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Gas Turbine Assessment for Air Management of Pressurized SOFC/GT Hybrid Systems

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This paper analyzes and compares transient and steady-state performance characteristics of different types of single-shaft turbo-machinery for controlling the air through a pressurized solid oxide fuel cell (SOFC) stack that is integrated into a SOFC/GT pressurized hybrid system. Analyses are focused on the bottoming part of the cycle, where the gas turbine (GT) has the role of properly managing airflow to the SOFC stack for various loads and at different ambient conditions. Analyses were accomplished using two disparate computer programs, which each modeled a similar SOFC/GT cycle using identical generic gas turbine performance maps. The models are shown to provide consistent results, and they are used to assess: (1) the influence of SOFC exhaust composition on expander behavior for on-design conditions, (2) the off-design performance of the bypass, bleed, and variable speed controls for various part-load conditions and for different ambient conditions; (3) the features of such controls during abrupt transients such as load trip and bypass/bleed valve failure. The results show that a variable speed micro-turbine is the best option for off-design operation of a SOFC/GT hybrid system. For safety measures a bleed valve provides adequate control of the system during load trip. General specifications for a radial GT engine for integration with a 550 kW pressurized SOFC stack are identified, which allow operation under a wide range of ambient conditions as well as several different cycle configurations. [DOI: 10.1115/1.2714567]

Keywords: SOFC/GT hybrid system, air management, ambient conditions, off-design modeling, transient modeling, gas turbine

Introduction

The National Fuel Cell Research Center (NFCRC), California, and the Thermochemical Power Group (TPG), Italy, have been working several years in the solid oxide fuel cell/gas turbine (SOFC/GT) hybrid system field: some of the latest achievements are reported in Refs. [1–4]. Previous studies and experimental evidence on high temperature pressurized SOFC/GT hybrid systems for stationary applications (i.e., distributed power generation) showed the need of air management devices that could ensure flexible operation at different loads and ambient conditions. The air management devices would maintain safe SOFC operating temperature, and enhance the availability of the hybrid system especially for off-design operating conditions. Gas turbine engines, and especially microturbines (net power below 500 kW), have been identified, thus far, as the best candidates for the first generation of pressurized SOFC/GT hybrid systems. Microturbines can provide air at the right pressure and temperature range and can effectively recover energy from the fuel cell exhaust to enhance overall efficiency.

Many companies, including FuelCell Energy, General Electric, Siemens Power Corporation, and Rolls-Royce Fuel Cell Systems are advancing SOFC/GT systems. Siemens and Rolls-Royce have introduced two basic design options for a pressurized hybrid system: (1) integrating an existing microturbine [5]; or (2) designing a new piece of turbomachinery [6], respectively. Each of these options has merits, but both still require further investigation to develop control strategies for managing the air to the fuel cell stack at different operating conditions, as well as recovering power for enhancing system efficiency. Thus, three control strat-

egies are proposed and investigated in the present work for a single-shaft microturbine with radial turbomachinery, which is representative of several existing microturbines:

1. *Variable speed control:* airflow is controlled by manipulating the generator load to change the rotational speed (N) of the turbomachinery;
2. *Bypass valve control:* a bypass valve is installed on a bypass branch that connects the compressor outlet to the turbine inlet, allowing a partial bypass of the compressor intake air around the fuel cell; and
3. *Bleed valve control:* a bleed valve is installed just after the compressor outlet that can be used to reduce the air sent to the stack by venting

Even if combinations of the above control strategies are possible, they have been assessed separately in the current analyses to show their specific features. Any individual hybrid system design could require several control options, but, would garner cycle simplicity and reliability by adopting only one of the control strategies.

Gas Turbine Modeling

Approach. Gas turbine technology is much more mature than that of a fuel cell. Most of the new installed electricity generating capacity is being met today by natural gas fired “combined gas turbine-steam turbine plants (just “combined cycles” in the following). Today’s gas turbines are pushing technology limits of emissions control and operating techniques as well as materials constraints that would require innovation or discovery and development of new materials to significantly advance the technology further. The design turbine inlet temperature (TIT), compressor pressure ratio, compressor and turbine efficiencies, blade cooling techniques, and combined cycle integration strategies have dramatically improved cycle performance over the years of advance-

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Fig. 1 Basic schematic of a single shaft radial compressor and turbine

ment and development. Unfortunately, the turbomachinery requirements for SOFC/GT hybrid systems are so different from those required to advance simple cycle or combined cycle machines that SOFC/GT turbomachinery may not benefit from these dramatic improvements. For example, the TIT of SOFC/GT cycles tends to be low ($<950^{\circ}\text{C}$), which decreases the performance of the GT, and only relatively low pressure ratios are desired. On the other hand, these requirements are fundamentally less challenging to achieve than those of modern GT and combined cycle technologies. Thus, it is likely that durable, robust and less sophisticated but proven turbomachinery can be well suited for integration with high temperature fuel cells. While this is not yet proven in practice and several advancements are required, the promise of low-technology turbomachinery clearly could increase overall reliability and produce very high efficiency when well integrated into a hybrid SOFC/GT system.

