Predictive Algorithms for Microturbine Performance for BCHP Systems

S. D. Labinov **A. Zaltash** **D. T. Rizy**
P. D. Fairchild **R. C. DeVault** **E. A. Vineyard**
Cooling, Heating and Power Group
Engineering Science and Technology Division
Oak Ridge National Laboratory

ABSTRACT

This paper documents the development and validation of predictive algorithms for modeling the microturbine in a building combined cooling, heating and power (BCHP) system. The mathematical model developed in this study is based on a 30-kW natural gas-fired microturbine; however, it can be extended to encompass a microturbine of any capacity and fuel-type. Both experimental and theoretical data are being used to model the BCHP system consisting of a combined microturbine and heat-recovery components and to determine ways to improve the overall BCHP system efficiency. The purpose of this paper is to outline the basis for the development of the BCHP model and to report on progress that has been made in regards to modeling microturbine operation with thermal recovery.

The basic steps being used to develop the modular structure of the BCHP system mathematical model are described, and the model developed to date has been validated by experimental data. The first step has been the mathematical modeling of the natural gas-fired microturbine. The process involved developing the thermodynamic equations that describe the polytropic processes of compression and expansion in the compressor and turbine, respectively, and developing the heat balance and mechanical energy balance equations. A linear analysis method was used to derive the equations that relate the change in the microturbine exhaust backpressure to the change in its output power and efficiency. The mathematical model was applied to the baseline performance data collected on the 30-kW natural gas-fired microturbine unit under steady-state conditions at various loads (10 to 30 kW or one-third to full output power settings, in 5 kW increments) and at various exhaust backpressures ($2.7x10^{-4}$ to $1.7x10^{-2}$ atm). Under these modes of operation, the basic operating parameters (temperatures, pressures, flows, voltages, currents, etc.) and the output power of the microturbine were measured, and its energy efficiency was calculated.

Without any external backpressure applied to the microturbine exhaust (damper fully open), the average measured power output ranged from 10.0 to 27.8 kW. The microturbine efficiency averaged from 18.0% to 23.0% for the one-third to full output power demand settings. With maximum externally applied backpressure ($1.70x10^{-2}$ atm), the model shows that the output power losses (decrease in power output) due to backpressure range from 3.5% for full output to 5.5% for one-third power (lowest value of ~3% occurs at 25 kW), while the efficiency losses (decrease in efficiency) range from 2.5 to 4% (lowest value of ~2.0% occurs at 25 kW), correspondingly. The internal energy losses were calculated to be approximately 30% of the total power produced.

INTRODUCTION

The goal of the Buildings Combined Cooling, Heating and Power (BCHP) program is to optimize the integration of electric power generation and heating, ventilating, and air-conditioning (HVAC) systems with other energy-efficient building technologies. The objective is to maximize energy efficiency, reduce energy use and emissions, increase the power available for critical loads by providing an option to central power generation, and improve electrical power reliability and quality (Banetta et al. 2001).

The traditional energy cycle in the United States and most other developed countries is the combustion of fossil fuels and/or the use of nuclear fuels in a large central power plant to generate electricity. The electricity is then delivered to users over a high-voltage transmission and lower-voltage distribution network. At least 50 to 70% of the energy content of the fuel is lost at the power plant alone through energy conversion inefficiencies and is discharged in the form of waste heat into the environment. Further losses (~8%) occur in the electric power transmission and distribution network in the form of electric current losses and power transformation losses (step-up and step-down transformer losses).

Distributed energy resources (DER), such as microturbines, are small, modular power generation systems located on or near the site where the energy that is generated is used (US DOE 2000). Unlike centralized energy resources, such as large power plants, they provide an opportunity for local control of power generation and more efficient use of waste heat to boost overall efficiency and reduce emissions. DER comprise a portfolio of technologies, both supply-side and demand-side. The DER technologies that can benefit the most from BCHP include gas turbines, reciprocating engines, microturbines, and fuel cells. In a BCHP system, waste heat from these DER technologies can be used as input power for heat-activated air conditioners, chillers, and desiccant dehumidifiers; to generate steam for space heating; and/or to provide hot water. By making use of heat energy that is normally wasted, BCHP systems can meet a building's electrical and thermal loads with a lower input of fossil fuel, yielding resource efficiencies of 40 to 70% or more.

The experimental and theoretical study of various BCHP configurations allows the evaluation of the optimal operational modes of these systems. Current BCHP system configurations consist of equipment originally developed for stand-alone use. One of the objectives of BCHP research is to determine how to integrate the hardware, controls, and operation of these separate pieces of equipment so that the system operates at optimum efficiency under both steady-state and transient conditions. A BCHP system needs to operate at maximum efficiency under different thermal (cooling and/or heating) and electrical load distributions. The nature of these distributions depends on climatic conditions, season and time of day, and the characteristics (such as size) and number of the thermal and electrical loads at the site. Depending on the nature of loading, the operation of each individual unit will be different so that thermal and energy balances are kept optimal. For a system consisting of five or more units, optimizing all the parameters of the individual units under various operating conditions is too complex to be accomplished by direct experiment. It is the goal of this study to build a BCHP model, based on both experimental data and mathematical modeling, that will allow more efficient analysis of a BCHP system and make it easier to determine its optimal equipment configurations and operational modes.

