

A HYBRID SOFC-MICROTURBINE COMBINED-CYCLE SYSTEM:
MODELING, EFFICIENCY EVALUATION AND POWER MANAGEMENT

by

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of the requirements for the degree

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in

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DEDICATION

For Laura

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NOMENCLATURE

Symbol	Description	Units
Cp_i	Specific heat capacity of species i	J/(mol K) or J/(g K)
D_{ij}	Effective binary diffusivity of i - j pair	m ² /s
F	Faraday constant	C/mol
ΔH	Enthalpy change	kJ/mol
$\Delta_f H_i^o$	Enthalpy of formation of species i	kJ/mol
i, I	Current	A
i_{den}	Current density	A/m ²
m_i, M_i	Molar mass of species i	g/mol
$\dot{m}_i$	Mass flow rate of species i	g/s
N_i	Stoichiometric coefficient of species i	-
N_p	Number of parallel connected fuel cells	-
N_s	Number of series connected fuel cells	-
$\dot{n}_i$	Molar flow rate of species i	mol/s
P	Pressure	atm or Pa
p	Partial pressure of species i	atm or Pa
Q	Thermal energy	kW
q	Heat transfer rate	J/s
R	Gas constant	J/(mol K)
T	Temperature	°F or K
T_i	Time constant of device i	s
V	Volume, or voltage	m ³ or V
Wf	Fuel demand signal	pu
w_i	Mass fraction of species i	-
χ_i	Mole fraction of species i	-
η	Efficiency	-

ABSTRACT

As centralized electricity generation and transmission issues continue to complicate electricity demand, interest in distributed generation solutions is increasing. Solid oxide fuel cells are high temperature and efficiency electrochemical devices that can operate on natural gas as well as hydrogen. When in combined cycle operation with a microturbine, the system has the ability to utilize the unused fuel from the solid oxide fuel cell and waste heat to increase the electrical energy, overall efficiency, and feasibility of market penetration of the system. The waste heat can also be repurposed outside the system, known as combined heat and power, for heating residential water supplies. This thesis presents the modeling, efficiency evaluation and power management of a hybrid solid oxide fuel cell/microturbine system in combined cycle operation with combined heat and power functionality for residential applications in islanded and grid-connected modes. The response of the system to load changes is also examined. The dynamic models of the solid oxide fuel cell and microturbine are integrated using power electronic interfacing and simulated in Matlab/Simulink. Simulation results demonstrate an efficiency increase of the system in combined cycle operation and the dynamic behavior of the system in stand-alone operation under different load conditions.

INTRODUCTION

The increased demand for electricity in the United States over the last few decades has led to amplified interest in distributed generation due to disproportional growth in central generation capacity over the same time period. The electrical infrastructure was built, over the past century, by vertically integrated utilities that generated electricity in large capacities near fuel supplies that relied on transmission facilities to transport the electricity to end-users. Interconnections between neighboring utilities were constructed to exchange power for increased reliability and shared excess generation. In 1996, the Federal Energy Regulatory Commission (FERC) issued Orders 888 and 889, which allowed non-utilities, or independent power producers (IPPs), open access to utility transmission systems. Through the deregulation of monopoly owned transmission lines, all transmission users are provided access without discrimination to more competitively priced electricity. As open access to transmission systems continued, the need for regulatory oversight began to build. In 1999, FERC Order 2000 mandated that the transmission owners relinquish control of their systems to regional transmission organizations (RTOs). The Energy Policy Act of 2005 was signed into law, which required compliance of all power system participants to the reliability standards of an electric reliability organization (ERO). In 2006, FERC designated the National Energy Reliability Council (NERC) to be the U.S. ERO. The expansion of IPPs led to the consideration of distributed generation (DG) sources as additional power generation.