In this work, two models were employed: the NFCRC model [7,8] and the TPG tool "TRANSEO" [9,10]. Both tools are Matlab based and have been previously used for assessing the dynamic performance of high temperature fuel cell gas turbine hybrid systems. Here the focus is on the turbomachinery side, hence a few details about the compressor and turbine modeling are provided.

Fig. 2 (a) Compressor pressure ratio versus corrected mass flow (nondimensional); and (b) compressor efficiency versus pressure ratio (nondimensional)

Fig. 3 (a) Turbine expansion ratio versus corrected mass flow (nondimensional); and (b) turbine efficiency versus expansion ratio (nondimensional)

The description of the models presents an overview of the common modeling approach used in both models, which only differs in the implementation strategy, the specific thermo-physical property correlations used, and in the time-dependent representation of single components.

The models predict the behavior of a lumped parameter compressor and turbine attached via rotating shaft, as shown in Fig. 1. A load can be applied to the shaft and the system dynamic behavior can be predicted.

Because of the wide variety of designs and operating parameters inherent to gas turbine design, it is important to characterize both the compressor and turbine of a gas turbine system. Manufacturers typically flow test their gas turbine components to determine the actual performance of any particular design. The performance characteristics of both the turbine and compressor can be quite complex functions of operating conditions that are very challenging to predict. In an effort to "map" component performance over a wide array of operating conditions, components are typically characterized using nondimensional variables related to rotational speed, mass flow, and pressure ratio.

Performance maps of compressors and turbines are generally presented in a two-dimensional (2D) format with additional lines that correspond to constant conditions of a third variable.

The generic maps of the radial compressor and radial turbine used in this work are shown in Figs. 2 and 3, respectively. Figure 2(a) presents the nondimensional pressure ratio (β/β_{nom}) and Fig. 2(b) presents the efficiency (η/η_{nom}) of the compressor versus nondimensional mass flow rate with lines corresponding to constant nondimensional (or corrected) shaft rotational speed (N_{corr}). Figure 3 presents the same information for the turbine.

For the corrected shaft rotational speeds (N_{corr}), in between those presented in Figs. 2 and 3 must be interpolated. NFCRC generated a 3D surface to represent the parameter (efficiency or mass flow) as a function of two variables (shaft speed, pressure ratio, or mass flow); the Matlab function called "griddata" was

used to interpolate this surface. TRANSEO used instead a specially designed interpolation subroutine that was developed to interpolate turbine and compressor maps for gas turbine applications.

The “surge line” in Fig. 2(a) is used to define the surge margin K_p of the compressor. Compressor surge is caused by a complex combination of factors that leads to a stalling or breakdown of flow around the compressor blades, which can lead to air accumulation in the compressor followed by large pressure transients, loud bangs, and even engine failure. Compressor surge should always be avoided in actual hybrid systems, and proper safety measures should be implemented in the plant (e.g., blow-off valve) to prevent surge from happening. This is usually done by keeping the surge margin within safe limits. The surge margin, K_p , is defined by Eq. (1)

$$K_p = \frac{\beta_{\text{surge}} m_{\text{surge}}}{\beta/m} > 1.05\text{--}1.10 \quad (1)$$

Commonly, the surge margin is maintained above 5–10% (i.e., $K_p > 1.05\text{--}1.10$) to ensure safe and stable compressor operation.

Finally, due to the particular design of radial microturbines with a very small distance between the compressor and turbine, it was found [11] that heat exchange between the compressor and the expander is not negligible as it is in axial machines (those with many compressor and turbine stages). This heat transfer from turbine to compressor was calculated in the current models by conduction heat transfer based on the average compressor and turbine operating temperatures with a conduction heat transfer coefficient. Such a coefficient can be calculated once the temperature rise in the compressor outlet due to the heat exchange (or the TOT decrease) is known.

Turbine Modeling. Turbomachinery performance curves, such as those presented in Figs. 2 and 3, are developed for a given working fluid and a fixed set of boundary conditions. In order to use these maps over a broader range of conditions, such as different inlet conditions or working fluid chemical compositions, similitude considerations are required. This is particularly true for the turbine, because the exhaust of the fuel cell presents thermodynamic properties that are significantly different from those of ambient air, which is uncommon for conventional recuperated microturbines.

A complete approach to properly deal with chemical composition variations of the exhaust is described in Ref. [9]. It is worth recalling that the final formulation of the corrected mass flow and corrected rotational speed that should be employed in the performance maps. Equation (2) provides the general dependency of turbine expansion ratio (ε) and isentropic efficiency (η) on the corrected mass flow and corrected speed

$$\varepsilon, \eta_{\text{st}} = f(m_{\text{corr}}, N_{\text{corr}}) = f\left(\frac{m \sqrt{RT_{0\text{in}}}}{P_{0\text{in}}} \middle/ m_{\text{crit}}, \frac{N}{\sqrt{kRT_{0\text{in}}}}\right) \quad (2)$$

$$m_{\text{crit}} = \frac{\sqrt{k}}{\left(\frac{k+1}{2}\right)^{\chi}} \quad (3)$$

$$\chi = \frac{k+1}{2(k-1)} \quad (4)$$

The standard efficiency obtained from Eq. (2) should finally be corrected with Eq. (5) to obtain the actual efficiency. Equation (5) represents the correction applied to η due to a variation in the ratio of specific heats, k

$$\frac{\eta}{\eta_{\text{st}}} = \frac{\left(1 - \frac{1}{\varepsilon^{k_{\text{st}}-1/k_{\text{st}}}}\right) \sqrt{\frac{k_{\text{st}}+1}{2}} (k-1)}{\left(1 - \frac{1}{\varepsilon^{k-1/k}}\right) \sqrt{\frac{k+1}{2}} (k_{\text{st}}-1)} \quad (5)$$