The semi-empirical BCHP model that is being developed will provide a means of analyzing optimal system operation. The mathematical aspect of the model allows the user to choose optimal operating conditions and predict parameter values for individual units and entire systems under various loads in both steady-state and dynamic operating modes. The experimental aspect of the model makes it possible to measure key parameters from a real-world BCHP system and its individual pieces of equipment and adjust the model to accurately represent the operation of the system. The experimental research also provides a way to verify model simulations that analyze recommended equipment design modifications and operational recommendations (Bejan 1996).

For the experimental phase of this study, a flexible test bed consisting of a 30-kW microturbine was instrumented and configured to operate without and with waste heat recovery from the microturbine exhaust. The thermal recovery components consist of an air-to-water heat exchanger, both an indirect-fired and a direct-fired desiccant dehumidifier, and an indirect-fired single-effect absorption chiller. An air duct network from the microturbine exhaust to the heat exchanger and/or the direct-fired thermal equipment, a water loop from the heat exchanger to the indirect-fired thermal equipment, and an air mixer (for mixing outside air with exhaust air) provide for flexible testing of various waste heat recovery conditions and loadings. In addition, the microturbine exhaust heat output can be varied by changing the power output to test different waste heat source conditions. The absorption chiller output can be used to cool the inlet air to the microturbine to increase its efficiency. (This cooling may be particularly useful when outside ambient temperatures are high, because the microturbine is located outdoors and its efficiency drops off with higher temperatures.) A detailed description of the experimental test bed and the results of baseline performance tests conducted with the 30-kW natural gas-fired microturbine are discussed in an earlier report (ORNL 2001). This paper provides a general discussion of the BCHP model and its development, along with a detailed discussion of the microturbine model and the use of the model (verified with experimental data from the test bed) to analyze the microturbine-based BCHP system. The experimental testing in this phase of the study involved emulating heat recovery from the microturbine by placing a throttle damper at its exhaust to apply external backpressure.

MODEL DEVELOPMENT

General Structure Of BCHP Model

The BCHP mathematical model has a modular two-level structure (Figure 1). The first level of the model consists of mathematical models of the individual units, such as a natural gas-fired microturbine; a chiller; a heat exchanger; desiccant dehumidifiers; and any next generation products with improved efficiency. The basic structure of each model consists of equations describing thermodynamic, heat exchange, hydraulic, and thermophysical processes that are typical of a given type of unit. In addition, the individual model has an initial database that includes the following:

- the results of experimental study of the unit and its elements,
- the unit's basic functional characteristics, depending on its design,
- the unit's manufacturer information,
- the unit's working fluids physical and chemical property data, and
- input and output parameter descriptions to provide for interaction of the model with other models of the BCHP system.

Each individual model incorporates a set of equations that represents the individual component, such as a natural gas-fired microturbine.

The second level of the model includes the matrix of possible BCHP system configurations and provides for the combined solution of energy, material, and mechanical balances of the entire BCHP system. The *MathCad* software (MathSoft 1999) is used to solve for the steady-state conditions of the model equations, and the *VisSim* software (Visual Solutions 1999) is used to solve for the dynamic conditions of the BCHP system.

The model's second level also includes the following:

- the data library for the heating, cooling, and energy loads for different users,
- the dynamic characteristics of the equipment,
- the special solution methods for optimizing under-defined sets of multi-parameter equations with free terms, and
- the interface that allows for interaction among levels and customer control over the configuration matrix.

The mathematical model is being developed step by step in parallel with the experimental part of this project: the first-level models of the individual units will be developed, and then the second level. To date only the microturbine mathematical model has been developed and is discussed in further detail below.

Model of The 30-kW Gas Microturbine

Gas Microturbine Unit. The microturbine is a three-phase 480-VAC/30-kW rated unit that can operate at 50 or 60 Hz (cycles/s) when connected to the grid. A stand-alone option (allows the microturbine to start and generate power without electric utility service) for the unit is also available from the manufacturer, although this feature was not included with the unit employed in our tests. The turbine-generator, which is designed to operate at a maximum speed of 96,000 rpm, produces high-frequency AC power that is rectified to DC and converted to 50 or 60-Hz AC power by the unit's power conditioning electronics. The gas turbine and the electrical generator are on the same shaft and rotate rapidly to produce the correspondingly high-frequency AC current. Subsequently, the microturbine has a digital power controller (DPC) to control its operation and all power conversion functions. The DPC converts the variable-frequency power from the generator into grid-quality power at the output terminals. The variable-frequency power is converted to constant-voltage DC power, which is then inverted to constant-frequency AC power. The DPC is cooled by three fans, each with its own electric motor.

The unit is designed to produce a continuous phase current of 36 A at 480 VAC and to produce unity power factor (the amount of real power divided by the total power) when the unit is grid connected. The unit nominal phase-to-neutral voltage is 277 VAC. The microturbine is connected to the grid (through a 480-VAC electrical panel which is connected to the local distribution system) via a 480-VAC/75-kVA three-phase isolation transformer. The transformer is connected wye-delta with the wye-side connected to the microturbine. The delta connection provides an additional measure for preventing harmonics from entering the grid from the microturbine. The microturbine acts as a current-source and thus has no direct affect on the grid voltage or frequency. The microturbine power controller incorporates protection functions that will shut down the unit if the phase-to-neutral voltage sags (or drops) to less than 208 VAC for more than 10 seconds. Islanding of the microturbine (or separation of the unit from the grid) is detected within milliseconds from the loss of current control. The microturbine also includes over voltage, over/under frequency, and rate of frequency protection functions to protect the unit and to prevent islanding of the unit. The 30-kW natural gas-fired microturbine was found to produce electricity with a maximum output power of ~28-kW (full load). The rejected heat of the microturbine exhaust gases can be employed to drive various thermally-activated units of a BCHP system. The schematic flow chart and the thermodynamic cycle of the gas microturbine unit are shown in Figures 2a and 2b. The primary components of the unit include the air compressor, recuperator, combustion chamber, turbine, and permanent magnet generator. The rotating components are mounted on a single shaft, supported by air bearings that rotate at up to 96,000 rpm at full load. The generator is air-cooled. The fuel (natural gas) is fed to the combustion chamber with the help of a gas compressor driven by an independent electric motor. The compressor module is air-cooled by a fan that also has an independent electric motor.

