Modulo per la presentazione delle osservazioni per i piani/programmi/progetti sottoposti a procedimenti di valutazione ambientale di competenza statale

Presentazione di osservazioni relative alla procedura di:

❑ Valutazione Ambientale Strategica (VAS) – *art.14 co.3 D.Lgs.152/2006 e s.m.i.*

✓ Valutazione di Impatto Ambientale (VIA) – *art.24 co.3 D.Lgs.152/2006 e s.m.i.*

❑ Verifica di Assoggettabilità alla VIA – *art.19 co.4 D.Lgs.152/2006 e s.m.i.*

(Barrare la casella di interesse)

Il Sottoscritto GABRIELLO CASTELLAZZI (portavoce dei Verdi della provincia di Savona)

La Sottoscritta MARIA CRISTINA FERRANDO (coordinatore del consiglio direttivo dell'Associazione Movimento politico VivereVado)

PRESENTANO

ai sensi del D.Lgs.152/2006, le **seguenti osservazioni** al

❑ Piano/Programma, sotto indicato

✓ Progetto, sotto indicato.

(Barrare la casella di interesse)

Progetto di realizzazione di una nuova unità a ciclo combinato nella centrale termoelettrica di Vado Ligure (codice procedura 5658)

___

(inserire la denominazione completa del piano/programma (procedure di VAS) o del progetto (procedure di VIA, Verifica di Assoggettabilità a VIA)

OGGETTO DELLE OSSERVAZIONI

(Barrare le caselle di interesse; è possibile selezionare più caselle):

✓ Aspetti di carattere generale *(es. struttura e contenuti della documentazione, finalità, aspetti procedurali)*

❑ Aspetti programmatici *(coerenza tra piano/programma/progetto e gli atti di pianificazione/programmazione territoriale/settoriale)*

✓ Aspetti progettuali *(proposte progettuali o proposte di azioni del Piano/Programma in funzione delle probabili ricadute ambientali)*

✓ Aspetti ambientali *(relazioni/impatti tra il piano/programma/progetto e fattori/componenti ambientali)*

❑ Altro *(specificare)* ___

ASPETTI AMBIENTALI OGGETTO DELLE OSSERVAZIONI

(Barrare le caselle di interesse; è possibile selezionare più caselle):

✓ Atmosfera

❑ Ambiente idrico

❑ Suolo e sottosuolo

❑ Rumore, vibrazioni, radiazioni

❑ Biodiversità (vegetazione, flora, fauna, ecosistemi)

✓ Salute pubblica

- ❑ Beni culturali e paesaggio
- ❑ Monitoraggio ambientale
- ❑ Altro *(specificare)* ____

TESTO DELL' OSSERVAZIONE

VEDI ALLEGATO 3

Il/La Sottoscritto/a dichiara di essere consapevole che, ai sensi dell'art. 24, comma 7 e dell'art.19 comma 13, del D.Lgs. 152/2006 e s.m.i., le presenti osservazioni e gli eventuali allegati tecnici saranno pubblicati sul Portale delle valutazioni ambientali VAS-VIA del Ministero dell'Ambiente e della Tutela del Territorio e del Mare (www.va.minambiente.it).

Tutti i campi del presente modulo devono essere debitamente compilati. In assenza di completa compilazione del modulo l'Amministrazione si riserva la facoltà di verificare se i dati forniti risultano sufficienti al fine di dare seguito alle successive azioni di competenza.

ELENCO ALLEGATI

Allegato 1 e 1 bis - Dati personali dei soggetti che presentano l'osservazione

Allegato 2 - Copia patente Cristina Ferrando

Allegato 2bis Carta id G Castellazzi

Allegato 3 – Osservazioni raddoppio Tirreno Power VerdiSV-VivereVado

Allegato 4 – Nota del CIRPS – Centro Interuniversitario di Ricerca per lo Sviluppo sostenibile

Allegato 5 – Allegato 1 alla nota CIRPS su benefici e fattibilità alimentazione ad idrogeno

Allegato 6 – Allegato 2 alla nota CIRPS

Finale Ligure, 21/01/2021

I Dichiaranti

Gabriello Castellazzi

(Gabriello Castellazzi)

Maria Cristina Ferrando

(Maria Cristina Ferrando)

OSSERVAZIONI GENERALI

sulla CENTRALE di VADO LIGURE

Riferimento: Avviso al Pubblico del 24/11/2020 MATTM-2020-009666

La scelta di realizzare una ulteriore centrale è palesemente in controtendenza con le linee di indirizzo e i finanziamenti previsti dall'Unione Europea (New Green Deal) per la transizione energetica, che prevede il progressivo abbandono delle fonti fossili, con le conseguenze negative del loro utilizzo per la produzione di energia elettrica, a partire dall'abbattimento delle emissioni climalteranti.

Nell'ipotesi di un via libera (condizionato) al progetto del nuovo impianto a ciclo combinato di cui TIRRENO POWER chiede l'autorizzazione al MATTM per l'installazione, va con fermezza sottolineato che esso deve comportare il **contestuale smantellamento dei due Gruppi alimentati a carbone** la cui utilizzazione è ora interdetta dall'Autorità giudiziaria.
Questa va intesa come condizione necessaria ed ineludibile. Possibilmente, va preteso che gli spazi che l'eventuale nuova Centrale dovrà occupare siano ricavati dallo smantellamento dei Gruppi a carbone.
Ciò anche in considerazione di pluriennali disagi ed effetti deleteri sulla salute delle persone che vivono a Vado Ligure, attestati dallo studio prodotto dal CNR, che indica come i danni alla salute siano stati indotti dalla alta densità di impianti industriali a consistente impatto ambientale, visivo, paesaggistico, che hanno inciso costantemente sulla normale conduzione delle attività quotidiane. Non si dimentichi che la città è, contemporaneamente ai previsti lavori del progetto, "minacciata" dall'ampliamento della attuale discarica per rifiuti urbani e speciali.

La proposta di Tirreno Power di costruire una ulteriore centrale per "garantire la la stabilità di rete" va confrontata con il diritto della popolazione – già penalizzata da decenni di carico ambientale elevatissimo – di ottenere lo smantellamento dei due gruppi a carbone senza nuovi insediamenti - **ipotesi zero**.
Un nuovo impianto a combustione produrrà ulteriori emissioni inquinanti. Che siano contenute nei limiti di legge non è sufficiente dal momento che comunque saranno aggiuntive rispetto alla situazione attuale.

Si chiede che, nel caso dell' accettazione dell' **ipotesi 1,** Tirreno Power si impegni a realizzare , per la nuova centrale a ciclo combinato (e per quella già esistente) il ricorso ad una alimentazione mista: gas metano addizionato con una percentuale di un **10 % di IDROGENO VERDE** (ovvero prodotto da elettrolisi verde o da biomasse); ciò per fare delle nuova centrale un **esempio avanzato e virtuoso sul piano energetico e ambientale.** A questo proposito si veda nota del CIRPS – Centro Interuniversitario di Ricerca Per lo

Sviluppo sostenibile.

In questa ottica la società Tirreno Power dovrà includere nel progetto la realizzazione di un impianto per la produzione di idrogeno verde, ovvero in grado di produrre energia elettrica da fonti rinnovabili che potrà essere, in ipotesi, anche utilizzata dalla rete delle industrie dall'area di Vado Ligure, incluse le attività portuali e di futuro rifornimento delle navi (*ricordiamo che per il 2023 è previsto il varo della prima nave passeggeri completamente alimentata a idrogeno liquido e tra cinque anni il collegamento dei fiordi norvegesi sarà vietato alle navi con motori termici)*.

<u>Peraltro, un impianto di questo tipo godrebbe degli incentivi che il RECOVERY PLAN ESPLICITAMENTE PREVEDE PER LA PROMOZIONE DELLE TECNOLOGIE DELL'IDROGENO.</u>

Maria Cristina Ferd

Gabriella Castellazzi

NOTA SUL POSSIBILE IMPIEGO DI IDROGENO VERDE IN COMMISTIONE CON GAS METANO PER L'ALIMENTAZIONE DI UNA CENTRALE A CICLO COMBINATO

In relazione all'alimentazione della Centrale a ciclo combinato in funzione a Vado Ligure (e, a maggior ragione per la eventuale seconda Centrale di cui è richiesta l'autorizzazione), si ritiene che **un deciso contributo alla riduzione del carico inquinante globale possa venire dall'adozione di una alimentazione mista** (gas metano-idrogeno).

Questa soluzione avanzata è in via di attuazione in diversi insediamenti in Europa, anche nell'ambito del grande **Programma EUROPEAN GREEN DEAL**. Inoltre, per incentivare la realizzazione di impianti, anche dimostrativi, inerenti alle tecnologie dell'idrogeno, **ingenti risorse sono previste anche dal testo del Recovery Plan**, nella versione che oggi circola, adottata dal Consiglio dei ministri tenuto nella notte.

Va messo in particolare evidenza, tuttavia, anche tenuto conto del contesto nel quale l'iniziativa si propone, che **si deve assolutamente pretendere che la soluzione cui si ricorrerà preveda utilizzo esclusivo di idrogeno cosiddetto "verde"**, cioè prodotto partendo da materie prime sostenibili e rinnovabili (acqua o biomasse) e ricorrendo esclusivamente ad energie rinnovabili per il ciclo di produzione e utilizzo. Non si accetteranno soluzioni commerciali, quali la produzione di idrogeno da combustibili fossili (es.: steam methane reforming), presentata quale soluzione transitoria, in attesa della "futura" produzione verde.

I principali benefici comportati dalle soluzioni che qui si prospettano sono quelli della **ingente riduzione degli inquinanti**, garantita dalla combustione dell'idrogeno in sostituzione dell'equivalente termico (energetico) di gas metano; da una **migliore combustione**; e dalla **promozione di una tecnologia verde** che, in prospettiva, potrà risultare autonoma e decisamente molto più sostenibile anche della migliore soluzione oggi offerta dai combustibili fossili, che è, appunto, quella dell'impiego del solo metano.

In via del tutto preliminare, salvo verifiche in loco, si può ipotizzare l'**immissione fino al 10% di idrogeno miscelato al gas naturale impiegato per l'alimentazione del gruppo turbogas di testa dell'impianto combinato**. Si deve trattare, però, di idrogeno prodotto in loco (preferibilmente proveniente da eccesso di disponibilità di energia da fonti rinnovabili, così da agire da accumulo delle stesse). Si sceglie la proposta di una frazione non superiore al 10% di idrogeno per ancorare la stessa a valori già collaudati e ormai definitivamente accettati; ma l'adozione di proporzioni maggiori di idrogeno è già sufficientemente sperimentata.

In realtà le nuove turbine a gas di ultima generazione accettano più alte percentuali di idrogeno e addirittura sono disponibili studi e sperimentazioni di gruppi commerciali alimentati interamente a idrogeno. Per valutare i benefici e la fattibilità, a titolo di esempio, si allega lo studio di Lozza e Mazzocchia del Politecnico di Milano, incentrato su una alimentazione al 100% di idrogeno, con il confronto rispetto al caso di alimentazione al 100% di gas naturale (Allegato 1). Si guardi anche lo stato dell'arte riportato nell'Allegato 2, che dimostra come molteplici Gruppi a ciclo combinato nel mondo siano alimentati già da miscele fino al 60% di idrogeno con grandi benefici per le emissioni di CO_2; le emissioni di NO_x sono ugualmente contenute.

Una soluzione alternativa molto promettente può poi essere costituita dall' **immissione -in miscela con il gas naturale- di biogas e/o syngas prodotto in loco da biomasse di scarto**. Anche per questa applicazione sono disponibili studi e applicazioni che dimostrano come sia il biogas, sia il syngas, possano alimentare anche al 100% un impianto di turbine a gas, **riducendo significativamente le emissioni ed impiegando biomasse di scarto**. Per una prima valutazione tecnico-economica, si veda questo articolo del 2016 https://www.sciencedirect.com/science/article/abs/pii/S1750583616301128.

La fattibilità della proposta che qui si avanza, il dettaglio della sua realizzazione va studiata per il caso specifico, tuttavia sia la produzione di idrogeno verde, sia il suo utilizzo in centrali di potenza sono documentati ampiamente nella letteratura scientifica, come dimostrato dagli Allegati e dal link.