The ratio of specific heats is calculated using Shomate coefficients that account for both the contributions of individual species and the overall temperature to the specific heat at constant pressure, c_p .

Shaft Dynamic Balance. The turbine shaft that mechanically connects the components and determines the speed and output work relationships of a gas turbine is modeled by the basic principle of sum of torques. Each component that is mechanically connected to the shaft (compressor, turbine, generator load) and bearing losses contributes a torque such that a positive torque balance increases shaft/rotor speed and a negative balance decreases speed as in Eq. (6)

$$\Delta\tau = \tau_{\text{turbine}} - \tau_{\text{compressor}} - \tau_{\text{load}} - \tau_{\text{loss}} \quad (6)$$

The time dependant shaft balance equation is modeled by Eq. (7)

$$J \frac{d\omega}{dt} = \Delta\tau \quad (7)$$

where J is the combined shaft and attached rotating component polar moment of inertia. The gas turbine accelerates if $\Delta\tau$ is positive and decelerates when $\Delta\tau$ is negative. Once equilibrium of torques is reached, the gas turbine is at a new steady state.

Torque is related to power by Eq. (8)

$$P = \tau\omega \quad (8)$$

Referring to Eqs. (6) and (8), the “load” power is the actual load requested by the generator, which differs from the electrical power produced by the gas turbine because of electrical losses. These losses can be significant because high speed generators normally employ rectification followed by power inversion to match grid voltage and frequency.

The power “loss” is due to bearing losses that are modeled as reported in Eq. (9)

$$P_{\text{loss}} = P_{\text{loss}}^{\text{ref}} \left(\frac{N}{N_{\text{ref}}}\right)^2 \quad (9)$$

Gas Turbine Design Point

The gas turbine design basis is fixed to one set of data for all the results presented. That is, the size and performance characteristics of the compressor, turbine, and shaft are completely fixed for a single gas turbine engine design that is used for all cycle configurations A (variable speed control), B (bypass valve control), and C (bleed valve control), mentioned above. The reason this is done is twofold: first, initial hybrid cycles must rely on existing turbomachinery that must be analyzed without significant modification for its capability to regulate the air to the fuel cell stack in different cycle configurations; second, we hope to identify a gas turbine design that could act as a general purpose radial microturbine for integration into a SOFC/GT hybrid system.

The compressor and expander design point parameters are reported in Table 1, while the design-point mechanical and electrical losses are shown in Table 2.

The design parameters have been chosen in order to:

1. Fit the SOFC hybrid system application, being able to provide the stack design airflow at all the different ambient conditions studied; and
2. Allow the microturbine to properly work in a stand-alone configuration, consistent with the desire to use “existing” microturbine hardware. In simple cycle operation, at 1173 K

Table 1 Gas turbine design parameters

Compressor design data			Turbine design data		
p_{in}	(10^5 Pa)	0.99	p_{in}	(Pa)	4.50
T_{in}	(K)	288.15	T_{in}	(K)	1173
k	(-)	1.399	k	(-)	1.317
R	(J/kg K)	288.1	R	(J/kg K)	292.9
β	(-)	4.8	ε	(-)	4.4
m	(kg/s)	1.3	m	(kg/s)	1.27
N	(rpm)	50,000	N	(rpm)	50,000
η	(-)	0.8	η	(-)	0.85
ΔT heat exch	(K)	+8	ΔT heat exch	(K)	-7

TIT and 50,000 rpm rotational speed, the current, nonrecuperated microturbine generates 180 kW net power at 18% low heating value (LHV) efficiency with K_p equal to 1.20.

With regard to the parameters in Tables 1 and 2 it should be noted that:

1. Compressor pressure ratio is a bit higher than usual for microturbines in order to obtain an operating pressure of about 400 kPa for the fuel cell in the cycle configurations *A* (variable speed) and *C* (bleed control). The chosen design-point value represents capabilities that are between aluminum–alloy and titanium–alloy radial impellers;
2. The flow mismatch between the compressor and turbine is due to the assumption of a 2% leakage flow between them;
3. The design point turbine flow composition is reported in Table 4, and it represents the composition of the SOFC exhaust;
4. Mechanical power losses are modeled according to Eq. (9), while generator efficiency and inverter power losses are kept constant [11], and
5. “ ΔT heat exch” parameters refer to the design temperature increase and decrease in the compressor and expander outlet flows, respectively, due to mutual heat exchange [10]