Microturbine Flow Chart and Thermodynamic Cycle. As shown in Figure 2b, air from state point A, at ambient temperature T_a and ambient pressure P_a, goes to the entrance duct of the air compressor, where it is heated to temperature T_1 at point 1 by the heat from the electrical generator. Disregarding the hydraulic losses between the points A and 1, let us assume that P_1 equals P_a. The air from state point 1, at temperature T_1 and pressure P_1, goes to the air compressor, where it is compressed polytropically to state point 2, at T_2 and P_2.

The ratio of the work of adiabatic compression from point 1 to point 2′, to the work of polytropic compression from point 1 to point 2, is the air compressor efficiency η_c. It depends on the compressor design and operational conditions. The ratio P_2/P_1 is called the compressor pressure ratio ε_c and depends on the backpressure at the compressor outlet and the conditions of the compressor operation.

After the compressor, air goes to the recuperator, where it is heated from temperature T_2 at point 2 to T_3 at point 3 by the heat of the rejected exhaust gases. Because of the hydraulic resistance of the recuperator in the high-pressure line, the air pressure decreases to P_2' at point 3. After the recuperator, the air goes to the combustion chamber, where it is heated by the chemical energy of fuel oxidation from temperature T_3 at point 3 to T_4 at point 4. After the combustion chamber, the hot gas is fed to the gas turbine, where it is expanded polytropically from the state of point 4, at temperature T_4 and pressure P_2', to the state of point 5, at T_5 and P_1'. The ratio of the work of polytropic expansion from point 4 to point 5, to the work of adiabatic expansion from point 4 to point 5N, is called the turbine efficiency η_t and depends on the turbine design and its operating conditions. The ratio P_2N/ P_1N is called the rate of expansion ε_t. After the turbine, the rejected gas goes to the recuperator, where it is cooled from temperature T_5 to T_6. Because of the hydraulic resistance of the recuperator in the low-pressure line, the pressure at point 6 is lower than P_1'. When the rejected gases are released to the atmosphere, the pressure at point 6 is P_1. If the rejected gases are employed to drive other units of the BCHP system, the pressure at point 6 is higher than P_1 by the value of the hydraulic loss in the BCHP units.

Microturbine Unit Basic Processes and Their Mathematical Descriptions. The mathematical equations representing the basic processes of the natural-gas fired microturbine are provided below (Bejan 1996; ORNL 2001; Reid et al. 1977; EPRI 1983; Burghardt 1982).

Electrical generator cooling. The ambient air passes through the electrical generator cooling system before it reaches the air compressor. The generator cooling system is an air duct into which the fins of the electrical generator stator casing extend, so that water is not needed to cool the electrical generator. At the same time, the air temperature increase at the compressor inlet from T_a to T_1 results in an increase in the compressor drive power consumption. The heat loss in the electrical generator cooling is defined as:

$$Q_{eg} = G_a \, Cp_a \, (T_1 - T_a) \tag{1}$$

Air compression. Air compression takes place in the single-stage centrifugal compressor. No air cooling is provided in this process, so the compression process efficiency is lower than optimal. Power to drive the compressor is defined as (Burghardt 1982):

$$W_c = \frac{k_c}{k_c - 1} T_1 \frac{R}{\mu_a} G_a \left(\varepsilon_c^{\frac{k_c - 1}{k_c}} - 1\right) \frac{1}{\eta_c} \tag{2}$$

The temperature of the air at the end of the polytropic compression process is (Burghardt 1982):

$$T_2 = T_1 \left[1 + \left(\varepsilon_c^{\frac{k_c - 1}{k_c}} - 1\right) \frac{1}{\eta_c} \right] \tag{3}$$

Recuperation. The efficiency of a recuperation process depends on the amount of heat loss to the environment. The recuperator is a ring-shaped container encircling the turbine and the combustion chamber, so it has the largest possible diameter of the external shell and consequently the surface area. This configuration results in high heat losses to the environment, which results in a lower recuperation process efficiency. Heat transferred from the rejected gases is defined as:

$$Q_r = G_a\, Cp_a\, (T_3 - T_2) = G_g\, Cp_g\, (T_5 - T_6)\, \eta_r \tag{4}$$

where

$$\eta_r = (1 - \alpha)$$

Natural gas compression. As the natural gas pressure of distribution lines usually does not exceed atmospheric pressure, an additional gas compressor is necessary to feed pressurized natural gas to the combustion chamber. This arrangement results in additional internal power consumption and causes a decrease in the microturbine output power and efficiency. Power necessary to drive the natural gas compressor is defined as (Burghardt 1982):

$$W_{ng} = \frac{k_{ng}}{k_{ng} - 1} \frac{R}{\mu_{ng}} T_a G_{ng} \left[\left(\frac{\varepsilon_c \cdot a_1}{\frac{P_{ng}}{P_1}} \right)^{\frac{k_{ng} - 1}{k_{ng}}} - 1 \right] \frac{1}{\eta_{ng}} \tag{5}$$

The parameter a_1 is assumed to have a value of 1.1 to take into account the expected hydraulic losses in the natural gas feed line to the combustion chamber.

