Proceedings of IMECE04
2004 ASME International Mechanical Engineering Congress and Exposition
November 13-20, 2004, Anaheim, California USA

IMECE2004-60265

GENERIC PERFORMANCE CHARACTERIZATIONS OF DISTRIBUTED ENERGY CHP SYSTEMS

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ABSTRACT

This paper describes work underway to quantify the amount and quality of thermal resource that is potentially available from a given amount of electrical distributed generation with combined heat and power (CHP) capability. The method of analysis has been to generically characterize the operating parameters of the various components involved in a CHP system (e.g., prime mover, exhaust heat exchanger, absorption chiller) and to determine expected steady-state conditions based upon thermodynamic behavior of the system. The goal is to develop a means to quickly estimate the amount of thermal energy available (e.g., MJ/hr) and quality of the thermal stream (e.g., temperature) based upon a minimum number of required inputs regarding the CHP system elements. From this effort, CHP system configurations can be evaluated for their potential use in a given application. Thus far, gas turbines, ranging from 30 kW to 14 MW of electric output, have been evaluated as the prime mover. The model has been benchmarked using operations data from a 30-kW CHP system at the CHP Integration Laboratory at Oak Ridge National Laboratory (ORNL). The comparison to actual system operations and the on-going research activities in this area will also be discussed in the paper.

INTRODUCTION

Use of combined heat and power (CHP) systems provides roughly twice the amount of useful benefit from a primary fuel source as compared to generation of electrical power alone. As each potential site for CHP will have its own unique electricity and thermal demands, it is useful to know the amount of thermal energy that can be obtained from various distributed energy prime movers and associated CHP equipment in order to provide a suitable match of supply to demand. It was felt that a simple tool would be useful to assist in initially quantifying the amount and quality of thermal energy resources available from a distributed energy prime mover. Such a tool is under development at Oak Ridge National Laboratory, and the material that follows provides details on its approach and preliminary results.

NOMENCLATURE

c_{pa}	Specific heat of air (J/kg K)
c_{pw}	Specific heat of water (J/kg K)
e_f	Energy content of fuel (MJ/kg) HHV
k	Specific heat ratio for air
m_a	Mass flow of air (kg/min)
m_f	Mass flow of fuel (kg/min)
m_w	Mass flow of water (kg/min)
P	Pressure (atm)
p_e	Net electric power output (kW)
p_p	Parasitic power losses (e.g., power electronics) (kW)
Q_t	Heat input to the turbine (GJ/hr)
r_c	Compression ratio
T	Absolute temperature (K)
w_c	Work per unit mass on compressor (kJ/kg)
w_t	Work per unit mass by turbine (kJ/kg)

Greek letters

η_b	Combustor efficiency
η_c	Compressor efficiency
η_e	Net electric efficiency on HHV basis
η_{hx}	Exhaust/water heat exchanger efficiency
η_r	Recuperator efficiency
η_t	Turbine efficiency

MODEL DEVELOPMENT

In order to determine the amount of useful thermal energy that is available from a given CHP design, one must establish

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the thermodynamic state conditions for all the components in the system. For example, a microturbine-based CHP system might consist of the microturbine, an exhaust-to-water heat exchanger, an absorption chiller, and/or a desiccant unit. The physical interconnections of these components establish a dependency relationship in which the thermal energy created by the prime mover can be envisioned as cascading through the downstream components. Thus, starting with primary fuel energy, the thermodynamic behavior of each component in the system can be calculated and used as the amount of energy input to the next component downstream.

A goal of the current modeling effort is to calculate the amount of useful thermal resource with a minimum of required user input. This has been accomplished by utilizing an efficiency term for each component or major subcomponent to account for irreversibilities and other losses. As an example, the equations related to the system shown in Figure 1 will be discussed.

The fuel mass flow into the combustor can be determined as

$$m_f = p_e / \eta_e / e_f \tag{1}$$

where the efficiency and heat content of the fuel are expressed on a higher heating value (HHV) basis.

