Int. J. Therm. Sci. (2001) 40, 372–384

S1290-0729(01)01225-X/FLA

Thermodynamic analysis of chemically recuperated gas turbines

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(Received 12 November 1999, accepted 21 July 2000)

Abstract — Significant research effort is currently centred on developing advanced aero-derivative gas turbine systems for electric power generation applications, in particular for intermediate duty operation. Compared to industrial gas turbines, aero-derivatives offer high simple cycle efficiency, a quick and frequent start capability without significant maintenance cost penalty. A key element for high system performance (efficiency and power output) is the development of improved *heat recovery systems*, leading to advanced cycles such as the *STeam Injected Gas Turbine* (STIG) cycle, *Humid Air Turbine* (HAT) cycle or the *Chemically Recuperated Gas Turbine* (CRGT) cycle. In this paper the chronology of development of this last technology and a detailed description of our research program "Thermodynamic analysis of chemically recuperated gas turbines" is presented. A comparative study of the performance potentials of CRGT cycles and the other advanced cycles for design and off-design mode is presented. The analysis method accounts for turbine blade cooling requirements, which have a decisive impact on cycle performance. Exergy calculations are included in the analysis method. Research perspectives for this technology are suggested. © 2001 Éditions scientifiques et médicales Elsevier SAS

gas turbine / chemical recuperation / methane steam reforming / part-load performance / humid air turbine / steam injection / exergy analysis

Nomenclature

c_p	specific heat	$\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$
ex^{ch}	molar chemical exergy	$\mathrm{J \cdot mol^{-1}}$
h	local heat transfer coefficient	$\mathrm{W \cdot m^{-2} \cdot K^{-1}}$
h	specific enthalpy	$\mathrm{J \cdot kg^{-1}}$
K	global heat transfer coefficient	$\mathrm{W \cdot m^{-2} \cdot K^{-1}}$
$\dot{m}$	mass flow rate	$\mathrm{kg \cdot s^{-1}}$
n	power law coefficient	
$\dot{N}$	molar flow rate	$\mathrm{mol \cdot s^{-1}}$
LHV	lower heating value	$\mathrm{J \cdot kg^{-1}}$
p	pressure	Pa
Q	heat	J
S	heat exchanger surface area	$\mathrm{m^2}$
s	specific entropy	$\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$
St	Stanton number	
T	temperature	K
TIT	turbine inlet temperature	K
ΔT_{eq}	chemical approach to equilibrium	K
X	molar fraction	

Greek symbols

β	compressor pressure ratio	
ε_{b}	blade cooling efficiency	
ϕ	coolant characteristic parameter	
η	density	$\mathrm{kg \cdot m^{-3}}$
χ	steam-to-methane ratio	$\mathrm{mol \cdot mol^{-1}}$
μ	molar chemical potential	$\mathrm{J \cdot mol^{-1}}$

Subscripts

air	compressor inlet air
c	cold side flow
cool	turbine cooling air
D	desig
ec	economizer
ev	evaporator
exit	reformer cold side outlet
h	hot gas flow
MSR	methane steam reformer
sh	superheater

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1. INTRODUCTION

Chemical recuperation is one of several innovative concepts applicable to natural gas fired gas turbine based power generation cycles. *Figure 1* illustrates the CRGT concept. Exhaust heat is recovered in a Heat Recovery Steam Generator (HRSG) where the superheater section is replaced by a Methane Steam Reformer (MSR).

The natural gas fuel is mixed with the generated steam and fed into the MSR. In the reformer, the mixture of natural gas and steam is heated by the combustion turbine exhaust, and an endothermic reaction occurs between the methane and the steam. The reaction requires the presence of a nickel based catalyst, and results in the production of CO, CO_2, H_2, excess steam and unconverted methane. The methane/steam mixture absorbs heat thermally (as it is heated), and chemically (as the endothermic reaction proceeds), resulting in a larger potential recuperation of exhaust energy than can be obtained by conventional recuperation. The reformed fuel is then fed into the turbine combustor. The basic CRGT cycle shown in *figure 1* can be improved by using one or more of the following performance enhancement concepts: compression intercooling (dry or by water injection), reheat combustion, and multipressure heat recovery system. One of the key advantages of CRGT cycle is its potential for ultra-low NO_x emissions due to the presence of a significant amount of steam in the reformate gas. Temperatures can thus be kept low in the primary zone of the gas turbine combustor, greatly reducing the formation of thermal NO_x.

NO_x emissions predictions for these cycles are as low as 1 ppm, without using selective catalytic reduction of the exhaust gas stream, as discussed by Lloyd [1].

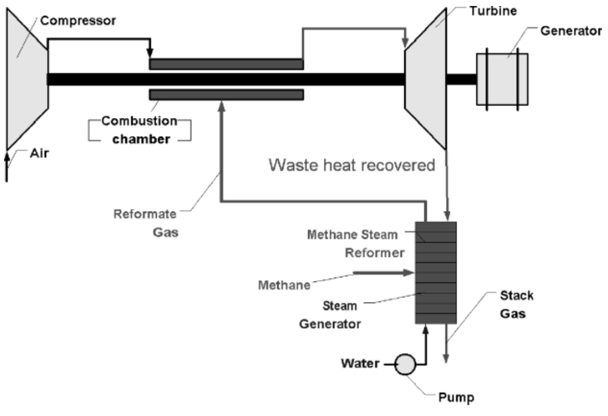

Figure 1. Chemically recuperated gas turbine cycle schematic.

2. REVIEW OF RESEARCH UP TO DATE

Several innovative cycle configurations have been proposed as alternatives to the well-established combined cycle configuration (gas turbine power plant with steam bottoming cycle). *Table I* presents a summary and description of the more relevant cycle configurations that have been proposed. A paradoxal aspect of the development of the stationary gas turbine is that most of these well-known cycle modifications available for improving performance, e.g., intercooling in the compressor (ICAD cycle), evaporative regeneration (HAT cycle), steam injection in the combustion chamber (STIG cycle) and steam reforming of fuel (CRGT cycle), are still largely not exploited, even though enormous "high-technology" advances have been made in turbine blade materials, design, and manufacturing. This is because such cycle modifications involve modification of the gas turbine design through the use of large quantities of steam or water in the cycle or the introduction of large heat exchangers between the gas turbine's modules, neither of which is relevant to aircraft applications. This situation presents an enormous opportunity because it means that major improvements can be made in the performance of gas turbines for stationary power applications with reasonable R&D efforts such as with the CRGT cycle.