Vincenzo NASO
Massimo GUERRA

Using Hydrogen as Gas Turbine Fuel

Article *in* Journal of Engineering for Gas Turbines and Power · January 2005

DOI: 10.1115/1.1787513

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Using Hydrogen as Gas Turbine Fuel

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This paper addresses the possibility to burn hydrogen in a large size, heavy-duty gas turbine designed to run on natural gas as a possible short-term measure to reduce greenhouse emissions of the power industry. The process used to produce hydrogen is not discussed here: we mainly focus on the behavior of the gas turbine by analyzing the main operational aspects related to switching from natural gas to hydrogen. We will consider the effects of variations of volume flow rate and of thermophysical properties on the matching between turbine and compressor and on the blade cooling of the hot rows of the gas turbine. In the analysis we will take into account that those effects are largely emphasized by the abundant dilution of the fuel by inert gases (steam or nitrogen), necessary to control the NO_x emissions. Three strategies will be considered to adapt the original machine, designed to run on natural gas, to operate properly with diluted hydrogen: variable guide vane (VGV) operations, increased pressure ratio, re-engineered machine. The performance analysis, carried out by a calculation method including a detailed model of the cooled gas turbine expansion, shows that moderate efficiency decays can be predicted with elevated dilution rates (nitrogen is preferable to steam under this point of view). The combined cycle power output substantially increases if not controlled by VGV operations. It represents an opportunity if some moderate re-design is accepted (turbine blade height modifications or high-pressure compressor stages addition).
[DOI: 10.1115/1.1787513]

1 Introduction

Hydrogen, as a carbon-free energy carrier, is likely to play a important role in a world with severe constraints on greenhouse gas emissions. In the power industry, its utilization as gas turbine fuel can be proposed under several possible scenarios, depending on the mode of H_2 production. For instance, hydrogen can be produced remotely from renewable energy sources (solar or wind) or from nuclear energy (via direct thermal conversion or by electrolysis), but in a more realistic and near-term vision it will be derived from conventional fossil fuels by conversion processes including CO_2 sequestration. Possible solutions include: (i) remote coal conversion to hydrogen (via gasification, shift, and separation from CO_2) and H_2 pipeline transport to the power station, (ii) integrated hydrogen and electricity production from coal or natural gas, exporting pure hydrogen to remote users, and using on-site low-grade hydrogen to produce power [1], (iii) electricity generation from combined cycles integrated to fossil fuel decarbonization (applicable to coal, oil, or gas) and to CO_2 capture [2]. Fuel cells and H_2-O_2 semiclosed cycles may represent future options for power generation, but combined cycles coupled to H_2 production/CO_2 sequestration processes can be proposed as a short/mid-term solution for massive greenhouse gas emission reduction.

This paper addresses the possibility to burn hydrogen in a large size, heavy-duty gas turbine designed to run on natural gas, for a prompt application of the above general concepts, regardless of the process used to produce hydrogen and its integrations with the combined cycle. We will focus on the behavior of the gas turbine, by considering the effects of the variation of volume flow rates and of thermophysical properties, related to switching from natural gas to hydrogen. These effects are emphasized by the fact that NO_x emission control relies on fuel dilution with large quantities of inert gases, like steam or nitrogen, as discussed in Sec. 2. The consequent variation of the operating conditions is therefore much larger than for the mere fuel substitution, calling for an analysis of the opportunity (or necessity) of design modifications to the gas turbine. The paper discusses these issues by considering some possible adaptation techniques, by discussing their operational limits and, mostly, by predicting the resulting combined cycle efficiency and power output.

2 NO_x Control

Generally speaking, three methods have been used to reduce NO_x emissions from gas turbine power plants: (i) premixed combustion, including catalytic combustion, (ii) fuel dilution, mostly by steam, water or nitrogen; (iii) removal from exhaust gases. For natural gas applications, the first technique is the preferred one: at present, the "dry low-emission" combustors are proposed by manufacturers for virtually any gas turbine model. Their basic principle is to achieve a moderate flame temperature by forcing more air than stoichiometric in the primary zone; this is obtained by mixing air to fuel before the combustion. Catalytic combustors, often referenced as the future technology for extremely low emissions, just enhance the same principle, allowing for a much larger rate of premixing, no longer limited by flame stability limits. When switching to hydrogen (or to hydrogenated fuels, such as the coal syngas used in IGCC plants) premixing becomes a very questionable practice, due to the much larger flammability limits and the lower ignition temperatures of hydrogen with respect to natural gas [3]. Therefore both dry low-emission and catalytic combustors cannot be safely proposed for large industrial applications, to the authors' knowledge, simply because hydrogen promptly reacts when mixed to air at typical gas turbine conditions, at virtually any rate. In fact, IGCC combustors, handling a CO-H_2 mixture with H_2 content from 25 to 40%, are diffusion burners and pre-mixed combustion was never attempted. Massive steam or nitrogen dilution is extensively used in these combustors [4] to control NO_x. In diffusion burners, the stoichiometric flame temperature (SFT) is representative of the actual flame temperature, strictly related to the NO formation rate.

Figure 1 shows a collection of literature data, mostly retrieved from a GE experimental investigation with hydrogenated fuels

Contributed by the International Gas Turbine Institute (IGTI) of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS for publication in the ASME JOURNAL OF ENGINEERING FOR GAS TURBINES AND POWER. Paper presented at the International Gas Turbine and Aeroengine Congress and Exhibition, Atlanta, GA, June 16–19, 2003, Paper No. 2003-GT-38205. Manuscript received by IGTI October 2002; final revision March 2003. Associate Editor: H. R. Simmons.

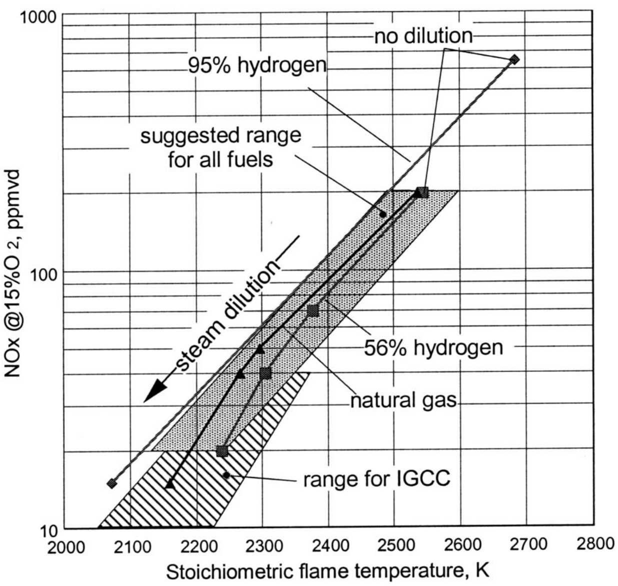

Fig. 1 Relation between NO_x emission and stoichiometric flame temperature, progressively reduced by steam dilution, for gas turbine diffusive combustion at 12–16 bar with different fuels. Nitrogen is the balance gas for 56% and 95% hydrogen.

reported by Todd and Battista [5], showing a relation between SFT and NO_x emission for various fuels in typical gas turbine conditions. It is clear that the utilization of undiluted H_2 brings about unacceptable levels of emission and that the SFT must be greatly reduced to have emissions comparable to power industry standards (25–45 ppmvd). A reasonable value of 2300 K for SFT can be stipulated to meet this standards, even if more experience must be gained to set precise indications.

The third technique mentioned above (removal from exhausts) includes: (i) selective catalytic reduction (SCR) by means of ammonia injection (a very well-known method), (ii) the Sconox process, recently proposed for extremely elevated removal rates, using adsorption-desorption on potassium carbonate beds. These techniques can be used downstream of H_2-fueled gas turbines, as well as for conventional units. However, their cost and size are basically related to the amount of NO_x removed (about 2000 $/ton for SCR, 6000–8000 for Sconox (Major and Powers [6])). Their utilization can be proposed to reduce emissions starting from a moderate concentration (for instance from 100 to 10 ppm), but the incidence on the electricity cost would be excessive when starting from many hundreds of ppm, as for H_2 combustion (especially for Sconox). Therefore, excluding premixed combustors and limiting the SCR to "finishing" applications, dilution techniques seem mandatory for hydrogen utilization in gas turbine combustors.

The selection of steam and nitrogen as the possible diluents is quite straightforward. Steam is always available in a combined cycle and can be extracted from the steam turbine at any pressure and at any reasonable rate. Nitrogen is available "for free" in processes including air separation, i.e., in any coal or refinery residual gasification plant: if the hydrogen used by the gas turbine is produced on site from decarbonization of syngas from heavy fuels, nitrogen will be surely present in large quantities[1] (see, for instance, Lozza and Chiesa [2]). In such plants, it is also possible to use saturation of the hydrogen-rich gas by means of warm water coming from the syngas cooling: it makes available a steam-diluted fuel without extractions from the steam turbine. In some other hydrogen production processes, nitrogen is "naturally" available: this is the case of natural gas decarbonization by means of an air-blown autothermal reformer [7,8], producing a synthesis fuel consisting of a 50–50% (approximately, by volume) mixture of H_2 and N_2, perfectly suited for NO_x abatement.

[1]Throughout the discussion we will assume that nitrogen for dilution is available at no energy cost at atmospheric pressure. This assumption is actually verified if the plant incorporates a low-pressure air separation unit.

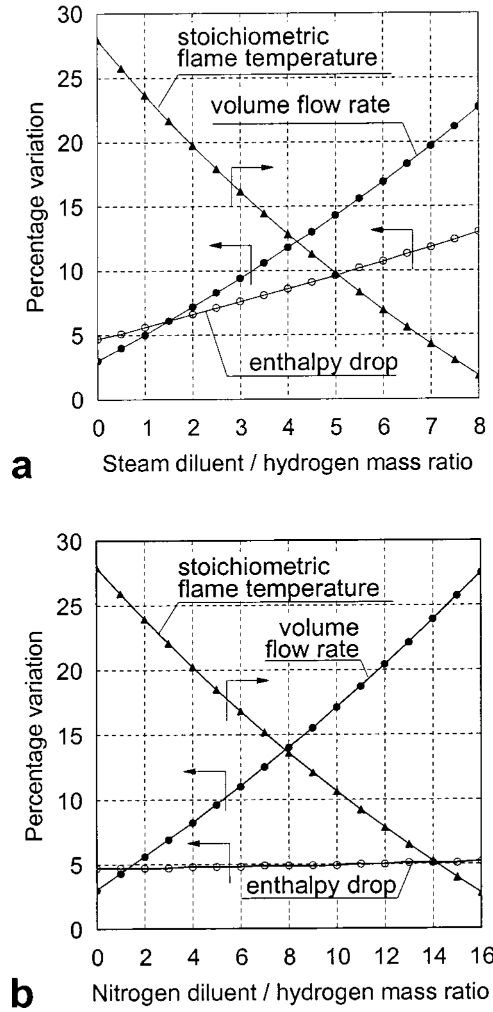

Fig. 2 Variation of the SFT and of the inlet volume flow rate and isentropic enthalpy drop of a hydrogen fueled gas turbine with respect to the reference natural gas case. Curves are drawn as a function of the added diluent flow rate: the upper diagram refers to steam, the lower diagram to nitrogen.

3 Effects of Hydrogen Combustion on Turbomachinery

Compared to natural gas, hydrogen combustion leads to a lower mass flow rate and to a different composition of the product gases, with an higher water content that in turn influences the molecular weight and the specific heat of the mixture. The most relevant effects on the operation of a gas turbine are: (i) a variation of the enthalpy drop in the expansion, (ii) a variation of the flow rate at the turbine inlet which, in turn, affects the turbine/compressor matching, (iii) a variation of the heat-transfer coefficient on the outer side of the turbine blades, affecting the cooling system performance.

3.1 Influence of Fluid Composition Variation on Turbine Enthalpy Drop and Inlet Volume Flow Rate. Figure 2(*a*) shows the influence of hydrogen combustion (in presence of a variable flow of diluting steam) on the isentropic enthalpy drop of a turbine at a given inlet condition ($T = 1450°C$, $p = 17$ bar) and atmospheric outlet pressure, compared to the corresponding natural gas fired case. Compared to natural gas, the simple hydrogen combustion increases the enthalpy drop by about 5%, a variation that increases as long as the amount of added steam rises. Assuming the working fluid as an ideal gas, since the isentropic enthalpy drop can be evaluated through the expression

$$\Delta h_{\mathrm{is}}=\int_{T_{\mathrm{FIN,IS}}}^{T_{\mathrm{IN}}} c_P(T)\,dT=\overline{c_P}(T_{\mathrm{IN}}-T_{\mathrm{FIN,IS}}),$$

it is possible to distinguish the effect due to the variation of average c_P and the one due to the variation of the temperature drop through the expansion (being the latter influenced by the exponent of the isentropic transformation γ, i.e., the specific-heat ratio). Increasing steam dilution entails an enhancement of the mixture specific heat but a simultaneous decrease of the exponent γ that reduces the temperature drop and consequently increases the turbine outlet temperature.

The second y axis reports the stoichiometric flame temperature resulting from the combustion. It shows that a diluent to H_2 mass ratio of about 7 is required to keep this temperature at 2300 K: correspondingly the enthalpy drop increases by about 12% with respect to the natural gas case.

Figure 2(*a*) also quantifies the variation of the volume flow rate at turbine inlet resulting from the hydrogen combustion (always in comparison with the natural gas fired case). The same amount of combustion air and the same combustion temperature (1450°C) are assumed for all the cases and therefore also the H_2 flow rate increases as long as the diluent flow rate increases. Notice that in the case of no dilution, although the mass flow rate of combustion products reduces (about 2%, considered that hydrogen LHV is 119.95 MJ/kg versus 44.77 of natural gas), the volume flow rate increases by about 3% due to the change in composition (molecular weight of this mixture reduces from 28.27 to 26.93 kg/kmol). This effect amplifies when dilution is considered. At steam to H_2 mass ratio of 7 the mass and volume flow rate increase by 11% and 20%, respectively.

Figure 2(*b*) reports the analysis as far as nitrogen is considered for dilution. The different scale on the abscissa reflects that a much larger diluent to fuel ratio is required to determine a given SFT abatement (about twice, since c_P of N_2 is approximately one-half of c_P of steam). Therefore dilution greatly affects the mass flow rate and, consequently, the volume flow rate (black dotted line). On the contrary the effect of nitrogen dilution on the turbine enthalpy drop is virtually negligible since a large amount of nitrogen (from combustion air) is already contained in the mixture so that even a large diluent addition does not substantially modify the fluid properties.