The amount of heat energy available to the microturbine is influenced by the efficiency of the combustor such that

$$Q_t = m_f \times e_f \times \eta_b \tag{2}$$

The temperature/pressure relationship for an open-cycle turbine under ideal, isentropic conditions can be expressed as

$$\left(T_{in} / T_{out}\right)^{k/(k-1)} = P_{in} / P_{out} = r_c \tag{3}$$

where r_c is the compression ratio of the machine [1]. Similarly, the ideal relationship for the compressor is

$$\left(T_{out} / T_{in}\right)^{k/(k-1)} = P_{out} / P_{in} = r_c \tag{4}$$

As both the compressor and turbine are subject to irreversibility, the outlet temperatures are corrected from the ideal process through the use of an efficiency factor. For the turbine,

$$\eta_t = \Delta T_{actual} / \Delta T_{ideal} \tag{5}$$

where ΔT is the temperature difference between inlet and outlet.

Similarly, the temperature increase in the compressor is corrected for irreversibilities through

$$\eta_c = \Delta T_{ideal} / \Delta T_{actual} \tag{6}$$

The work done by the turbine per unit mass is simply the product of the actual temperature drop across the turbine and the specific heat of air, averaged over the inlet to outlet range. The work done to run the compressor per unit of mass is similarly determined as the product of the actual temperature increase across the compressor and the specific heat of air, averaged over the inlet to outlet range.

The mass flow of air through the compressor can be determined through a work/energy balance such that

$$w_t\left(m_a + m_f\right) - \left(w_c \times m_a\right) = p_e + p_p \tag{7}$$

A turbine system may also employ a recuperator to improve net electric efficiency (internal heat recovery). The recuperator serves as a combustion air preheater by transferring exhaust heat energy to the compressor outlet air flow. The energy balance is

$$\left(m_a + m_f\right) \times c_{pa} \times \eta_r \times \Delta T_{exhaust} = m_a \times c_{pa} \times \Delta T_{air} \tag{8}$$

The exhaust from the microturbine can be routed through an exhaust/water heat exchanger to transfer heat energy to a water loop. Similar to the recuperator, the energy balance for the exhaust/water heat exchanger can be expressed as

$$\left(m_a + m_f\right) \times c_{pa} \times \eta_{hx} \times \Delta T_{exhaust} = m_w \times c_{pw} \times \Delta T_{water} \tag{9}$$

If an absorption chiller is utilized as shown in Figure 1, the amount of available cooling is the product of the heat energy in the hot water loop (equation 9) and the chiller coefficient of performance (COP).

Using the cascading thermal energy approach to determine the useful thermal benefit for a desiccant unit is somewhat problematic in that there are several influencing factors that determine the degree of humidity reduction. Additional work is needed to complete the desiccant system models.

Calibration of the Model

In order to calibrate the model and determine the efficiency factors for the various components, data from existing systems were needed. One source of data was the Integrated Energy Systems test laboratory. The lab, a National User Facility, conducts benchmarking and experimental studies on various combinations of thermal-utilization equipment, driven by a 30 kW microturbine [2-4]. Using data collected from operations of CHP equipment at the lab, the values for the various efficiency factors were adjusted to obtain output parameters consistent with the actual operation data. As an example of the calibration process, the data in Table 1 were used as input to the model. The efficiency variables in Table 2 were then adjusted

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to the values shown in order to produce output values consistent with the laboratory data (Table 3).

Table 1. Sample input parameters from laboratory data

Microturbine	
Net electric power output (kW)	22.8
Internal parasitic losses (kW)	6.0
Compressor inlet temp (°C)	32.1
Turbine inlet temp (°C)	842.9
Compression ratio	3.4
Air/Water Heat Exchanger	
Exhaust inlet temp (°C)	279.3
Exhaust outlet temp (°C)	102.9
Water inlet temp (°C)	78.9
Water mass flow (kg/s)	2.35
Internal electrical load - kW	1.97
Absorption Chiller	
Chilled water supply temp (°C)	10.8
Chilled water flow (kg/s)	1.78
Internal electrical load - kW	3.6