As a final remark, the SOFC stack flow design point is assumed to be 1.0 kg/s (see next section), which is about 23% lower than the compressor design flow. The sizing of the compressor at this level provides the necessary margin to the microturbine for ensuring the nominal stack flow rate at standard conditions and at low

Table 2 Mechanical and electrical losses at design point

Bearing mechanical loss	(kW)	3.0
Electrical generator efficiency	(-)	0.98
Inverter loss (constant)	(kW)	14.0

Table 3 Assumptions for nonideal microturbine behavior and hybrid system components

Air inlet pressure drop	(-)	2%
Recuperator air side pressure drop	(-)	2%
Recuperator exhausts side pressure drop	(-)	2%
Fuel cell stack pressure drop	(-)	3%
Fuel cell stack outlet T	(K)	1160

air density ambient conditions, such as those with high ambient temperature or low ambient pressure (high altitude), as demonstrated in the off-design analysis.

Reference Gas Turbine SOFC Hybrid System

The SOFC/GT hybrid configuration is illustrated in Fig. 4. It is a pressurized SOFC/GT system employing a recuperator to pre-heat the cathode air delivered to the fuel cell stack. The microturbine is modeled as described before. The assumptions made for pressure drops through the hybrid system components are provided in Table 3.

The recuperator is modeled as a quasi-2D transient model, with ten nodes each on the air, exhaust, internal matrix, and external vessel sides.

The valves are modeled with generic opening characteristics: each fractional opening (0–100%) corresponds a pressure drop coefficient, which is used to calculate the actual mass flow rate.

The focus of the present paper is on the turbomachinery behavior coupled with the fuel cell at different operating conditions. In this regard, the fuel cell model used assumes that it contains a control strategy capable of maintaining the outlet temperature constant (T_5 in Fig. 4 is assumed equal to 1160 K in all the conditions considered in the off-design analysis). This is a reasonable assumption since the fuel cell temperature is usually precisely controlled and utilization is held constant resulting in a similar amount of heat release in the anode off-gas oxidizer. This allows the current research to focus on the air delivery conditions and on the flexibility of the air management provided by the “bottoming” cycle (turbomachinery and control valves). The simplified fuel cell model is based on a voltage–current curve taken from a single-cell atmospheric test rig [3] shown in Fig. 5. This voltage–current relationship is assumed to be the high-pressure operating characteristic of the fuel cell stack. This approximation is considered to be acceptable for this work because this paper is not focused on fuel cell performance. Since no particular fuel cell is simulated herein, this approximation allows reasonable prediction of fuel cell behavior without affecting conclusions that are made regarding turbomachinery or other system components. Moreover, since the electrical potential increases with pressurization and decreases due to cell stack connection ohmic resistance, these countervailing effects may mitigate the discrepancies be-

Fig. 4 SOFC/GT hybrid system schematic

Fig. 5 V/I curve for the single-cell SOFC model

tween predicted performance and actual fuel cell stack performance. A detailed analysis of the fuel cell system has already been performed by the authors in Refs. [1,3], but this is not required by the scope of the present paper.

The fuel cell power is derived from the voltage–current curve for a given current density. The electrochemical heat generation and reformation heat absorption are balanced in the SOFC taking into account heat transfer and the thermal capacitance of the SOFC to produce a dynamic fuel cell response.

Three different design points were defined for each control strategy *A*, *B*, *C*. The main cycle data are reported in Tables 4–6. Note that the turbomachinery never operates exactly at its design point, as defined in Table 1, because of the need for various amounts of airflow sent to the fuel cell stack, while also being able to deliver the on-design value of 1.0 kg/s in all the ambient conditions considered in the off-design analysis. Such tables also report the comparison between the NFCRC and TPG models, which produce consistent results, despite the slightly different assumptions in the ambient air composition. With this favorable comparison between the models demonstrated, further comparison

results are omitted from the off-design and transient analyses, since these comparisons were also favorable and thus of little interest to the scope of the present work.

With regard to the fuel cell air feeding conditions (point 4 of Fig. 4), it should be noted that the bypass control (case *B*) provides air at about 100 K less than cases *A* and *C*, because of the quenching effect due to mixing in front of the turbine inlet. On the other hand, design pressure is the highest for case *B*, which could turn out to be an advantage in terms of fuel cell performance.

Off-Design Analysis

Each of the three control cases was simulated for various cathode mass flow rates requirements in the SOFC. It should be noted that a mean of controlling the cathode inlet temperature is not attempted in this comparison, but additional control of the cathode inlet temperature would complement the mass flow control presented herein.

Case A: Variable Speed Microturbine. A variable speed microturbine is assumed as the control actuator for mass flow in the cathode of the SOFC and the entire system. The alternator load is manipulated in order to change the microturbine speed and mass flow. This control approach is probably the most “elegant” solution, as it does not require modifications to the air cycle that can be provided by a typical microturbine generator. Figure 6 presents the microturbine speed for various mass flows. The ambient pressure was 1 atm and three ambient temperatures were evaluated. Figure 7 presents the electrical power produced by the microturbine for the same conditions. The electrical power is not much affected by ambient temperature, but the speed increase required to meet cathode flow requirements for a 30°C increase in ambient temperature is 9.3% (from 43 krpm to 47 krpm).