3.2 Compressor/Turbine Matching. Because of the variation of the volume flow rate caused by the different fuel (and additional diluent), using hydrogen affects the original matching between compressor and expander in a gas turbine originally designed to run on natural gas. A different running point will be set where mass flow rate and pressure ratio will restore the fluid-dynamic equilibrium between the two turbomachines. Typical operational curves are shown in Figs. 3 and 4, for a single shaft arrangement operating at fixed rotational speed (the only solution adopted for large industrial gas turbines). For high performance axial compressors with several transonic stages, used in advanced gas turbines, the characteristics show that the mass flow rate is virtually constant when the inlet is choked. To improve partial load operations, variable geometry guide vanes (VGV's) are used on several stator rows, affecting the characteristic lines as shown by Fig. 3.

The operating line of the expander at constant speed is reported in Fig. 4. When the machine is fueled with hydrogen, having a higher heating value than natural gas, the mass flow rate $G_{T,\mathrm{IN}}$ reduces for a give compressor airflow. Nevertheless the nondimensional flow $G_{R,\mathrm{T,IN}}$ slightly increases, because of the molecular mass reduction. This increase becomes more and more important when $G_{T,\mathrm{IN}}$ grows up due to the diluent addition. Therefore switching from natural gas to hydrogen makes impossible to operate the gas turbine on the same running point (i.e., at the same VGV angle, pressure ratio, $G_{C,\mathrm{IN}}$ and $T_{T,\mathrm{IN}}$).

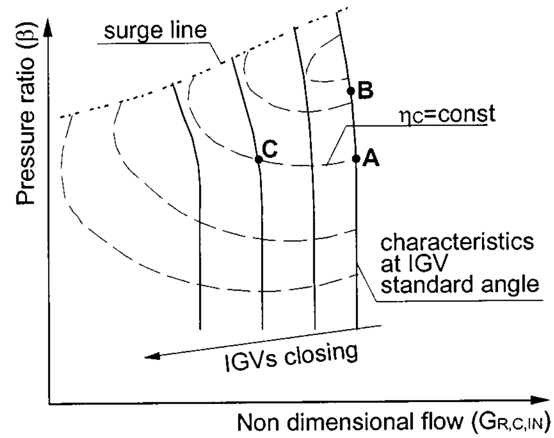

Fig. 3 Typical compressor characteristic curve at constant rotational speed. Different lines correspond to different settings of the variable guide vanes angle.

Assuming that A and A' are the design points on the compressor and turbine maps (Figs. 3 and 4) of the natural gas fueled machine, three different regulation strategies can be envisaged:

(i) Letting the compressor to work at the same point (A, at the same VGV angle) and reducing the $T_{T,\mathrm{IN}}$, to restore the fluid-dynamic matching between compressor and turbine. The expander runs at the design point A':

(ii) Letting the VGV angle and $T_{\mathrm{T,IN}}$ at their original value, $G_{R,T,\mathrm{IN}}$ can be adjusted by increasing the pressure ratio, (i.e. moving from A to B on the compressor characteristics while the turbine running point moves from A' to B'). If the compressor is not choked, the higher β also reduces the mass flow rate and helps to reset the matching. If the required pressure ratio exceeds the available surge margin, one or more high-pressure stages must be added to the compressor.

(iii) Letting $T_{T,\mathrm{IN}}$ and β at their value, equilibrium can be found by closing the VGV's and reducing $G_{C,\mathrm{IN}}$. The corresponding running point moves from A to C in Fig. 3. If condition depicted by point C exceeds the available surge margin, additional stages are required. The turbine running point remains unaffected so that C' overlaps A'.

Remarking that the actual regulation can be carried out by adopting all the three strategies at the same time, it is evident that the first one seems the least interesting since the performance of a combined cycle substantially decays when $T_{T,\mathrm{IN}}$ reduces. The effects of the latter strategies on the cycle performance will be dis-

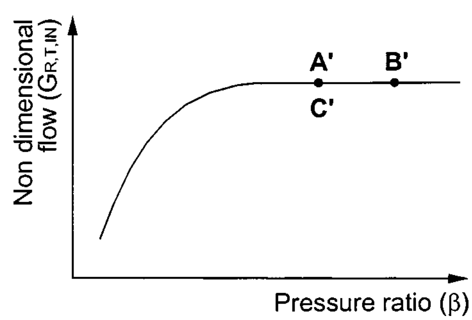

Fig. 4 Typical turbine characteristic curve at constant rotational speed

Table 1 Thermophysical properties at 1000°C and 10 bars

	ρ, kg/m^3	c_P, kJ/kg K	$\mu\times10^6$, Pa s	k$\times10^3$, W/m K	$\rho^{0.63}c_P^{1/3}k^{2/3}/\mu^{0.7}$
Air	2.736	1.183	50.109	83.164	73.84
Steam	1.702	2.482	48.241	135.465	98.15
CO_2	4.158	1.289	49.524	81.696	98.09

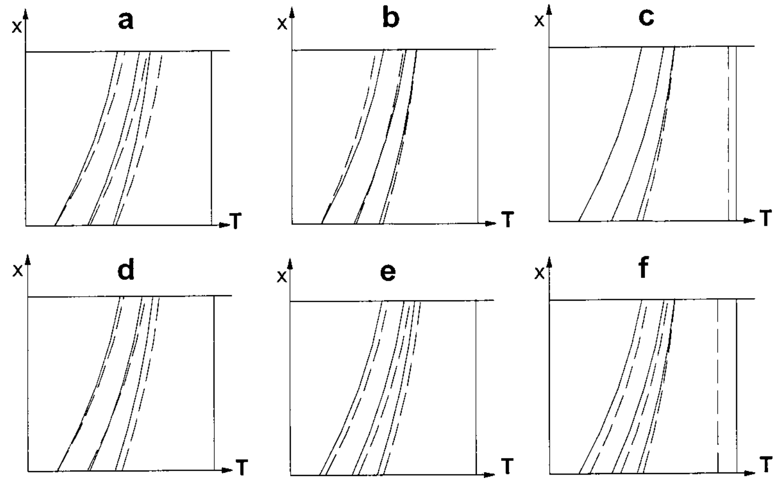

Fig. 6 Temperature–heat-transfer area diagrams showing different situations in the simplified cooling circuit of Fig. 5. Temperature profiles have the same meaning of Fig. 5: from left, they refer to coolant, inner blade wall, outer blade wall, main gas stream. The continuous lines refer to the original situation, the dashed ones to modified conditions.

cussed later (Sec. 5), but it can be anticipated that off-design operations imply a substantial change of the gas turbine power output. Therefore considerations about mechanical stresses can heavily influence the regulation strategy: dealing with such limits is beyond the scope of the present analysis but they must be carefully considered.

The hydrogen combustion (and related dilution) also entails substantial changes in the shape of the velocity triangles, due to the increase of the enthalpy drop and volume flow rate that influence the flow velocity and its axial component, respectively. Given that the flow is accelerated along the gas path, the turbine blades can operate efficiently even for incidence angles sensibly different from the design value and these changes in the velocity triangles consequently have small effects on the turbine performance. A more relevant efficiency decay can be caused by the increase of the kinetic energy loss at the exhaust due to the increased flow rate for the same exhaust area.

3.3 Blade Cooling. Hydrogen combustion and additional dilution affect the cooling system under two different aspects:

• the varied composition of the hot stream enhances the convective heat-transfer coefficient on the outer side of the blade increasing the thermal flux with negative consequences on the performance of the cooling circuit;

• the higher pressure ratio increases the convective heat-transfer coefficients on both blade sides and the temperature of air used in the cooling circuit whose performance decays.

Effect of Flow Composition. The correlation proposed by Louis [9] allows us to evaluate the average heat-transfer coefficient on the outer side of the blade:

$$h_{\mathrm{OUT}}=0.285\,\frac{(\rho v)^{0.63}c_P^{1/3}k^{2/3}}{D^{0.37}\mu^{0.7}},$$

where v is the main stream speed referred to the cascade exit. Replacing steam to CO_2 (as it actually occurs when H_2 replaces natural gas as fuel) has no significant consequences on the heat flux imposed on the blade outer surface, as it can be argued from Table 1. On the contrary, steam dilution determines an increase of the thermal flux since the heat-transfer coefficient for steam is higher than for air. A secondary effect is an increase of h_{OUT} due to the higher average velocity of the gas stream along the flow path related to the higher available enthalpy drop.

Although the calculation model used for the final discussion (Sec. 5) considers the current gas turbine cooling circuits including film cooling and multipass channels, the behavior of a cooling circuit in consequence of a change in the main stream composition can be better discussed by considering a very simplified convective cooling circuit. It consists of a single internal duct run by the cooling fluid whose blade transverse section is schematically shown on the left side of Fig. 5. The temperature profiles along the blade height are shown in Fig. 5, right side. The blade can be considered a cross-flow heat exchanger, where the cooling flow ensures that the highest metal temperature remains within the stipulated limit.

Enhancing the thermal flux on the blade at constant cooling flow rate causes an increase of the temperatures along the profiles as shown in Fig. 6(*a*). The constraint on the maximum metal temperature can be restored either by increasing the cooling flow

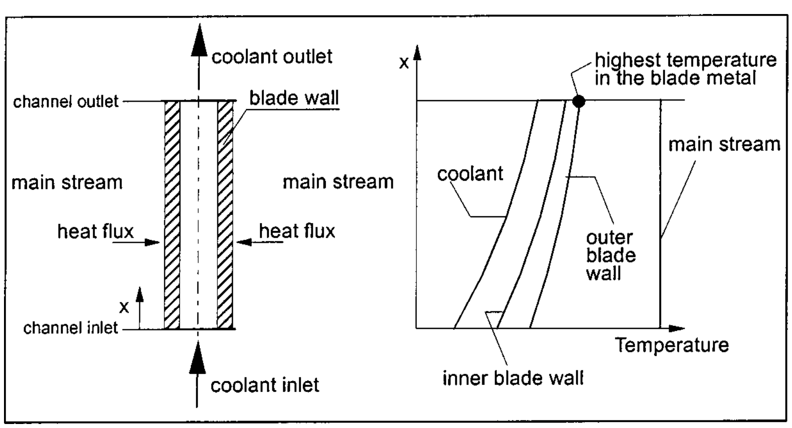

Fig. 5 Simplified blade cooling model. Blade is assumed as a cross-flow heat exchanger where heat capacity of the outer stream is infinitely larger than the one of the inner stream. Main temperature profiles are reported in the right diagram.

rate (Fig. 6(*b*)) or by reducing the temperature of the outer stream (Fig. 6(*c*)). In hydrogen operation of a gas turbine designed to run on natural gas, it seems straightforward that the cooling circuit does not change and therefore the solution of Fig. 6(*b*) cannot be adopted. Decreasing the turbine inlet temperature (Fig. 6(*c*)) appears the only feasible alternative.[2]

Effect of Pressure Ratio. An increase of the cycle pressure ratio influences the blade cooling mechanisms in three main aspects: (i) the heat-transfer coefficients enhance on both the inner and the outer blade side due to the fluid density increase; (ii) the temperature of the cooling air from compressor increases; (iii) the coolant mass flow rate increases because of coolant density increase for a given circuit geometry. About the first point, we already discussed the negative effects of an h_{OUT} enhancement. On the contrary, an enhancement of h_{COOL} has positive effects because it reduces the temperature difference between the fluid and the metal blade. Nevertheless, the simultaneous and proportional enhancement of the heat-transfer coefficient on both the blade sides due to the pressure ratio increase is not neutral because it increases the heat flux (and consequently the temperature drop) across the blade wall, bringing the maximum metal temperature beyond its admissible value (Fig. 6(*d*)). Finally, a coolant temperature increase causes the shift of all the temperature profiles as shown in Fig. 6(*e*). A temperature decrease of the flow at the turbine inlet is then required to restore the capability of the cooling circuit to meet the imposed limits (Fig. 6(*f*)) although this effect is somehow mitigated by the coolant flow increase allowed by the higher β.

4 Calculation Methodology

The performance prediction was carried out by a computer code developed by the authors' research group during several years of activities about gas turbine power plants. For a comprehensive description, see Chiesa and Macchi [10]. As a brief reminder, the main features of the code include the capability of reproducing very complex plant schemes by assembling basic modules (such as turbine, compressor, combustor, steam section, heat exchanger, etc.) and an effective prediction of the efficiency of turbomachines (gas and steam turbine stages, compressors) at their design point by means of built-in correlations. The calculation process also includes the one-dimensional design of the gas turbine stages, useful to establish all the aerodynamic, thermodynamic, and geometric characteristics of each blade row necessary for an accurate estimation of the cooling flows and the evolution of the cooled expansion. The cooling model accounts for film cooling, thermal coatings, and multipassage internal channels with enhanced heat-transfer surfaces. These effects are evaluated by means of some parameters, calibrated to reproduce the performance of advanced gas turbines. The complete procedure is reported in Ref. [10].