The main promoter for the CRGT cycle has been the California Energy Commission (CEC), which in the early 90s was interested in CRGT as a means of meeting new NO_x emission standards proposed by the South Air Quality Management District. In a CEC report [2] the efficiency of a CRGT cycle derived from the CF6-80C2 aero-engine with intercooling and reheat (*figure 2*) was estimated to be as high as 59 %. The CRGT cycle concept proposed by the CEC group includes a number of state-of-the-art performance enhancement features. The turbine is cooled by steam extracted from the steam generator, which is dual pressure. The HP and LP steam reformers are in a parallel configuration. The CEC group suggested that independent studies by several groups should be carried out for a new concept such as the CRGT, particularly in the early stages. Based on the CEC recommendation, a parametric comparison of simple CRGT and STIG cycles was conducted by Lloyd at Princeton University [1]. He showed, for a simple gas turbine configuration, superior efficiency but reduced work for a CRGT cycle compared to a STIG. Lloyd also investigated complex configurations using intercooling and reheat. The efficiency of an intercooled reheat CRGT cycle based on a hypothetical General Electric LM8000 machine, which could be derived from the GE aircraft jet engine, the −80C2, was estimated at about 52.7 %. The

TABLE I
Gas turbine Power Generation Technology summary.

Acronyms	Definition	Main components	Main feature	Market and status
STIG	STeam Injected Gas turbine	GT (mostly AD) + HRSG	The GT exhaust heat is recovered to produce steam directly injected in the combustion chamber. Compared to CCGT: lower efficiency & power output but lower cost for midsize power plant (no condenser, no steam turbine) $\Rightarrow$ intermediate load.	Commercially available for intermediate load cogeneration applications. LM5000-STIG: 51.16 MW, $\eta = 43.8\,\%$, 272 $\$\cdot\text{kW}^{-1}$.
HAT	Humid Air Turbine	GT + Regenerator + Saturator	The heat contained in the exhaust gas is recovered to saturate and then reheat the air at the outlet of the compressor. Compared to CCGT and STIG, Power output, efficiency & cost mid way CCGT and STIG cycle $\Rightarrow$ base and intermediate load application.	One machine available on the demonstration scale "CHAT" developed by Westinghouse.
ICAD GT	InterCooled AeroDerivative Gas Turbine	Modified AD GT + Compressor Intercooler	An intercooler is added at the outlet of the LP compressor. Compared to AD GT: decreased compressor power consumption & decreased turbine air cooling flow $\Rightarrow$ increased power output $+100\,\%$ (AD), increased efficiency $+15\,\%$ (AD).	No machine currently available. Main promoter: the CAGT programme for intermediate load application.
CRGT	Chemically Recuperated Gas Turbine	GT + HRSG + MSR (Methane Steam Reformer)	The exhaust heat is recovered to produce steam which instead of being superheated as in STIG cycle, is mixed with methane to drive a highly endothermic reaction in the MSR (chemical recuperation). Primary advantage: NO_x are reduced to 1–3 ppm and the efficiency reaches 50 %.	No machine currently available. Main promoter: California Energy Commission. Intermediate load application midsize power station. Its market depends on the future regulations of NO_x emissions.

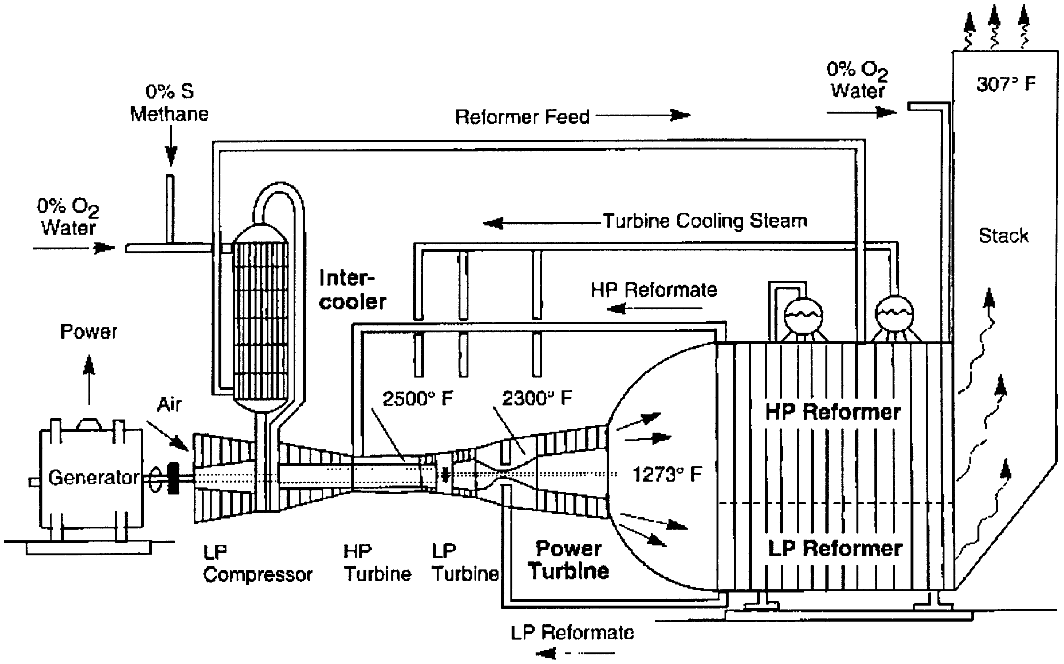

Figure 2. Intercooled Reheat CRGT cycle based on the CF6-80C2 considered by the California Energy Commission.

efficiency of a reheat combined cycle based on the same gas turbine engine was estimated to be 53.6 %. He also pointed out that:

- the location of the intercooler and the reheat combustion chamber were not optimised;
- the cooling model (especially in reheat cycles) seems to greatly influence the performance estimation;
- part load operation has to be addressed especially if CRGT cycles are to operate at intermediate load.

The performance of CRGT cycles were compared to combined cycles assuming both heavy-duty and aeroderivative gas turbine engines by Kesser et al. [3] from the University of California. They used more detailed turbine blade air cooling flow estimations than Lloyd. The combined cycle was shown to have the highest efficiency. The basic CRGT cycle with a compressor ratio of 15 : 1 and TIT = 1 200 °C was shown to have a thermal efficiency of about 47.3 % compared to 49.7 % for the combined cycle. Kesser et al. strongly recommended that in order to obtain a deeper insight into the origin of the losses in a CRGT cycle, an exergy analysis should be performed.