Figure 8 presents the recuperator exit (potential cathode inlet) temperature. The minimum cathode inlet temperature was set at 773 K (500°C). To check whether this criterion is met, the high pressure recuperator exit temperature is calculated at the operating point to ensure that it is equal to or higher than 773 K. Figure 8

Table 4 Design point data for variable speed control (Case A). Input parameters are in *italic*.

Design point A (variable speed control)							
NFCRC				TPG			
Cycle data				Cycle data			
	m (kg/s)	p (10 ⁵ Pa)	T (K)		m (kg/s)	p (10 ⁵ Pa)	T (K)
1	1.032	0.992	288	1	1.025	0.992	288
2	1.011	3.82	470	2	1.005	3.80	469
3	1.011	3.82	470	3	1.005	3.80	469
4	1.011	3.74	839	4	1.005	3.72	840
5	1.032	3.63	1160	5	1.025	3.60	1160
6	1.032	3.63	1160	6	1.025	3.60	1160
7	1.032	1.03	903	7	1.025	1.03	899
8	1.032	1.02	560	8	1.025	1.02	558
9				9			
10				10			
Microturbine data				Microturbine data			
N	(rpm)	45,300		N	(rpm)	45,300	
K_p	(-)	1.30		K_p	(-)	1.31	
P_{el}	(kW)	115.4		P_{el}	(kW)	117.1	
Exhaust gas composition				Exhaust gas composition			
H ₂ O	Mol. fract.	0.0707		H ₂ O	Mol. fract.	0.0830	
N ₂	Mol. fract.	0.7610		N ₂	Mol. fract.	0.7447	
CO ₂	Mol. fract.	0.0354		CO ₂	Mol. fract.	0.0369	
O ₂	Mol. fract.	0.1319		O ₂	Mol. fract.	0.1265	
Ar	Mol. fract.	0.0000		Ar	Mol. fract.	0.0089	

Table 5 Design point data for bypass control (Case B). Input parameters are in italic.

Design point B (bypass control)							
NFCRC			TPG				
Cycle data			Cycle data				
	m (kg/s)	p (10 ⁵ Pa)	T (K)		m (kg/s)	p (10 ⁵ Pa)	T (K)
1	1.323	0.992	288	1	1.324	0.992	288
2	1.297	4.61	490	2	1.297	4.59	491
3	1.000	4.61	490	3	1.003	4.59	491
4	1.000	4.52	737	4	1.003	4.50	737
5	1.020	4.37	1160	5	1.023	4.36	1160
6	1.317	4.37	1022	6	1.318	4.36	1023
7	1.317	1.03	756	7	1.318	1.03	755
8	1.317	1.02	578	8	1.318	1.02	577
9	0.297	4.61	490	9	0.295	4.59	491
10				10			
Microturbine data			Microturbine data				
N	(rpm)	50,000		N	(rpm)	50,000	
K_p	(-)	1.22		K_p	(-)	1.23	
P_{el}	(kW)	119.9		P_{el}	(kW)	119.9	
Exhaust gas composition			Exhaust gas composition				
H ₂ O	Mol. fract.	0.0707		H ₂ O	Mol. fract.	0.0830	
N ₂	Mol. fract.	0.7610		N ₂	Mol. fract.	0.7447	
CO ₂	Mol. fract.	0.0354		CO ₂	Mol. fract.	0.0369	
O ₂	Mol. fract.	0.1319		O ₂	Mol. fract.	0.1265	
Ar	Mol. fract.	0.0000		Ar	Mol. fract.	0.0089	

suggests that a problem could arise during low-flow operating conditions due to temperatures that are too high in the recuperator ($\sim 950^\circ\text{C}$) in Fig. 8. This could require constraining the case A system to operation in a limited range of part load conditions (i.e., establishing a minimum part load that is achievable). Figure 9

presents the surge margin for the microturbine, which is reasonably high (>1.3) for all conditions suggesting safe operation away from compressor surge.

Analyses were further conducted for variations in ambient pressure, assuming the ambient temperature is 15°C . To simulate el-

Table 6 Design point data for bleed control (Case C). Input parameters are in italic.

