Fermin 'Vic' Viteri

Clean Energy Systems, Inc.
11330 Sunco Drive, Suite A
Rancho Cordova, CA 95742

Bus. (916) 379-9143
Fax (916) 379-9146
fviteri@cleanenergysystems.com

## 1.3 TURBINE - BASED ZERO EMISSIONS PLANTS

1.3.1 Oxy-Fuel

1.3.1.2 Clean Energy Systems

1.3.1.2.1 Introduction

Clean Energy Systems, Inc. (CES) of Sacramento, CA and DOE's National Energy Technology Laboratories (NETL) have developed and demonstrated unique technologies that will enable construction and operation of efficient zero-emission power plants (ZEPP). The enabling technologies are an oxy-fueled combustor developed under a DOE/NETL Vision 21 program, an oxy-fueled reheater (RH) designed by NETL and tested at a NASA test facility, and oxy-syngas combustor being developed under a DOE/NETL program. The CES process involves burning high purity oxygen with a hydrocarbon fuel, e.g., natural gas (NG), coal syngas, gasified biomass, etc., in the presence of water to generate a high pressure, high temperature gas comprising approximately 90 % steam, 10 % carbon dioxide (CO_2), and a small amount of oxygen (O_2). This gas is used to drive steam turbo-generators. CES power plants use cryogenic air separation units (ASU) to provide oxygen. These ASU plants can be made more efficient by the use of axial-flow-type compressors, typical of those found in gas turbines. This section discusses the integration of oxy-fueled combustors and reheaters with steam and gas turbines, gas turbine air compressors, a steam/CO_2 condenser, and CO_2 compressors/intercoolers. The resulting integrated ZEPPs produces power; generate high quality water, and conditioned CO_2, ready for beneficial uses or sequestration. Key issues include ASU/gas turbine compressor flow matching, gas turbine blade cooling using steam rather than air, turbine material compatibility, and gas turbine temperature differences between steam/CO_2 and air combustion. Various CES-type ZEPP concepts are illustrated and their performance characteristics defined for a range of operating conditions that are achievable with present day steam and gas turbines.
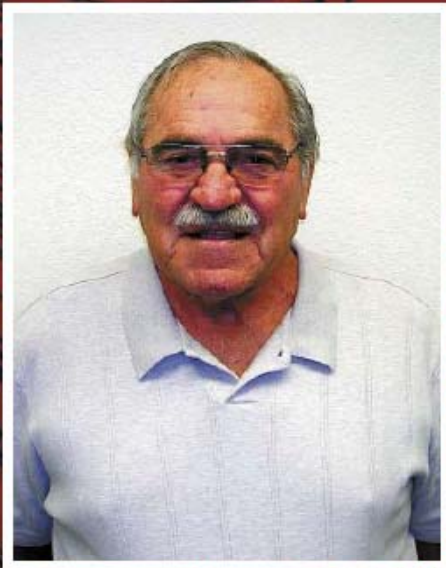
1.3.1.2.2 The CES Zero Emissions Power Plant

Recent test programs by CES and DOE/NETL have successfully demonstrated the enabling combustion technologies of an oxy-fueled (NG) combustor 816 ºC, 104 bar (1500 ºF, 1500 psia)[1,2,3,4,5,6] and an oxy-fueled (NG) reheater 1200 ºC, 10 bar (2200 ºF, 145 psia)[7,8,9]. A simplified schematic diagram of a CES plant that incorporates these new components is shown in Figure 1.

The following discussion explores the integration of oxy-fueled combustion technologies with gas turbine and steam turbine technologies.

CES power plants use the basic Rankine power cycle and consist of four basic systems as described by Martinez-Frias, et al.[10]. However, there are other alternative cycles that use CES technology and these are discussed in Section 1.3.1.2.8.
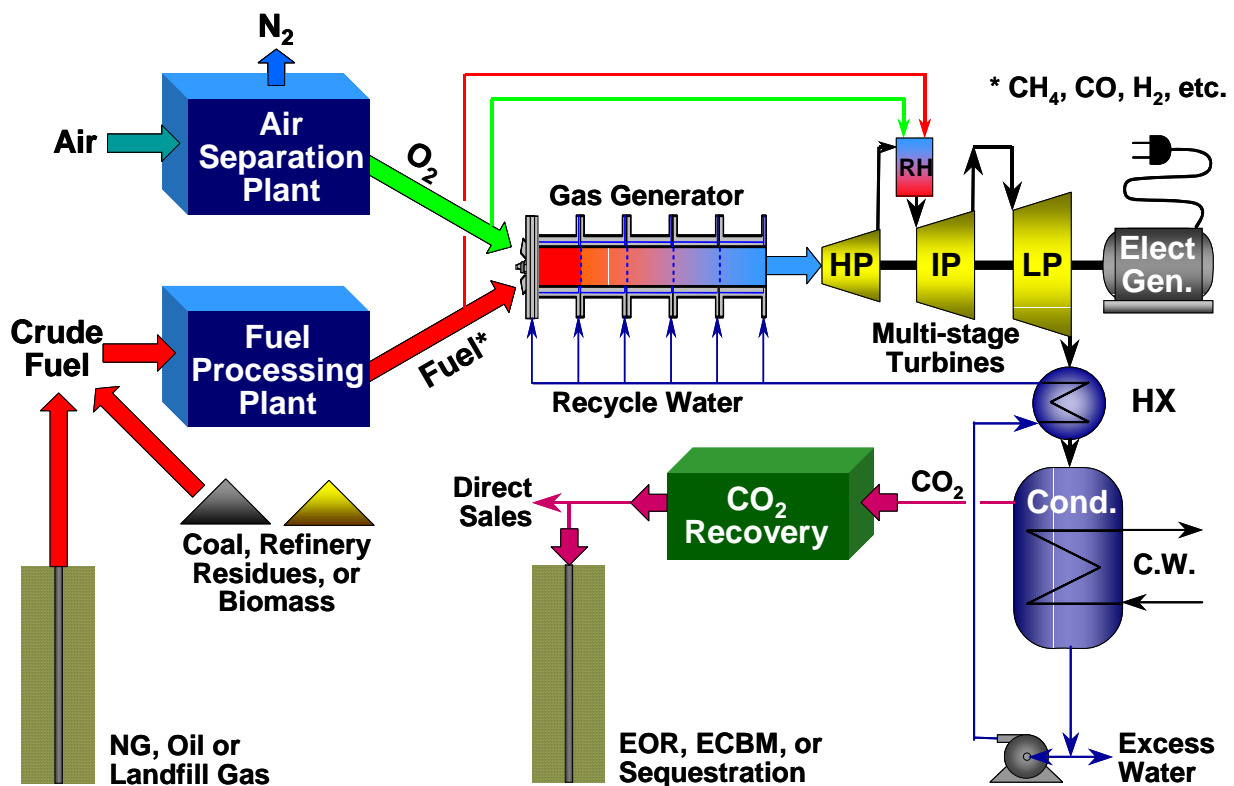

Fig.1. The Basic CES Zero-Emissions Power Plant

1. Fuel Processing and Gas Compression: Gaseous fuel, derived from virtually any organic source, e.g., natural gas, gasified coal, biomass or refinery residue, is processed by cleansing any undesirable substances (e.g. sulfur, nitrogen, etc.) and compressed to the combustor injection pressures.

2. Air Separation and Oxygen Compression: Nearly pure oxygen is derived from large cryogenic air separation unit (ASU) and compressed to the combustor injection pressure.

A NG-fired CES plant typically comprises four subsystems:
1. Fuel processing and gas compression
2. Air separation and oxygen compression
3. Power generation (power-train)
4. Carbon dioxide separation and conditioning

3. Power Generation (Power-Train): The power generation system includes three turbines in series driven by high temperature gases consisting of approximately 90 %v steam, 10 %v CO_2 and a small amount of oxygen. The excess oxygen suppresses CO_2 dissociation and drives the combustion reactions to completion. High temperature gases are generated by an oxy-combustor at approximately 540 °C.– 760 ºC (1000 °F – 1400 °F) and with one or two reheaters operating at 1240 °C – 1760 ºC (2240 °F - 3200 °F). These gases drive multi-stage turbines. The turbines, in turn, drive an electric generator through a common or multiple shaft system, depending upon the selected plant configuration.