Combustion. The external heat is supplied by the combustion of the natural gas. The heat transferred in the combustion chamber is defined as:

$$Q_g\, \eta_{cc} = G_g\, Cp_g\, (T_4 - T_3) \tag{6}$$

Expansion. The expansion process takes place in a single high-revolution turbine. Power developed by the turbine is defined as:

$$W_t = \frac{k_t}{k_t - 1} \frac{R}{\mu_g} T_4 \, G_g \left(1 - \frac{1}{\varepsilon_t^{\frac{k_t - 1}{k_t}}} \right) \eta_t \tag{7}$$

Gas temperature at the end of the expansion process is:

$$T_5 = T_4 \left[1 - \left(1 - \frac{1}{\varepsilon_t^{\frac{k_t - 1}{k_t}}} \right) \eta_t \right] \tag{8}$$

Heat and work balance. According to the First Law of Thermodynamics, for the turbine cycle, the heat and work balance is described by the equations:

$$Q_g \, \eta_{cc} - G_g \, Cp_g \, [(T_6 - T_1) + (T_5 - T_6)\, \alpha] - W_l = W_p \tag{9}$$

where

$$W_p = W_t - W_c - W_l$$

Electrical controller cooling. The electrical controller is heated in the process as a result of the electrical resistance, and its heat is removed by air passing through three vents. The total quantity of heat loss from the controller cooling is defined as:

$$Q_c = G_c \, Cp_a \, (T_1 - T_a)\, \theta = (W_p + W_l)\, \eta_{cn} \tag{10}$$

Equations (1) - (10) that describe the basic processes of the gas microturbine unit utilize average values of specific heat ratios determined from the real gas equations in the range of pressure and temperature change for the processes.

Dependence of Microturbine Output Power and Efficiency on Backpressure

The combined operation of the natural gas-fired microturbine and the other units of a BCHP system is characterized by a growing backpressure at the outlet of the microturbine caused by the hydraulic resistance of the heat exchangers and other heat recovery equipment. The magnitude of the backpressure depends on the operational conditions both of the microturbine and the individual thermal recovery units in the duct loop. As a result of the backpressure, the microturbine useful power and efficiency are reduced. The change in the microturbine output power and efficiency as a function of the change in backpressure (for gas subsonic flow in the turbine) is described by the following equations (Fairchild et al. 2001):

$$-\frac{\Delta W_p}{W_p} = Z\frac{\Delta P_1}{P_1} - \left(\frac{\Delta T_4}{T_4} + \frac{\Delta G_g}{G_g}\right)\frac{W_t}{W_p} + \left(\frac{\Delta T_1}{T_1} + \frac{\Delta G_a}{G_a}\right)\frac{W_c}{W_p} + \frac{\Delta W_l}{W_l}\frac{W_l}{W_p}$$

(11)

$$-\frac{\Delta E}{E} = Z1\frac{\Delta P_1}{P_1} - \left(\frac{\Delta T_4}{T_4} + \frac{\Delta G_g}{G_g}\right)$$

where

$$Z = \frac{k_c - 1}{k_c}\left[\frac{\varepsilon_c^{\frac{k_c-1}{k_c}}}{\varepsilon_c^{\frac{k_c-1}{k_c}} - 1}\right]\frac{W_c}{(W_t - W_c)}\frac{1}{\varepsilon_c} + \frac{k_t - 1}{k_t}\left[\frac{1 - \frac{1}{\varepsilon_t}}{\varepsilon_t^{\frac{k_t-1}{k_t}} - 1}\right]\frac{W_t}{(W_t - W_c)} \quad \textbf{(12)}$$

$$Z1 = Z - \frac{k_t - 1}{k_t}\left[\frac{1 - \frac{1}{\varepsilon_t}}{\varepsilon_t^{\frac{k_t-1}{k_t}} - 1}\right] \quad \textbf{(13)}$$

$$E = \frac{\text{Microturbine Output Power}}{\text{Natural Gas Heat Input}} = \frac{W_P}{Q_g} \quad \textbf{(14)}$$

It should be noted that the efficiency reported in this study is based on the higher heating value of natural gas (HHV) instead of its lower heating value (LHV). The efficiency based on LHV would be approximately 10% higher on a relative basis than the one based on HHV or approximately 25% instead of 23% at full power output. It should be noted that the efficiency of 26% (+/-2%) specified by the microturbine manufacturer is based on the LHV of natural gas.

Solving The Model Equations

Equations (1) – (10) are used to determine the gas turbine unit parameters T_4, ε_c, G_a, η_t, η_c, η_{cc}, η_r, α, and W_l from the experimental data (Table 1) and the information provided by the microturbine manufacturer (Table 2). The calculation procedure is as follows:

- On the basis of experimental data from the microturbine operation under full load and the manufacturer data on temperature at the turbine inlet under full load, and the unit air flow rate, power consumption to drive the natural gas compressor, vents, and heat loss with DPC cooling; the system is initially solved for η_t, η_c, η_{cc}, η_r, and α.
- With the values of η_t, η_c, η_r fixed for subsequent modes of the microturbine operation, the system is solved iteratively for T_4, g_c, η_{cc}, α, G_a, and W_l.

Additional calculations show that the values of η_t, η_c, and η_r are not significantly influenced by change in the microturbine load, while the values of η_{cc} and " vary within the narrow interval 0.5–2%.