Even if the code is conceived for prediction of gas turbine performance at the design point only, introduction of convenient hypotheses on off-design behavior of turbomachines has made possible calculating the performance of hydrogen fueled combined cycles. We suppose that off-design operations are limited to the gas turbine because of the extreme rigidity of its design. Heat recovery steam generator and steam turbine can be easily adapted to run at the different conditions resulting from H_2 combustion, due to a more flexible manufacturing.

The ''reference'' natural gas combined cycle was calculated by using a set of assumptions reported by Table 2. The main data for the gas turbine are tuned to describe a Siemens V94.3A unit, representative of a state-of-the-art, heavy-duty, single-shaft machine [11]. The assumptions for the steam cycle calculation reproduce the present technological standards. The efficiency (38.17 and 57.57%) predicted for the gas turbine and the combined cycle, respectively, are in good agreement with declared data (38.20 and 57.30%). The same holds for power output (259.4 and 387.2 MW versus 260 and 390 MW). The assumptions of Table 2 were used for all the cases considered, apart from the gas turbine air flow, pressure ratio, and TIT, varied according to the following discussion. In fact, according to Sec. 3, different approaches can be adopted to use hydrogen as the fuel; three alternatives will be considered in the paper:

[2]The rationale underlying this strategy is that the same lifetime of a machine running on natural gas can be preserved in hydrogen operations by maintaining the same maximum metal temperature.

Table 2 Main assumptions for reference cycle calculations

Gas turbine
Ambient condition: 15°C, 1.0132 bar, 60% RH
Inlet/outlet pressure losses=1/3 kPa
Air/exhaust gas flow=633.8/644 kg/s
Pressure ratio=17, TIT=1350°C
Natural gas LHV=44.769 MJ/kg, preheated at 185°C
Steam cycle (three pressure levels, reheat)
Evaporation pressures: 166/36/4 bar
Condensing pressure: 0.0406 bar
Maximum steam temperature at SH/RH outlet=565°C
ΔT at pinch point=8°C, at SH approach point=25°C
Auxiliaries consumption=1% of heat rejected

VGV Operation. In this case no major modifications are required to the gas turbine provided that the stall margin is guaranteed. Additional high-pressure compressor stages can help to recover this margin. Calculation proceeds keeping the pressure ratio at the design value, with an inlet airflow reduced to recover the matching between compressor and turbine. Given the shape of efficiency curves on the compressor map (Fig. 3), it has been assumed to keep the compressor efficiency at the design value (the actual variation of efficiency depends on specific design criteria and cannot be generalized). The turbine maintains the original geometry (diameters, blade heights, angles) and cooling circuit characteristics but runs on a lower TIT in order to maintain the same blade metal temperature of the natural gas case. The different enthalpy drop is accommodated by varying the load on each stage at constant degree of reaction: according to Sec. 3, effects of loading on the stage efficiency have been neglected, but variations of the kinetic energy at the turbine outlet were kept into account.

Increased β. The second approach assumes that the VGV's remain full open and compressor/turbine matching is reset by increasing the operating pressure ratio. Calculation proceeds by assuming that the compressor characteristics is vertical (constant airflow). Given the stall margins available on the actual machines, it is really doubtful that this strategy can be adopted without any modification to the machine design, especially when SFT of 2300 K are demanded. Probably, one or more high-pressure compressor stages must be added [12,13] shifting upward the surge limit. In this case every compressor stage operates very close to the design point so that their efficiency can be correctly predicted by the code built-in correlations. Assumptions for turbine calculation are the same used in the previous case. TIT experiences a more significant decrease, justified by the warmer cooling flows and the higher heat-transfer coefficients related to the higher β.

Re-engineered Machine. In this case the standard machine is re-designed to comply with the larger flow rate at the turbine inlet. The compressor is virtually unchanged and the turbine blade height is increased to accommodate the larger gas flow. In this approach, turbine geometry and blade cooling flows are adapted to operate the gas turbine at the same β and TIT of the standard machine. Since the calculation is based on the very same assumptions used for the natural gas fired machine, this case represents the highest performance limit attainable with a hydrogen fueled gas turbine of the assigned technology level.

Table 3 Main results of the investigation (GT: gas turbine, SC: steam cycle)

Fuel	Nat. gas	Hydrogen, VGV operation			Hydrogen, increased β			Hydrogen, re-engineered		
Diluent	none	none	steam	nitrogen	none	steam	nitrogen	none	steam	nitrogen
Dil./fuel mass ratio	0.00	0.00	6.78	14.44	0.00	6.92	15.36	0.00	6.83	14.45
SFT, K	2545	2745	2300	2300	2746	2300	2300	2745	2300	2300
Pressure ratio	17.00	17.00	17.00	17.00	17.05	18.47	19.73	17.00	17.00	17.00
TIT, °C	1350	1339	1316	1340	1339	1305	1319	1350	1350	1350
TOT, °C	585.1	574.7	577.2	574.2	574.1	562.7	548.6	584.0	591.4	569.5
Air flow, kg/s	633.8	631.9	584.1	550.7	633.8	633.8	633.8	633.8	633.8	633.8
Gas flow, kg/s	644.0	632.7	623.5	631.1	634.6	676.5	728.2	634.7	678.1	725.9
Fuel flow, kg/s	15.02	5.58	5.67	5.52	5.59	6.02	6.11	5.66	6.31	6.31
Diluent flow, kg/s	0.00	0.00	38.44	79.67	0.00	41.71	93.78	0.00	43.10	91.21
Ma_{AX}	0.441	0.437	0.442	0.437	0.439	0.479	0.504	0.441	0.441	0.441
Cooling flows, kg/s	139.8	138.0	138.4	138.1	138.3	146.2	149.0	143.6	168.9	163.1
GT output, MW	256.8	264.5	292.0	297.6	265.1	314.4	340.5	266.3	323.8	342.7
SC net output, MW	130.4	125.6	91.5	125.3	125.7	92.1	132.4	130.1	104.9	142.1
N_2 compressor, MW	0.0	0.0	0.0	42.7	0.0	0.0	54.3	0.0	0.0	48.9
Total output, MW	387.2	390.1	383.5	380.2	390.9	406.4	418.6	396.4	428.7	436.0
LHV efficiency, %	57.57	58.32	56.38	57.46	58.32	56.25	57.15	58.35	56.60	57.57

5 Discussion of Results

The general results of the investigation are reported in Table 3, showing details of (i) the reference natural gas cycle, (ii) the three pure hydrogen fueled cases, calculated according to the strategies described in Sec. 4, (iii) the three hydrogen cases with steam dilution to achieve 2300-K SFT, (iv) the same cases repeated for nitrogen dilution. Figures 7(*a*)–(*f*) reports the most relevant parameters of the calculated cycles as a function of the SFT, i.e., by varying the hydrogen dilution rate.

5.1 Results With VGV Operations (Constant β). As discussed in Sec. 3, pure hydrogen combustion products show superior heat-transfer capabilities and a lower TIT must be selected (11 K—see Table 3). To keep the same pressure ratio, the airflow remains almost unchanged, as well as the heat input (LHV), for a number of reasons related to the variations of molecular mass, inlet temperature, nozzle cooling flow. The gas turbine power increases (3%), due to a larger turbine enthalpy drop, but the steam cycle loses some power (5 MW), due to a lower TOT (about 8 K) and gas flow. The total power slightly increases (0.7%) and a better efficiency is predicted. Note that this efficiency increase is not related to any improvement in the power cycle. It just depends on the thermodynamic properties and on the different lower heating value of the fuels (in fact, the higher heating value reduces efficiency).

When using steam dilution, we obtain (with respect to the undiluted H_2 case) (i) a lower TIT (Fig. 7(*c*), i.e., 23 K at dilution for SFT=2300 K), due to the higher heat-transfer capabilities of hot gases with larger water content, (ii) a reduced air flow (Fig. 7(*d*)), to accommodate for the added diluent flow, (iii) a relevant improvement of the gas turbine output (lower compressor power due to lower air flow, elevated turbine power due to a larger enthalpy drop), (iv) a reduced steam turbine output, due to the steam extraction. Therefore the total output does not change dramatically (Fig. 7(*a*)) but a different gas to steam turbine power ratio can be depicted (Fig. 7(*f*)). A loss of efficiency is predicted (Fig. 7(*b*): 2 percentage points at elevated dilution), because of the detrimental effects of steam/air mixing (typical of mixed gas/steam cycles, as discussed by Macchi et al. [14]).

The situation is different with nitrogen dilution, because: (i) the TIT and the TOT do not change significantly, the gas properties being very little affected by N_2 addition, (ii) the compressor air flow must be reduced because of nitrogen injection, to keep the $G_{R,T,\mathrm{IN}}$ unchanged, (iii) the gas turbine power increases due to the lower compression power, (iv) the steam turbine power remains unchanged (same TOT and gas flow), (v) the N_2 compressor power requirement is larger than the power augmentation of the gas turbine (42.7 MW versus 33 for the cases reported in Table 3), because it is less efficient than the gas turbine compressor (85.0% versus 92.4 on a polytropic basis) and brings the nitrogen (from atmospheric pressure) to a larger pressure than combustion air (1.2 times), sufficient for fuel mixing. Therefore the power output and the efficiency reduce with N_2 injection, mostly due to the above quoted effects regarding N_2 compression: the cycle thermodynamics is practically unmodified (differently from steam injection, strongly affecting the cycle with larger efficiency losses).

5.2 Results at Increased β (Constant Air Flow). When using pure hydrogen as the fuel, results are very similar to the previous case (a negligible variation of β here, of airflow there). On the contrary, significant differences arise with large dilution ratios: a larger pressure ratio is required to accommodate for the larger gas flow at the same airflow and turbine nozzle area. TIT must be reduced to keep into account for the higher coolant temperature (a consequence of the larger β), in addition to different heat-transfer properties of steam-rich mixtures. Figure 7(*e*) and Table 3 show that β must be increased to 18.5 for steam and to 19.7 for N_2 if the SFT should be kept at 2300 K, requiring the addition of at least one compressor stage. Compared to VGV operations, the TIT reduction is much larger (Fig. 7(*c*)) because of the higher coolant temperature (436°C at β=19.7 versus 406 at β=17), even if slightly larger cooling flow rates result from increased β, assuming that coolant passages are unmodified.

The lower TIT is the main reason for the lower efficiency obtained for the present cases (Fig. 7(*b*)); another reason is the increased kinetic energy loss at the turbine exhaust because of the higher flow rate through the same annulus area. The power output (Fig. 7(*a*)) is much higher than for the cases with VGV operations, because the air flow rate is no longer reduced and full advantage is taken from the added diluent flow. For the 2300-K dilution the gas turbine power rises to 314 MW (steam) and 340 MW (N_2) from an original value of 257 MW. Such a large modification will require a number of mechanical adaptations and a larger generator in addition to a modified compressor. Similar situations were encountered in the development of gas turbines for IGCC applications [12,13].

5.3 Results for the Re-engineered Machine. This is the situation showing the minimum impact on the cycle efficiency and the maximum improvement of the power output. With respect to the previous case, a larger power output is accomplished because a TIT reduction is no longer necessary, due to the same coolant temperature (unmodified pressure ratio) and to the adaptation of the cooling circuit to the different heat transfer capabilities (Table 3 shows that cooling flows vary according to the turbine flow, determining the blade surface). On another side, keeping the design pressure ratio allows for the optimum cycle configurations:

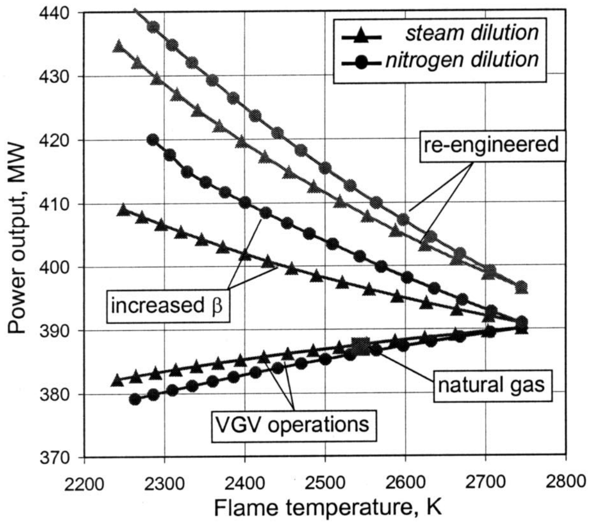

Fig.7a: Combined cycle net power output

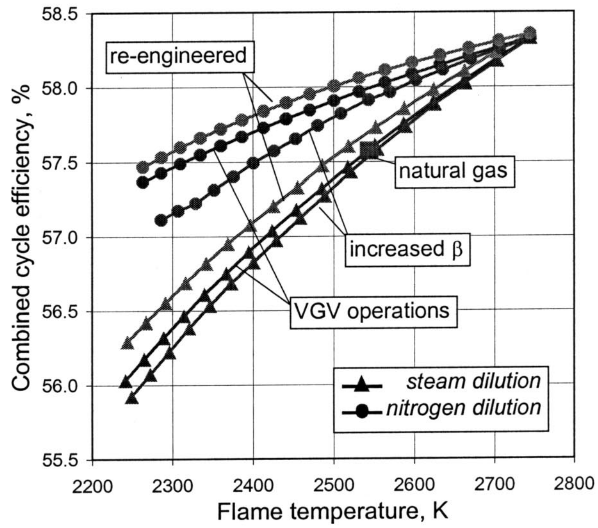

Fig.7b: Combined cycle net efficiency

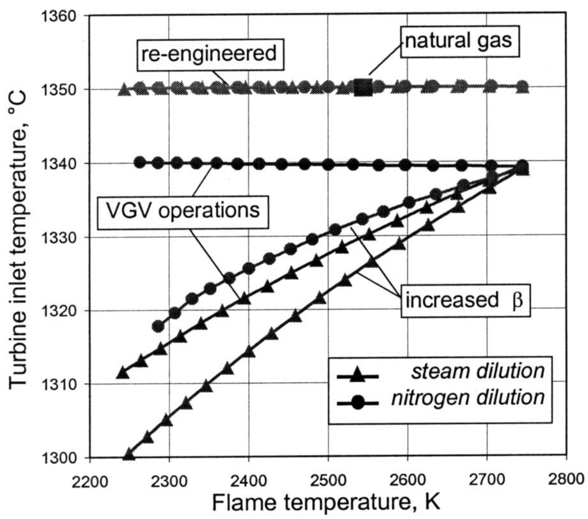

Fig.7c: Total temperature at first rotor inlet (TIT)

