideal Brayton cycle intercooling increases specific work but definitely impairs efficiency. However, this situation changes substantially for real cycles, not only because there is a positive influence of intercooling on the efficiency penalties due to fluid-dynamic losses in turbomachines, but especially due to the presence of relevant coolant flows. In this case, intercooling brings about lower coolant temperatures, which in turn allow reducing the coolant flow required to keep the blades below a given temperature. Since cooling flows constitute a source of relevant efficiency penalties (coolant throttling, heat transfer, mixing, etc., see Consonni, 1992), their reduction is definitely beneficial. Therefore, for a heavily cooled gas turbine the overall impact of intercooling on efficiency can be highly positive, especially when considering the possibility to take advantage of lower coolant temperatures to increase TIT, rather than to decrease cooling flows.

Tab.3: *Cooling flows calculated for the "reference" engines representative of state-of-the-art aero-derivative and heavy-duty technology. It is assumed that, for each engine type, the volumetric flow ratios $V_{cl,nz}/V_g$ and $V_{cl,1r}/V_g$ can be increased only by improving the cooling technology.*

Engine Type	Aero-derivative	Heavy-duty
$\dot{m}_{cl,nz}/\dot{m}_a$, %	7.06	6.28
$\dot{m}_{chrg}/\dot{m}_a$, %	10.01	6.23
$V_{cl,nz}/V_g$, %	3.33	2.46
$V_{cl,1r}/V_g$, %	2.75	2.37

4.1 Intercooling in Modified Modern Aero-engines

Let us discuss the effects of intercooling by referring to a cycle with β=46, approximately corresponding to the intercooled version of current aero-derivatives under study by some manufacturer (Stambler, 1993). Gas turbine power output has been evaluated by assuming that the flow cross-section at the nozzle exit is the same of the "reference" aero-engine reported in Tab.1 (TIT=1250°C, β=30) and operates in chocked conditions. Consequently, the predictions discussed in this chapter approximately represent what could be achieved by implementing intercooling to actual commercial engines without modifying the hot section of the turbine[3]. Besides the simple cycle, let's also consider the performances of a combined cycle where (i) the heat available in the exhaust gases is recovered in a three-pressure-level bottoming cycle condensing at 32.9°C (0.05 bar); (ii) the heat discharged by the intercooler and/or the aftercooler above 100°C (heat below 100°C is wasted to ambient) is used to produce power at 50% second-law efficiency – i.e. producing half the power of a reversible cycle driven by the intercooler heat and discharging heat to ambient. The work output resulting from the latter hypothesis may be produced by a separate heat recovery cycle (e.g. an organic Rankine cycle) or, more plausibly, by "recycling" the heat to other parts of the cycle by pre-heating the fuel, pre-heating the make-up water or generating LP steam.

The situation is depicted in the diagrams of efficiency, power and cooling flows reported in fig.1, which shows three types of curves:

[3] In order to accommodate the larger enthalpy drop, the LP section of the turbine must be modified by adding one or more stages. More substantial changes are required for the compressor: a new LP section ahead of the intercooler and an adjustment of the HP cross-sections to warrant the desired turbine nozzle area.

- "allowed" conditions (continuous lines), for which the ratio V_{cl}/V_g is below its critical value;
- "not allowed" conditions (dashed lines), for which such ratio is above its critical value, thus representing unfeasible situations;
- "critical" conditions (dashed-dotted lines), for which the ratio V_{cl}/V_g at either the nozzle or the first rotor equals its upper bound, i.e. the values listed in Tab.3.

With no intercooling ($\beta_{LPC}=1$) the cycle is unfeasible because for $\beta=46$ the coolant temperature is higher than that encountered in the reference engine with $\beta=30$, thus requiring a V_{cl}/V_g well above the critical value. Moving toward higher β_{LPC} there is, at first, a beneficial effect on η, implying that the "technological" benefits brought about by lower cooling flows overcome the thermodynamic drawbacks. At high β_{LPC} thermodynamics eventually prevails, thus decreasing the cycle efficiency. At optimum β_{LPC} efficiency approaches 45%, a significant improvement over the reference case. As for power output, there is a very large increase even without increasing TIT (at 1250°C P=70-90 MW_e vs $\simeq$40 MW_e of the base case in Tab.1), partly due to the increase of specific work and partly due to higher β which, for the same turbine nozzle area, increases the mass flow and thus power output. The figure also shows that:

- While higher TITs are always beneficial to power output, they produce significant efficiency benefits only for combined cycles, which can take advantage of higher TOT. As a matter of fact, the intercooled gas turbine with no heat recovery achieves maximum efficiency at the moderate TIT of 1280°C (and $\beta\simeq 80$, see Fig.15 of Part B).
- At TIT=1500°C and optimum $\beta_{LPC}\simeq 3.5$, the combined cycle efficiency reaches values ($\simeq$55%) fully comparable with those of large heavy-duty-based systems.
- The range spanned by efficiency and power output covers the performances projected for the intercooled version of the GE LM6000 which, as indicated in Stambler (1993), should reach a power output of 90 MW with an efficiency close to 45-46%, going up to 110 MW and 54% in combined cycle.
- "not allowed" cooling flow situations occur only at low intercooler pressures ($\beta_{LPC}<1.8$ for TIT=1250 °C, $\beta_{LPC}<3.4$ for TIT=1500 °C), while critical conditions are always very close to those giving the highest efficiency. Although critical cooling flows are first established in the nozzle, they occur almost simultaneously also in the first rotor.
- At TIT=1500°C, the chargeable mass flow is always much higher, although not critical, than in the "reference" engine, basically because the number of stages – and thus the area – to be cooled is much higher.

Since critical cooling conditions first occur in the nozzle, we investigated the possibility of removing this barrier by "aftercooling" the coolant bled at the compressor exit before using it in the nozzle, a practice often adopted in heavy-duties; similarly to the intercooler heat, the heat made available by aftercooling at T>100°C was assumed to be converted to power with 50% second-law efficiency.

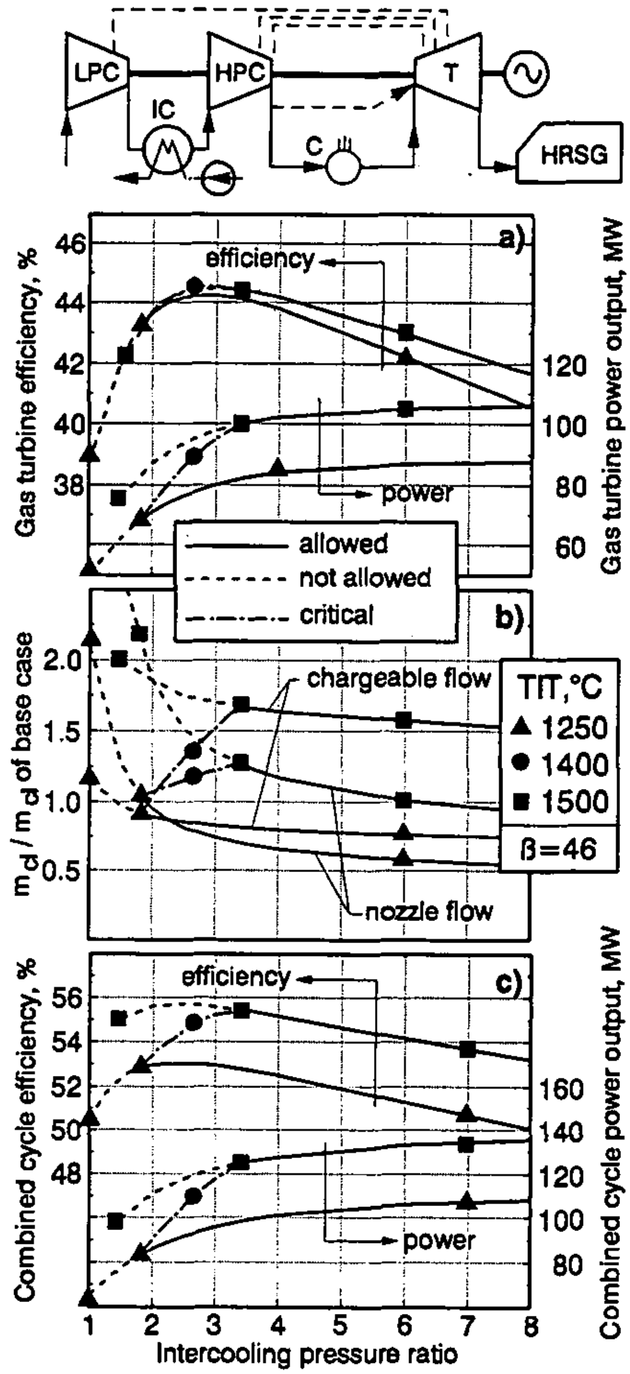

Fig.1: *Efficiency and power output of an intercooled unmixed cycle based on a modified aero-engine. The upper figure refers to the simple cycle, the lower one to the combined cycle. The middle diagram shows the ratio between the cooling flows of this case and the ones of the "reference" aero-engine quoted in Tab.1 and Tab.3. The air mass flow is varied so to preserve the same turbine nozzle cross-section of the reference engine.*

Like for intercooling, the outcome of aftercooling depends on the trade-off between lower coolant flow and the related thermodynamic penalties: irreversible aftercooler heat recovery and larger ΔT between the coolant and the blade wall. The plant scheme now includes a heat exchanger cooling the nozzle coolant down to 25°C (like at the intercooler exit), while the coolant for the turbine is bled at variable pressure like in the "reference" engine. The final temperature of 25°C has been assumed to emphasize the influence of coolant aftercooling; in practice higher final temperatures may provide slightly better performance. As shown in fig.2, results are very similar to the ones with intercooling only, with these minor differences:

- the lower allowable β_{LPC} corresponds now to critical cooling conditions in the first rotor rather than in the nozzle, because aftercooling cuts the nozzle cooling flow by more than 50% (see diagram of cooling flows);
- there is a slight increase in power output and a slight decrease in efficiency.