DG sources are small in size, relative to the power capacity of the attached system, normally less than 10 MW and modular in structure. Additional local and small

scale power generation allows for optimal placement within distribution systems for increased grid stability, diminished transmission losses, and overall improved system reliability and efficiency. Although DG holds significant promise in the area of infrastructure improvement, impediments still exist that prevent widespread market penetration. Some DG technologies have been employed, others tested, and others still merely exist in research and development. Alongside the lack of implementation for all DG technologies, the issues associated with DG deployment arise. Three major concerns with DG installation are safety, reliability and power quality. If maintenance needs to be performed on a system with a DG attached, the grid power could be disconnected while the DG is not, thus the line would still be live. The requirements of DG disconnection with current electric power systems (EPS) have been addressed by IEEE Standard 1547 [1], determining disconnection from the grid within two seconds for islanded DGs. A DG is said to be islanded if the grid power is disconnected while the DGs maintain a connection to the subsystem. The reliability of variable generation sources such as wind or solar photovoltaic (PV) provide operational challenges of meeting power demand. However, generation sources such as fuel cells and gas turbines depend on a continuous fuel source and can thereby maintain a constant output power. These DG sources require power electronics interfacing, namely dc/dc and dc/ac converters, have the potential to inject imperfect-sinusoidal current into the grid, creating distortion. Filtering techniques must be employed to eliminate any harmonics generated by the DG to prevent any operational issues from occurring.

Another issue with DG deployment involves the interconnection and interaction with the existing grid. DG sources can operate in stand-alone, grid-connected, and islanded modes. Stand-alone operation is independent of the grid and as such must provide voltage and frequency regulation within the system [2]. Stand-alone operation is typically used for remote and mobile applications to supply electricity. In grid-connected operation, the DG sources are connected to the grid and operate as an additional source of energy. Should the DG sources fail the end-user can rely upon the utility system to provide electricity. Conversely, if the utility system should fail the end-user can be disconnected from the grid and rely on the DG sources. This connection scheme is referred to as islanded operation. In this sense, the islanded subsystem becomes a small scale version of the centralized grid, and is aptly referred to as a microgrid. Broadening this definition, a power subsystem that contains electricity generation, storage, and loads and can be operated in islanded mode is considered to be a microgrid.

As interest in small-scale DG continues to increase, so too will the research on optimal operation of DG sources. The complementary technologies of DGs have led to generation sources with similar capabilities along with the added benefits of increased efficiency, reliability, and the potential of decreased costs. In particular, the coupling of a fuel cell and small scale gas turbine has shown the most promise in the area of base generation for DG deployment. Relying on DGs for dispatchable power generation due to their fast ramp up and ramp down characteristics has also increased their popularity. Any generation source that has the ability to be turned on or off upon demand, or

dispatched, at the request of a power grid operator, typically within a few minutes, is known as a dispatchable generation source. Dispatchable generation sources are commonly utilized by power grid operators for voltage and frequency stabilization of the grid. Traditional base load generation sources, such as coal-based or nuclear power plants, which generate constant power at a consistent level, have a limited range of power output and require large time spans to differ their power within said range. On the other hand, renewable energy sources such as PV or wind power are intermittent and unable to provide reliable or readily available dispatchable power. Although energy storage techniques have been used to resolve the intermittency of PV and wind power, the storage methods require additional equipment, maintenance, and cost. A solid oxide fuel cell (SOFC) and microturbine (MT) system, on the other hand, has the ability to provide constant power at variable levels with fast responses to demand and is a robust option for less carbon emitting and more efficient dispatchable power generation.

Current State of Hybrid Fuel Cell/Gas-turbine Systems

Solid oxide fuel cells (SOFCs) are energy conversion devices that convert the chemical energy of a fuel into electrical energy [3]-[5]. SOFCs high operating temperatures of 600-1000°C allow for internal fuel reforming within the electrode gas channels, which facilitates the use of a variety of fuels including hydrogen, natural gas, and hydrocarbon gases. Microturbines (MTs) are small scale gas turbines that typically operate with a variety of gaseous or liquid fuels at nameplate capacities between 10 and 400 kW of electricity [6], [7]. MTs are also capable of operating at high and low

pressure levels. MTs have a short start-up time and thus are valuable when used for peak power generation for grid support [8].