Design point C (bleed control)							
NFCRC				TPG			
Cycle data				Cycle data			
	m (kg/s)	p (10 ⁵ Pa)	T (K)		m (kg/s)	p (10 ⁵ Pa)	T (K)
1	1.345	0.992	288	1	1.336	0.992	288
2	1.318	3.81	498	2	1.309	3.82	501
3	0.998	3.81	498	3	1.000	3.82	501
4	0.998	3.73	843	4	1.000	3.74	844
5	1.019	3.62	1160	5	1.019	3.62	1160
6	1.019	3.62	1160	6	1.019	3.62	1160
7	1.019	1.03	903	7	1.019	1.03	898
8	1.019	1.02	580	8	1.019	1.02	583
9				9			
10	0.320	3.81	498	10	0.31	3.82	501
Microturbine data				Microturbine data			
N	(rpm)	50,000		N	(rpm)	50,000	
K_p	(-)	1.48		K_p	(-)	1.49	
P_{el}	(kW)	17.2		P_{el}	(kW)	16.7	
Exhaust gas composition				Exhaust gas composition			
H ₂ O	Mol. fract.	0.0707		H ₂ O	Mol. fract.	0.0830	
N ₂	Mol. fract.	0.7610		N ₂	Mol. fract.	0.7447	
CO ₂	Mol. fract.	0.0354		CO ₂	Mol. fract.	0.0369	
O ₂	Mol. fract.	0.1319		O ₂	Mol. fract.	0.1265	
Ar	Mol. fract.	0.0000		Ar	Mol. fract.	0.0089

Fig. 6 Speed of microturbine for 1 atm pressure and various ambient temperatures

evated or high altitude operation (mountain) the pressure values of 0.8, 0.9, and 1.0 atm were evaluated. Figure 10 presents the microturbine shaft speed for various pressures and cathode mass flow rates. Higher shaft speed (Fig. 10) is required at lower ambient pressures to produce adequate mass flow, and the surge margin is significantly reduced for the low-pressure cases: in general, low-density cases require a higher “stress” to the turbomachinery. Figure 11 presents the microturbine power production for the vari-

Fig. 7 Microturbine power for 1 atm pressure and various ambient temperatures

Fig. 8 High-pressure recuperator exit temperature for 1 atm pressure and various ambient temperatures

Fig. 9 Surge margin for 1 atm pressure and various ambient temperatures

Fig. 10 Microturbine speed for 15°C and various pressures (atm)

ous pressures and mass flows. The power production of the microturbine improves with lower ambient pressures (at the same mass flow). There is approximately a 25 kW increase (20%) in power between the 0.8 atm and 1 atm cases at 1.0 kg/s mass flow. This can be explained by the fact that, at the same compressor intake air flow and same TIT, the compressor delivery pressure is almost the same: hence, the lower pressure cases present a higher compressor pressure ratio than the reference case.

Case B: Bypass Valve. A bypass valve was used to control the mass flow through the SOFC. Mass flow was bypassed around the cathode side of the SOFC. Figure 12 presents the percent bypass mass flow, based on the total compressor air intake flow. To achieve 0.5 kg/s through the SOFC, which is 50% of the nominal flow, more than 60% of the compressor mass flow is bypassed. Figure 13 presents the net microturbine power for various ambient temperatures and SOFC mass flows. Note that at 0.5 kg/s, the microturbine is being motored in order to maintain a shaft speed of 50 krpm.

Figure 14 presents recuperator exit (potential cathode inlet) temperatures where only in one case, $T_{amb}=30^{\circ}\text{C}$, does this reach the minimum requirement of 773 K. The other data points are kept to illustrate the trend even though they are not consistent with the assumptions that were made. The bypass valve control

Fig. 11 Microturbine power for 15°C and various pressures (atm)

Fig. 12 Bypassed percent mass flow for 1 atm pressure and various ambient temperatures

Fig. 13 Microturbine power for 1 atm pressure and various ambient temperatures

alone cannot adequately maintain SOFC cathode inlet temperature, unless a combustor is fired at the turbine inlet to elevate TIT, TOT, and, as a final result, elevate cathode inlet temperature.

The bypass control approach (case *B*) was analyzed for various ambient pressures. Figure 15 presents percent flow bypassed around the SOFC. Figure 16 shows the net power production for the microturbine. The microturbine is motored at lower mass flows through the SOFC, only for the higher-pressure case. As was the case for the various ambient temperatures, the bypass valve does not properly manage the hybrid system, because the cathode inlet temperature for the SOFC is generally too low. Nonetheless, one advantage of the bypass control of case *B* is that it can better maintain fuel cell operating pressure during off-design operation than the controls of cases *A* and *C*. In fact, while the *A* and *C* controls almost halve the pressure at 50% stack air flow, this case *B* control results in only a 10% pressure decrease at 50% flow [12].

Case C: Bleed Valve. A bleed valve was used to control the mass flow through the SOFC in this case. Mass flow from the compressor outlet was bled off until the desired mass flow through the SOFC was achieved. Figure 17 gives the percent mass flow bleed, based on compressor intake airflow. Not all of the operating

Fig. 14 High-pressure recuperator exit temperature for 1 atm pressure and various ambient temperatures

Fig. 15 Bypassed percent mass flow for 15°C and various pressures (atm)

Fig. 16 Microturbine power for 15°C and various pressures (atm)

points tested were valid, because it was decided not to motor the microturbine at more than 50 kW. This is the reason why the bleed valve control scheme could not achieve 50% mass flow to the stack for any of the temperature conditions investigated (Fig. 17). Figure 18 presents the net microturbine power for various ambient temperatures, which indicate a need to motor the microturbine for almost all conditions investigated with bleed valve control only.

For all of the operating points that were studied, the recuperator exit (potential cathode inlet) temperature is acceptable with bleed valve control. In addition, the microturbine operates above the surge margin for all valid operating points.