In conclusion, the addition of heat exchangers on the coolant flow path does not offer substantial advantages. Intercooling appears the most efficient way to limit cooling flows and increase TIT without penalizing efficiency, for both simple and combined cycles.

To confirm this statement, we also considered the option of simply lowering the coolant temperature of the "reference" engine of Tab.1 (β=30). Since cooling only the nozzle coolant would be ineffective (it would simply shift the problem to the first rotor) and since it is unrealistic to assume that each coolant bleed would have its own heat exchanger, we assumed that the entire coolant flow is bled at the compressor exit and then cooled. As before, for the combined cycle it is assumed that the heat released in the aftercooler above 100°C is recovered with 50% 2nd-law efficiency.

Fig.3 shows that the severe throttling losses incurred by the chargeable flow override the gain brought about by lower coolant flows. Despite the approximate 50% reduction of cooling flows, at TIT=1250°C the efficiency of both the simple and the combined cycle is more than 1 percentage point lower than that attainable with the reference engine. At TIT=1500°C the simple cycle suffers dramatic penalties due to large cooling flows – and thus large throttling losses; the combined cycle makes up for such losses with heat recovery, although efficiency never goes above the one achieved with the reference engine. The only benefit of higher TIT is therefore a substantial increase of power output. It is worth mentioning that these results are particularly unfavorable because the reference engine of Tab.1 makes best use of the coolant by bleeding it at many points along the compressor; if the configuration considered for fig.3 were compared with a reference engine with only 2 or 3 coolant bleeds, the outlook of aftercooling would be somewhat less grim. In any case, the relevance of throttling losses unquestionably hinders the idea of using high-pressure, low-temperature coolant for the turbine section downstream the nozzle.

The results discussed in this chapter can be summarized by saying that, for a given cooling technology, intercooling is definitely the most convenient practice to raise the performance of aero-derived engines.

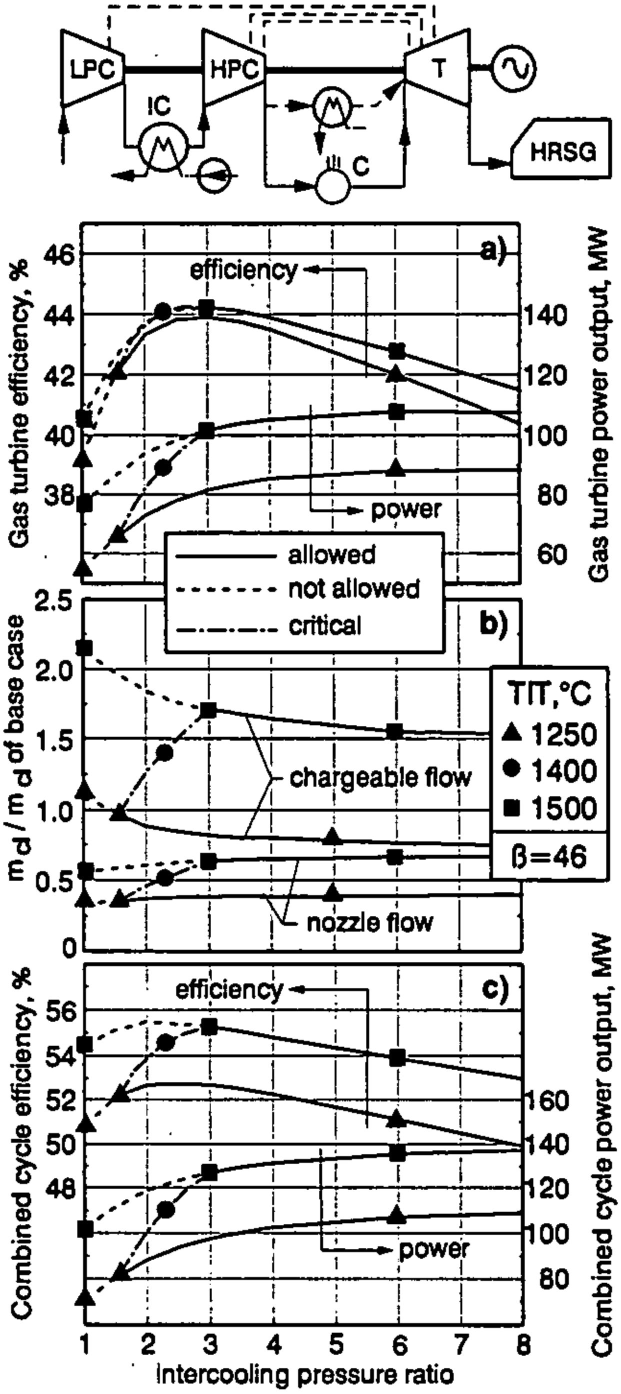

Fig.2: *Efficiency and power output of an intercooled unmixed cycle based on a modified aero-engine with aftercooling of the first nozzle coolant. The upper figure refers to the simple cycle, the lower one to the combined cycle. The middle diagram shows the ratio between the cooling flows of this case and the ones of the reference aero-engine quoted in Tab.1 and Tab.3.*