4. Carbon Dioxide Separation and Conditioning: This subsystem cools the turbine exhaust in a condenser at atmospheric or sub-atmospheric pressures to condense the steam and separate the CO_2. Most of the condensed water is preheated in a feed water heater, located at the turbine exhaust, to recover any residual heat before recirculation back to the combustor. The separated humid CO_2 exiting the condenser is extracted and compressed to approximately 145 bar (2100 psia) with multi-stage compressors. Intercoolers between stages remove the remaining water vapor and condition the CO_2 for ultimate sequestration.

A coal-fired ZEPP is similar to a NG-fired plant except it includes an oxygen-blown coal gasification and syngas cleanup and compression system in place of the NG processing and compression system. Such a power plant is described and shown diagrammatically by Martinez-Frias, et. al.[11].

1.3.1.2.3 ASU and Turbine Compressor Matching

CES and integrated coal gasification combined cycle (IGCC) power plants require large dedicated air separation units (ASU) to provide the oxygen required for combustion of the fuel or gasification of the coal. The capacities of ASUs required to support three types of 400 MW_e power plants are in the range of 2600 - 6700 metric tons O_2/day as shown in Table I.

Table I. Typical ASU Plant Sizes Required for Three Types of 400 MW_e Power Plants

Plant Type	ASU Size, metric tons O_2/day
NG-fired CES ZEPP	6500
Coal Syngas-fired CES ZEPP	6700
IGCC with O_2-blown coal gasifier	2600

Cryogenic air separation is currently the most efficient and cost effective technology for producing large quantities of oxygen[12]. NG and coal syngas fired CES power plants larger than about 200 MW_e require ASUs with air compressor capacities exceeding that of existing conventional industrial centrifugal and axial-centrifugal compressor equipment. Only large gas turbine compressors can provide the necessary air from a single unit. The capacity of present-day conventional ASUs is limited to about 3600 metric tons O_2/day[13] and closely matches the

compressed air supply capabilities of a 6F class gas turbine. The very large 9F class gas turbines can meet the compressed air requirements of an ASU that produces about 12,000 metric O_2/day[13] and could support a 700 MW_e CES power plant. Also, the cost of the air compression system increases from 25% of the total ASU plant cost at 1800 metric tons O_2/day to 35% at 3200 metric tons O_2/day [12]. Therefore, the air compression system is a key cost element of an ASU. Because large ASU's require large air compressors available in gas turbines, the economics of large power plants favor the integration of relatively lower cost gas turbine/air compressor unit into CES/ASU systems. Integration eliminates the need for large gearboxes, electric motors, or steam drive motors, and other associated equipment, including the electric motor starting apparatus which can approach the cost and size of the motor itself[14]. Also, integration simplifies the controls and the control system for improved plant reliability.

Gas turbines are available in a wide range of capacities[15]. Typical gas turbines ranging in compressed airflow rates of 40 - 600 kg/sec (88 -1320 lb/sec) are listed in Table II along with the approximate sizes of equivalent ASU and CES ZEPP power plants such turbines could support.

Table II. Typical Gas Turbine Compressor Capacities versus Equivalent ASU and ZEPP Sizes

Gas Turbine Mfg	Comp. Cap., kg air/sec	Equiv. ASU & ZEPP Sizes	
		ASU MetricTons/Day	Plant Size, MW_e at 42% Eff
Alstom/Cyclone	40	800	50
GE/LM1600	50	1000	60
GE/LM2500	86	1,800	100
P&W/FT-8	86	1,800	100
GE/LM6000	130	2,600	150
RR/Trent	175	3,500	200
SW/V64.3A	200	4,000	230
GE/7EA	295	6,000	350
GE/7FA	455	9,100	530
GE&SW/H, Al.GT26	600	12,000	700

From Table II it can be seen that current commercial gas turbine compressors can potentially accommodate integration of ASUs and CES power plants with capacities ranging from approximately 790-12,000 metric tons O_2/day and about 50 to 700 MW_e, respectively

1.3.1.2.4 Effect of Drive Gases on Gas Turbine Operating Parameters

Analysis was performed to define the comparative nominal operating parameters of existing aero-derivative and industrial gas turbines driven by normal air-breathing combustion gases and by CES combustion gases. The assessment involved one-dimensional analyses and the assumptions listed below. A more thorough assessment was conducted by Fern Engineering[16] , however, further work should be conducted by turbine manufacturing teams that possess precise turbine design information (e.g., exact blade angles, stresses, materials and design limits). Turbine manufacturers are most qualified to accurately determine the design and off-design performance when substituting CES drive gases for air-breathing combustion drive gases.

Assumptions:
- Inlet pressure unchanged
- Pressure ratio unchanged
- Design based on 50% reaction
- Blade cooling flow rate equal to or less than design
- Inlet temperature equal to or less than design value
- Turbine efficiency unchanged and approximately equal to 90%
- Blade cooling effectiveness equal to 100% (transpiration cooling)
- Blade coolant temperature equal to compressor discharge temperature for air cooling ~510 ℃ (~ 950 ℉) aero-derivative and ~230 ℃ (~ 450 ℉) industrial) and 230-270 ℃ (450-500 ℉) slightly superheated for steam cooling.

Criteria for validation:
- Aerodynamic similarity is nearly maintained (blade angles and Mach No.)
- Operating speed is within the recommended range
- Heat transfer and blade temperatures are equal to or less than estimated design limits
- Blade root stresses remain approximately the same [proportional to $(speed)^2$ and torque]

Because both aero-derivative (high-pressure-ratio) and industrial (low-pressure-ratio) turbines are widely used, the following two designs were considered representative of units for medium size 150 MW$_e$ and large 700 MWe ZEPP plants.

Baseline Aero-derivative Turbine (150 MW$_e$ ZEPP):

• Two-stage design	• Inlet temperature = 1245 ℃(2273 ℉)
• Inlet pressure = 29.93 bar (434 psia)	• Exit pressure = 7.03 bar (102 psia)
• Design Speed = 9586 rpm	• Speed range = 9500 to 10,800 rpm
• Design flow rate = 126.0 kg/sec	• Turbine mean diameter 77.47 cm
• (277 lb/sec)	• (30.5 inches)
• Compressor pressure ratio = 29.4	• Cooling air temperature 504 ℃(940 ℉)
• Cooling air flow rate 9.31 % of main gas flow	

Baseline Industrial Gas Turbine (700 MW$_e$ ZEPP):

• Four stage design	• Inlet Temperature = 1427 ℃ (2600 ℉)
• Inlet Pressure = 19.31 bar (280 psia)	• Exit Pressure = 1.10 bar (16 psia)
• Compressor pressure ratio = 19.1	• Design flow rate = 583 kg/sec (1282 lb/sec)
• Design speed = 3600 rpm	• Turbine mean dia. = 211 cm (83 inches)
• Cooling air 232 ℃ (450 ℉) flow rate 4.8 % of turbine design flow rate.	

The resulting analyses, comparing nominal operating parameters using air-breathing combustion gases or CES gases at baseline and at a lower temperature, are shown in Table III for both an

aero-derivative and an industrial gas turbine. Various typical operating parameters for the first stage of the turbines and the exit temperature from the last stage are illustrated in the table.

The operating parameters for the typical aero-derivative turbine given in Table III shows that parameter matching with the different drive gases is favored by increasing turbine speed by about 12-14 % when using CES drive gases. This speed increase permits close fluid flow angle matching and, when the gas inlet temperature is also decreased slightly, fluid flow angles coincide with the baseline case and last stage exit gas temperature closely approximates the baseline case. It can also be seen that replacing air-breathing combustion gases with CES gases provide 11-16 % higher 1st-stage power output and lowers turbine coolant flow rate by 70-75%. This reduction in flow rate is due to changing the blade coolant from air to steam.