Various ambient pressure conditions produced results for the bleed valve control case that are similar to those for various ambient temperature conditions. There are a few more valid operating points at the lower pressure. Less mass flow was bled as shown in Fig. 19 for the 0.8 atm case. Figure 20 shows that there is better performance in the net microturbine power for the 0.8 atm inlet pressure condition.

For bleed valve control cases with various operating pressures,

Fig. 17 Bleed percent mass flow for 1 atm pressure and various ambient temperatures

Fig. 18 Microturbine power for 1 atm pressure and various ambient temperatures

Fig. 19 Bleed percent mass flow for 15°C and various pressures (atm)

the potential cathode inlet temperatures are acceptable for all of the operating points and the microturbine operates above the surge margin for all cases studied.

Finally, it is interesting to compare the three control cases from the SOFC operating pressure point of view. Figure 21 shows the variation of compressor delivery pressure at standard ambient conditions for the three control strategies: it is evident the similar pressure behavior of the variable speed control (case A) with the bleed control (case C), while the bypass control (case B) tends to reduce the variation in operating pressure.

Transient Analysis of Failure Response

The complete characterization of the air management cycle configurations A, B, and C also requires the time-dependent analysis of how the system responds to failures. In fact, the overall system availability and reliability are as important as cycle efficiency and operational flexibility. Reliability and availability significantly depend on the system response to failures that occur internally (e.g., damaged components) or externally (e.g., electrical grid trip).

Fig. 20 Microturbine power for 15°C and various pressures (atm)

Fig. 21 Delivery pressure at compressor exit for standard control cases ($T_{amb}=15^\circ\text{C}$ and $P_{amb}=1\text{ atm}$)

Table 7 Main assumptions for the transient analysis

Recuperator internal matrix weight	(kg)	100.0
Recuperator vessel weight	(kg)	5.0
Recuperator metal specific heat	(kJ/kg K)	0.7535
Compressor weight (for heat storage)	(kg)	15.0
Compressor metal specific heat	(kJ/kg K)	0.7535
Turbine weight (for heat storage)	(kg)	20.0
Turbine metal specific heat	(kJ/kg K)	0.7535
Microturbine shaft rotational inertia	(kg m ²)	0.024
SOFC system empty volume	(m ³)	15.0

To initiate analysis of failure response, two failure occurrences are studied: (1) load trip and (2) valve failure. The first applies to all cycle configurations (A, B, C), and it implies the sudden loss of power on the alternator; the second applies only to cycle configurations B and C, and it implies an abrupt complete opening of the valve with subsequent increase in air bypassed or bled. The valve failure results in a peak flow that is achieved in the initial moment when the valve abruptly opens, and flow is 20% more than the initial (design) flow once the valve is blown. Valve behavior during the transient is modeled as a concentrated pressure drop, which is caused by a variable area nozzle. Five cases in total are studied, and they all start from the hybrid system design points reported in Tables 4–6.

For the two cases of valve failure, the power to the alternator is automatically cut when 50% of the nominal rotational speed is reached, that is 25,000 rpm. The dynamic simulations were set to stop in the case of 20% overspeed (i.e., $N=60,000\text{ rpm}$), because it is considered to be the working limit of the turbomachinery, which is also often engineered into microturbines to prevent serious damage.

Since all the simulations carried out in this section are for relatively fast transient phenomena (about 3 min maximum), the fuel cell temperature is considered constant during the entire simulation period [1,4]. The major additional assumptions for the time-dependent simulations of this section are reported in Table 7.

Figure 22 presents the power loss failure. Load trip causes the shaft to accelerate, according to Eq. (7): due to the relatively small rotational inertia of the single-spool microturbine, the safety control system has very short time to react. For the variable speed (A) and bypass (B) cases the system overspeeds beyond 20% nominal speed in a very short time (about 20 s). The bleed valve control (case C) is the only mean able to prevent the turbomachinery from being seriously damaged during a loss of generator load (full time span not shown in Fig. 22). The bleed valve successfully removed the enthalpy from the system (reducing power production of the expander) quick enough to prevent the GT from overspeeding.

Fig. 22 Microturbine rotational speed versus time

Fig. 23 Trace of the compressor operating point in the non-dimensional mass flow–compression ratio map

Figure 22 also presents the valve failure for cases *B* and *C*. The microturbine stops in about 1 min for both the bypass or bleed valve failure cases. In fact, valve failures cause a decrease in expander power production, due either to lower TIT (bypass failure) or lower mass flow rate (bleed failure), which resulted in deceleration of the shaft. The main problems are related to the abrupt depressurization of the SOFC system, which if not prevented, loses about 3–3.5 bars in this short time (1 min).

An additional problem is present for the bypass valve failure. Since the pressure is kept high by the mass flow bypass in this case, compared to the bleed case, the compressor quickly (15 s) runs into surge (Fig. 23). Even if one assumes the compressor can recover from this surge event, unacceptable mechanical stress or even damage to the SOFC stack could occur during this surge event.