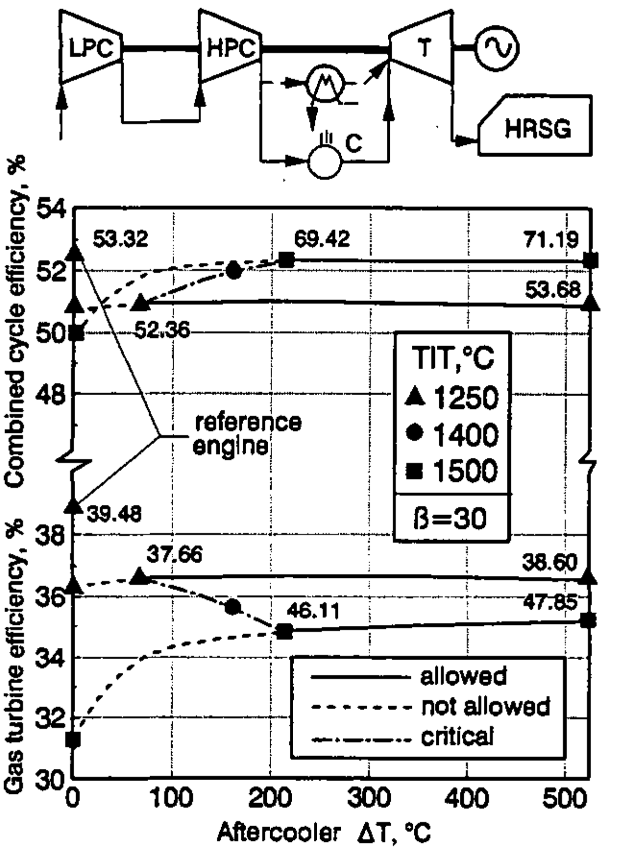

Fig.3: *Efficiency of simple and combined cycles with external cooling of the air for turbine cooling, as a function of the coolant temperature drop. The entire cooling flow is extracted at the discharge of the HP compressor. Numbers on top of markers represent the cycle power output [MW].*

Aftercooling is generally detrimental, except when applied solely to the nozzle coolant (as in fig.2). The effect of aftercooling the nozzle coolant depends on how efficiently the aftercooler heat is recovered: in the more complex cycles investigated in Part B (RWI and HAT), the heat recovery mechanism is efficient enough to produce small performance gains when bleeding the nozzle coolant at the lower available temperature of the HP air circuit. However, since the gain is always very small, all cases considered hereafter do not include aftercooling of nozzle coolant; whenever possible they simply take advantage of the enhanced cooling capabilities of air/vapour mixtures by bleeding the coolant downstream water or steam injection.

4.2 Intercooling in Combined Cycles with Heavy-duty Gas Turbines

The question may arise whether intercooling and the related TIT enhancements can also improve the efficiency of combined cycles based on large heavy-duties. This case is inherently different from that of aero-engines, which are characterized by high pressure ratios and poor steam cycle efficiencies (due to small size). Opposite to aero-engines, in heavy-duties intercooling faces several unfavorable circumstances: (i) lower β produce lower initial coolant temperature and thus higher ΔT between the coolant and the blade ($\simeq$480°C vs. $\simeq$260°C for aero-derivatives): the relative increase of this ΔT brought about by intercooling is much smaller, and so are the benefits; (ii) at the low β of heavy-duties the increase of combustion irreversibilities caused by intercooling (in the combustor dilution air is heated under larger ΔT) is more severe; (iii) specific work augmentation is small, because at low β compression work is a much smaller fraction of net work: for combined cycles based on the reference engines in Tab.1 the ratio between compression work and net work is $\simeq$66% for the heavy-duty, vs. $\simeq$128% for the aero-derivative.

Let us discuss these issues by considering the results of calculations performed for the "reference" heavy-duty machine of Tab.1. The bottoming steam cycle is a three-pressure reheat cycle, condensing at 0.05 bar with a maximum steam temperature of 565°C; the maximum pressure has been optimized in the range 100-300 bar (the optimum value depends on the gas turbine outlet temperature, see Lozza, 1993). The intercooling pressure ratio β_{LPC} has been set to the minimum value that meets the volumetric flow ratios of Tab.3, with the constraint $\beta_{LPC} \geq 2$. As usual, intercooling heat above 100°C is recovered with 50% 2nd-law efficiency. The results summarized in fig.4 show that:

- The non-intercooled cycle efficiency of 55.8% adequately portrays the performance of large plants based on latest gas turbine and steam cycle technology. 54-55% CC efficiencies are now quoted by several manufacturers (e.g. Tomlinson et al., 1993) and have been measured on an ABB plant based on an older generation gas turbine (Werner et al., 1993).
- At constant TIT=1280°C and β=15, intercooling at the minimum allowed β_{LPC}=2 decreases efficiency by about 0.7 percentage points. Such gap can be eliminated by increasing TIT to about 1350°C.
- Higher TITs give marginal efficiency gains, mostly because they require higher β_{LPC} (e.g. 2.8 at TIT=1500°C) to lower the coolant temperature and thus meet the limit on V_{cl}/V_g: this overcomes the benefits of higher TIT even when recovering the intercooler heat (continuos line).
- At pressure ratios much higher than usually adopted in heavy-duties (the figure shows β=30) the outcome is worse, because the higher β_{LPC} required to meet the limits on V_{cl} amplify the negative effect on the cycle thermodynamics – even with intercooling heat recovery. This means that the handicap ensuing from non-optimal β (for current combined cycle technology β_{opt}= 15) cannot be fully neutralized even by TIT=1500°C[4].

The only benefit of intercooling is therefore an augmentation of power output: for the same inlet air flow of 600 kg/s,

[4] Given the superiority of cycles with moderate pressure ratios, one might think of improving the performance of aero-engine-based combined cycles simply by reducing the pressure ratio. Although this would increase the combined cycle efficiency, it would also dramatically decrease power output due to much lower air mass flow. The consequent strong increase of specific costs makes this proposition highly unrealistic.

the intercooled cycles with TIT=1500°C provide an electric power of about 480 MW, vs 350 MW of the reference case.

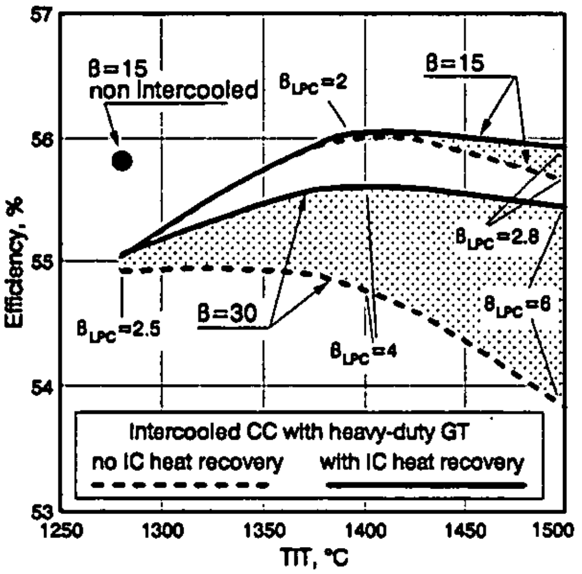

Fig.4: *Influence of intercooling on the efficiency of combined cycles based on heavy-duty gas turbines with $\beta=15$ and $\beta=30$. Dashed lines refer to cases where the intercooling heat is wasted to ambient; continuous lines refer to cases where the fraction of intercooling heat available at T> 100°C is converted to power with 50% second-law efficiency.*

5. STEAM INJECTED CYCLES

Injecting the steam generated in the Heat Recovery Steam Generator into the gas turbine rather than using it in a closed-loop bottoming cycle is a well established practice. The cycle has been extensively discussed in the technical literature since its appearance (Cheng, 1978) and still recently (Rice, 1993). Fully-steam-injected (STIG) versions of few gas turbine engines (slight modifications to turbine bladings may be required to accommodate larger flow rates) have been commercially available for several years (Oganowski, 1987). Rather than high electrical efficiency alone, the rationale behind this scheme is based on considerations like operational flexibility in cogeneration applications, investment cost (lower than for combined cycles), effective NO_x abatement due to massive steam injection into the combustnr, etc. As an example of the trade-offs involved, let's mention the most efficient commercial STIG package based on the large aero-derivative GE LM5000, which features a power output of 49.6 MW, with a net electric efficiency of 43.8% (Anon., 1993b); the combined cycle version of the same engine exhibits an 8% lower power output (45.9 MW) and much higher investment cost, but achieves an efficiency of 49%.

Efficiencies and power output of steam injected cycles can be enhanced by a more radical re-design of the engine. General Electric has been proposing the intercooled LM8000 ISTIG version for several years (Horner, 1989), but the project never reached the commercial phase.

5.1 Plant arrangement

Fig.5 depicts the plant arrangement considered for steam injected cycles where intercooling is carried out by a surface heat exchanger. In most cases make-up water and fuel pre-heat in the intercooler do not improve overall heat recovery, because the same heat can be transferred to water and the fuel in the HRSG by reducing the exhaust gas temperature (except for extremely high steam injection rates). However, make-up pre-heat in the intercooler allows some savings in the heat transfer devices of the whole plant: (i) water heaters are removed from the HRSG; (ii) only part of the intercooling heat is rejected to ambient, therefore reducing the size and cost of cooling towers (or air coolers), as well as their consumptions (power, water). Additionally, fresh make-up water may help in achieving the lowest possible compressed air temperature.