Table III. Operating Parameters of a Typical Aero-derivative Turbine and a Typical
Industrial Gas Turbine with Air-Breathing and CES Drive Gases

Parameter	Aero-derivative Turbine			Industrial Gas Turbine		
	Air-Breath	CES Gases		Air-Breath	CES Gases	
Turbine Stage		1st	1st	1st	1st	1st
Inlet Gas Temp., ºC (ºF)	**1245 (2273)**	**1245 (2273)**	**1121 (2050)**	**1427 (2600)**	**1427 (2600)**	**1427 (2100)**
Inlet Pressure, bar (psia)	29.93 (434)	29.93 (434)	29.93 (434)	19.31 (280)	19.31 (280)	19.31 (280)
Exit Pressure, bar (psia)	14.55 (211)	14.55 (211)	14.55 (211)	9.24 (134)	9.24 (134)	9.24 (134)
Exit Gas Temp., ºC (°F)	1026 (1878)	1084 (1983)	973(1784)	1176(2149)	1248(2279)	999(1831)
Weight Flow, kg/sec (lb/sec)	126.0 (277.1)	104.7 (230.3)	109.2 (240.3)	582.7 (1282)	485.0 (1067)	531.4 (1169)
Speed, rpm	9,586	10,717	10,858	3600	3600	3600
Power/Stage, MW	**27.64**	**32.09**	**30.75**	**145.8**	**167.6**	**153.7**
Coolant Temp., ºC (ºF)	504 (940)	260 (500)	232 (450)	232 (450)	232 (450)	232 (450)
Coolant C_p, kJ/kg-ºC (Btu/lb-°F)	1.033 (0.247)	2.395 (0.551)	2.305 (0.551)	1.033 (0.247)	2.305 (0.551)	2.305 (0.551)
Coolant Flow, % gas flow	9.31	2.78	2.23	4.80	3.90	1.90
Blade Temp., ºC (ºF)	**816 (1500)**	**816 (1500)**	**816 (1500)**	**816 (1500)**	**816 (1500)**	**816 (1500)**
Nozzle Exit Vel.,m/sec (ft/sec)	699 (2293)	824 (2703)	792 2597)	744 (2442)	907 (2975)	802 (2632)
Rotor Exit Vel., m/sec (ft/sec)	435 (1423)	455 (1494)	491 (1611)	460 (1510)	597 (1939)	498 (1633)
Mean Blade Speed, m/se (ft/sec)	389 (1276)	435 (1426)	440 (1445)	396 (1300)	396 (1300)	396 (1300)
Main Gas C_p, kJ/kg-ºC (Btu/lb-°F)	1.230 (0.294)	2.385 (0.570)	2.343 (0.560)	1.230 (0.294)	2.385 (0.570)	2.343 (0.560)
Specific Heat Ratio	1.32	1.21	1.21	1.32	1.20	1.20
Nozzle Incid. Angle, Deg.	**0**	**-4.7**	**0**	**0**	**-9.0**	**-3.8**
Rotor Incid. Angle, Deg.	**0**	**-2.3**	**0**	**0**	**-6.9**	**-2.8**
Last Stage Exit Temp., ºC (°F)	833 (1532)	937 (1719)	834 (1534)	637 (1178)	829 (1524)	649 (1200)

The operating parameters for the typical industrial turbine given in Table III shows that when turbine speed is set by the generator speed, the change in fluid angles is somewhat larger when switching to CES drive gases, but is considered to be within the range of capabilities of a high-efficiency, reaction-type turbine. Alternatively, reducing the turbine inlet temperature at constant speed provides closer matching of fluid flow angles at the nozzle and rotor exits and last-stage gas exit temperature. At similar turbine inlet temperatures and speeds, replacing air-breathing combustion gases with CES gases provide 15 % higher 1st-stage power output and lowers blade coolant flow about 20% while maintaining constant turbine blade operating temperature. When the turbine inlet temperature is reduced 260 °C (500 °F) and speed is maintained constant, 1st-stage power output is still increased about 5 % over the baseline and turbine blade coolant flow is reduced by 60%.

A subsequent study by Fern Engineering[16] on a similar, but slightly smaller aeroderivative gas turbine, resulted in the following conclusions:
At the design "firing temperature" of 1280 ° (2336 ºF), the CES cycle yields:
- *Slightly higher power output (~6%)*
- *A lower overall turbine pressure ratio (17.6 vs. 20.3)*
- *Much lower mass flow of working fluid*
- *Significantly cooler HP turbine nozzle metal temperature*
- *Slightly lower power turbine inlet pressure*
- *Slightly hotter power turbine inlet temperature, but a cooler power turbine nozzle metal temperature due to the use of steam cooling*
- *Smaller turbine jet velocity ratios => slightly lower turbine efficiencies*
- *Much hotter power turbine exhaust temp*

1.3.1.2.5 Effect of Coolant on Gas Turbine Blade Temperatures

Heat transfer study results based on models similar to those of DePaepe and Dick[17,18] are presented in Table IV and Figure 2. Table IV shows the 1st-stage blade baseline temperature of 816 °C (1500 ºF) in a high-pressure-ratio aero-derivative type turbine can be reduced by 246 °C (475 ºF)[(1)] using steam at 260 °C (500 ºF) for cooling rather than air at 504 °C (940 ºF) at constant volumetric flow rates. Alternatively, the inlet temperature of CES gases to the turbine could be increased approximately 166 °C (330 ºF) when using 260 °C (500 ºF) steam at constant volumetric coolant flow rates and still decrease turbine blade temperature nearly 204 °C (400 ºF) compared to baseline air-breathing case. The lower blade operating temperature, using steam as

(1) The study by Reference[16] calculated the temperature reduction as 143 °C (290 ºF) and felt the 246 °C (475 ºF) prediction by CES was too optimistic. However, CES assumed a transpiration cooling effectiveness of 1.0 while Reference[16] assumed a factor 0.58. Measured blade surface temperatures by reference[19] for 1/2 scale model tests indicated the cooling effectiveness ranged from 0 .60 to 0.80 , for a steam cooled stator and rotor blade operating with 1700 °C (3092 ºF) hot gases(steam) at (355 psia). The measured blade metal temperatures were in the range of 595 to 816 °C (1100 to 1500 ° F).