A consideration for a DG source is their ability to handle transients as a measure of system reliability and power quality. The SOFC transient response is slow, on the order of minutes, due to the electrochemical reactions taking place [9], [10]. As a result, SOFCs are normally coupled with a device capable of fast response to transient occurrences to prevent long term damage to the SOFC and maintain frequency stabilization [11], [12]. The utilization of the unused fuel and thermal energy of the SOFC can provide additional electrical energy or efficiency through the implementation of a gas turbine, or a combined cycle or cogeneration operation [13].

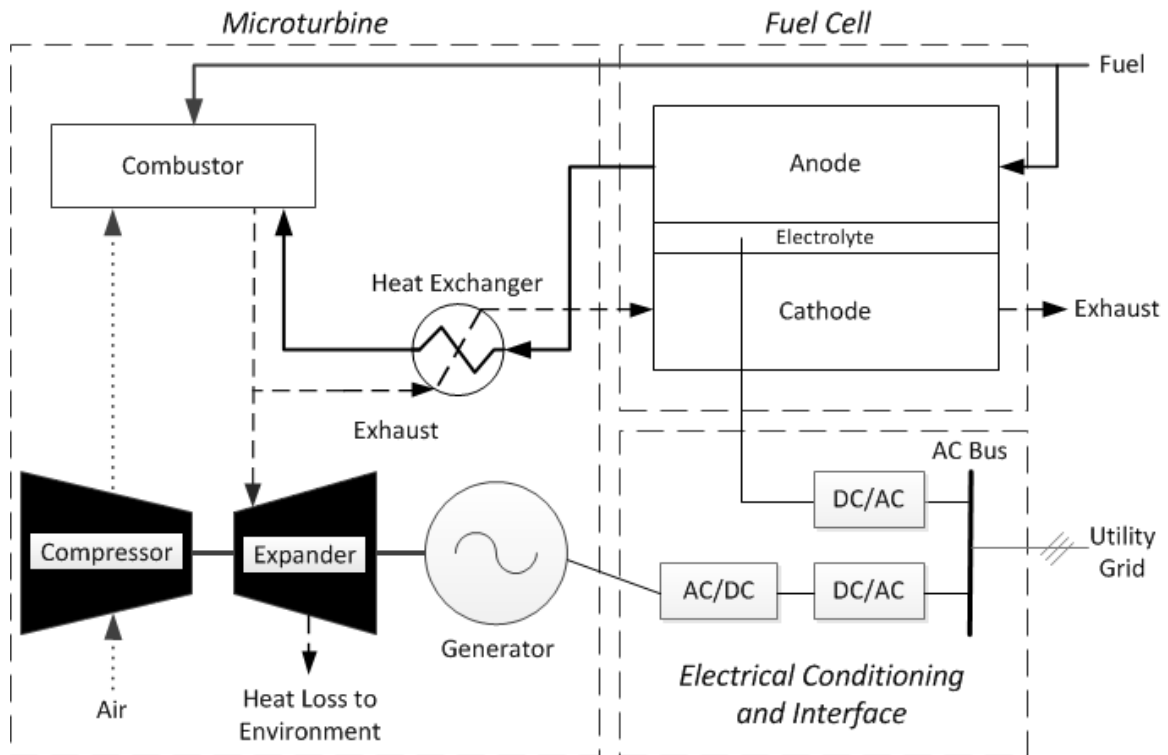


Fig. 1.1 Combined cycle operation of SOFC-MT system.

Combined cycle (CC) operation refers to the coupling of heat streams for purposes within the system itself. For the research of this thesis, CC operations refer to the coupling of the MT exhaust gas stream and the unused SOFC fuel stream from the anode channel, as shown in Fig.1.1. Such an implementation of CC operation is typical in SOFC/MT systems, utilizing the unused exhaust from the MT in a heat recovery operation to preheat the fuel into the MT to increase efficiency. CC operation will be discussed in further detail in Chapter 3.