More recently a thermodynamic and environmental assessment of a CRGT system was carried out for a RB211 based compressor station by NOVA Chemicals in Canada [4]. The cycle was evaluated using ASPEN Plus. Their results confirmed that the application of the CRGT concept to the RB211 showed reasonable increases in efficiency (from 36 to 42 %) and power output (from 18.5 to 24.0 MW) compared to the simple cycle gas turbine. The CRGT cycle efficiency is always higher than that of the corresponding STIG cycle. The study includes also some results of off-design performance calculations. However, their work assumes that the design and off-design performance behaviour of the base engine are unaffected by the cycle modifications implemented for CRGT operation. A preliminary economic evaluation of the CRGT system indicated that it is economically feasible only if a low-cost source of water is available. The cash flow analysis of the evaluated system indicated that a return on capital investment could be achieved in 9 years.

Chemical engineering aspects of the steam reforming reaction in a CRGT cycle have been discussed by several authors. Rostrup-Nielsen et al. [5] discuss the severe constraints imposed by thermodynamics of steam reforming of natural gas on CRGT performance potential. The kinetics of the steam reforming reactions were also studied by Murray and Snyder [6] who developed a kinetic model of a steam/methane reformer as a design and analytical tool for a fuel cell system conditioner.

3. GOAL OF THE PRESENT WORK

In 1995 the École des Mines de Nantes (Nantes, France) initiated two projects in parallel in collaboration with the University of Florence:

1. Design and off-design operation issues of the CRGT cycle based on the LM2500 aeroderivative gas turbine, including comparisons with a STIG cycle based on the same engine [7, 8].
2. Thermodynamic analysis of the CRGT cycle performance potential at full and part load operation, considering various gas turbine configurations in order to identify a feasible and high performance configuration.

The present paper is a synthesis of the second project and its final conclusions. The goals of the project were:

(1) to understand the thermodynamics of chemical recuperation through exergy analysis tools;
(2) to optimise the thermodynamic performance of CRGT cycles based on advanced configurations such as reheat and intercooling through a parameter sensitivity analysis and the investigation of alternative fuels such as methanol;
(3) to identify a high-efficiency, low-cost configuration based on an existing gas turbine and compare it to an optimal combined cycle based on the same gas turbine;
(4) to simulate off-design performance of the CRGT cycle.

4. COMPUTER MODEL

All cycle evaluations were computed using Aspen Technology's ASPEN+ process simulation code. However, the following extensions had to be made to the base ASPEN+ code in order to adapt it to simulation of gas turbine cycles at nominal and part-load operating conditions.

4.1. Turbine cooling model

High temperature gas turbine performance levels are very sensitive to blade cooling requirements. Different cooling models have been developed in the past for blade cooling. Elmasri's model [9] is by far the most complete empirical approach to blade cooling. It represented the turbine as an expansion path with continuous, rather than discrete, work extraction. If the aim is to analyse the global performance of the cycle rather than focus on the blade cooling, a discrete model is computationally more convenient since it reduces considerably the simulation time. Discrete models were also developed by several authors, the most relevant is Consonni's model [10] which reduces the expansion path into a large number of discrete elementary operations containing each a mixing process between the hot gas and the cooling fluid followed by the expansion process. For our study, we considered the cooled turbine model proposed by Stecco and Facchini [11][1]. The model considers the turbine stage-by-stage, and estimates the cooling flow necessary for the stator and rotor of each stage. The stator flow is assumed to mix with the main gas flow prior to flow through the turbine, whereas the rotor coolant flow is mixed into the main stream at the rotor exit. For internal convection cooling, the coolant flow can be estimated using

$$\dot{m}_{\mathrm{cool}} = \dot{m}_{\mathrm{g}} \frac{c_{p,\mathrm{g}}}{c_{p,\mathrm{cool}}} St_{\mathrm{g}} \frac{Q_{\mathrm{b}}}{Q_{\mathrm{g}}} \frac{1}{\varepsilon_{\mathrm{g}}} \frac{\phi}{1-\phi}$$

where subscripts g and cool refer to the main gas stream and the blade cooling stream, respectively. For an advanced gas turbine generation, standard values for St_{g}, and $Q_{\mathrm{b}}/Q_{\mathrm{g}}$ and ε_{b} are 0.005, 4 and 0.3, respectively [9]. ϕ is defined by

$$\phi = \frac{T_{\mathrm{g}} - T_{\mathrm{b}}}{T_{\mathrm{g}} - T_{\mathrm{cool}}}$$

in which T_{b} refers to the turbine blade metal temperature. If full or partial film cooling is associated with internal convection on the external blade surface, the overall cooling flowrate requirements are reduced, typically by 10–30 % [11]. We derived sets of input parameters (cooling parameters, polytropic efficiencies, combustor efficiency etc.) to represent the current and advanced generation of gas turbines. The used data were obtained by calibrating the model against the published performance data of some representative gas turbine engines. Given these data, the model was then validated by reproducing approximately the performance of six gas turbines by running the model with the same pressure ratio, turbine inlet temperature (if provided) and mass flow rate of the exhaust gas.

Figure 3 shows the (T, s) diagram for the LM6000PA simulated with our model with a typical pressure ratio 30 : 1 and turbine inlet temperature 1 575 K for an aeroderivative gas turbine. We can note on the diagram the three successive bleeds of air from the compressor to cool the turbine and its effect on the (T, s) diagram.

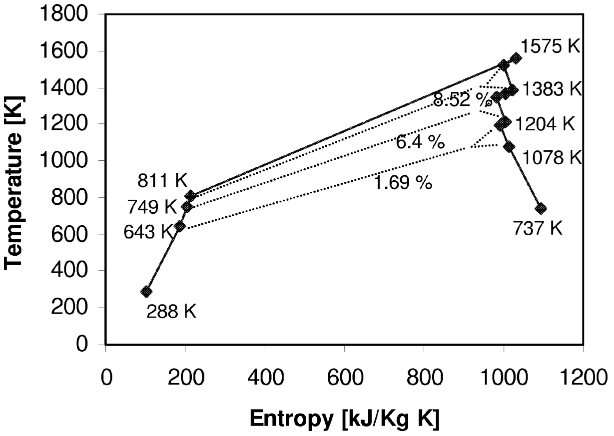

Figure 3. (T, s) diagram of an aeroderivative gas turbine cycle.

4.2. Methane and methanol steam reforming reactions

The reaction between steam and natural gas, often termed "steam reforming", is widely used in the chemical process industry for hydrogen production for ammonia or methanol synthesis. Typical reactor conditions are 700 to 900 °C and up to 35 bar. The steam-to-methane ratio is in the range 3–5 to prevent carbon formation that can reduce the effectiveness of the nickel-based catalyst. Methane reacts with steam via two independent reactions:

$$CH_4 + H_2O \rightarrow CO + 3\,H_2 \tag{1}$$

$$CO_2 + H_2 \rightarrow CO + H_2O \tag{2}$$

Reaction (1) is highly endothermic, while the second, often known as the water–gas shift reaction, is exothermic. This reaction is undesirable since it reduces the net endothermicity, but is unavoidable. Low pressure, high temperature and high steam-to-methane molar ratio as shown in *figure 4* favour methane conversion which corresponds to a molar ratio of water to methane equal to 6 : 1.