Table IV. Comparison of Turbine Blade Temperature at Constant Coolant Flow Rates

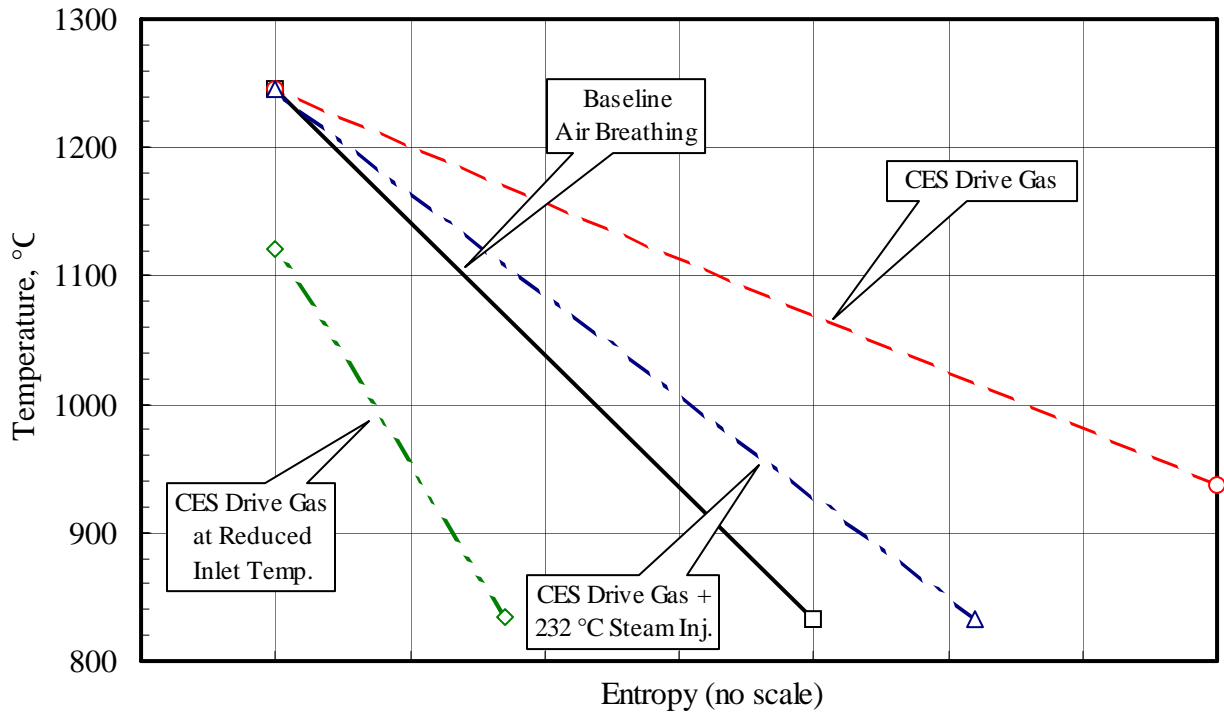| Type of Turbine and Drive Gas	Gas Temp., T_g, °C (°F)	Blade Temp., T_b, °C (°F)	Coolant	Coolant Temp., T_c, °C (°F)	Coolant Flow Rate, m^3/min (ft^3/min)
Aeroder.-Air	1245 (2273)	816,(1500)	Air	504 (940)	0.872 (30.8)
Aeroder.-CES	1245 (2273)	552 (1025)	Steam	260 (500)	0.872 (30.8)
Aeroder.-CES	1427 (2600)	599 (1110)	Steam	260 (500)	0.872 (30.8)

Fig. 2. Gas Temperature Across Two Stages of a Typical Aero-Derivative Turbine

coolant, could result in increased blade stress safety margins and increased life or permit higher gas inlet temperatures, the use of simpler coolant passage designs, or possibly lower-cost blades.

The Japanese[20] investigated various cooling methods for an intermediate pressure steam turbine operating with1700 °C (3100 °F) steam while using open loop and closed loop cooling circuits with water and 300 °C (572 °F) steam. These studies indicated the following cooling losses for a 500MW steam plant: (1) closed-circuit water cooling of the combustor, nozzle and stators vanes, and steam cooling of the rotor blades (CCWCN-SCR) has the lowest cycle loss of 10MW; (2) closed-circuit steam cooling of the nozzles and rotor blades (CCSCN-R) has a 21 MW loss; and (3) open-circuit steam cooling for nozzle and rotor blades (OCSCN-R) has a 52 MW loss.

1.3.1.2.6 Gas Turbine Operation with CES Gases versus Air-Breathing Gases

The temperature drops across turbine stages change when the nature of the drive gas changes from air-breathing combustion gases to the gases produced by CES gas generators or reheaters because the specific heat ratios of these gases varies from 1.32 to 1.20, Table III. This effect is shown in Figure 2. Where temperature drops are compared for a typical two-stage aero-derivative turbine. Comparing the baseline air-cooled air-breathing case with the CES drive gas case, each with an assumed turbine inlet temperature of 1245 °C (2273 °F), the turbine exit gas temperature for the CES drive gas is higher than for the baseline case by 88 °C (158 °F). However, this difference is negated or markedly reduced when open-loop steam cooling is considered. Assuming steam cooling with 232 °C (418 °F) steam, the temperature drop can be made to coincide with the baseline air-cooled air-breathing case by additional steam injection. Alternatively, the turbine inlet temperature of the CES drive gas can be reduced and made to coincide with the baseline air-cooled air-breathing case.

From the preceding examples it can be seen that stage-wise temperature drops or exit temperatures of air-cooled air-breathing turbines can be replicated when the turbines are switched to CES gases with open-loop steam cooling operating practice requires the use of more expensive materials and/or fabrication techniques, the technology base is well established. The major hindrance to increasing steam turbine operating temperatures has been with boiler limitations rather than with turbine limitations. The CES gas generator in CES ZEPP's removes the boiler and its temperature constraints. Intermediate pressure turbines power plants would not operate at temperatures beyond current commercial gas turbine practice and may operate at even lower blade temperatures because very effective open-loop steam cooling becomes practical.

1.3.1.2.7 Turbine Materials Issues

The turbine materials issues addressed in this section concern steam and gas turbine materials operating in high temperature, high-pressure steam environments. The issues primarily involve: 1) matching materials mechanical properties with turbine operating temperatures and stresses and (2) defining materials that are compatible with CES gases (~90% steam, ~10% CO_2, and a slight amount of oxygen) at high temperatures and pressures and with weak carbonic acid condensate.

Typical materials for various steam and gas turbine components are listed in Table V. Increasing steam turbine operating temperatures beyond ~ 540 to 565 °C (~1000 to1050 °F) requires the use of materials such as high chromium-ferritic steels and austenitic stainless steels[21]. The temperature limits for this type of materials is near 649 °C (1200 °F) but may extend to slightly higher values. For even higher metal operating temperatures, i.e., 816 °C (1500 °F) and above, high nickel and cobalt alloys become necessary and turbine blades may require even more exotic single-crystal materials.

Although increasing steam turbine operating temperatures above current operating practice requires the use of more expensive materials and/or fabrication techniques, the technology base is well established. The major hindrance to increasing steam turbine operating temperatures has been with boiler limitations rather than with turbine limitations. The CES gas generator in CES

Table V. Typical Materials of Construction for Steam and Gas Turbine Components

Component	Typical Alloys		
	Eddystone Steam Turbine 593 - 649°C (1100-1200 °F)[22]	Solar 816°C, 103 bar (1500 °F, 1500 psia) Steam Turbine[23]	Gas Turbines[24,25,26,27]
Casings	St. 316 (inner), 2 ¼ % Cr-steel (outer)	Inconel 939 (inner), 2 ¼ % Cr-steel (outer)	Inconel 718, Rene'41, Hastelloy X, Haynes 188
Comb. Liner	N.A.	N.A	Hastelloy X, Haynes 188
Transition Duct	St. 316	Inconel 617	Inconel 617, Inconel 939
Discs	Discalloy	Inconel 718	Inconel 718, Inconel 738, Waspaloy, Udimet 700
Vanes and Blades	K42B, St. 422 (blades) St. 316 (nozzles)	Inconel 718 (blades), Inconel 939 (nozzles)	Inconel 713C, 718, & 738, Rene'80, Udimet 500 & 700, CMSX-4, FSX-414, M-252, Multimet (N-155)

ZEPP's removes the boiler and its temperature constraints. Intermediate pressure turbines in CES power plants would not operate at temperatures beyond current commercial gas turbine practice and may operate at even lower blade temperatures because very effective open-loop steam cooling becomes practical.

The materials used in both current and advanced steam turbines have demonstrated good compatibility with pure steam but the effects of CES gases, which also contain a minor amount of CO_2 and a small amount of oxygen, are not well established. Preliminary compatibility studies with simulated CES gases and typical turbine materials are underway. No significant problems have been encountered in the absence of an aqueous liquid phase but more comprehensive work is required, including the effects of carbonic acid formation in regions subject to condensation.

The compatibility of a number of nickel and/or cobalt based alloys with steam at 816 °C, 103 bar (1500 °F and 1500 psia) has been studied by Solar Turbines, Inc. and compared with the behavior of the same alloys in air at 816 °C, 1.01 bar (1500 °F, 14.7 psia). The results of 1000-hour tests are summarized in Table VI.