TABLE 1.
Experimental data on microturbine performance for various output power levels

a. Damper at microturbine exhaust is fully open (no externally-applied backpressure)

T_a (K)	n (rpm)	W_p (kW)	T_5 (K)	T_6 (K)	P_1 (atm)	G_{ng} (kg/s)	Q_g (kW)	Phase Currents A, B, C (A)	Phase Voltages A, B, C (V)
272.2	89558	27.75	873.0	532.6	7.9×10^{-4}	2.571×10^{-3}	118.76	32, 32, 32	287, 285, 285
273.5	86475	24.94	876.4	526.2	7.1×10^{-4}	2.369×10^{-3}	109.44	28, 29, 29	287, 286, 285
276.0	81414	19.95	881.8	514.6	5.7×10^{-4}	1.967×10^{-3}	90.86	23,23, 23	286, 285, 285
275.8	75048	14.96	888.8	499.0	4.4×10^{-4}	1.567×10^{-3}	72.36	17, 17, 17	285, 284, 284
277.2	67757	9.99	897.2	484.3	2.7×10^{-4}	1.202×10^{-3}	55.48	11, 12, 12	284, 283, 283

b. Damper at microturbine exhaust is partially closed to produce maximum allowable backpressure on the microturbine

T_a (K)	n (rpm)	W_p (kW)	T_5 (K)	T_6 (K)	P_1 (atm)	G_{ng} (kg/s)	Q_g (kW)	Phase Currents A, B, C (A)	Phase Voltages A, B, C (V)
277.5	90974	26.49	871.2	538.1	1.7×10^{-2}	2.592×10^{-3}	119.71	30, 31, 30	288, 287, 287
280.6	86923	22.17	876.2	530.3	1.6×10^{-2}	2.251×10^{-3}	104.12	25, 25, 25	289, 289, 288
270.9	78905	18.37	884.7	505.2	1.7×10^{-2}	1.884×10^{-3}	87.23	21, 21, 21	287, 286, 285
275.2	74698	14.32	889.3	497.8	1.7×10^{-2}	1.561×10^{-3}	72.17	16, 17, 17	286, 285, 284
275.8	67578	9.59	897.1	482.3	1.7×10^{-2}	1.197×10^{-3}	55.26	11, 11, 11	285, 283, 283

c. Damper at microturbine exhaust is fully open and the unit's speed is adjusted to match the speed for the maximum backpressure tests below

T_a (K)	n (rpm)	W_p (kW)	T_5 (K)	T_6 (K)	P_1 (atm)	G_{ng} (kg/s)	Q_g (kW)	Phase Currents A, B, C (A)	Phase Voltages A, B, C (V)
273.5	90799	27.93	871.3	535.9	8.4×10^{-4}	2.666×10^{-3}	123.07	32, 32, 32	288, 287, 287
270.4	86624	24.96	876.1	524.4	7.6×10^{-4}	2.393×10^{-3}	110.66	28, 29, 29	288, 287, 286
261.7	78547	20.01	871.5	495.9	5.9×10^{-4}	1.976×10^{-3}	91.48	23, 23, 23	286, 284, 284
273.2	74730	14.96	889.2	496.5	4.2×10^{-4}	1.601×10^{-3}	73.99	17, 17, 17	285, 284, 284
274.8	67429	9.97	897.1	480.9	2.7×10^{-4}	1.212×10^{-3}	55.69	11, 12, 12	284, 283, 283

TABLE 2.
Microturbine information supplied by the manufacturer for the low pressure natural gas unit at full load, under ISO conditions (288K at sea level)

Parameter	Description	Value
T_4	Turbine inlet temperature (K)	1116
g_c	Compressor pressure ratio	3.4
G_a	Air compressor air flow rate (kg/s)	0.31
W_f	Total power of the electrical fan motors (kW)	0.122
W_{ng}	Gas compressor power (kW)	2.1
0_{cn}	Digital controller efficiency	0.94
W_p	Microturbine full load power (kW net)	28 +/-1
E	Microturbine efficiency (LHV)* (%) Microturbine efficiency (HHV)* (%)	26 +/-2 23.6 +/-1.8
Q_g	Natural gas heat input (kW)	122.5
T_6	Exhaust gas temperature at the recuperator outlet (K)	534
$Q_{exhaust}$	Total exhaust energy (kW)	85
n	Microturbine engine speed (rpm)	96,000

*Note that calculations presented in this study are based on the higher heating value (HHV) of natural gas rather than the lower heating value (LHV).

Determination of Linear Multipliers for Gas Microturbine Unit Output Power and Efficiency. The values of linear multipliers Z and Z1 are obtained from Equation (11) after the values of g_c, W_t, W_c, W_l, T_4, T_1, G_a and G_g have been determined as a result of solving the set of equations for each mode of turbine operation, including the modes with changing backpressure.

Determination of Gas Microturbine Unit Power Loss. The microturbine power losses under nominal load are determined as a result of experimental measurements and on the basis of information provided by the manufacturer. When the losses are calculated for partial load modes, the following conditions are assumed:

- for all loads, the power consumed to drive fans is constant and is equal to the power consumption in the nominal mode,
- the cooling airflow rate is the same for all load conditions and is equal to the flow rate in the nominal mode,
- the power consumption to drive the natural gas compressor under partial-load conditions depends on natural gas flow rate and air compressor pressure ratio, and
- the efficiency of the natural gas compressor remains constant in all modes of operation.