Table 2. Efficiency model calibration factors

Efficiency - overall (%) HHV	19.0
Efficiency - combustor (%)	94.0
Efficiency - compressor (%)	72.5
Efficiency - turbine (%)	75.0
Efficiency - recuperator (%)	86.0
Efficiency - HX (%)	95.0
COP - Absorption chiller	0.65

A similar approach was used with the larger gas turbines, where in this case, a manufacturer of gas turbines (OEM) provided detailed operations data to permit a similar calibration. Of particular interest in the large turbine calibration effort was the availability of data covering a range of turbine capacities. As shown in Figure 2, specific equipment will have unique performance parameters, exhibited in this case by available exhaust energy. As an example, the specific 7500 kW unit has a higher electrical efficiency than neighboring units, therefore producing less available exhaust energy. In order to produce a generic model not tied to a particular manufacturer or product, generic response curves were generated. Referring to Figure 2, the generic exhaust energy curve captures the overall trend without requiring manufacturer-specific data. It is anticipated that generic response curves will be embedded in the model to provide default data. It should also be noted that the OEM and specific model curves in Figure 2 appear superimposed due to the close agreement of the model when using OEM-specific data.

Table 3. Equipment operating parameters

	Model	**Lab data**
Compressor		
Inlet air flow (kg/m)	16.3	16.3
Turbine		
Turbine outlet temp (°C)	595.9	593.1
Recuperator		
Recuperator exhaust temp (°C)	289.9	289.4
Air/Water Heat Exchanger		
Water outlet temp (°C)	83.7	83.6
Thermal energy in water (kJ/hr)	167011	164589
Absorption Chiller		
Total available cooling (kJ/hr)	108557	109345
Chilled water return temp (°C)	14.9	14.9

Current Status and Future Plans

Work on the model is currently focused on completion of a double-effect absorption chiller module. The graphical user interface has been completed and is shown in Figure 3. Through the graphical interface a user can select various equipment items to configure a particular CHP system. Selection boxes route the user to specific areas of the spreadsheet model where input parameters can be entered or modified.

Future work will include beta testing, expanding the range and type of thermal equipment available in the model, as well as adding reciprocating engine prime movers.

CONCLUSIONS

A model for evaluating the steady-state thermal behavior of distributed energy combined heat and power systems has been successfully developed and validated against experimental data for microturbines and larger gas turbines, ranging from 30 kW to 14 MW. The model provides insight into the amount and quality of useful thermal energy that can be obtained from the exhaust flows of distributed generation prime movers. A graphical interface makes the system user-friendly with selection boxes that route the user to specific equipment component areas of the model. The model has value to prospective end-users, educators, and package system vendors seeking to understand the amount and quality of useful thermal energy available from distributed generation.

ACKNOWLEDGMENTS

The authors acknowledge the funding support of the U.S. Department of Energy, Office of Distributed Energy and the wise guidance of Dr. Abdi Zaltash at ORNL.