Reaction (1) is rate-limited by reaction kinetics, whereas the shift reaction can be assumed to be at equilibrium for the conditions considered. The steam-to-methane ratio must be chosen sufficiently high to prevent carbon coking. Typical steam-to-methane ratios are in the 3–6 range. Reaction (1) proceeds in the presence of a catalyst, usually nickel-based. For the low temperature reforming considered in this case, it is necessary

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[1] A detailed review of these cooling models could be found in the PhD document [12].

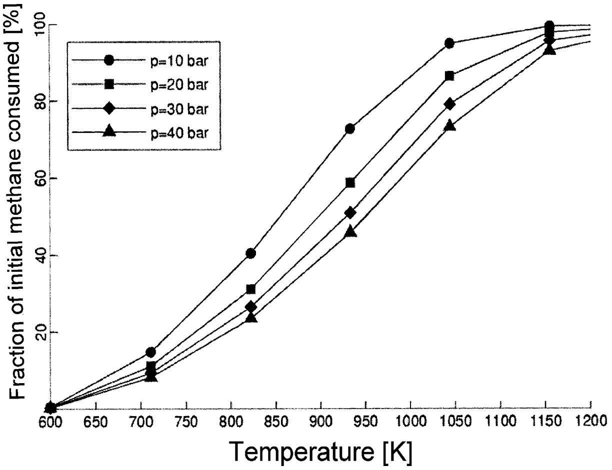

Figure 4. Methane conversion in steam reforming reaction for a molar steam-to-methane ratio 6.

to use catalysts that are active at low temperatures. According to information provided by Janes [2], catalysts can be found that are sufficiently active for temperatures above 600 K. For an overall cycle analysis, the chemical nonequilibrium effects due to reaction kinetics can be modelled using the chemical approach to equilibrium ΔT_{eq}. The approach to equilibrium indicates how close a reaction is to reaching equilibrium, and is a measure of catalyst performance. An approach to equilibrium of 10 K signifies that the composition is the equilibrium composition at $(T - 10)$. Its value depends upon the catalyst operating temperature. Based on information on the available catalysts for methane steam reforming provided by Lloyd [1], we developed the following correlation for ΔT_{eq}:

$$\Delta T_{\mathrm{eq}} = 250\mathrm{e}^{-B(T-573)}$$

$$\text{for } T_{\mathrm{exit}} > 573\ \mathrm{K},\ B = 0.016094$$

For $T_{\mathrm{exit}} < 573$ K, no conversion is observed at the reformer exit. In such cases, the reformer can be modelled as a simple heat exchanger. *Figure 5* shows the methane conversion ratio versus the reforming equilibrium temperature with and without ΔT_{eq}. It shows clearly that for low temperatures the use of the approach to equilibrium temperature is essential.

In a gas turbine, the steam reforming temperature will be that of the exhaust gas at the outlet of the turbine (500–800 K) and the pressure at which the steam reforming occurs will be that of the compressor outlet (10–35 bar). Therefore, equilibrium constraints will prevent complete consumption of the reactants. This restricts the amount of chemical heat recovery possible.

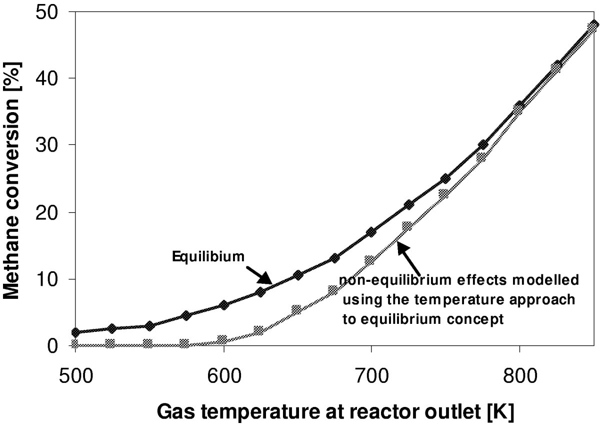

Figure 5. Effect of the use of the temperature approach to equilibrium.

4.3. Off-design model

The basic approach used to describe the different components of the gas turbine system is that described by Dixon [13]. The complete turbine system model, which is integrated into the gas turbine performance simulation programme ESMS [14], was then coupled to ASPEN+ process simulation software where we developed the HRSG and the methane steam reformer MSR model in order to simulate STIG and CRGT cycles. *Figure 6* shows how the two codes are coupled for the case of a basic CRGT cycle. The ESMS code is used for

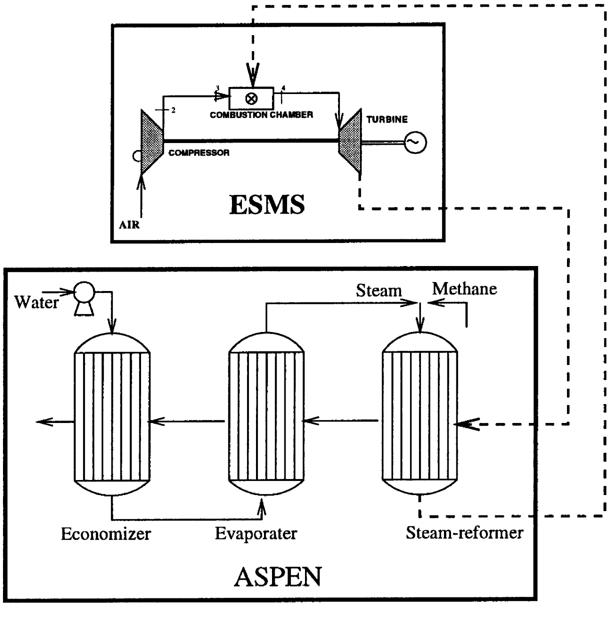

Figure 6. Coupling of the modified ESMS code with ASPEN+.