The arrangement assumed for the calculation of cycles with intercooling by direct-contact evaporative heat exchangers is almost the same: the only difference is a water-air mixer that substitutes the surface intercooler, with no make-up water and fuel pre-heat ahead of the HRSG.

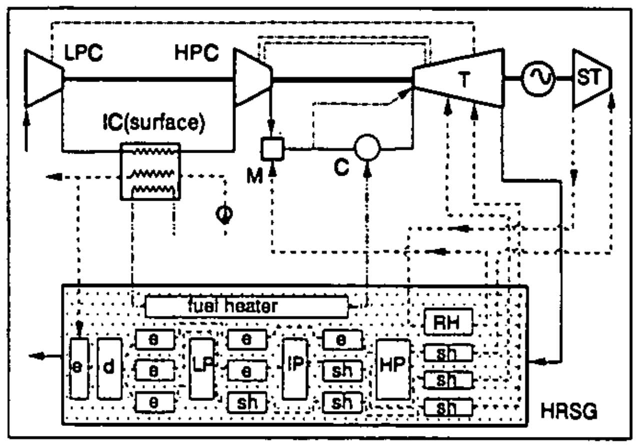

Fig.5: *Conceptual plant scheme of intercooled steam injected cycles with surface intercooler (the HRSG banks are: "e" economizers, "sh" superheaters, "d" deaerator).*

The rather sophisticated HRSG arrangement shown in the figure gives the highest efficiencies:

• three-pressure-level steam generation for injection of HP steam into the combustor and IP and LP steam into the gas turbine accomplishes thorough heat recovery (even if it entails the highest water consumption);

• at each pressure, maximum superheating is accomplished by parallel heat transfer banks: this is beneficial to efficiency, minimizing the temperature gap between steam and gas;

• whenever possible, a steam turbine (ST) expands steam between the highest drum pressure and the injection pressure (after the expansion steam is reheated before being injected). This allows full optimization of all steam generation pressures, thus reducing HRSG heat transfer irreversibilities.

Turbine cooling flows (dotted lines in the figure) are bled from the compressor like in a simple-cycle, while the flow for nozzle cooling is bled after HP steam injection to take advantage of the increased moisture content[5].

Like for other mixed cycle configurations considered in Part B, the scheme of fig.5 includes full fuel pre-heat up to the temperature made possible by the gas turbine discharge conditions. Fuel pre-heat is always beneficial to efficiency, because it substantially reduces combustion irreversibilities without significant drawbacks on other processes (heat transfer entails minor irreversibilities, while the small decrease of steam production yields only marginal reductions of specific work): for the configurations considered in the paper it generally gives a 1 percentage point efficiency increase. Despite this appealing benefit, emission concerns would presumably prevent from adopting fuel pre-heat in simple and combined cycles, as well as in ICRs, because without steam or water injection the achievement of low-NO_x emissions most likely requires premixed burners; to avoid preignition, such burners would be fed with fuel at ambient temperature. For these reasons, we have considered fuel pre-heat only for mixed cycles.

5.2 Variables to be optimized

The cycle parameters to be optimized are (i) overall pressure ratio, (ii) intercooling pressure and (iii) steam evaporation pressures. On the contrary, steam injection rates are determined by the HRSG energy balance and the imposed pinch-point ΔTs.

Let us first discuss the influence on efficiency of the intercooling pressure. Fig.6 depicts the situation for the two options of mixing and surface intercooling. Calculations were performed at fixed β and optimized steam pressures: 39.3/13.2/4.3 bar at TIT=1250°C, β=30; 58.9/15.9/5.6 bar at TIT=1500°C, β=45; both cases do not require the HP steam turbine. With surface intercooling and TIT=1250°C the highest efficiency is reached for very low LPC pressure ratios (about 2). For β_{LPC} above optimum, the efficiency benefits brought about by intercooling are offset by the higher losses in the combustor (lower inlet temperature) and in the intercooler (heat discharge). Compared to the simple STIG cycle intercooling improves efficiency by about one percentage point. The gain reaches 3 percentage points only by taking full advantage of lower coolant temperatures to increase TIT up to 1500°C. In this case the optimum β_{LPC} is about 4, enough to meet the limits on V_{cl} (Tab.3) but much lower than the value which minimizes compression work.