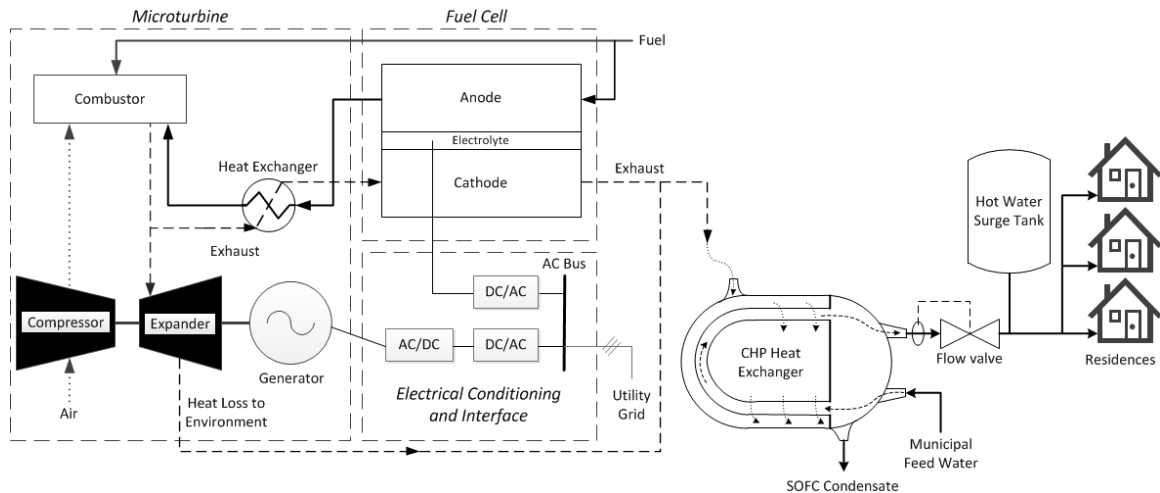


Fig. 1.2 Configuration of SOFC-MT-CC system in CHP operation for residential hot water production.

Combined heat and power (CHP), or cogeneration, systems combine on-site power generation with the use of byproduct heat. For the system presented within this research, the CHP operation utilizes the exhaust gas from the SOFC to aid in residential or commercial heating, in particular, residential hot water production, while generating electricity, shown in Fig. 1.2. CHP operation will be discussed in further detail in Chapter 3.

Hybrid SOFC-MT systems benefit from combined cycle operation by increasing the efficiency and transient capabilities of the system along with decreasing fuel costs and emissions [14]. Efficiencies for hybrid SOFC-MT configurations have been shown to be greater than 60% [12], [15]. Whereas traditional coal-fired and natural gas-combustion power plants in the United States operate with efficiency ranges from 20-38% and 30-50%, respectively [16]-[18]. SOFC-MT hybrids have a great potential to function as effective electricity sources for the future, while more efficient fuels, i.e. hydrogen, can subsequently be integrated into the SOFC-MT installations, further improving generation efficiency and lowering associated costs. The hybrid SOFC-MT generation systems of today can be viewed as an intermediary generation source providing a cleaner and more efficient alternative to traditional generation methods until a robust hydrogen supply and distribution economy has been implemented.

Currently, the research on hybrid renewable systems involving fuel cells, PV, wind power, and microturbines is growing in the area of distributed generation, due to increased interest in cleaner and cheaper alternative methods of generation, along with a proliferation of new technologies. The modeling of such a system consisting of a robust thermodynamic-based SOFC plant with a machine and thermodynamic-based microturbine, however, has not. Hybrid SOFC-MT systems have been researched with a transfer function model [19] or thermodynamic model [12], [20], [21], of a gas turbine but an integrated model has yet to be developed to our knowledge. Machine, or transfer function, based models of gas turbines are readily available and commonly used [22]. Thermodynamic based models for gas turbines have also been exhaustively investigated

over the past 20 years [23]. For the lack of a combined model, the interconnection of a transfer function and thermodynamic microturbine model is presented within this thesis. The ability to analyze efficiency of a system based on thermodynamic modeling, e.g. reduction of fuel, and transfer function modeling, e.g. power generation and management, led to this research.

Thesis Organization

The research described in this thesis seeks to concentrate on the modeling, efficiency evaluation, and power management of a hybrid SOFC-MT system in combined cycle (CC) operation with combined heat and power (CHP) functionality for residential applications in islanded or grid-connected modes within a microgrid environment. Chapter two presents the operation, modeling, and simulation of the SOFC, MT, combustor, and heat exchanger components of the system. Chapter three provides the operation and efficiency evaluation of a hybrid SOFC-MT-CC system. Chapter four presents the power management of the system during transient occurrences. The thesis concludes in chapter five with a discussion of the results and recommendations for future work.