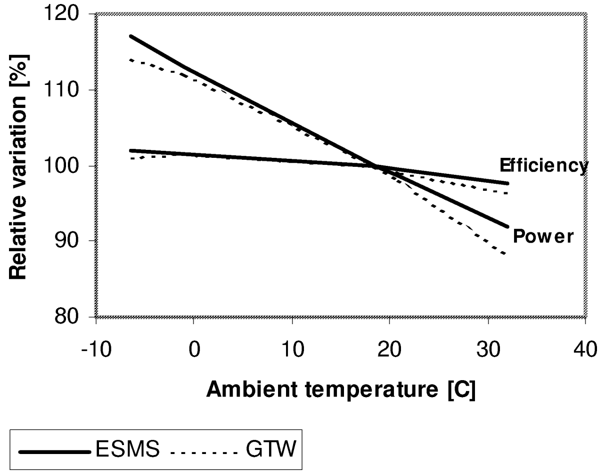

Figure 7. Comparison of ESMS results with published data in the Gas Turbine World (GTW) performance list.

simulation of the gas turbine, whereas ASPEN+ is used for the remainder of the cycle. The gas turbine model was validated by reproducing, for a simple gas turbine, published performance data (efficiency and power versus ambient temperature, see *figure 7*) for a heavy duty gas turbine operating in design and off-design modes with a maximum error of 2 to 3% on the efficiency and power output.

Minor modifications were made to the original code to adapt it for STIG and CRGT cycle simulation. Thus, the combustion chamber reaction equations were modified to allow a fuel stream containing a mix of H_2, H_2O, CO, CO_2 and CH_4, such as encountered in a CRGT cycle. The compressor and turbine models were not modified.

The turbine section was increased to accommodate the increased flowrate due to the low heating value reformate fuel stream used in the CRGT cycle. Indeed, we assumed a turbine optimally designed for the cycle in question. Another option could be considering the original turbine design and section which would limit the amount of water injected into the cycle due to the compressor surge margin. However, since the aim of this paper is to investigate the potential of the CRGT cycle, we only considered the first option. The parameters used to characterise blade cooling are representative of established heavy-duty gas turbines such as the GE Frame 6 series.

For combined cycle simulations, the ESMS code developed at the University of Florence also includes a steam bottoming cycle module that can be interfaced with the gas turbine module. The steam cycle simulation program developed by Bettagli and Facchini is described in detail in Bettagli's work [15]. The heat recovery model for STIG and CRGT cycles was developed in ASPEN+. The program computes, in design mode, the required surface of the different sections of the heat recovery device, based on pressure drops, required steam flows and conditions, pinch-point temperature differences, and approach subcooling values specified by the user. In off-design mode, the overall heat transfer coefficient is varied according to the variation of the gas-side heat transfer coefficient, using the following polynomial law recommended by Kehlhofer [16]:

$$\frac{K}{K_{\mathrm{D}}} = \frac{h}{h_{\mathrm{D}}} = \left(\frac{\dot{m}_{\mathrm{D}}}{\dot{m}_{\mathrm{h,d}}}\right)^{n} \left[1 - \left(\overline{T}_{\mathrm{D}} - \overline{T}\right) \cdot 5{\cdot}10^{-4}\right]$$

with $n = 0.56$–0.58 for staggered tubes and 0.59–0.65 for in-line tubes. $\overline{T}_{\mathrm{D}}$ and $\overline{T}$ denote respectively the mean gas temperatures at design and off-design conditions.

The pressure drop is determined using the following equation:

$$\frac{\Delta p}{\Delta p_{\mathrm{D}}} = \frac{\rho_{\mathrm{D}}}{\rho}\left(\frac{\dot{m}^2}{\dot{m}_{\mathrm{D}}^2}\right)$$

This equation assumes that the variation of the frictional loss coefficient is negligible.

4.4. Fortran subroutine to perform exergy analysis

ASPEN+ does not have a built-in exergy analysis function, and thus, computations of stream exergies and component exergy losses are tedious. We built in an exergy subroutine [17] in which we generalised the chemical exergy concept of fuel introduced by Moran [18] to every $C_\alpha H_\beta N_\gamma O_\delta$ component in order to calculate the exergy of various fuels such as methanol.

The exergy flows of the material streams in the process were computed according to Moran:

$$\dot{Ex} = \dot{m}\left[(h - h_0) - T_0(s - s_0)\right] + \dot{N} ex_{\mathrm{mix}}^{\mathrm{ch}}$$

where subscript 0 refers to the reference conditions, chosen as 298.15 K, 101 325 Pa for this study, $\dot{m}$ the mass flow rate, and $\dot{N}$ the molar flow rate. H_2O is in the liquid state at reference conditions. The last term refers to the molar mixture chemical exergy of a single-phase flow:

$$ex_{\mathrm{mix}}^{\mathrm{ch}} = \sum_{i=1}^{n} X_i ex_i^{\mathrm{ch}} + RT_0 \sum_{i=1}^{n} X_i \ln X_i + G^{\mathrm{E}}$$

The last term, G^E, is the excess free Gibbs energy, negligible at low pressure in a gas mixture. We generalised the chemical exergy concept of fuel to every $C_\alpha H_\beta N_\gamma O_\delta$ component. The molar chemical exergy ex^{ch} of such a component will be

$$ex_c^{ch} = \left(\mu_{c,0} - \mu_c^e\right)$$

μ_c^e refers to the chemical potential of the component in the restricted dead state "e" derived by using a Van't Hoff equilibrium box. We found that a practical computation form for the chemical potential is

$$\mu_c^e = \alpha\mu_{CO_2}^e + \frac{\beta}{2}\mu_{H_2O}^e + \frac{\gamma}{2}\mu_{N_2}^e + \left(-\alpha - \frac{\beta}{4} + \frac{\delta}{2}\right)\mu_{O_2}^e$$

$\mu_{c,0}$ represents the chemical potential of the components in their thermo-mechanical equilibrium state ($T = T_0$, $p = p_0$) with the standard environment.

We can then apply the exergy analysis concept to every component of the gas turbine cycles by determining the sum of the exergy streams into every module $\sum Ex_i$, the sum of the exergy streams out of every module $\sum Ex_0$. The difference between these two streams is the exergy destruction Ex_D which corresponds to the irreversibilities of the process occurring inside the modules:

$$\left|\sum Ex_i - \sum Ex_0\right| = Ex_D$$

The exergy streams include the exergy associated to the heat transfer and work. For more details on how to apply the exergy concept, the reader is referred to Moran [18] and Kotas [19].

5. RESULTS

5.1. Exergy analysis of CRGT cycles

Table II presents the key results for the two cycles STIG and CRGT. We considered a heavy duty gas turbine configuration ($\beta = 15$) for both cycles. The TIT was fixed to 1 575 K and the steam to methane ratio was fixed to 6 for the CRGT cycle. The pressure of the reformate fuel gas stream is 18 bar. We note in the table that the introduction of the chemical recuperation has two major effects:

(1) increase in the thermal efficiency,
(2) decrease in the power output.