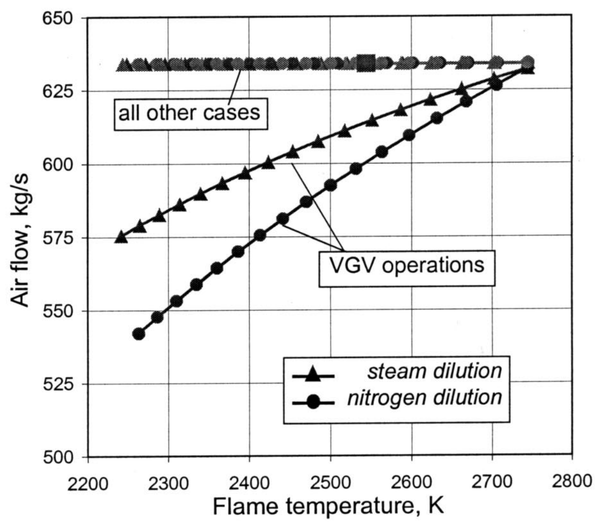

Fig.7d: Air flow at compressor inlet

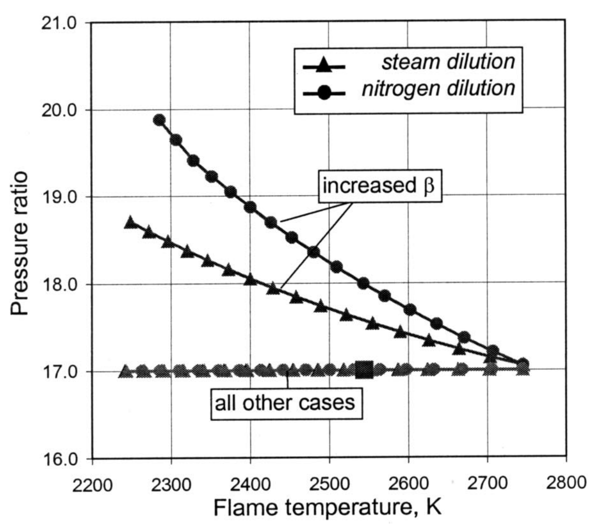

Fig.7e: Gas turbine pressure ratio

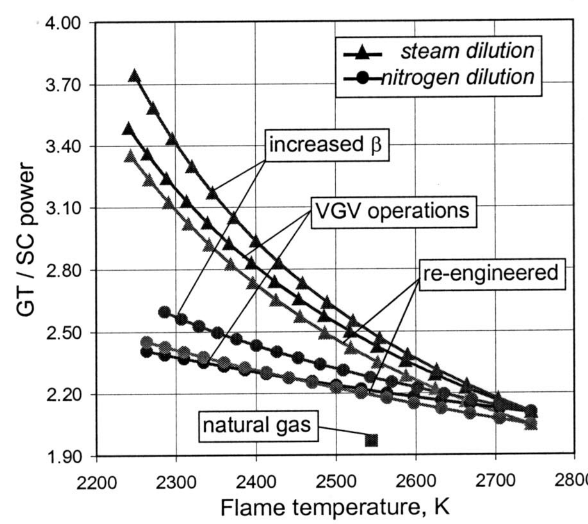

Fig.7f: Gas turbine / steam cycle power output

Fig. 7 (*a*) Combined cycle net power output, (*b*) combined cycle net efficiency, (*c*) total temperature at first rotor inlet (TIT), (*d*) air flow at compressor inlet, (*e*) gas turbine pressure ratio, (*f*) gas turbine/steam cycle power output

the efficiency decays for the same reasons described for VGV operations (steam mixing or higher pressure of compressed N_2, depending on the diluent).

6 Conclusions

The simulations carried out in this work allow a positive answer to the issues related to hydrogen combustion in modern gas turbines. However, a SFT abatement to about 2300 K seems necessary to comply with NO_x emission limits without incurring excessive operating costs of the end-of-pipe de-nitrification systems. This is possible without dramatic performance losses by means of a massive fuel dilution with steam or nitrogen (the latter providing minor losses of efficiency). Different strategies have been envisaged to operate the gas turbine in presence of dilution. Looking at the VGV operated solution (which appears the most likely for the first realizations) the efficiency loss is limited to 0.9 points for nitrogen dilution and 1.9 for steam dilution. Equally small is the influence on the combined cycle power output provided that the gas turbine power output can be increased (by about 10%) in consequence of the compressor airflow reduction. The other solutions here investigated (increased pressure ratio and re-engineered machine) are not particularly attractive in terms of efficiency but provide a much larger power output, an opportunity to reduce the specific costs provided that engineering costs are divided upon a sufficient number of units. It must also be noticed that VGV operations reduce the part-load capabilities of the machine, but make the gas turbine rather insensitive to elevated ambient temperatures (the "natural" power loss can be compensated by re-opening the VGV's). As a final consideration on system costs, it can be said that steam dilution allows for reduced capital cost compared to nitrogen, even if providing lower efficiency. In fact, a smaller steam turbine and condenser can be adopted, while the N_2 dilution requires a bulky and expensive additional compressor.

Acknowledgment

The work has been performed within the research on the Italian Electrical System "Ricerca di Sistema" Ministerial Decrees of January 26, 2000, and April 17, 2001.

Nomenclature and Acronyms

c_P = specific heat at constant pressure, J/kg K
D = reference blade dimension (chord), m
G = mass flow rate, kg/s
G_R = nondimensional mass flow rate ($G\sqrt{RT}/p$)
h = heat transfer coefficient, W/m^2 K
k = thermal conductivity, W/m K
LHV = lower heating value, MJ/kg
Ma_{AX} = axial Mach number at turbine outlet
p = pressure, Pa
SFT = stoichiometric flame temperature, K
T = temperature, °C or K
TIT = first rotor total inlet temperature, °C
TOT = turbine outlet temperature, °C
v = flow velocity, m/s
VGV = variable guide vanes
β = compressor pressure ratio
Δh = enthalpy drop, J/kg
γ = specific heat ratio
η = efficiency
μ = dynamic viscosity, Pa s
ρ = density of the gas stream, kg/m^3

Subscripts

C = relative to the compressor
COOL = coolant side of the blade wall
FIN = final condition
IN = inlet condition
IS = isentropic
OUT = outer side of the blade wall
T = relative to the turbine

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Demonstrated Applicability of Hydrogen Fuel for Gas Turbines

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Abstract:

In recent years significant progress has been made in the development of market applications for hydrogen fuel use in gas turbines. These applications include integrated gasification combined cycle (IGCC) and other types of process/power plants. Development of a new application using gas turbines for significant reduction of power plant CO_2 emissions has initiated extensive efforts to expand the range of hydrogen combustion capabilities. This paper reports on leading gasification systems producing hydrogen fuel, their technology background, and the results of a recent hydrogen combustion-testing program including resultant affects on gas turbine cycles.

Testing program results show the feasibility of hydrogen use for 20-90% CO_2 emission reduction with control of NO_X emissions to below 10 ppmvd at 15% oxygen. Operating flexibility, turndown and gas turbine life criteria are also discussed. Testing program data suggest that reliability, availability and maintenance (RAM) statistics for power blocks using hydrogen fuel can be maintained at equivalent levels to those of Natural Gas power plants.

Power Plants Using Hydrogen for CO_2 Emission Abatement

Widespread concern regarding global warming has initiated numerous studies of the economics of various solutions for reducing CO_2 emissions from power plants. Many of these studies focus on removing CO_2, either in the pre-combustion process, which affects the small, high-pressure fuel stream, or in the post-combustion process, which affects the very large atmospheric exhaust stream. Pre-combustion decarbonization schemes that rely on the use of hydrogen as a gas turbine/combined cycle fuel are considered to be more economical at the current stage of combustor development. The power block cycle is usually integrated with the process plant; incorporating gasification/reforming to separate the hydrogen from the carbon and removing the carbon as CO_2 for enhanced oil recovery or sequestration.

In plants fueled by natural gas the gasification block may use air- or oxygen-blown processes and may include catalytic partial oxidation (Cat Pox). In this case the plant is referred to as an integrated reformer combined cycle (IRCC) [Figure 1]. Following the production of H_2 and CO, syngas enters shift reactor and CO_2 removal system process blocks. The shift reactor block uses water to produce CO_2 and additional hydrogen. The CO_2 removal system block separates the CO_2 for enhanced oil recovery or sequestration. The hydrogen is then used as gas turbine fuel. The blocks outlined are needed only when CO_2 reduction is required. They amount to an additional plant cost of about 45% as compared to a conventional combined cycle power plant. New configurations such as reformers using heat from conventional Pox gasification are called Gas Heated Reformers. These combinations can reduce plant cost and improve economics.

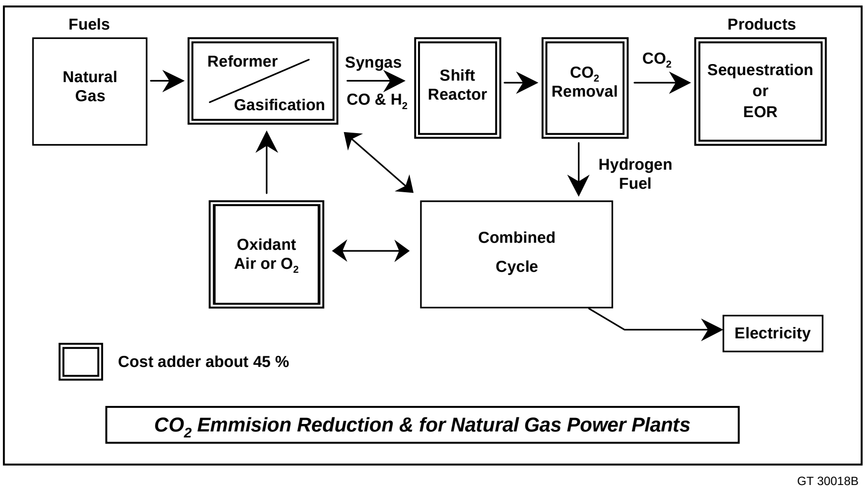

Figure 1: IRCC

Heavy oil and coal integrated gasification combined cycle (IGCC) plants already use many of the same process blocks to gasify fuel [Figure 2]. This reduces the incremental cost of CO_2 reduction. Cat Pox is not generally considered due to contaminants in heavy fuels. The blocks outlined in Figure 2 are only needed when CO_2 reduction is required, and amount to an additional plant cost of about 15%.

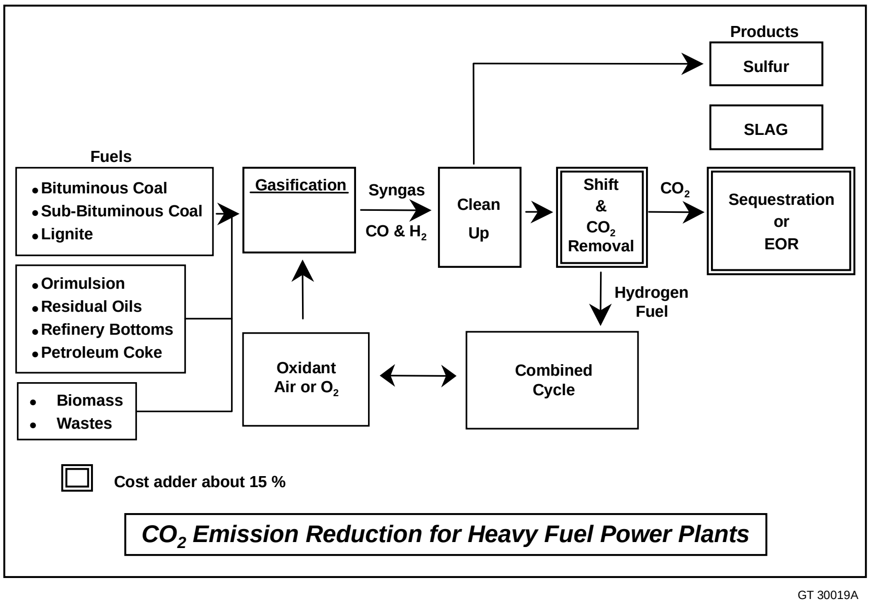

Figure 2: IGCC

Power Block/Process Block Integration

Gas turbines and gasification systems can be coupled by integrating the gas turbine with the steam- or air-side or both sides of the gasification system to improve economics.