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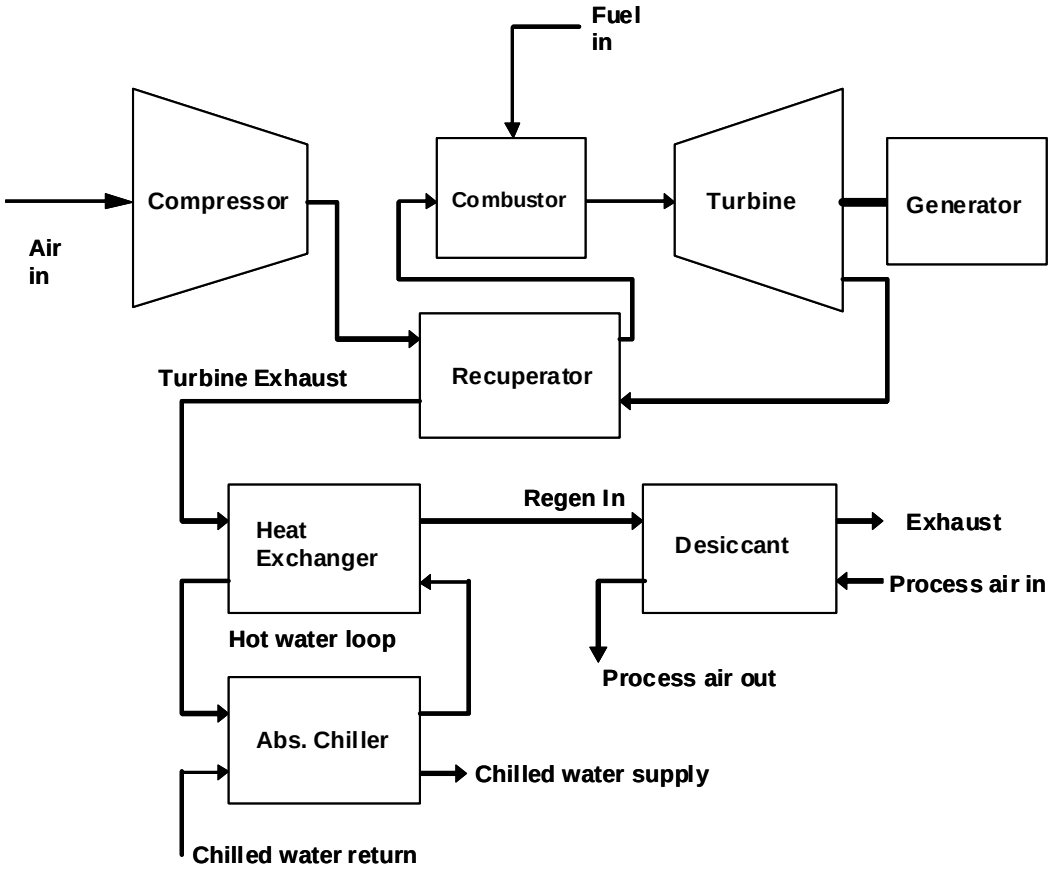

Figure 1. Example microturbine-based CHP System

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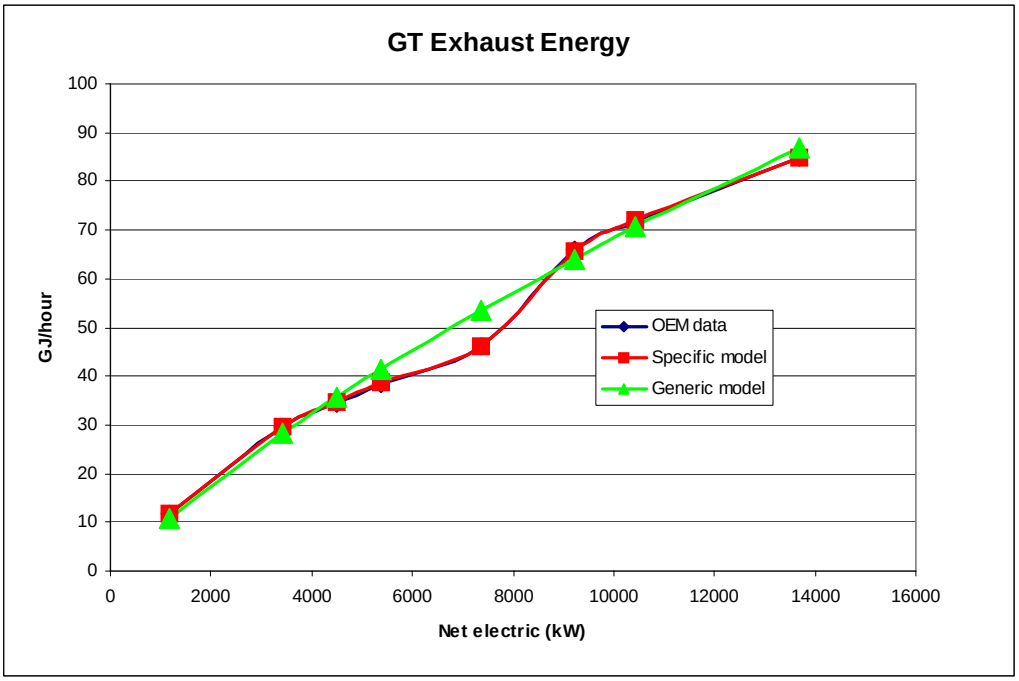