TABLE II
Key results for simple STIG and CRGT cycles.

	CRGT	STIG
Pressure ratio β	15	15
TIT [K]	1 575	1 575
η_{el} [%]	49.86	47.88
W [kJ·kg$_{air}^{-1}$]	565	633
Reformed gas pressure	18	–
Steam-to-methane ratio	6	–
Methane conversion [%]	40.5	0
$\dot{m}_{H_2O}$ [% of $\dot{m}_{air}$]	14.37	19.8

TABLE III
Exergetic comparison of STIG and CRGT cycles.
TIT = 1 575 K, $\beta = 15$.

	Exergy losses [%]	
	CRGT	STIG
Combustor	27.48	28.55
Heat exchangers	7.45	8.84
Stack gas	6.89	7.11
Turbine	5.06	4.79
Compressor	2.45	2.08
Auxiliaries	2.43	2.45
Exergy efficiency	48.05	46.18

The decrease in the power is due to the decrease of the mass flow rate in the turbine $\dot{m}_{H_2O}$ (expressed in percentage of the compressor inlet air flow). However, it should be noted that no restriction was imposed on the amount of water that could be injected in the STIG cycle since in each case the turbine section was enlarged in order to accommodate this flow increase. If the two cycles were compared using the same turbine section (therefore, the same mass flow rate), the CRGT cycle would achieve comparable power output to the STIG cycle and higher efficiency. The latter was shown in a previous paper by one of the authors [7]. *Table III* shows the exergy losses, presented as percentages of the exergy inlet flow to the cycle (fuel) in order of decreasing magnitude.

As expected, the largest exergy loss in each cycle occurs in the combustor followed by those of the heat exchangers and the stack gas. The increase in the efficiency, when chemical recuperation is introduced, is due to the decrease in these exergy losses and there are two reasons for this:

1. The reduction of the amount of water used in the cycle has two direct implications:

- exhaust gas exergy is reduced since it contains less latent heat associated with the water vapour (7.11 %

losses for the STIG cycle compared to 6.89 for the CRGT cycle);

- the exergy losses associated with the mixing of the water vapour with the main stream in the combustor is reduced (28.55 % losses for the STIG cycle compared to 27.48 % for the CRGT cycle).

2. By replacing the superheater with a steam reformer the average temperature difference between the hot and cold streams in the heat recovery device is reduced, and therefore, the exergy losses are also reduced (8.84 % losses for the STIG cycle compared to 7.45 % for the CRGT cycle).

The difference between the two cycles decreases when the pressure ratio is increased. This is because high pressure ratios (resulting in low temperature exhaust gas) reduce the methane conversion ratio (see *figure 4*), and therefore, the methane steam reformer gradually becomes a superheater as in a STIG cycle.

5.2. Parametric analysis of advanced CRGT configurations

Performance of the CRGT cycle was computed for the following cycle configurations:

- Cycles associated with single pressure steam reforming:
  (1) simple gas turbines $10 < \beta < 35$,
  (2) intercooled gas turbines $15 < \beta < 35$.
- Cycles associated with dual pressure steam reforming:
  (1) reheat gas turbines $25 < \beta < 35$,
  (2) reheat intercooled gas turbines $25 < \beta < 35$.

The results are shown in *figure 8*. For reheat configurations, only high overall pressure ratios were considered, typical of aero-derivative gas turbines. The location of the intercooler was optimised by identifying the intercooling pressure that maximises cycle efficiency. Also, in order to maximise cycle efficiency, the reheat temperature should be as high as possible. In this study we assume that the same temperature can be reached in the reheat combustor as that reached in the main combustor. The reheat pressure is set such that the turbine exhaust gas temperature is 900 K, which allows conventional HRSG materials to be used in the methane steam reformer. We compared the performance of these configurations to the performance of an intercooled humid air turbine called REVAP. The REVAP cycle is an innovative evaporative cycle in which the saturation tower has been eliminated and replaced by a two-phase flow heat exchanger recently proposed by DeRuyck et al. [20] and identified to be close to the optimum Humid Air Turbine (HAT) cycle configuration for the given pressure ratio and TIT. *Figure 8* shows the overall results of the analysis (cycle efficiency versus specific work) for the different cycles considered. Specific work is reported per kg of air at the compressor inlet.

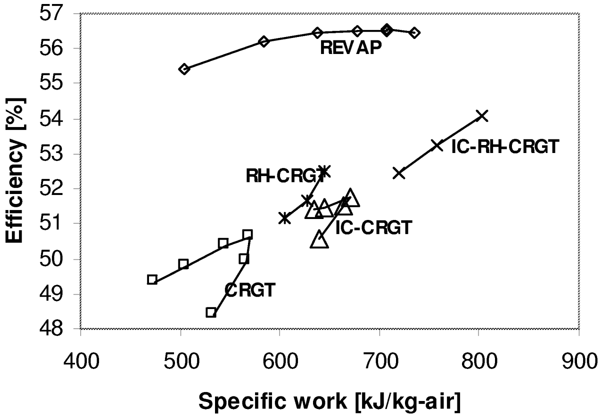

Figure 8. Performance potential of advanced CRGT cycles compared to REVAP cycle.

5.2.1. Intercooler heat recovery

We investigated the possibility of recovering the heat released in the intercooler through thermal or chemical recuperation [21]. The simulations showed that the low-grade heat released by the intercooler could not be put to use in the cycle where low-grade heat is already plentifully available in the low temperature stack gas.

5.2.2. Basic CRGT configuration

For the basic CRGT cycle, *figure 8* shows that the efficiency first increases with cycle pressure ratio, then passes through a maximum and thereafter decreases. This is due to the minimum temperature difference imposed at the pinch point in the boiler. The stack temperature increases from 383 K for $\beta = 20$ up to 427 K for $\beta = 30$.

5.2.3. Intercooled CRGT configuration (IC-CRGT)

Intercooling improves the efficiency of the CRGT cycle only if its location is optimised. This is because intercooling has two effects on the cycle:

- Negative impacts:

(1) The required fuel flowrate to achieve the TIT increases so the efficiency will tend to decrease.

(2) The quantity of steam needed to achieve $\chi = 6$ must be increased. However, for a given inlet temperature and expansion ratio, intercooling has little effect on the turbine outlet temperature, and therefore, there is not enough heat to raise the supplementary amount of steam required. Thus, χ decreases (for $\beta = 15$, $\chi = 4.5$) which in turn decreases the methane conversion in the reformer, therefore, the positive effect on efficiency of the chemical recuperation is reduced.

- Positive impact:
  (1) the compressor power requirement is reduced,
  (2) the turbine blade air cooling is reduced.