The data in Table VI show that the high temperature, high-pressure steam attacks the alloys to a relatively minor extent and with few exceptions similar to the attack of air at a lower pressure. The tests in the steam environment were extended to 4000-hour exposures to provide a firmer basis for selecting materials for a steam turbine (see Table V) that operated successfully at 816 °C, 103 bar (1500 °F, 1500 psia). The results of these latter tests are summarized in Table VII.

The data in Tables VI and VII, along with the experience gained with gas turbines operating in an oxidizing environment at moderate pressures and very high temperatures, suggest that CES

Table VI. Metallographic Measurements on Alloys Exposed for 1000 Hours to Steam at 816 °C , 103 bar (1500 °F, 1500 psia) Compared to Air at 816 °C, 1.01 bar (1500 °F, 14.7 psia) [23]

Alloy	Outer Scale Thickness, microns		Depth of Internal Oxidation, microns		Depth of Alloy Depletion, microns	
	Steam	Air	Steam	Air	Steam	Air
Inconel 718	3.8-7.6	7.6-17.8	10.2-20.3	5.1-15.2	10.2-22.9	0.5-20.3
Inconel 625	2.5-10.2	2.5-6.4	0.0-20.3	0.0–10.2	30.5-40.6	10.2 15.2
Inconel 800	4.1-6.1	5.1-10.2	10.2-20.3	3.8-25.4	0.0	20.3-50.8
Hastelloy X	0.0-2.5	6.4-22.9	0.0-16,5	0.0- 5.1	0.0	6.1-20.3
Hastelloy S	2.5-7.6	2.5-5.1	0.0-6.4	3.8-12.7	0.0	5.1-10.2
Waspaloy	2.5-5.1	5.1-12.7	7.6-20.3	12.7-33.0	15.2-30.5	12.7-25.4

Table VII. Metallographic Measurements on Alloys Exposed for 4000 Hours to Steam At 816 °C, 103 bar (1500 °F, 1500 psia)[28]

Alloy	Outer Scale Thickness, microns	Depth of Internal Oxidation, microns	Depth of Alloy Depletion, microns
Inconel 718	Nil	5.1-55.9	15.7-50.8
Inconel 713	Nil	12.7-24.4	25.4-38.1
Inconel 625	Nil	6.4-12.7	19.1-20.3
Inconel 617	0.0-2.5	12.7-30.5	30.5-40.6
Incoloy 800	1.3-3.8	0.0-6.4	2.5-3.8
Hastelloy X	1.0-2.0	12.7-16.5	6.1-10.2
Waspaloy	9.4-15.7	31.8-44.5	63.5-81.3

gases will not pose major compatibility problems when used to drive gas turbines. This outlook remains, however, to be demonstrated.

1.3.1.2.8 Integrated Plant Concepts

Integration of CES' technology with ASU's, gas turbines, steam turbines and CO_2 conditioning equipment to build environmentally friendly zero emissions power plants appears feasible and highly beneficial[29]. Since CES ZEPP plants involve a number of subsystems, there are many possible concepts. The following two configurations are representative of only a few of the many possible combinations.

ZEPP #1, shown in Figure 3, consists of a high pressure 80-100 bar (1200-1500 psia) oxy-combustor feeding a high pressure steam/CO_2 turbine (HPT) at 600-760 °C (1100-1400 ºF); and an intermediate pressure reheat combustor at 30-40 bar (430-600 psia) feeding an intermediate pressure steam/CO_2 turbine (IPT) at 1240-1760 °C (2240-3200 ºF). The IPT exhausts to sub-atmospheric pressures in the range of 0.15-0.4 bar (2.2-5.8 psia). Residual heat in the LPT exhaust is used to raise cooling steam for the IPT, and to preheat combustor feedwater.
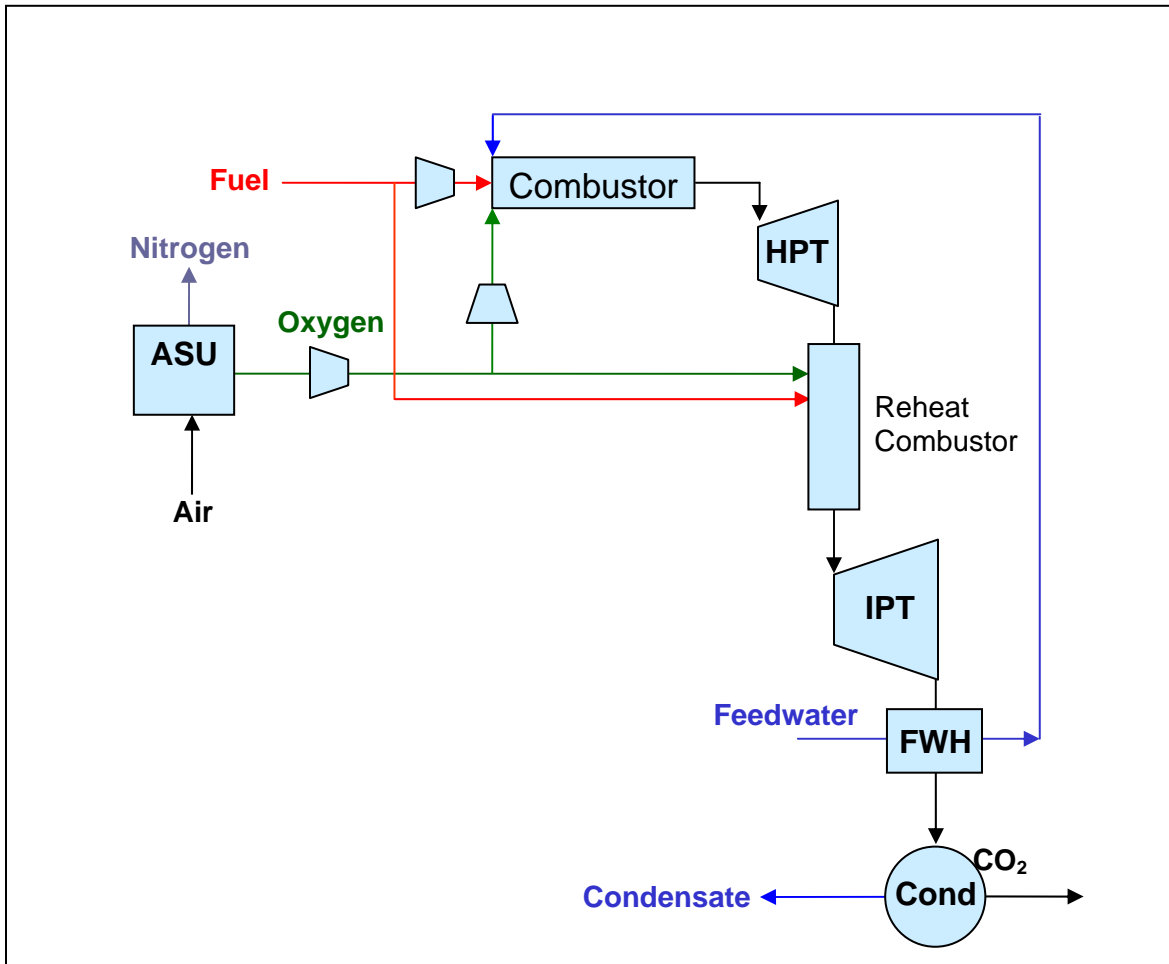
Fig. 3. Process Flow Diagram of ZEPP #1

ZEPP #2, shown in Figure 4, consists of an intermediate pressure 30-40 bar (430-600 psia) oxy-combustor feeding an intermediate pressure steam/CO_2 turbine (IPT) at 1240-1760 °C (2240-3200 ºF) that exhausts to approximately 1 atm. (14.7 psia). The IPT exhaust stream enters a HRSG (heat recovery steam generator) that raises high-pressure steam for a back-pressure HP steam turbine (HPT). Most of the HPT steam exhaust is delivered to the combustor as diluent, and some is used as cooling steam for the IPT. If desired, the HRSG may be fired with an oxy-fuel burner to generate additional HP steam. Finally, some of the latent heat in the IPT exhaust is recovered by raising sub-atmospheric steam for a low-pressure steam turbine (LPT).