The concept of both steam- and air-side integration is common in IGCC systems. However, there are new possibilities for integrating the gasification plant air-side with natural gas plants using auto thermal reformers (ATR) [Figure 3]. These Cat Pox processes operate at 30 bar pressure and high temperatures so integrating gas turbine air extraction with a booster compressor can be beneficial. Oxygen blown ATRs may also be used by inclusion of an Air Separation Plant (ASU). Alternatively, non-catalytic oxygen gasifiers may be used for large plants with air being supplied from the gas turbine.

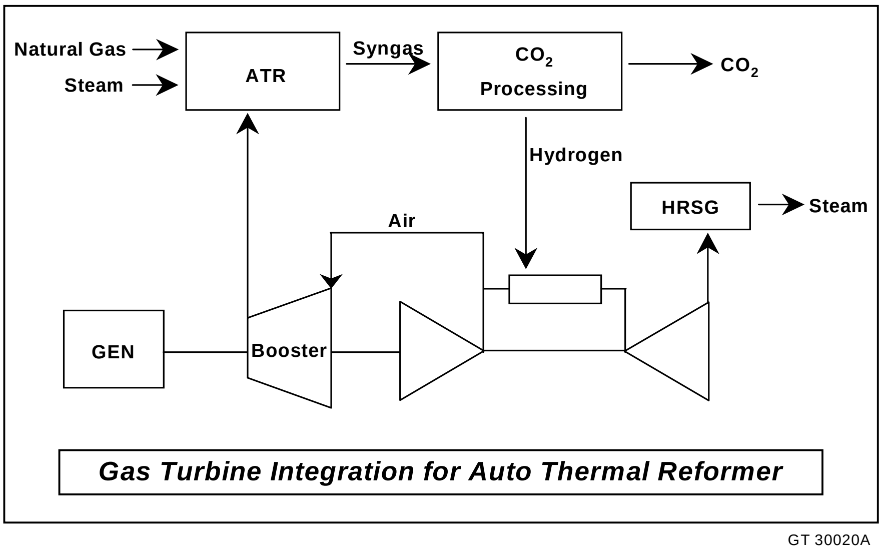

Figure 3: ATR Air Integration

Air may also be supplied from membrane separation as proposed in the Starchem Process [Figure 4]. In this process, air extracted from the gas turbine is separated into an oxygen-rich stream for syngas production and a depleted-air stream that can be returned to the gas turbine. The Starchem Process, as shown, produces methanol but the front end can also be used for CO_2 reduction cycles.

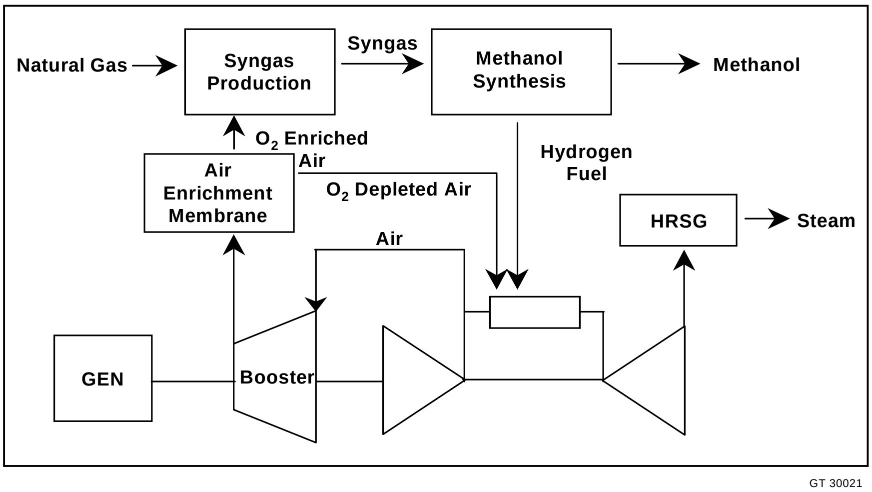

Figure 4: Starchem Process Air Integration

Optimization of these and other integration schemes can frequently produce a 20-30% improvement in economics compared with separate systems.

Hydrogen Fuel for Gas Turbines

Many existing gas turbine combined cycle (GTCC) plants, including refinery gas applications and all IGCC plants, use hydrogen-based fuels. IGCC plants use fuel ranging from 9 to 60% hydrogen by volume. Figure 5 shows the fuel constituents for plants in operation and under construction. A middle hydrogen range is shown for the IRCC plant. It is important to note that with IRCC the fuel is essentially 100% hydrogen since other combustibles are less than 2% by volume. Each of these fuels has been demonstrated successfully in full-scale lab testing and most tests were followed-up by field verification in operating units.

Syngas	PSI	Tampa	El Dorado	Pernis	Sierra Pacific	ILVA	Schwarze Pumpe	Sarlux	Fife	Exxon Singapore	Motiva Delaware	IRCC	Star Chem
H_2	24.8	37.2	35.4	34.4	14.5	8.6	61.9	22.7	34.4	44.5	32.0	46.84	40.0
CO	39.5	46.6	45.0	35.1	23.6	26.2	26.2	30.6	55.4	35.4	49.5	1.13	1.0
CH_4	1.5	0.1	0.0	0.3	1.3	8.2	6.9	0.2	5.1	0.5	0.1	0.75	9.0
CO_2	9.3	13.3	17.1	30.0	5.6	14.0	2.8	5.6	1.6	17.9	15.8	0.06	6.0
N_2 + AR	2.3	2.5	2.1	0.2	49.3	42.5	1.8	1.1	3.1	1.4	2.15	40.82	43.0
H_2O	22.7	0.3	0.4	--	5.7	--	--	39.8	--	0.1	0.44	10.40	--
LHV, - Btu/ft³	209	253	242	210	128	183	317	163	319	241	248	139	203
- kJ/m³	8224	9962	9528	8274	5024	7191	12,492	6403	12,568	9,477	9,768	5,480	8000
T_{fuel}, °F/°C	570/300	700/371	250/121	200/98	1000/538	400/204	100/38	392/200	100/38	350/177	570/299	--	450
H_2/CO Ratio	.63	.80	.79	.98	.61	.33	2.36	.74	.62	1.26	.65	41.5	40
Diluent	Steam	N_2	N_2/Steam	Steam	Steam	--	Steam	Moisture	H_2O	Steam	H_2O/N_2	NA	--
Equivalent LHV													
- Btu/ft³	150	118	113*	198	110	--	200	--	*	116	150	139	--
- kJ/m³	5910	4649	4452	7801	4334	--	7880	--	--	4600	5910	5480	--

* Always co-fired with 50% natural gas

High Hydrogen Content Syngas for IRCC Falls in IGCC Operating Range

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Figure 5: Syngas Comparison

Maximizing the net output from a system using production gas turbines sets cost of electricity (COE) optimization parameters. GE has used technology studies to establish parameters for optimizing COE for IGCC. Established criteria indicates that the use of hydrogen alone, while technically possible, is not economically sound and that some diluent, such as nitrogen, can significantly improve economics. In air-blown systems nitrogen is already in the syngas fuel supply, while in oxygen-blown systems the waste nitrogen stream from the air separation unit (ASU) can be used as the diluent.

Currently there are almost 5,000 MW of GE IGCC type plants in operation or on order, with a cumulative total of over 200,000 operating hours. These statistics contribute to the hypothesis that this type of integrated system can be operated safely and with equipment life characteristics similar to those of natural gas combined cycle [Figure 6]. Data gleaned from this extensive IGCC operating experience can also be used in developing suitable CO_2 emission reduction cycles.

Customer	Type	MW	Syngas Start Date	Hours of Operation		
				Syngas	N.G.	Dist.
Cool Water	107E	120	5/84	27,000	-	1,000
PSI	7FA	262	11/95	12,300	-	3,000
Tampa	107FA	250	9/96	12,800	-	3,800
Texaco El Dorado	6B	40	9/96	11,600	17,100	-
Sierra Pacific	106FA	100	-	0	20,500	-
SUV Vresova	209E	350	12/96	42,000	1,200	-
Schwarze Pumpe	6B	40	9/96	15,500	-	3,400
Shell Pernis	2x6B	120	11/97	18,600	17,900	-
ISE / ILVA	3x109E	540	11/96	62,200	2,200	-
Fife Energy	6FA	80	-	0	5,600	-
			Total	*202,000*		

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Figure 6: GE Syngas Experience – June 1999

To date, combined cycle operating experience with IGCC plants shows good reliability, availability and maintenance (RAM) results for a wide variety of syngases [Figure 7]. Experience also indicates that a reduction in firing temperatures of high hydrogen fuels will provide gas turbine metal temperatures consistent with those of machines operating on natural gas. The effects of higher flow and high moisture content without an appropriate control system can increase metal temperatures and significantly shorten equipment life. An IGCC system operating with a low NO_X requirement and using 45% hydrogen fuel by volume may have as high as 26% moisture in the working fluid. GE's practice in IGCC applications is to reduce firing temperature, thus keeping gas turbine material temperatures and component life similar to those in natural gas applications.

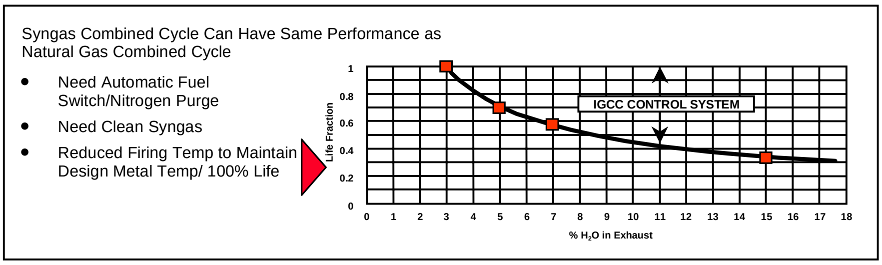

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Figure 7: Syngas – Reliability, Availability and Maintenance

The IRCC process differs from IGCC applications in that gas turbine fuel is hydrogen-only as compared to a mixed syngas. Studies have indicated a need for demonstration of hydrogen-only fuel for gas turbines. GE and Norsk Hydro have collaborated to provide the necessary demonstration.

Norsk Hydro Program

Norsk Hydro (NH) is a leading energy company headquartered in Norway and is the largest producer of hydrogen in Europe. Their studies of CO_2 emission reduction strategies with reference to COE optimization have focused on the IRCC process. GE and NH have collaborated to perform full-scale combustion system testing for a modern gas turbine with combustion exit temperatures of about 1400°C.

Combustion Test Program

The combustion test had three purposes:

1. To evaluate operability and emissions of the GE IGCC multi-nozzle combustor burning the Norsk Hydro primary design case gas. Operation testing was performed throughout the load range.
2. To evaluate component metal temperatures throughout the load range.
3. To determine sensitivity of major performance parameters (operability, emissions, steam effectiveness for NO_X control and component temperatures) to variations in hydrogen content.

Combustion system performance and operability is evaluated on the basis of measured exhaust emissions, combustion dynamics or pressure fluctuations, combustor metal temperatures, combustion system pressure drop, and flame stability and retention, particularly at low combustion exhaust temperatures and high inert gas injection rates.

Test Configuration

Testing was performed in GE's 6FA test stand because the 6FA IGCC combustion system is the basis for all current GE high temperature gas turbine IGCC combustion systems [Figure 8].

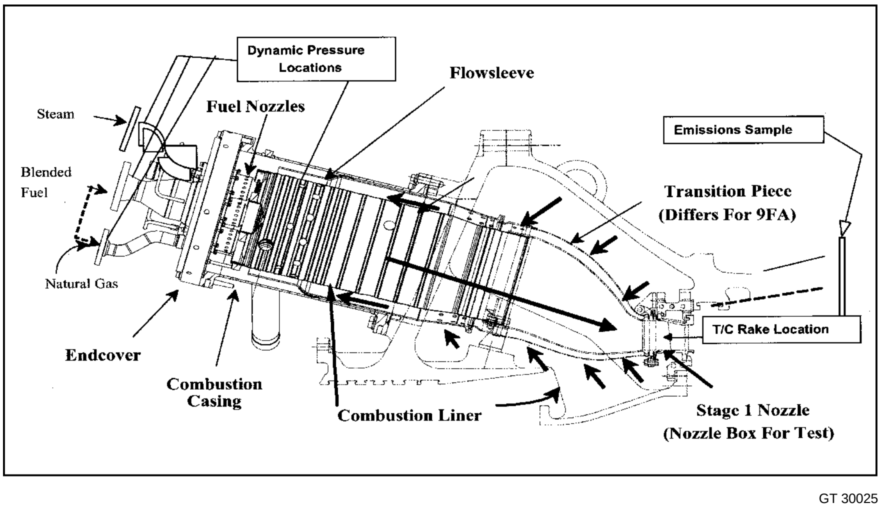

Figure 8: 6FA IGCC Multi-nozzle Combustion Systems

The 6FA test stand duplicates one combustion chamber on a gas turbine. Instead of a first stage nozzle section, the test stand is equipped with a "nozzle box," which simulates the open flow area of one nozzle segment. The nozzle box is instrumented with eight thermocouple "rakes," each containing five thermocouples located at 10, 30, 50, 70 and 90 percent radial height. In addition to the rakes, this section also contains a total pressure impact probe and emissions probes if needed. For most combustion tests, including those performed on the 6FA, emissions are taken downstream of the nozzle box in order to obtain a well-mixed sample that most closely represents actual turbine exhaust emissions.