The total value of energy losses (W_l) was determined as a sum of all losses, that is W_{ng}, Q_c, Q_{eg}, and W_f.

EXPERIMENTAL STUDY RESULTS

Microturbine Operational Modes

Experimental study of the 30-kW natural gas-fired microturbine is described in detail in an earlier report (Fairchild et al. 2001). The 30-kW microturbine was tested over power output settings of one-third to full power (10 to 30 kW) in 5-kW increments. At this point, only some peculiarities related to the operational conditions with backpressure change at the turbine outlet are outlined. The task at this stage of the research was to find experimentally how the microturbine output power and efficiency would change with increase in levels of backpressure at the unit exhaust. Equation (11) predicts that the function should be linear under a small change (no more than 2%) in backpressure. This linear dependence is expected if temperatures at the turbine and compressor inlets and the gas flow rate remain constant with varying backpressure. Under actual experimental conditions, the turbine inlet temperature and the exhaust gas flow rate are not measured directly, so instead of these parameters being kept constant, the engine speed (rpm) was kept constant when backpressure was changed.

The mode of measurement was established as follows:

- Under steady-state microturbine operating conditions, the backpressure was increased by partially closing the damper at the unit exhaust. Under this arrangement, the microturbine control system, trying to maintain output power at a fixed value, increased its engine speed and then increased the compressor pressure ratio and turbine inlet temperature. As a result, the output power value was kept practically constant irrespective of the backpressure (Table 1b).
- Upon reaching a new steady-state mode of fixed backpressure, the microturbine output power was gradually decreased until the initial turbine speed (turbine speed measured with no externally applied backpressure) was reached (Table 1c). After a steady-state condition had been reached with a speed equal to the initial value, measurements were accomplished: 15 minutes for each mode. The arithmetic mean was computed over this interval. The averaged data are given in Table 1. These data along with the mathematical model were used for further evaluation of the microturbine.

Experimental Results With The Mathematical Model

Figures 3-9 show the various microturbine parameters calculated using Equations (1) - (10). The turbine inlet temperature (T_4) versus the microturbine output power (W_P) is shown in Figure 3. It is a nonlinear relationship and shows a variation of inlet temperature of ~1050 to 1115K for the turbine power demand settings of one-third to full power output. The compressor pressure ratio (P_2/P_1) versus the microturbine output power is shown in Figure 4. The compressor pressure ratio was found to vary from 2.2 to 3.4 over the microturbine power demand settings of one-third to full power. Figure 5 shows both the air flow rate (G_a) through the air compressor and the natural gas flow rate (G_{ng}). The air flow rate was found to vary from ~0.18 to 0.31 kg/s and the natural gas flow rate varied from 0.0012 to 0.0026 kg/s for the one-third to full power settings of the microturbine. Both vary in a fairly linear fashion with the microturbine output power. Figure 6 shows the heat input of the natural gas (Q_g) to the microturbine, the heat output from the microturbine (Q_{ut}) for utilization by thermal recovery components, and the engine speed (n) versus the microturbine power output (W_P). The heat input of the natural gas and heat output from the microturbine were found to vary from ~55 to 120 kW and from ~18 to 41 kW respectively, over the microturbine power test operating range. Although the engine speed is rated at 96,000 rpm, the speed for the particular operating conditions of the tests varied from ~68,000 to 90,000 rpm over the one-third to full power output range. It should be noted that the operating engine speed of the microturbine is not only a function of power output but outside ambient temperature. All three of these parameters vary in a fairly linear fashion with the microturbine output power.

Figure 7 shows the energy losses for the microturbine and how they vary with the microturbine output power. The values shown include the total losses (W_l), the controller heat loss (Q_c), the electrical generator heat loss (Q_{eg}), the gas compressor power (W_{ng}), and the total power of the fan motors (W_f). The total losses were found to vary from ~2.9 to 6.1 kW over the one-third to full output power range of the microturbine. Figure 8 shows the power to the turbine (W_t), the power to the air compressor (W_c), and the engine speed versus the microturbine output power (W_P). These functions feature strong linear behavior.

The recuperation rate (R_r) as a function of the microturbine output power is shown in Figure 9. Results of the microturbine model show that R_r increases from 0.817 to 0.842 with decreasing output power. The increase in the recuperation rate is caused by the decrease in the compressor pressure ratio.

During the tests, measurements of the microturbine parameters were taken with the exhaust backpressure changing from 2.7×10^{-4} (without externally applied backpressure) to 1.7×10^{-2} atm (maximum externally applied backpressure) to determine the dependence of the microturbine output power and efficiency on increasing backpressure. Although the values of the turbine inlet temperature (T_4), the compressor inlet temperature (T_1), the air flow (G_a), and the gas flow through the turbine (G_g) were not constant, they did not change significantly (less than 5%) during these tests. However, the influence of these parameters on Equation (11) is considerable. In order to find the direct dependence of these parameters on backpressure, the measured values for these parameters were used to deduce corrections in Equation (11). Next, these corrections were subtracted from the measured values for) W_P/W_P and) E/E, correspondingly. Thus, corrected experimental data were compared with the model calculations based on constant temperatures and flow rates as shown in Figure 10. It should be noted that in Figure 10, only the maximum backpressure was used to illustrate the maximum difference between the model and the measurements for relative changes in power and efficiency over the tested power operating range. As shown, the corrected experimental and calculated data agree quite well. The data show that the output power losses (decrease in power output) due to maximum backpressure range from 3.5% for full output to 5.5% for one-third output (note that the lowest value of ~3% occurs at 25 kW). Further, the data show that the efficiency losses (decrease in efficiency) due to the maximum backpressure range from 2.5 to 4% (it should be noted that the lowest value of ~2.0% occurs at 25 kW).