Figure 2. Model results for gas turbine

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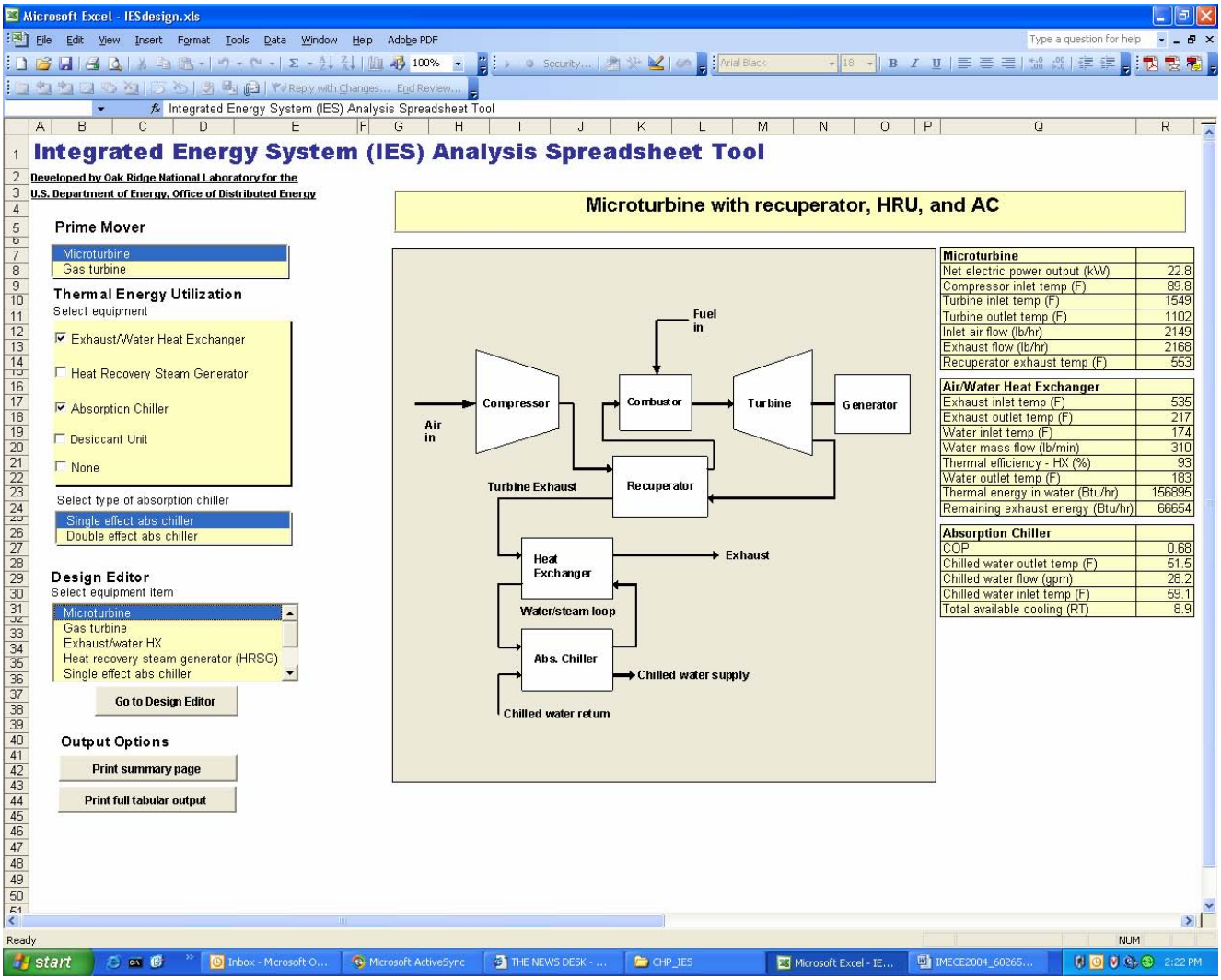

Figure 3. IES model front screen

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