Test Conditions

Natural gas fuel is used to fire the test stand and bring it to full airflow and firing temperature conditions for typical IGCC operation. After the inlet air temperature stabilizes, a series of test points fired on 100% natural gas are taken, followed by the addition of steam injection, to establish a baseline for combustor performance.

The transfer to hydrogen rich syngas fuel (H_2 syngas) begins with natural gas flowing through a fuel passage that is separate from the H_2 syngas passage in the fuel nozzle. While the combustor fires on natural gas, inert gas in this case, N_2 is introduced through the H_2 syngas fuel passage in order to purge the lines of any air and also to preheat the H_2 syngas lines. Next in the test sequence, steam is introduced through another separate passage. Following natural gas operation, the fuel supply is transferred from natural gas to the Norsk Hydro H_2 syngas fuel. Initiating the H_2 syngas blended fuel flow while natural gas is still flowing, but steam is turned off, prepares the transfer. After the individual components are set to the correct proportions, the controls are set to automatic and the transfer to H_2 syngas is complete.

Syngas Compositions

Table 1 shows H_2 syngas blend possible variations, depending on the configuration of the plant process. Only the major fuel constituents, H_2, N_2 and H_2O, were blended during testing, as indicated with the check marks in the last column of Table 1. The effects of the small amounts of remaining fuel constituents on flame stability and emissions were expected to be insignificant and in no way compromise the test objectives. In order to eliminate the confounding effect of CH_4 on emissions (particularly NO_X) it too was omitted from the other process gas compositions. Four nominal H_2 syngas blends were selected for testing: 46/41/13, 56/44/0, 77/23/0 and 95/5/0 ratios of $H_2/N_2/H_2O$. Note that in all cases the combustible portion of the mixture is 100% hydrogen.

Component	Component Molecular Weight	Gas C	Gas B	Gas A	Primary Design Gas	Represented in Test Syngas
H_2	2.016	53.9	76.96	94.69	45.5	✔
CO	28	0.15	0.39	0.38	1.1	
CH_4	16.04	3.07	3.28	3.64	0.4	
CO_2	44	0.1	0.1	0.1	0.6	
N_2 + AR	28	42.01	18.46	0.46	39.5	✔
H_2O	18.016	0.77	0.6	0.73	12.9	✔
Blend Mol. Wt.		13.57	7.51	2.90	14.94	
Blend LHV	(Btu/ft^3)	176.5	242.5	294.4	132.1	
	(kJ/m^3)	6952	9553	11599	5206	

Table 1: Matrix of Possible H_2 Syngas Composition Variations

A schematic of the gas blending facility, with hydrogen stored in tube trailers and blended with vaporized N_2 in controlled proportions to form H_2 syngas compositions is shown in Figure 9. For test conditions using steam, the steam is blended in the H_2 syngas line before entering the combustor fuel nozzle.

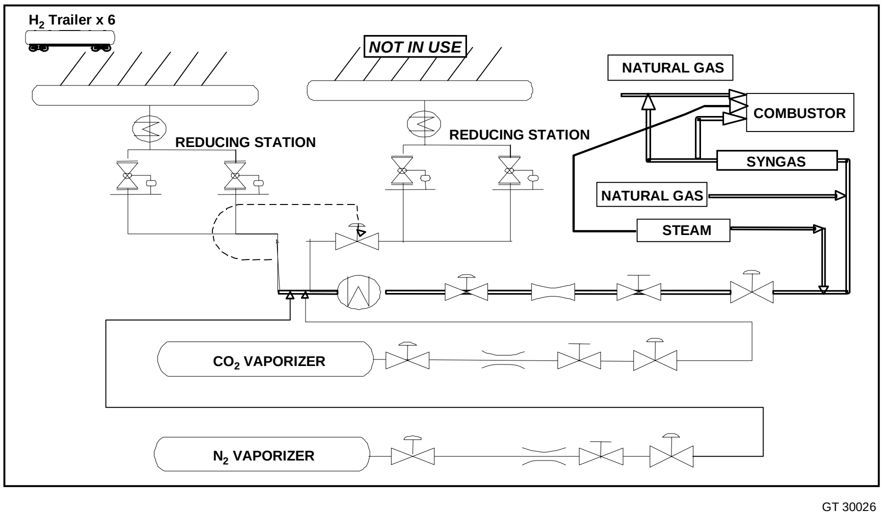

Figure 9: Schematic of Gas Blending System

Test Results

Natural Gas Baseline

Test results showed combustor performance on natural gas to be as expected. Emissions were within acceptable tolerance when compared to previous data recorded on the same combustor configuration. Figure 10 compares emissions from this test, as a function of the ratio of head-end steam injection to fuel flow by mass, with a previous test (indicated as Test 9068 in Figure 10). Both tests show low CO emissions and the same effectiveness of steam in reducing NO_X. Unburned hydrocarbons (UHC) for both tests were below 1 ppmvd. Dynamic pressure fluctuations on natural gas were also very low.

Norsk Hydro Syngas Performance

Actual H_2 syngas compositions blended during the combustion test are shown in Table 2. H_2 syngas compositions, as measured by a spectrometer, are compared with those calculated from the individual flows. The mass spectrometer was configured to measure only dry H_2 syngas. The comparison shows a high level of agreement between the two methods of measurement.

Test points SG1, SG1S1 and SG7 through SG9 were run on the primary Norsk Hydro H_2 syngas composition. While H_2 and N_2 content at any given test point differs from the desired values indicated in Table 2, the average composition for the five test points was within 5% of the target values.

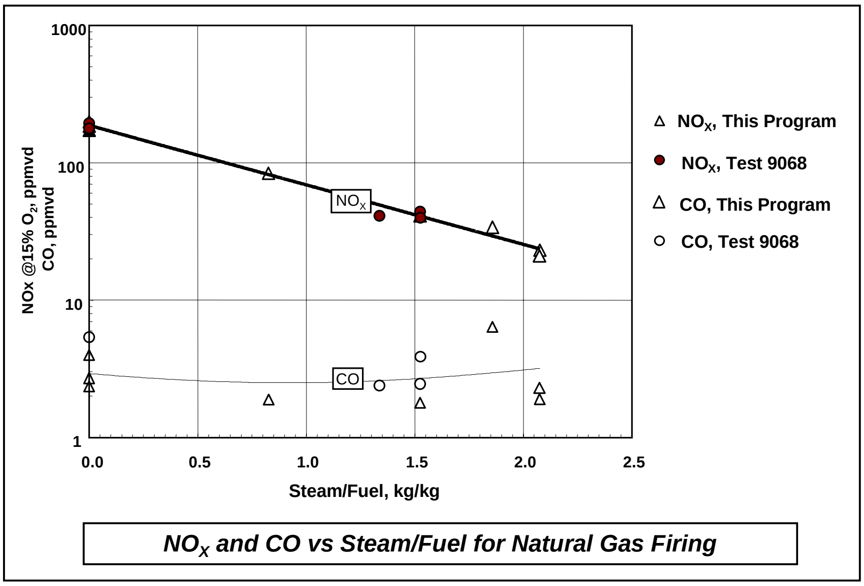

Figure 10: NO_X and CO Emissions During Natural Firing

Test Point	Mass Spectrometer		Flow Calculation			dH_2 +/- % of Ave	dN_2 +/- % of Ave
	H_2	N_2	H_2	N_2	H_2O		
SG1	-	-	53.59	33.06	13.35	N/A	N/A
SG1S1	-	-	54.70	31.99	13.31	N/A	N/A
SG2S1	-	-	58.88	41.12	0.00	N/A	N/A
SG3	73.70	26.24	76.82	23.18	0.00	2.08	6.20
SG3S1	74.30	25.73	77.35	22.65	0.00	2.01	6.36
SG4S1	88.28	11.85	89.23	10.77	0.00	0.54	4.76
SG4	85.75	14.36	87.00	13.00	0.00	0.72	4.98
SG7	46.99	52.79	50.08	49.92	0.00	3.18	2.79
SG8	45.81	53.98	50.13	49.87	0.00	4.50	3.96
SG9	41.73	58.00	43.46	56.54	0.00	2.02	1.27

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Table 2: Actual H_2 Syngas Composition Tested

Gas composition values showed significantly less variation than the variation for which the 6FA gas turbine combustor is designed and in no way invalidates the test. The remaining test points represent variations in H_2 and N_2 content for dry H_2 syngas.

Emissions:

Figure 11 shows the laboratory NO_X emissions as a function of steam-to-fuel mass ratio. The combustor exit temperature was low for the case of H_2 syngas containing 85-90% hydrogen and was in fact, falling while data were being taken, as noted in Figure 11. Therefore, the effect of steam injection on NO_X is not representative for this case, but is shown for completeness.

NO_X emissions may also be viewed in terms of equivalent calorific value, or the calculated calorific value one would obtain if the head-end steam injection were actually mixed with H_2 syngas. The results are illustrated in Figure 12. Note that the projected NO_X for 90% hydrogen is considerably greater than measured, again due to the falling combustor exit temperature.

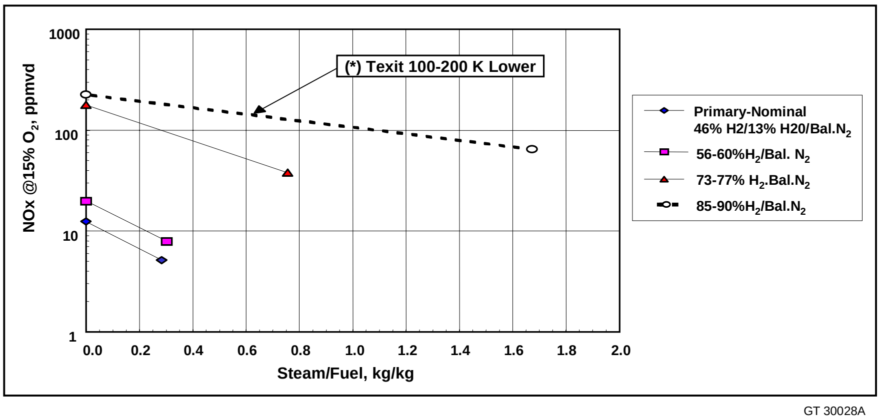

Figure 11: H_2 Syngas NO_X Emissions as a Function of Steam/Fuel Mass Ratio

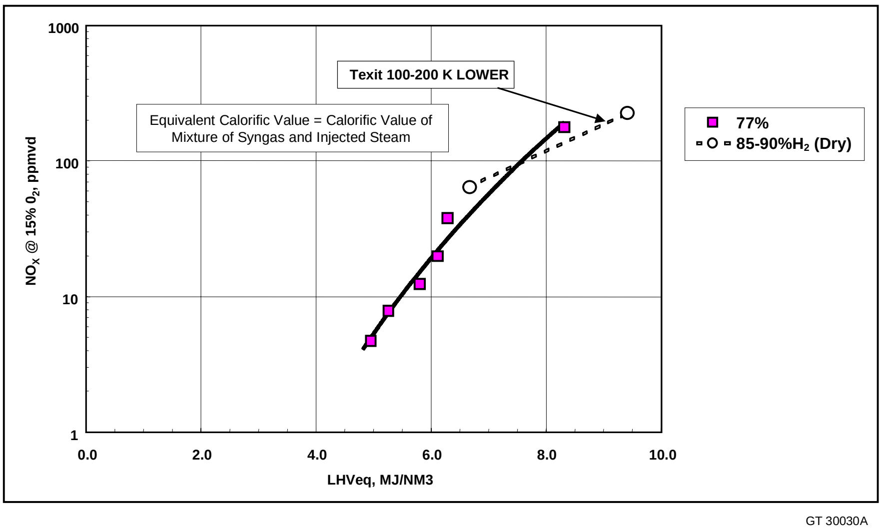

Figure 12: NO_X vs Equivalent Calorific Value for Several Fuel Compositions

Combustion Pressure Fluctuation:

Combustor test results showed very low combustion pressure fluctuation for both natural gas and H_2 syngas. The measurement is taken just inside the combustion liner in the primary reaction zone. A spectrum analyzer is used to record a dynamic pressure signal, typically at a frequency range of 0 to 800 Hz.

Figure 13 represents the overall root mean square (RMS) dynamic pressure levels as a function of steam-to-fuel ratio at full firing conditions. The data are expressed as a percentage of the maximum design amplitude. RMS levels are generally used to judge potential combustor life and stability issues. In all cases tested, the levels were less than 40% of the maximum design amplitude.

Although not shown, the maximum discrete amplitudes (pure tones at a given frequency) were less than 20% of design criteria on H_2 syngas and less than 50% during natural gas firing for all conditions tested. The maximum discrete amplitudes generally occurred at one of the combustion system's fundamental "organ pipe" frequencies, although at low levels it is often difficult to distinguish a maximum at any one frequency. In all cases, discrete levels were below the measurement threshold, or the level at which the data are considered meaningful. Higher frequency ranges were checked for at intervals throughout the test. Amplitudes in the higher frequencies were also below the measurement threshold.