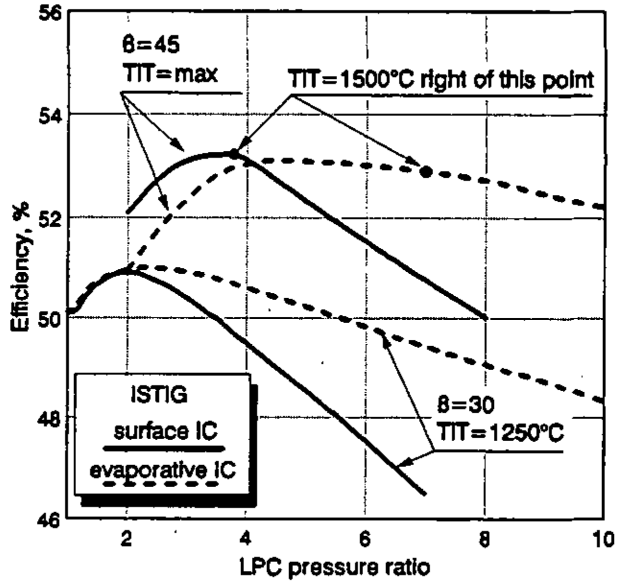

Fig.6: *Efficiency of ISTIG cycles at β=30, TIT=1250°C and β=45, TIT=max as a function of β_{LPC}. Continuous lines refer to intercooling by surface heat exchangers; dashed lines refer to spray intercoolers. The points with β_{LPC}=1 represent the non-intercooled STIG cycle; due to the coolant flow limitations discussed in Ch.3, in this case TIT cannot be increased.*

5.3 Surface vs Evaporative Intercoolers

At the optimum β_{LPC} spray intercoolers do not produce any efficiency advantage over surface heat exchangers. This is not surprising because, although the former do not waste heat to ambient, in both cases the heat released by air is transferred to a sink at low temperature, with large ΔT and thus large irreversibilities. In spray intercoolers the heat sink is low-pressure water evaporating at approximately the intercooler exit temperature (it would be the exit temperature if the exit flow were saturated). In surface intercoolers the heat sink is the ambient; notice that in this case reducing the intercooler heat transfer irreversibilities by increasing the heat transfer area (and thus the water exit temperature) doesn't help, because the gain in the intercooler would be completely lost by discharging water at higher temperature.

At large β_{LPC} evaporative intercooling becomes more efficient because, while the heat sink of surface intercoolers remains the same, the temperature after the spray – and thus the temperature of the evaporating water – increases significantly. For example, at β_{LPC}=6 the temperature at the outlet of the evaporative intercooler is 86.8°C: transferring the heat released by air to water evaporating at such temperature is much less irreversible than discharging it to ambient at 15°C.

Even if spray intercoolers suffer lower heat transfer losses due to water evaporating above the ambient temperature, there are additional losses not present in surface intercooling: (i) water/air mixing; (ii) higher compression losses due to higher temperatures at the inlet of the HP compressor; (iii) higher stack losses, due to higher exhaust moisture content. At optimum β_{LPC} these additional losses generally produce

[5] Due to its higher heat capacity and superior heat transfer properties, steam is a more effective coolant than air. The implications of steam cooling in non-intercooled cycles have been discussed by Chiesa et al. (1992): for a STIG cycle with TIT=1250°C the use of saturated steam for cooling would boost specific work by 12%, while efficiency would increase by only 0.1 percentage points. Due to the complications brought about by steam cooling (steam bleedings from HRSG drums to the turbine, more complex start-up and operation, erosion/corrosion problems, etc.) such possibility will not be considered here.

performances slightly worse than with surface heat exchangers. For these reasons all parametric calculations have been referred to surface heat exchangers.

As for power output, the larger mass flow rate ensuing from evaporative intercooling always gives more power per kg of air entering the LP compressor. However, if one refers power output to the mass flow in the HP compressor (or the turbine), he would obtain approximately the same specific work produced with surface heat exchangers. This means that for a given size of HP turbomachinery evaporative intercooling gives no power output increase.

5.4 Results for Optimized Cycles

The results of the overall cycle optimization are summarized in the efficiency-specific work plane reported in fig.7. At TIT=1250°C and $30<\beta<45$ efficiency is slightly above 51%; at these high pressure ratios the HP steam turbine gives no benefits, because the optimum HP steam evaporating pressure does not exceed the one necessary for injection into the combustor. The adoption of higher TITs made possible by intercooling increases efficiency by about 2 percentage points, while the corresponding optimum pressure ratio increases from about 36 to 45. The results of fig.7 fully agree with General Electric predictions of 52% efficiency at TIT=1371°C and β=34 (Horner, 1989): for the same cycle parameters our model gives η=52.2%. It is interesting to note that the thermodynamic "quality" of the two ISTIG cycles optimized for TIT=1250 and 1500°C is almost the same (slightly better for the latter): the ratios between ISTIG and Carnot efficiencies are 51.16/81.08 = 63.1% at 1250°C and 53.23/83.74 = 63.6% at 1500°C.