Figure 8 shows that, as for the basic CRGT cycle, as the pressure ratio increases above 25, the efficiency decreases as a result of the pinch point constraint in the heat recovery exchanger.

5.2.4. Reheat CRGT configuration

Introducing reheat into the cycle for high pressure ratios increases the specific work output as well as the efficiency. The increase in the power output is due to the increase in the exhaust gas temperature, and therefore, the amount of steam that can be generated and expanded in the turbine to produce power. As discussed for the intercooled cycle, reheat has both negative and positive impact on the efficiency of the cycle:

- Positive impacts. Reheating the main stream improves the cycle thermodynamics in many ways:

(1) The irreversibilities in the second combustion process are low since the main stream enters the combustor at a much higher temperature than in the first combustor. Globally the combustion process exergy losses are reduced.

(2) The use of the double pressure recovery allows the recovery of a greater portion of the heat contained in the exhaust gas (up to the maximum steam to methane ratio allowable). Therefore, the exergy losses associated with the exhaust gas are reduced. It can clearly be noted on *figure 8* that the efficiency does not decrease as in simple and intercooled CRGT cycles when the pressure ratio increases.

- Negative impacts. As a result of reheat, the air cooling requirement for the turbine blade increases, and therefore, the exergy losses due to mixing of the cooling air into the main stream increase.

Globally, the efficiency of the cycle is improved by 1.83 percentage points due to reheat and the same effect is observed when reheat is introduced to the intercooled CRGT.

So, we may conclude that intercooling and reheat do improve the efficiency of the CRGT cycle. However, the increase remains modest (the highest potential thermal efficiency achieved is 54 %) for a fairly high turbine inlet temperature (TIT = 1 573 K) compared to the best CCGT cycle which today achieves 60 %. The methane steam reforming is limited by thermodynamics (low conversion ratio). The figure shows clearly also that the REVAP cycle has a higher efficiency potential than the CRGT cycle.

5.2.5. Replacing methane with methanol

The thermodynamic constraint of the methane steam reforming disappears when using methanol as primary fuel. In a previous paper [17], we investigated the capability of the chemical recuperation considering methanol and methane as primary fuels for reheat intercooled chemically recuperated gas turbine cycles using exergy analysis. The analysis revealed that the exergetic efficiency was essentially the same for both fuels. We then concluded that the basic CRGT cannot be modified to achieve an efficiency competitive with that of a combined cycle for base load applications, so the development effort should concentrate on simple CRGT cycle for intermediate load applications.

5.2.6. Optimal gas turbine type for simple CRGT cycles

Simulations were conducted for simple CRGT cycle based on four simple aeroderivative gas turbines (LM 1600PA, LM 2500PE, LM 5000PC, LM 6000PA) and two heavy duty gas turbines (PG7121EA, PG7231FA). Their performance were compared to the performance of their respective combined cycle configuration (CC1-1600, CC1-2500, CC1-5000, CC1-6000) and (S-107EA, S-107FA) (*figure 9*) in terms of efficiency and power output.

The figure shows that for aeroderivative gas turbines, CRGT cycles are comparable to CCGT in terms of efficiency and higher power output. However, the efficiency and power output of the CRGT cycle are inferior to those of CCGT cycle for heavy-duty gas turbines. Aeroderivative gas turbine operating parameters are not optimised for CCGT operation, unlike heavy duty configurations, where the combination of high gas turbine exhaust temperature and triple pressure HRSG favour high-efficiency.

We may, therefore, conclude that CRGT cycle is a competitor to CCGT cycle only for intermediate load applications based on aeroderivative gas turbines.

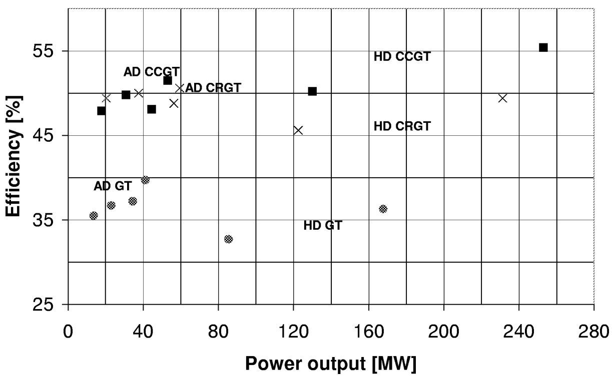

Figure 9. Performance comparison of CRGT and CCGT cycles for aeroderivative (AD) and heavy duty (HD) gas turbines.

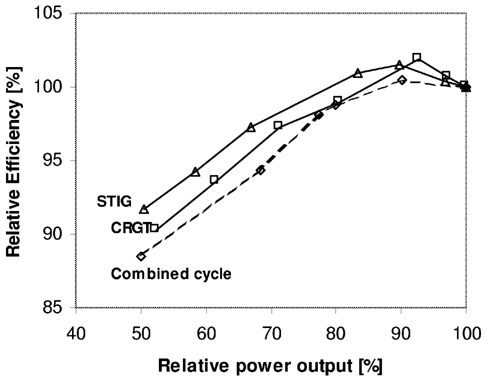

Figure 10. Relative part load performance of CRGT, STIG and combined cycles.

5.3. Off-design performance of simple CRGT cycle

We investigated the use of the various power control systems usually found on industrial gas turbines:

- Inlet Guide Vane (IGV): the inlet guide vane opening angle can be varied, thus slightly reducing the mass flow rate at the compressor inlet. The angle of the fluid entering the compressor is also varied.
- Mass Fuel Rate variation (MFR): reducing the mass flow rate of the fuel entering the combustion chamber reduces the power output of the turbine. The overall mass flow rate passing through the machine is essentially unaffected.

It is well known that reducing power by closing the inlet guide vanes maintains a higher efficiency than reducing the turbine inlet temperature by reducing the fuel flowrate [14]. However, the range of power reduction that can be achieved using IGV variation is limited due to the surge margin of the compressor. This is not the case for MFR power reduction, which allows a much greater range of power reduction without risks of engine operation instabilities. Thus, optimal engine operation usually involves combined use of the two systems. Small reductions in power are achieved by closing the inlet guide vanes. Larger power reductions are obtained by reducing the turbine inlet temperature, also modifying the inlet guide vanes in order to maintain acceptable efficiency. Clearly, for cycles with heat recovery, reducing the turbine inlet temperature also reduces the turbine exhaust temperature, which in turn has a negative impact on the heat recovery device efficiency. Closing the inlet guide vanes, on the other hand, has a small effect on the exhaust flow rate and temperature, and thus, this power reduction mode has a much smaller impact on heat recovery efficiency. Thus, in this study we consider combined use of the IGV and MFR power reduction systems for the STIG, CRGT and combined cycle power plants considered.