The only failure response case that reaches a new steady-state point is the load drop failure applied to the bleed valve control (case *C*) configuration. This is mainly due to the lower initial power obtained from the shaft (see Table 6), which enables the turbomachinery to accelerate only slightly when this smaller load (net load) is removed. These facts confirm that the bleed valve control is the most inefficient, but perhaps the safest (able to handle failure) control strategy. Actually, if the bleed valve were used only as a safety control measure, an alternative and more efficient strategy for part load would be employed for controlling the airflow to the fuel cell stack, hence resulting in a larger net power from the microturbine generator; in this case, power trip would result in a larger power unbalance onto the rotating shaft, hence requiring a bleed valve that would be larger if compared to the bleed control strategy (*C*) considered here.

Figures 24 and 25 show how the compressor air splits between the SOFC system (recuperator air side) and the control valve (bypass or bleed). The critical challenge is represented by the rapid decrease of flow to the fuel cell in the valve failure cases. This presents a safety and component damage issue apart from the issues related to fuel cell thermal management. In addition, the recuperator is also potentially adversely affected by the reduced cooling flow. A fast increase in the air temperature after the recuperator for the bleed valve failure case is to be expected, while the bypass valve failure case takes advantage of the cooling effect on TIT (and consequently on TOT). Moreover, high air temperatures achieved for the bleed valve failure case can also cause serious damage to the recuperator.

All the dynamic failure response cases analyzed here could cause serious damage to the hybrid system, except for the power loss failure in the bleed valve control case, where a new steady operating condition is achieved with a relatively smooth transient. Overall, from this transient study it is possible to conclude that:

Fig. 24 Air flow rate through the regulating valve (zero for the case of variable *N*) versus time

1. The variable speed control doesn't introduce additional components into the system, hence the failure possibilities are significantly reduced. Nevertheless, in the case of uncontrolled load trip, this control strategy alone quickly leads to turbomachinery overspeed. Auxiliary control devices, such as blow-off valves, become necessary to preserve system integrity.
2. The bypass valve control is the only configuration in this study that brought the compressor into surge, that is, when the bypass valve fails. This is certainly one of the most serious challenges for a hybrid system to handle. This type of control would also require some safety blow-off valves to maintain the compressor in a stable operating regime.
3. The bleed valve control case is shown to be the safest in responding to a power loss failure, while it could cause damage to the recuperator and fuel cell inlet in the case of valve failure, due to a significant increase in the temperature and decreased mass flow of the air delivered to the stack.

Conclusions

The design specifications for a general purpose microturbine to fit a 550 kW SOFC pressurized stack in a hybrid system have been identified. Such a microturbine needs to be oversized in terms of flow capacity in order to handle different ambient conditions while assuring the achievement of nominal power.

Three different cycle control strategies have been studied for assessing their features in managing and regulating the air to the fuel cell stack: (a) variable speed control; (b) bypass valve control; and (c) bleed valve control.

Fig. 25 Air flow rate in the recuperator cold side versus time

The results show that a variable speed microturbine is the best option for off-design operation of a SOFC/GT hybrid system, even if the recuperator temperatures, which tend to increase during part load, could provide serious operating limitations.

Bypass valve control alone, without the use of a post-burner in front of the turbine inlet, can cause too much thermal stress to the fuel cell stack, because of the low cathode inlet temperature, which is a consequence of the low TOT. Bleed valve control is not able to reduce significantly the flow to the stack, unless significant motoring power to the alternator is allowed.

For safety measures and response to failures, a bleed valve provides the only adequate control of the system in the case of load trip with respect to the microturbine. The bleed valve could cause overheating of the recuperator. Perhaps repositioning the bleed valve after the recuperator would correct this, but the response time would change.

In a fully controlled emergency shutdown, the SOFC load would be dramatically reduced or dropped. The SOFC would need to be isolated in a manner that would allow a safe depressurization while maintaining the correct environments (for example a reducing environment in the anode section) around the cells when the SOFC is above a critical temperature. These safety measures presented provide insight into the microturbine response. The findings from this analysis in combination with additional analyses with respect to the SOFC stack would aid the development of a comprehensive safety system.

Nomenclature

c_p	= constant pressure specific heat (kJ/kg K)
c_v	= constant volume specific heat (kJ/kg K)
GT	= gas turbine
J	= polar moment of inertia (kg m ²)
k	= c_p/c_v
K_p	= surge margin
LHV	= low heating value (kJ/kg K)
m	= mass flow (kg/s)
N	= rotational speed (rpm)
p	= pressure (Pa)
P	= power (kW)
R	= perfect gas constant (J/kg K)
SOFC	= solid oxide fuel cell
T	= temperature (K)
TIT	= turbine inlet temperature (K)
TOT	= turbine outlet temperature (K)

Greek Symbols

β	= compression ratio
ε	= expansion ratio
η	= isentropic efficiency
τ	= torque (N m)
ω	= angular velocity (rad/s)

Subscripts

amb	= ambient
corr	= corrected
crit	= critical
in	= inlet
nom	= nominal
ref	= reference
st	= standard
surge	= surge
0	= total

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