The low dynamic levels measured in the combustor test were typical of the performance of GE IGCC combustion systems on H_2 syngas and consistent with GE's extensive experience base of burning low calorific fuels. Levels were well below present design criteria. As expected, the combustion noise was lower on H_2 syngas than on natural gas, but at such low levels the differential effect is small.

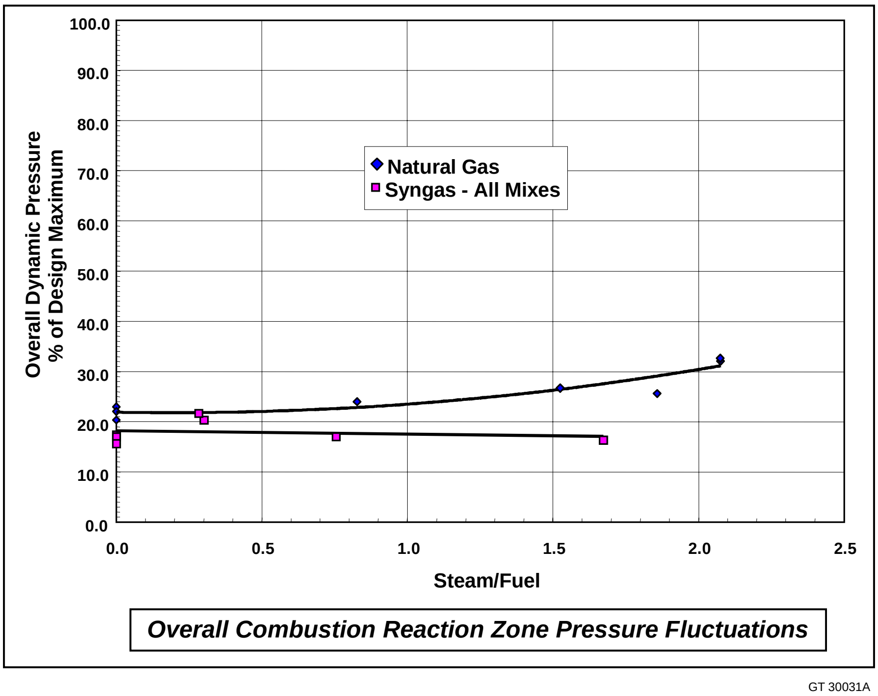

Figure 13: Overall Dynamic as a Function of Steam (Injected) Fuel Ratio

Combustor Metal Temperatures

Metal temperatures were measured along the length of the liner at locations both in-line with a fuel nozzle tip and in between two of the six tips. In all cases the metal temperatures were well below the upper design limit of about 780°C. Figure 14 shows the variation in axial temperature distribution with fuel composition (primarily hydrogen content) for the row of thermocouples in-line with a nozzle gas tip. The temperatures in this location are typically the highest. Temperature distribution is shown as a ratio of measured temperature during full firing conditions on H_2 syngas to the design limit for the 6FA gas turbine combustor. All temperatures measured on the liner were below the design maximum and are typical of IGCC combustors in production. This was also true when steam was injected for NO_X control.

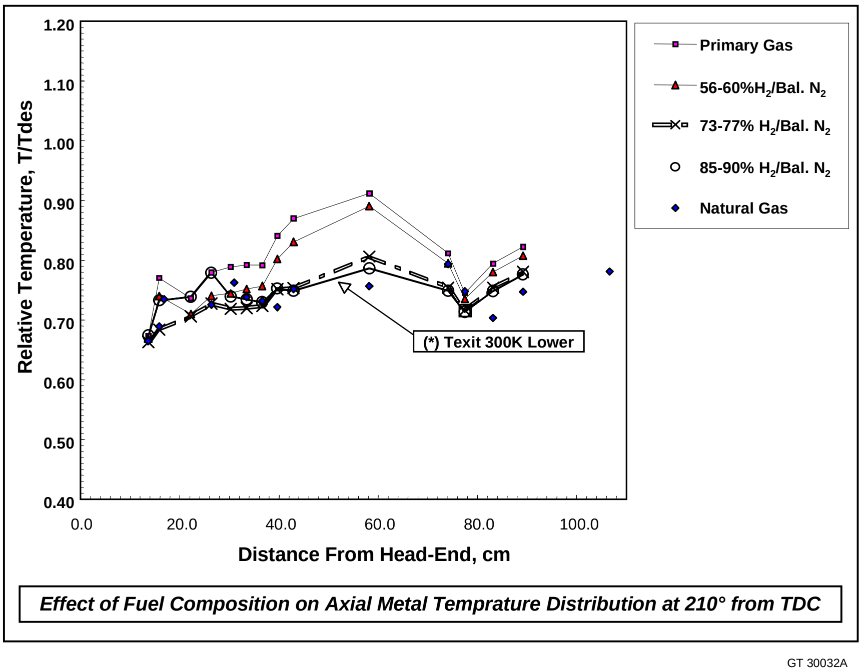

Figure 14: Combustor Metal Temperatures at 210º from TDC – Inline with Fuel Nozzles

Increasing the total hydrogen content of the mixture resulted in decreased liner metal temperatures, even though the stoichiometric flame temperature increased substantially with increasing hydrogen. This temperature reduction may be attributed to the burner design, which produces a very tight conical recirculation zone at the exit of each nozzle tip. As the amount of hydrogen is increased, the reaction occurs more rapidly and is contained further from the liner walls. The reduction in fuel-to-air ratio with increased hydrogen also contributes to a tighter flame. Video images of the flame zone confirm the flame structure as described [Figure 15]. A considerable reduction in wall temperature at 85-90% hydrogen is attributed more to the substantial reduction in combustor exit temperature than the actual hydrogen level.

Combustion Test Summary and Conclusions

The results presented here clearly demonstrate the feasibility of burning hydrogen as the only combustible, up to 90% by volume of the total fuel in GE's IGCC combustion systems. The impacts on combustion performance and expected hardware life, if any, were minimal within the parameters tested. The limited supply of hydrogen precluded firing at 100% hydrogen with no diluent. Combustion metal temperatures were well within acceptable limits and there was no apparent evidence of flame holding on the face of the fuel nozzle gas tips.

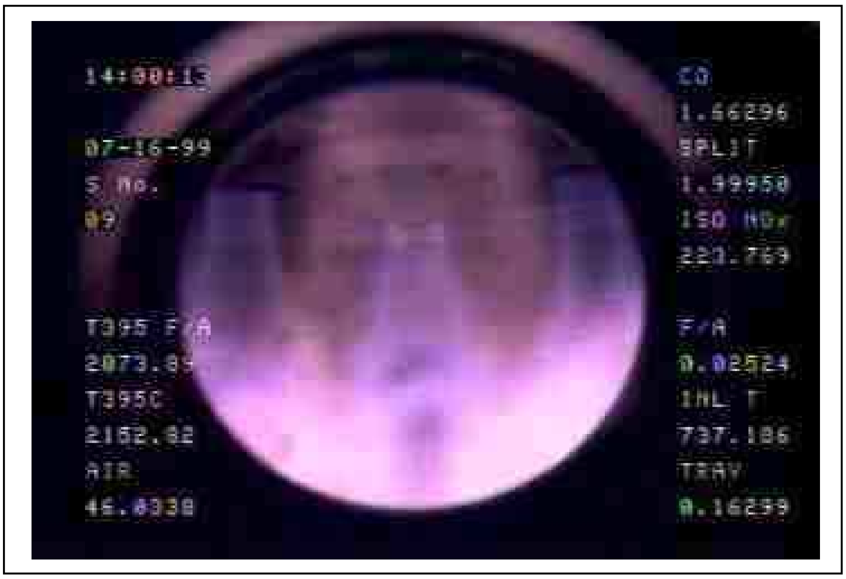

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Figure 15: Video Capture of Flame Structure, 85-90% Hydrogen

Combustion pressure fluctuation was very low on all fuels including natural gas. Head-end steam injection had very little effect on dynamics within the levels injected during testing. It is expected that higher levels of injection are possible during H_2 syngas operation without significantly affecting combustion noise.

The effect of variation in total hydrogen content on NO_X emissions was as expected, as was the amount of steam needed to suppress NO_X. The data appear to fit the previous NO_X correlation with respect to calculated flame temperature reasonably well [Figure 16]. The 7FA data used for comparison was taken from over seven separate tests with multiple combinations of H_2, CO, CO_2, H_2O, N_2 and CH_4.

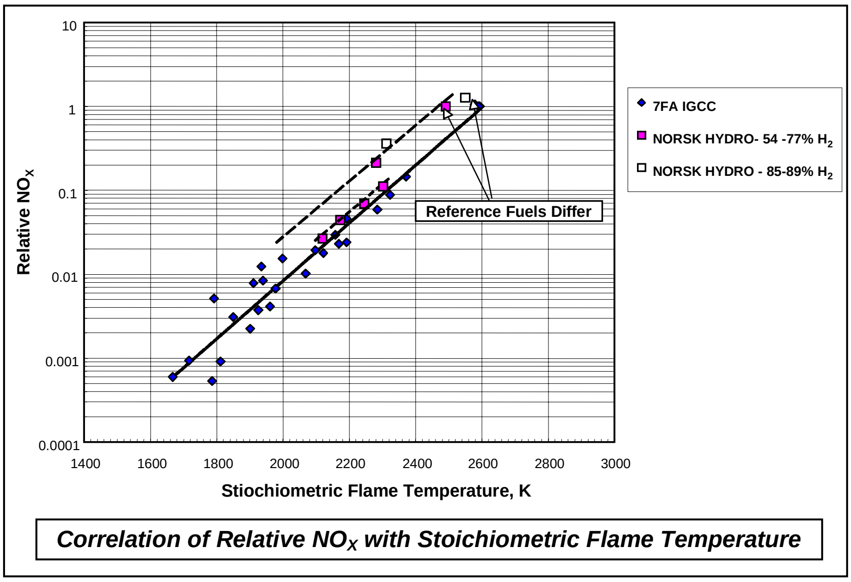

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Figure 16: Relative NO_X = NO_X/NO_X @ T Reference

While lower NO_X levels could have been achieved with higher steam injection rates, this may not be practical. Even at the moderate injection levels tested here, the exhaust moisture content is over 20% by volume. Such high levels of moisture significantly shorten turbine bucket life unless firing temperatures are substantially reduced, as discussed in the “Hydrogen Fuel for Gas Turbines” section of this paper.

H_2 syngas compositions simulated here were representative of an application where CO_2 in the gas stream was minimized. In cases where H_2 syngas contains substantial amounts of CO_2, similar NO_X levels may be achieved with considerably less steam. Figure 17 illustrates the calculated impact on NO_X emissions of CO_2 in H_2 syngas.

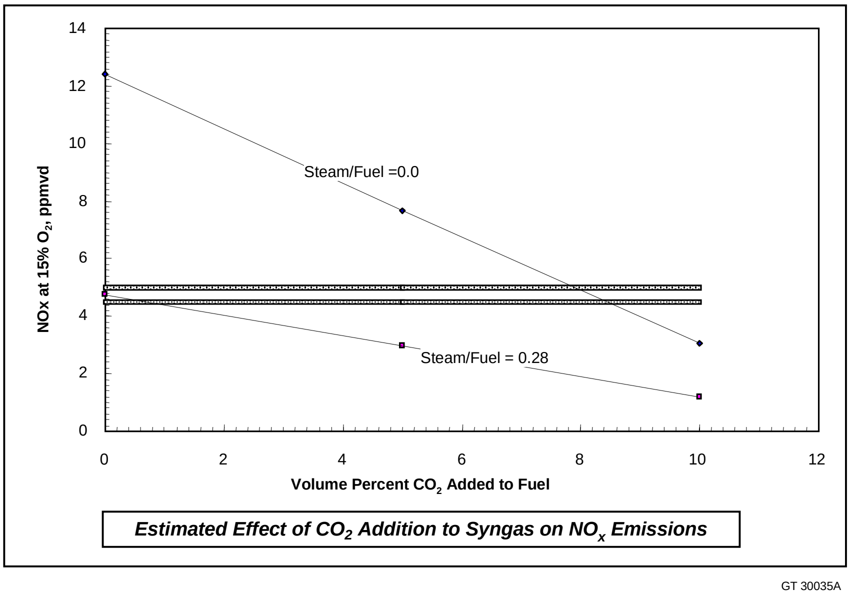

Figure 17: Primary Gas Composition

Effect of Test Results on Pre-Combustion Decarbonization – CO_2 Reduction Power Cycles

For coal and heavy oil IGCC plants where gasification costs are already included, test results show that CO_2 emission reduction can be accommodated without a large effect on COE, RAM statistics or operating parameters.

For natural gas power plants, the costs of gasification will tend to increase the COE but the effect on RAM statistics and operating parameters will be minimal. IGCC experience has shown that the integration of process and power blocks can be optimized to reduce extra costs. Normally it is expected that increased net output can be obtained during the optimization process, considerably reducing the cost of gasification.

The test results described above show that CO_2 emission reduction may be feasible with adherence to low emissions of NO_X and other pollutants but at extra cost.

Now that a combustion system technology has been developed for IRCC cycles, the next steps will be further product development by optimization and integration to reduce costs for CO_2 reduction applications.