Figure 11 shows the dependence of the microturbine efficiency on the output power of the unit. The baseline curve shows a maximum efficiency of 23.4% (based on HHV) for the microturbine, with its current design that includes air cooling system, a recuperator, and a natural gas compressor to boost the input gas pressure from 0.34 to 3.74 atm. From the microturbine-based BCHP model, it was determined that the overall plant system efficiency with its current design is approximately 59%. BCHP efficiency (with recuperator), E_{chp}, is defined as:

$$E_{chp} = \frac{W_P + G_g\ Cp_g\ \left(T_6 - T_{vent}\right)}{Q_g} \tag{15}$$

The parameter T_{vent} is assumed to be 400K. Therefore, Equation (15) is based on a 400K exhaust gas temperature to the atmosphere. The recuperator recovers part of the turbine rejected heat to the cycle which in turn increases the microturbine efficiency from ~15% (simple cycle or unrecuperated cycle) to ~23% (cycle with recuperator). However, the use of a recuperator in a microturbine-based BCHP system causes a decrease in the available exhaust heat from the microturbine, which is used to drive the other heat recovery components. As a result, with the recuperator present, the total quantity of exhaust heat from the microturbine is decreased which may cause a decrease in the overall BCHP efficiency. It should be noted that the recuperator represents ~25–30% of the microturbine's overall cost (McDonald 2000).

Startup, Shutdown, and Microturbine Response to Load Change

As mentioned earlier, tests on the microturbine were conducted at various output power demands and turbine backpressures (ORNL 2001). Figure 12a shows the startup of the microturbine from cold start to full power (data measurements every 5 s). The data show that the time required for the unit to reach full power is ~210 seconds. The data for the engine speed shows three step changes; from 0 to ~ 25,000 rpm, from 25,000 to ~45,000 rpm, and from 45,000 rpm to the speed required for the full power. Figure 12b shows the shutdown time to be ~520 seconds with the engine speed changing from full speed to ~ 45,000 rpm and then from 45,000 to 0 rpm. Figure 13 shows how quickly the microturbine can vary its output power. It shows the ramping down of the unit from one steady-state power level to another. The unit output power is decreased from full power to 25 kW. Results show that the time required for ramping down and ramping up was ~20 seconds. The 20-second response time for power variations was found to be quite consistent regardless of the microturbine power level and regardless of the change in the load level.

CONCLUSION

A general structure and basic steps for developing a building combined cooling, heating and power (BCHP) system model is presented. Although the BCHP model is still under development, the model will be invaluable for assessing the operation of numerous combinations of power generation and thermal recovery systems. Currently, the model is based on a microturbine-based power generation source. However, the modular form of the BCHP model would allow the addition of other power generation source models, such as fuel cells and reciprocating engines.

A test facility has been developed for testing combined power generation and thermal recovery components. The testing to date has included assessment of the performance of a microturbine with increasing external backpressure applied to its exhaust to emulate the application of thermal recovery systems. The data measurements taken during the microturbine testing have been used to develop a semi-empirical model for the microturbine in order to determine how its power output and efficiency would vary with thermal recovery. The next steps in the testing process will be to develop models for the thermal recovery systems and to incorporate these individual models into the overall BCHP model.

The 30-kW microturbine was tested over its power demand settings of one-third to full output power (10 to 30 kW) in 5-kW increments. The microturbine was tested both without any externally applied backpressure to its exhaust and with increasing levels of externally applied backpressure. Under these modes of operation, the basic parameters of the microturbine as well as the microturbine output power were measured and its energy efficiency was calculated.

The first step in the modeling process has been the development of a mathematical model for the 30-kW natural gas-fired microturbine. The process involved developing the thermodynamic equations that describe the polytropic processes of compression and expansion in the compressor and turbine correspondingly and developing the heat balance and mechanical energy balance equations for the microturbine. A linear analysis method was used to derive the equations relating the change in the microturbine output power and energy efficiency with the change in applied backpressure at the microturbine exhaust and change in its power output level. The model was applied to the data collected during the microturbine tests and was found to adequately describe the processes taking place in the 30-kW natural gas microturbine. The deviation between calculated and experimental data is within the accuracy of the test measurements. The experimental data processing, with the help of the model, also made it possible to determine numerical values of parameters that cannot be measured directly and to establish how they change with the microturbine output power.

During the tests, measurements of the microturbine parameters were taken with the exhaust backpressure changing from 2.7×10^{-4} to 1.7×10^{-2} atm to determine the dependence of the microturbine output power and efficiency on externally applied backpressure to emulate thermal recovery of waste heat. The model shows that the output power losses (decrease in power output) due to backpressure range from 3.5% for full output to 5.5% for one-third power (lowest value of ~3% occurs at 25 kW), while the efficiency losses (decrease in efficiency) range from 2.5 to 4% (lowest value of ~2.0% occurs at 25 kW), correspondingly.

ACKNOWLEDGMENTS

The authors would like to thank the U.S. Department of Energy (DOE) for supporting this work. This research was conducted by the Oak Ridge National Laboratory (ORNL) under DOE contract DE-AC05-00OR22725 with UT-Battelle, LLC.