1.3.1.2.9 Performance

Earlier cycle analyses were made by CES and other organizations, using a variety of modeling tools. These include: (1) CES's in-house code; (2) the commercially available AspenPlus® soft
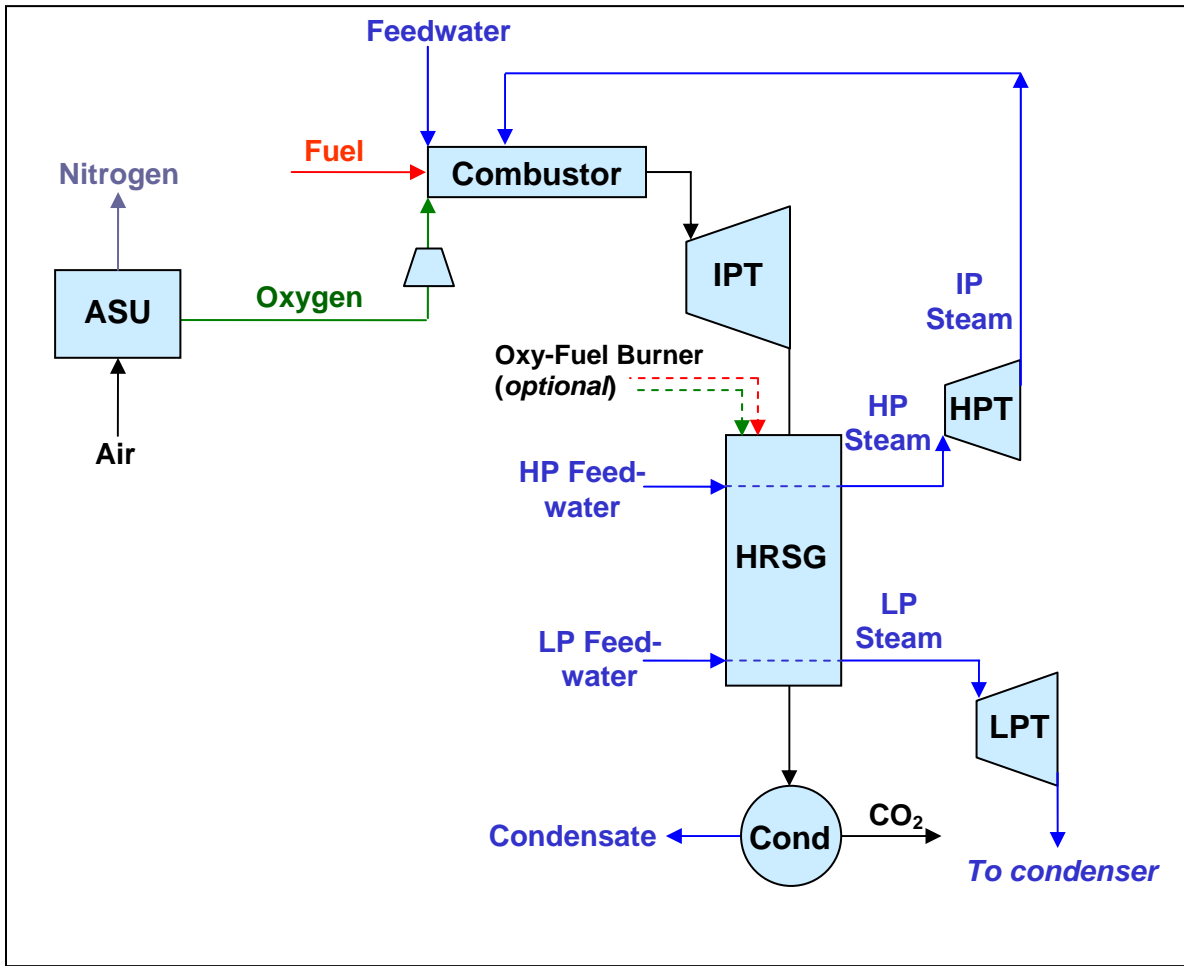
Fig. 4. Process Flow Diagram of ZEPP #2

ware; (3) the Lawrence Livermore National Laboratory's (LLNL) program developed by Martinez-Frias[10,11] using Engineering Equation Solver (EES) software; and (4) Gates used by Fern Engineering[16]. All four codes were checked against each other for verification when applied to a CES power plant test case and all showed close agreement, provided the same process assumptions were made.

Recent analyses have been made by CES personnel for both natural gas and coal-based plants, using the AspenPlus® software combined with updated process information for the ASU, steam/CO_2 turbines, steam turbines, CO_2 compression system, and gasifier (for coal-based systems). These analyses revealed that the efficiencies of the ZEPP #1 and ZEPP #2 cycles are very similar, provided the ZEPP #2 cycles include supplemental HRSG firing to boost the HP steam flowrate to the corresponding HP steam/CO_2 flowrates in the ZEPP #1 cycles.

Table VIII lists the key assumptions made in these analyses.

Table VIII. Key Assumptions for Natural Gas and Coal Cycle Analyses

ASU auxiliary load	0.20 kWh/kg O_2
IP turbine isentropic efficiency	91%
Temperature of IPT cooling steam	380 ºC
Steam turbine isentropic efficiency	88%
Compressor isentropic efficiency	82%
Compressor per-stage pressure ratio	2.5
Compressor inter-cooler temperature	31 ºC
Condenser temperature	31 ºC
HRSG pinch-point	20K
Turbine shaft losses	1%
Turbine generator losses	1%
Compressor motor losses	5%

Natural Gas Systems

With this set of assumptions, CES personnel calculated the expected efficiencies of various near- and long-term natural gas-fired CES plants. The key variables were (1) the HP and IP turbine inlet temperatures, and (2) the quantity of steam required for IP turbine cooling. Table IX lists the expected LHV cycle efficiencies for HPT/IPT inlet temperatures of 620/1240ºC, 620/1450ºC, and 760/1760ºC; and open-loop cooling steam flows of 10, 20 and 30%.

Table IX. LHV Cycle Efficiencies for CES Natural Gas-Fired Plants

HPT/IPT Inlet Temperatures	IPT Cooling Steam Flow (% of turbine inlet flow)			
	0%	**10%**	**20%**	**30%**
620ºC/1240ºC	43.0%	41.7%	40.8%	40.0%
620ºC/1450ºC	-	46.0%	44.8%	43.6%
760ºC/1760ºC	-	51.0%	49.3%	48.0%

The various HPT/IPT inlet temperatures, which were provided by Siemens, represent sequential advancements that may be made through 2015. Since these cycle studies did not include a detailed analysis of the IPT cooling requirements for each case, a wide range of IPT cooling steam flowrates were considered. Also, all IPT cooling was assumed to be by open-loop steam cooling, where the steam is injected into the drive gas.

As shown in the table, the flowrate of cooling steam has a significant impact on the cycle efficiency since it lowers the effective inlet temperature of the drive gas, particularly at the higher turbine inlet temperatures. This highlights the importance of optimizing the turbine cooling methodology to minimize its impact on the cycle performance.

Coal-Based Systems

Under award DE-FC26-05NT42645 ("Coal-Based Oxy-Fuel System Evaluation and Combustor Development"), CES personnel performed detailed cycle analyses on coal-based plants where an Illinois #6 coal is gasified and the clean syngas used as fuel in the CES oxy-combustors. These are referred to as IGCES (integrated gasification CES) plants. The studies incorporated input and interface information from Siemens (turbines), Air Products (ASU), Future-Energy (gasifier), and MAN-Turbo (compressors). Near-term (2010) and long-term (2015) cases were considered, each of which had its own set of assumptions. These assumptions, also provided by Siemens, appear in Table X.