SYSTEM OPERATION & MODELS

Solid Oxide Fuel Cell

Fuel cells are energy conversion devices that convert the chemical energy of a fuel into electrical energy [4]. Solid oxide fuel cells (SOFCs) high operating temperatures of 600-1000°C allow for fuel reforming within the electrode gas channels, which facilitates the use of a variety of fuels including hydrogen, natural gas, and hydrocarbon gases.

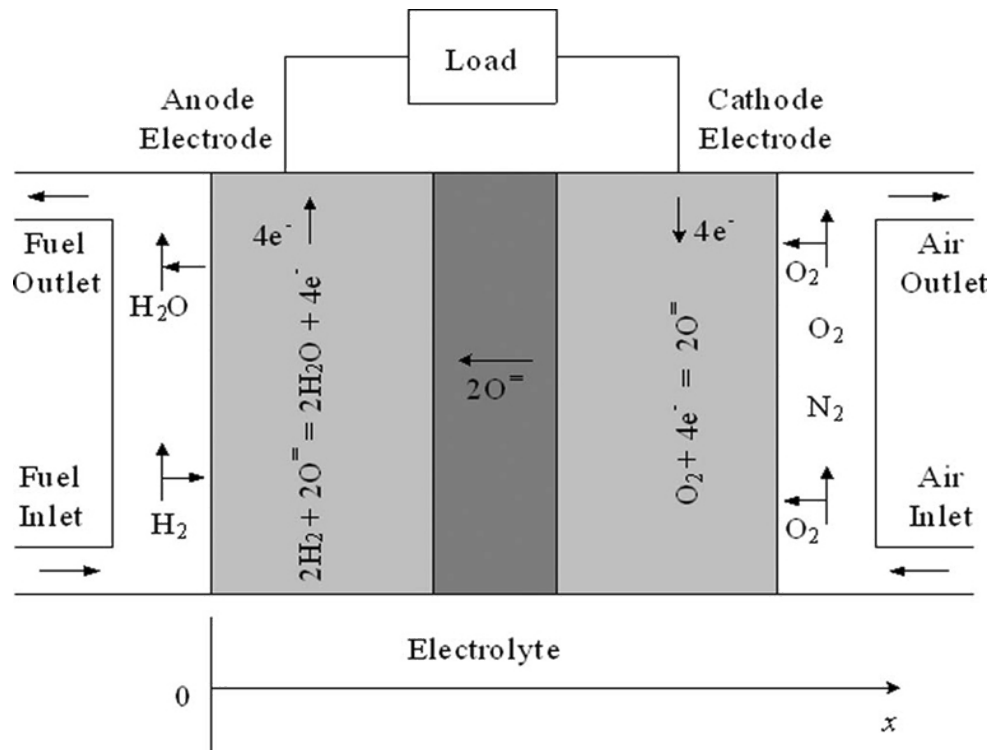
SOFC Operation

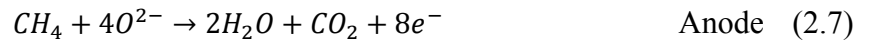
Fig. 2.3 Schematic diagram of a SOFC with H₂ as fuel [4]

SOFCS operate with a 45-65% efficiency range, and up to 80% in hybrid mode with CHP systems [13], [24]. The high efficiency of SOFCs is one reason why they show promise for distributed generation applications [11]. A schematic of a SOFC is shown in Fig. 2.3 depicting pure hydrogen and atmospheric air as reagents.

At the cathode, oxygen reacts with electrons to form oxygen ions (2.1). The oxygen ions flow through the electrolyte from cathode to anode. At the anode, the oxygen ions react with hydrogen to form H_2O and excess electrons travel to the external load (2.2), resulting in an overall reaction producing H_2O (2.3).



The chemical reactions when fuel other than hydrogen, such as natural gas (CH_4) is utilized are fundamentally the same with additional intermediate steps, shown in Eq. (2.4)-(2.7).