NOMENCLATURE

a_1	Multiplier to account for hydraulic losses
AC	Alternating current
Cp_a	Air heat capacity at constant pressure (kJ/kg.K)
Cp_g	Gas heat capacity at constant pressure (kJ/kg.K)
C_v	Heat capacity at constant volume (kJ/kg.K)
DC	Direct current
E	Microturbine efficiency (%)
E_{chp}	BCHP overall system efficiency (%)
G_a	Air flow rate through the air compressor (kg/s)
G_c	Cooling air flow rate (kg/s)
G_g	Gas flow rate through the turbine (kg/s)
G_{ng}	Natural gas flow rate (kg/s)
k_c	Average C_p/C_v in the air compressor
k_{ng}	Average C_p/C_v in the gas compressor
k_t	Average C_p/C_v in the turbine
n	Engine speed (rpm)
P_1	Pressure at the air compressor inlet (atm)

P_2	Pressure at the air compressor outlet (atm)
P_1N	Pressure at the turbine outlet (atm)
P_2N	Pressure at the recuperator outlet (atm)
P_a	Ambient pressure (atm)
P_{ng}	Natural gas pressure (atm)
Q_c	Controller heat loss (kW)
Q_{eg}	Electrical generator heat loss (kW)
$Q_{exhaust}$	Total exhaust energy (kW)
Q_g	Natural gas heat input (kW)
Q_r	Heat transferred in the recuperator (kW)
Q_{ut}	Heat utilized by the heat recovery system (kW)
R	Universal gas constant (8.314 kJ/kgmole.K)
R_r	Recuperation rate
T_1	Temperature at the air compressor inlet (K)
T_2	Temperature at the air compressor outlet (K)
T_2N	Temperature at the end of the adiabatic compression process (K)
T_3	Temperature at the recuperator outlet (K)
T_4	Temperature at the turbine inlet (K)
T_5	Temperature at the turbine outlet (K)
T_5N	Temperature at the end of adiabatic expansion process (K)
T_6	Exhaust gas temperature at the recuperator outlet (K)
T_a	Ambient temperature (K)
T_{vent}	Temperature of exhaust gas vented to the atmosphere (K)
VAC	AC voltage
W_l	Total power losses (kW)
W_c	Air compressor power (kW)
W_f	Total power of the electrical fan motors (kW)
W_{ng}	Gas compressor power (kW)
W_P	Microturbine Output power (kW)
W_t	Turbine power (kW)
Z	Turbine output power linear multiplier for subsonic gas flow
Z1	Turbine efficiency linear multiplier for subsonic gas flow
"	Recuperator heat loss coefficient
$	Coefficient of heat losses due to the non-ideal cycle
) P_1	Hydraulic losses of the BCHP System (atm)
P_1 +) P_1	Gas pressure at the recuperator outlet (atm)
g_c	Compressor pressure ratio
g_t	Expansion rate
0_c	Compressor efficiency
0_{cc}	Combustion chamber efficiency
0_{cn}	Controller efficiency
0_{ng}	Natural gas compressor efficiency
0_r	Recuperator efficiency
0_t	Turbine efficiency
2	Controller heat loss coefficient
$:_a$	Air molecular weight (kg/kgmole)
$:_g$	Exhaust gas molecular weight (kg/kgmole)
$:_{ng}$	Natural gas molecular weight (kg/kgmole)
>	Ratio of expansion rate to compressor pressure ratio (g_t /g_c)

REFERENCES

Banetta S., F. Paganucci, and R. Giglioli. 2001. System Description and Test Planning For a Combined Heat and Power (CHP) Plant Composed by a Micro Gas Turbine and an Absorption Chiller/Heater.

Proceedings of ASME TURBO EXPO 2001, 2001-GT-0102, June 4-7, 2001, New Orleans, Louisiana, pp. 1-5.

Bejan A. 1996. Thermal Design and Optimization. John Wiley and Sons, Inc., New York, New York.

Burghardt D. M. 1982. Engineering Thermodynamic and Applications. Harper and Row Publishers, New York, New York.

EPRI. 1983. Gas Turbine Evaluation (GATE) Computer Program. Research Project AP-2871-CCM, EPRI, Palo Alto, California.

Fairchild P. D., S. Labinov, A. Zaltash, and D. T. Rizy. 2001. Experimental and Theoretical Study of Gas Microturbine–Based BCHP System. Proceedings of the 2001 International ASME International Mechanical Engineering Congress and Exposition, IMECE2001/AES-23622, November 11-16, 2001, New York, New York, pp. 1-12.

MathSoft Inc. 1999. Mathcad User's Guide, Mathcad and Mathcad PLUS 6.0. Cambridge, Massachusetts.

McDonald C. F. 2000. Low Cost Recuperator Concept For Microturbine Applications. Proceedings of ASME TURBO EXPO 2000, 2000-GT-167, May 8-11, 2000, Munich, Germany, pp. 1-9.

ORNL. 2001. Baseline Performance Evaluation of Microturbine Distributed Power Generation (DG) Equipment. ORNL Report, ORNL/TM-2001/276, Oak Ridge National Laboratory, Oak Ridge, Tennessee, January 31, 2001.

Reid R. C., J. M. Prausnitz, and T. K. Sherwood. 1977. The Properties of Gases and Liquids. McGraw-Hill Book Company, New York, New York.

US DOE. 2000. Strategic Plan For Distributed Energy Resources, Office of Energy Efficiency and Renewable Energy and Office of Fossil Energy. U.S. Department of Energy, Washington, D.C., September 2000 (http://www.eren.doe.gov/der/documents_resources.html).

Visual Solutions, Inc. 1999. VisSim User's Guide, Version 4. Westford, Massachusetts.