Table X. Key Turbine Assumptions for Near- and Long-Term Coal-Based IGCES Plants

Parameter	Near-term (2010)	Long-term (2015)
HPT inlet temperature	620ºC	760ºC
IPT inlet temperature	1450ºC	1760ºC
IPT cooling steam flowrate	25%	15%
IPT exhaust pressure	1.0 bar	0.15 bar

This analysis was performed in more detail than the previous natural gas analyses as the study included an extensive information exchange with key equipment suppliers, particularly Siemens. For this reason, the analysis focused on a fewer number of cases than the natural gas study.

The cycle analyses revealed that the performance of an IGCES plant is sensitive to the gasifier configuration, particularly the mode of heat recovery from the hot syngas stream.

Most commercial gasifiers provide the option of cooling the hot syngas in a syngas cooler which raises saturated HP steam at pressures in the range of 100 bar (1,500 psia). In the ZEPP #1 cycle, the steam from the syngas cooler can be injected into the HP oxy-combustor, heated and expanded through the HPT, reheated in the reheat combustor, and expanded through the IPT. This represents efficient use of the steam.

An alternate approach is to quench the hot syngas via water injection to produce a cooler, saturated syngas stream. Some of the latent heat in this stream may be recovered by raising steam at a lower pressure than the partial pressure of moisture in the syngas, and expanding this steam through an LP turbine. Although this is a simpler technique with lower capital costs, the energy losses associated with syngas quenching have a negative impact on the overall cycle performance.

Table XI lists the expected HHV cycle efficiencies for the near-term and long-term cases, for plants with either syngas cooler or syngas quench systems.

Table XI. HHV cycle efficiencies for CES coal-based plants

Availability	Syngas Heat Recovery Method	
	Quench System	Syngas Cooler
Near-term cycle (2010)	27.2%	30.0%
Long-term cycle (2015)	34.2%	37.0%

As shown in the table, technical advancements that may be made by 2015 will have a significant impact on the cycle performance. Also, a syngas cooler is preferable to a syngas quench system to maximize cycle efficiency.

1.3.1.2.10 Conclusions

The use of modified IP (gas) turbines along with steam turbines in CES power plants enables high-efficiency, near-zero power generation. In some applications, the gas turbine compressor could be used as the air supply source for an ASU. This integration of systems eliminates the need for large electric drive motors, gearboxes, etc. to drive the compressors and, therefore, could significantly reduces plant capital cost and plant operational and maintenance costs. The high capacities of gas turbine compressors also permits construction of larger single train ASU's that could support integrated CES-type ZEPP's.

Gas turbines operating with CES drive gases have lower temperature drops per stage and this results in higher temperatures for the later stages. To alleviate these higher temperatures, additional steam at 204-260 °C (400–500 ºF) could be injected at each stage to reduce the temperature to that compatible with the stage. Alternatively as a temporary solution, turbine inlet temperature could be reduced 93 °C (167 ºF) for aero-derivative and 260 °C (470 ºF) for industrial turbines with consequent small reductions in plant efficiencies.

Matching aero-derivative gas turbines with CES drive gas is more easily accomplished when the turbine speed can be increased by about 12% and the inlet temperature is reduced by 93 °C (167 ºF). These changes permit almost exact fluid angle matching. Matching industrial gas turbines with CES drive gas where the turbine speed is set by the generator speed, i.e., 3600 rpm, causes modest changes in fluid angles. Reducing the inlet temperature by 260 °C (470 ºF) minimizes this mismatch and allows the fluid angles to remain in the range of high efficiency turbine operation. *Ultimately, gas turbines with CES drive gases should be able to operate at 1450 °C (2640 ºF), and higher temperatures, using water and warm steam for stationary components such as GG/RH, transition sections, nozzles, stators, etc., (this could be a separate closed circuit that operates before start-up and after shut-down) and the more effective CES open-loop transpiration steam cooling for rotating components such as rotor disks and blades.*

Blade temperatures of both high-pressure-ratio aero-derivative type turbines and low-pressure-ratio industrial gas turbines can be reduced appreciably using open-loop steam cooling rather than air. The lower blade operating temperatures increase blade stress margins of safety and life and could permit the use of lower cost blades or increase the turbine inlet temperature to achieve higher efficiencies.

Increasing steam turbine operating temperatures above current operating practice requires the use of more expensive materials and/or fabrication techniques but the technology base is well

established. Initially, intermediate pressure turbines in CES power plants would not operate at temperatures beyond current commercial gas turbine practice. Materials compatibility data and experience gained with gas turbines operating in an oxidizing environment at moderate pressures and very high temperatures, suggest that CES gases should not pose major compatibility problems when used to drive turbines or CES ZEPP's, however, this requires further confirmation, including full-scale testing.

The use of existing low-pressure steam turbines with CES drive gases may require design modifications to eliminate condensation and the compatibility issues presented by carbonic acid that would otherwise form.

1.3.1.2.11 References

[1] Anderson, R. E., Baxter, E. and Doyle, S.E. (Clean Energy Systems, Inc.), Final Report: <u>Fabricate and Test Advanced Non-Polluting Turbine Drive Gas Generator,</u> prepared for the United States Department of Energy National Energy Technology Laboratory; Cooperative Agreement No. DE-FC26-00NT 40804, September 1, 2000 to June 1, 2003.

[2] Anderson, R., Brandt, H., Doyle, S., and Viteri, F., "A Demonstrated 20 MW_t Gas Generator for a Clean Steam Power Plant", presented at 28th International Technical Conference on Coal Utilization & Fuel Systems, Clearwater, FL, March 10-13, 2003

[3] Anderson, R., Brandt, H., Doyle, S., Pronske, K., and Viteri, F., "Power Generation with 100% Carbon Capture and Sequestration", presented at the 2nd Annual Conference on Carbon Sequestration, Alexandria, VA, May 5-8, 2003

[4] Anderson, R.E., Doyle, S.E., and Pronske, K.L., "Demonstration and Commercialization of Zero-Emission Power Plants", <u>29th International Technical Conference on Coal Utilization and Fuel Systems</u>, Clearwater, Florida. April 18-22, 2004

[5] Anderson, R.E. and Pronske, K.L., "Kimberlina - A Zero-Emission Multi-Fuel Power Plant and Demonstration Facility**",** <u>30th International Technical Conference on Coal Utilization and Fuel Systems</u>, Clearwater, Florida. April 17-21, 2005

[6] Anderson, R.E. and Bischoff, R.W., "Durability and Reliability Demonstration of a Near-Zero-Emission, Gas-Fired Power Plant, California Energy Commission, Publication #CEC 500-2006-074, July 2006

[7] Chorpening, B., Richards, G.A., and Casleton, K.H., U.S. Dept. of Energy, National Energy Technology Laboratory (NETL); Woike, M., Willis, B., NASA Glen Research Center, Plum Brook Station, Hoffman, L. Clean Energy Systems, Inc.; " Demonstration of a Reheat Combustor for Power Production With CO_2 Sequestration"; ASME Turbo Expo Land, Sea, Air, June 16-19,2003, Atlanta, GA. USA

[8] Chorpening, B., Casleton, K.H., and Richards, G.A., U.S. Dept. of Energy, National Energy Technology Laboratory (NETL), Woike, M., Willis, B., NASA Glen Research Center, Plum Brook Station, " Stoichiometric Oxy-Fuel Combustion for Power Cycles with CO2 Sequestration", Proceedings of the Third Joint Meeting of the U.S. Sections of The Combustion Institute, March 16-19, 2003, Chicago, IL

[9] Richards, G.A., Casleton, K.H., and Chorpening, B., U.S. Dept. of Energy, National Energy Technology Laboratory (NETL); Morgantown, WV 26505, USA, "Dilute Oxy-Fuel Combustion Technology for Zero – Emission Power", 1st International Conference on Industrial Gas Turbine Technologies, Brussels, July 10-11, 2003