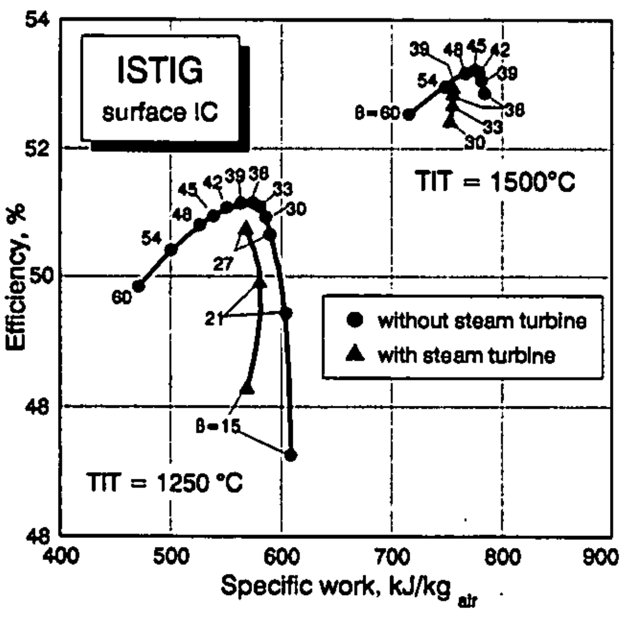

Fig.7: *Efficiency and specific work of ISTIG cycles with surface intercoolers and optimum β_{LPC}. For β higher than the ones indicated by the triangular markers the addition of an HP steam turbine (component ST of fig.5) has no beneficial effect.*

The operating conditions for the optimized cycle (β=45, TIT=1500°C) are given in fig.8: with reference to a simple cycle operating at a nearly optimum pressure ratio (β=30) and TIT=1250°C, the increase of specific work (referred to the inlet air flow) is as large as 138%. Moreover, if we stipulate that the flow area of the HP turbine remains unchanged, the engine is "supercharged" with an increase of inlet air flow of 31.6%: the cumulative result is a net power output increase of over 215%, which means that a "modified" ISTIG version of a 40 MW simple cycle machine would generate a power output of about 128 MW. For constant HP compressor exit flow areas (as assumed by Day and Rao, 1992), the increase in power output is even larger: about 435%, giving a power output of 217 MW_{el}.

Previous calculations (Macchi et al., 1991) demonstrated that the introduction of a reheat turbine would further improve the ISTIG cycle efficiency by about 3 percentage points, and almost double its specific work.

CONCLUSIONS

The analysis performed in this Part A points out that intercooling, coupled with the higher pressure ratios and the higher TITs made possible by the lower compressor temperature, can substantially enhance the efficiency and the power output of both simple and combined cycles based on current aero-derivative engines. On the contrary, intercooling does not lead to any efficiency improvement of heavy-duty-based combined cycles, although it still gives higher power outputs.

When the bottoming closed-loop steam cycle is replaced by steam injection, the cycle suffers an efficiency loss of 3-4 percentage points. Part B investigates other mixed cycle concepts able to reduce, or even reverse, this efficiency gap.

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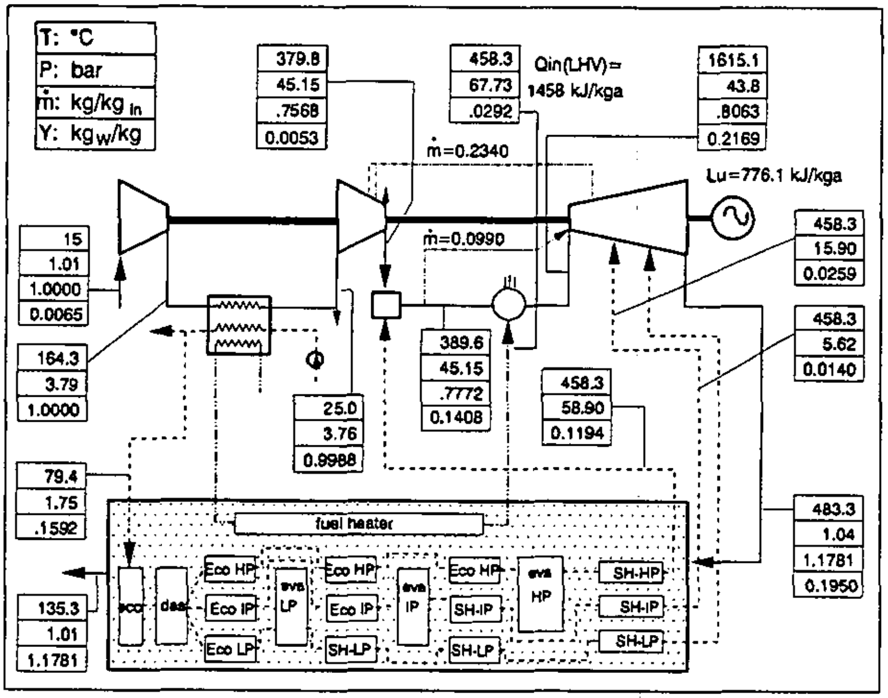

Fig.8: *Mass flow, pressure, temperature and water content at the most relevant points of an optimized ISTIG cycle with $\beta=45$ and TIT=1500°C.*

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