Figure 10 shows the part load performance of the CRGT, STIG and combined cycles. The results shown are for the optimal mixed IGV-MFR power reduction system.

The figure shows the increase in efficiency of the STIG and CRGT cycles for a small load reduction achievable with IGV control with a clear advantage for the CRGT cycle. However, with the MFR control system, the efficiency of the STIG cycle decreases less rapidly at part load than that of the CRGT cycle.

The difference between the two cycles is due to the variation of the turbine outlet temperature and its consequence on the methane conversion ratio. At part load with the MFR control system, for the CRGT cycle, methane conversion tends to drop off rapidly as the turbine exit temperature decreases, and therefore, the efficiency of the CRGT decreases faster than the STIG cycle. The inverse is valid when IGV system is used, since the turbine outlet temperature increases and, therefore, the methane conversion increases.

The combined cycle efficiency increases only slightly with IGV power reduction. However, a much greater range of power reduction can be achieved using only IGV control than for the STIG and CRGT cycles.

6. CONCLUSIONS AND RESEARCH PERSPECTIVES

Few studies have been published to date presenting performance predictions for CRGT cycles. Furthermore, several of the studies announce optimistic performance levels, and it is not clear from the studies whether the performance gains claimed are due to the nature of the CRGT cycle, or due to optimistic assumptions for future gas turbine technology levels. In this study, we have presented a comparative study with the other advanced cycles such as STIG, HAT and CCGT cycles. The results have shown clearly that CRGT cycle efficiency values are superior to those of STIG cycles. However, they remain inferior to those of HAT cycles. When advanced CRGT configurations are considered (intercooling and reheat), the efficiency achieved does not reach more than 54 % even with relatively high TIT (1 573 K) and remains inferior to those of the best combined cycle available today even when alternative fuels with a greater chemical recuperation potential were investigated. Therefore, the most realistic configuration from an economical point of view would probably be the simple CRGT cycle which shows comparable performance to CCGT cycle when applied to aeroderivative gas turbine engines. This last configuration combines comparable efficiency to CCGT cycle (based on the same aeroderivative gas turbine) and slightly higher power output. The results showed also the superiority of the simple CRGT cycle on CCGT at part load. Therefore, future development effort should be carried out on the feasibility of the simple CRGT cycle. In this intermediate load application, the main competitor would be advanced cycles based on simple gas turbine such as the STeam Injected Gas turbine STIG and the InterCooled AeroDerivative gas turbine ICAD. Compared to the CRGT cycle, these last two cycles achieve moderate capital and development cost. This could limit the application of the CRGT cycles to cogeneration and coproduction applications in the basic chemicals process industry. The thermodynamics of the CRGT cycle and its performance potential are now well understood. Future research should focus on the environmental, technical and economical issues.

6.1. Environmental issues

The formation of nitrogen oxides (NO_x) during combustion is the main pollutant problem associated with gas turbines. NO_x plays a role in both acid rain formation and local air pollution. Increasingly strict environmental regulation has led to rapid development of NO_x control technologies in recent years. Gas turbines typically have uncontrolled NO_x levels of 150–250 parts per million (ppm) by volume when running on natural gas. NO_x control technologies fall into three categories: exhaust gas clean-up systems such as selective catalytic reduction (SCR), steam or water injection, and dry low NO_x combustor designs. Combustion strategies which reduce thermal NO_x often lead to increased carbon monoxide emissions, and so the solution adopted must be a compromise achieving acceptable levels of both. Regulatory limits in some areas are as low as 9 ppm for NO_x and 50 ppm for carbon monoxide. NO_x levels depend primarily on the peak temperatures in the primary zone. Steam or water injection leads to reduced NO_x levels because the diluting effect of the water or steam lowers the stoichiometric adiabatic flame temperature. An upper bound to the amount of H_2O injection is set by the occurrence of flameout below a certain flame temperature and high carbon monoxide emissions. With CRGT, it is believed that the presence of a significant quantity of hydrogen in the fuel gas diluted with a significant quantity of steam enables these low temperatures to be reached resulting in potential ultra-low NO_x emissions. Indeed, hydrogen has a much lower flameout temperature than methane (around 1 300 K compared to 1 800–2 000 K for methane), so one would expect the flameout temperature of the reformed gas to be lower than that of methane. However, experiments are required to confidently predict NO_x levels for the reformed fuel gas, because of scientific uncertainty regarding prompt NO_x and it should be remembered that dry low NO_x combustors (not developed to burn reformed gas) are improving rapidly and are now capable of achieving below 10 ppm NO_x.

6.2. Economical issues

It is too early to make an accurate estimate of the cost of a CRGT at this stage. The economics issues that need to be addressed are as follows:

1. Since the main advantage of chemical recuperation is the ultra-low NO_x emissions, economic comparisons of the cycle should take into consideration the cost of the NO_x reduction systems such as dry low NO_x systems. Where dry NO_x systems are provided as standard equipment, the associated costs are relatively low compared to the overall cost of the engine, especially for the large heavy frames. However, for the aeroderivatives, dry low NO_x systems significantly increase the engine cost. For these machines in the nominal 20–50 MW power range, a dry low NO_x combustion system can easily add 10–15 % or more to the package price. Therefore, this additional cost should be taken into account when we compare CRGT to any cycle which would require DLN system.
2. All of the performance predictions for CRGT cycles assume a “rubber turbine”, that is a turbine optimally designed for the cycle in question. That obviously

involves some R&D costs that have to be evaluated. Their impacts on the overall capital cost of the cycle, which will depend on its overall market penetration, could be a big disincentive to the development of the cycle.

3. The development of CRGT cycles are likely to be incremental. Engine development would center around modification of an existing STIG cycle in order to avoid excessive development costs. The major component cost difference would be the replacement of the superheater by a Methane Steam Reformer (MSR) in the Heat Recovery Steam Generator. Further discussion of design issues for the MSR component requires a thermo-economic analysis of the cycle.

6.3. Technical issues

It is difficult to predict all possible limitations and difficulties that can arise when the system is put into practice. The crucial parameter is probably the chemical reaction kinetics. If the kinetics are too slow, methane conversion will be low in a reasonably sized reactor. As fast reaction as possible is desirable since the time of reaction is going to determine the necessary residence time in reactors and, therefore, the steam reformer size and pressure drop. Therefore, a key area for the future of the CRGT cycle is the development of low-cost methane–steam reforming catalysts that are effective at the temperature levels encountered in gas turbine exhaust flows.

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