[10] Martinez-Frias, J., Aceves, S., Smith, J.R. (Lawrence Livermore National Laboratory), and Brandt, H. (Clean Energy Systems, Inc.), "Thermodynamic Analysis of Zero-Atmospheric Emissions Power Plant", presented at ASME International Conference, New Orleans, LA, November, 2002

[11] Martinez-Frias, J., Aceves, S., Smith, J.R. (Lawrence Livermore National Laboratory), and Brandt, H. (Clean Energy Systems, Inc.), "A Coal-Fired Power Plant with Zero-Atmospheric Emissions", IMECE2003-43923, presented at 2003 ASME Mechanical Engineering Congress & Exposition, Washington, D.C., November 15-21, 2003

[12] Smith, A.R., Klosek, J., Sorensen, J.C., and Woodward, D.W.; " Air Separation Unit Integration for Alternative Fuel Projects", Air Products and Chemicals, Inc. Allentown, Pa 18195, ASME Paper No. 98-GT63

[13] Smith, A.R. and Dillon, J.L. IV; "Gas Turbine Applications for Large Air Separation Units", Air Products and Chemicals, Inc., Allentown, PA 18195, ASME Paper No. 99-GT-321, presented at the Int. Gas Turbine & Aerospace Congress & Exposition, Indianapolis, IN, June 7-10, 1999

[14] Scharle, W., J., Wilson, Air Products and Chemicals, "Oxygen Facilities for Synthetic Fuel Projects", ASME Journal of Engineering for Industry, November,1981, Vol. 103 pp 409-417

[15] <u>2003 GTW Handbook</u>, Gas Turbine World, Vol. 23, (2003), Pequot Publishing, Inc., Southport, CT

[16] Phillips, J. N. "Integration of Commercial Gas Turbine Technology into a Clean Energy Systems Zero Emission Power Plant", Report No. 5909-08-3 To Clean Energy Systems, Inc., June 7. 2004, by Fern Engineering, Inc., Pocasset, MA.

[17] DePaepe, M. and Dick, E., "Cycle Improvements to Steam Injected Gas Turbines" (University of Gent, Gent Belgium), Int'l J. of Energy Research, pp. 1081-1107, 24, (2000)

[18] DePaepe, M. and Dick, E., "Cycle Improvements to Steam Injected Gas Turbines" (University of Gent, Gent Belgium), Applied Thermal Engineering, pp. 135-156, 21, (2001)

[19] Tsukuda, Y. et al., (Mitsubishi Heavy Industries, Ltd.), "The Operating Experience Over the Next Generation M501G/M701G Gas Turbine", presented at the ASME Turbine Exposition, June 4-7, 2001, New Orleans, LA, paper 2001-GT-0546

[20] Kizuka, N., Sagae, K., Anzai, S., Marushima, S., Ikeguchi, T., Kawaike, K., "Conceptual Design of the Cooling System for 1700 °C Class, Hydrogen-Fueled Combustion Gas Turbines", Transactions of the ASME Journal of Engineering for Gas Turbines and Power, January 1999, Vol. 121, pp. 108-115

[21] Ernette, D., Zabrecky, J., and Heine, W-H., "Ultrasupercritical Steam Turbine Designs – The Next Generation of Coal-Fired Plants", presented at 28th Int'l Technical Conference on Coal Utilization & Fuel Systems, Clearwater, FL, March 10-13, 2003

[22] U.S. Development in Advanced Steam Turbine Materials", Materials & Components in Fossil Energy Components Newsletter, No. 146, June 1, 2000

[23] Duffy, T. and Schneider, P. (Solar Turbines, Inc., San Diego, CA), "High Performance Steam Development: 1500 °F Steam Plant for Industrial Cogeneration Prototype Development Tests", Phase III Final Report SR94-R-5527-101, Contract DE-AC02-87CD40812, January 1996.

[24] Okamura, T., Koga, A., Itoh, S., and Kagagishi, H., "Evaluation of 1700 C Class Turbine Blades in Hydrogen Fueled Combustion Turbine System" (Toshiba Corp.), ASME 2000-GT-615, presented at the Int'l Gas Turbine & Aeroengine Congress & Exhibition, Munich, Germany, May 8-11, 2000

[25] Industrial & Marine Gas Turbine Engines, First edition (1982-83), Pequot Publishing, Inc., Southport, CT

[26] Fullagar, K. et al., "Aero Engine Test Experience with CMSX-4® Alloy Single-Crystal Turbine Blades", Trans. ASME, pp. 380-88, 118, April 1996

[27] Tsukuda, Y. et al., (Mitsubishi Heavy Industries, Ltd.), "The Operating Experience Over the Next Generation M501G/M701G Gas Turbine", presented at the ASME Turbine Exposition, June 4-7, 2001, New Orleans, LA, paper 2001-GT-0546

[28] Duffy, T. and Knox, C. (Innovative Steam Technologies, San Diego, CA), "Preliminary Assessment of High-Temperature Steam Topping Turbine", Final Report SR-R-5659-20 to Electric Power Research Institute, Palo Alto, CA, Appendix C, "High Temperature and Pressure Steam Test Loop Description and Test Results", p. C-7, Oct. 1992

[29] Marin, O. and Bourhis,Y., Air Liquide, Chicago Research Center; Di Zanno, P. Air Liquide, Champigny, France; Viteri, F., and Anderson, R., Clean Energy Systems, Inc., Sacramento, CA; "High Efficiency, Zero Emission Power Generation Based on a High-Temperature Steam Cycle" 28th International Technical Conference on Coal Utilization & Fuel Systems, Clearwater, Florida, March 10-13, 2003

BIOGRAPHY

1.3.1.2 Clean Energy Systems
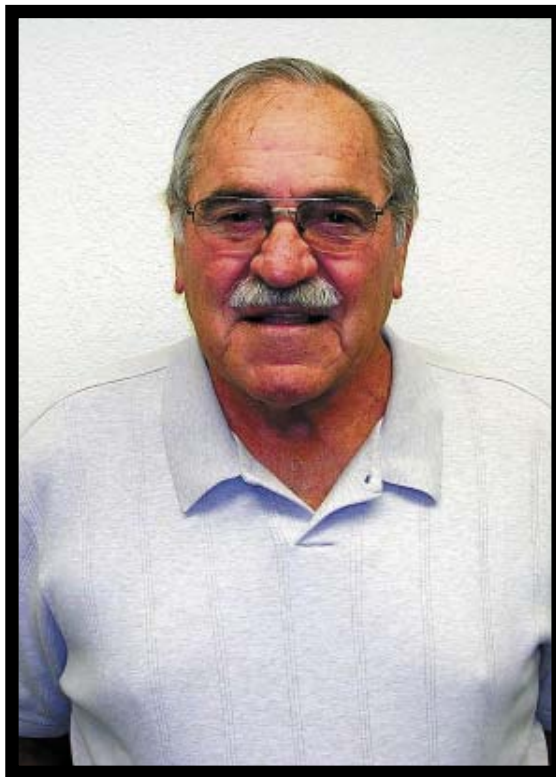
Fermin 'Vic' Viteri

Clean Energy Systems, Inc.
11330 Sunco Drive, Suite A
Rancho Cordova, CA 95742

Bus. (916) 379-9143
Fax (916) 379-9146
email: fviteri@cleanenergysystems.com

Mr. Viteri worked 36 years at Aerojet's Liquid Rocket Co., Sacramento, CA. and became manager of Rotating Machinery where he directed the analysis, design, and testing of high speed turbomachinery for pump fed rocket engines, waterjets for Navy patrol boats and submarine torpedo ejector pumps. All of these pumps were driven with specially built turbines or by gas turbines that were commercially purchased.

Mr. Viteri is one of the original founders of Clean Energy Systems, Inc., a company involved in the research and development of oxy-fuel Zero Emissions Power Plants (ZEPP). He served as President of the company from 1996 to 1999. Currently, Vic is involved with convertng a recently aquired 5 MW biomass plant in Bakersfield, CA. to an oxy-fuel ZEPP and supporting studies of similar plants for Norway and the Netherlands.