Since SOFCs are solid-state devices that may employ a ceramic material as the electrolyte, corrosion and management problems can be significantly curtailed. As compared to other high operating temperature fuel cells, e.g. molten carbonate fuel cells

(MCFCs), which endure higher manufacturing costs and standards due to their electrolytic material [5].

SOFC Model

A dynamic model for a 5 kW tubular SOFC stack, reported in [4], has been used in this study for the efficiency evaluation of the SOFC-MT system. The model was developed based on SOFC electrochemical, thermodynamic, and material diffusion properties, along with the mass and energy conservation laws with emphasis on the fuel cell terminal electrical measurements.

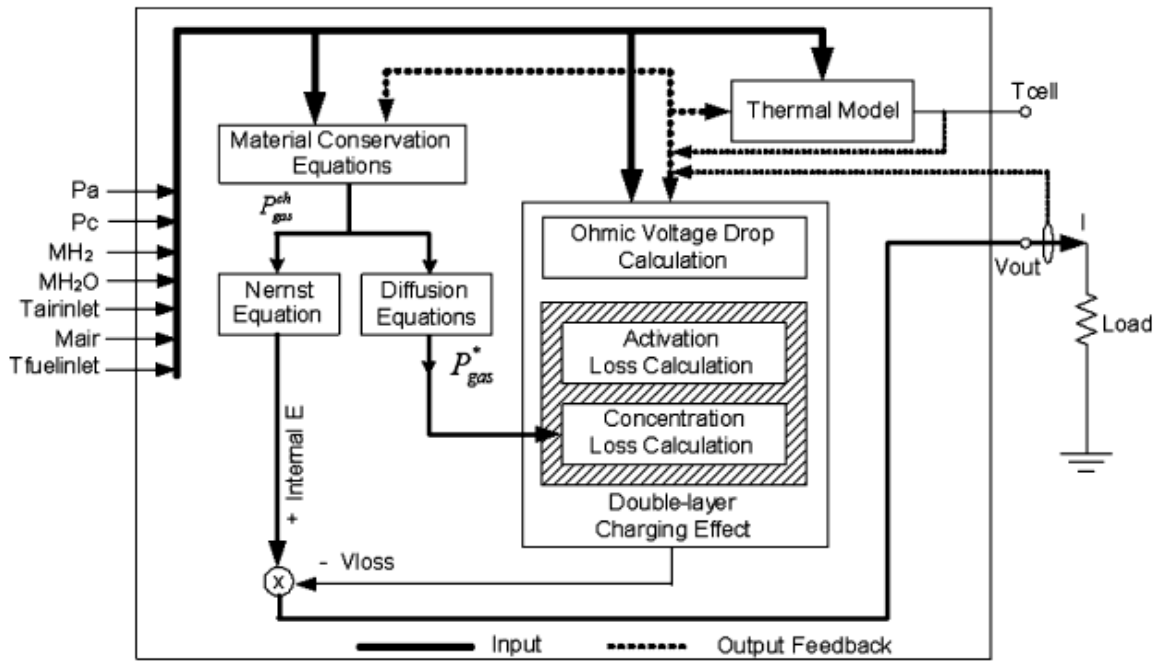


Fig. 2.4 Block diagram for building a dynamic model of SOFC [4].

Fig. 2.4 shows the block diagram of the dynamic SOFC model reported in [4]. The input quantities are anode and cathode pressures (P_a and P_c), hydrogen flow rate (M_{H_2}), water vapor flow rate (M_{H_2O}), air flow rate (M_{air}), and initial temperatures of air

($T_{air\ inlet}$) and fuel ($T_{fuel\ inlet}$) of the fuel cell cathode and anode channels, respectively.

At any given load current and time, the cell temperature T_{cell} is determined and both the current and temperature are fed back to different blocks, which take part in the calculation of the SOFC output voltage. The output quantities are the fuel cell voltage and cell temperature.

The terminal voltage versus current ($V-I$) and power versus current ($P-I$) curves of the SOFC model, obtained from the dynamic model, are shown in Fig. 2.5 [4]. The operating conditions of pressure, temperature, air/fuel mixture, and flow rates are specified within the figure.

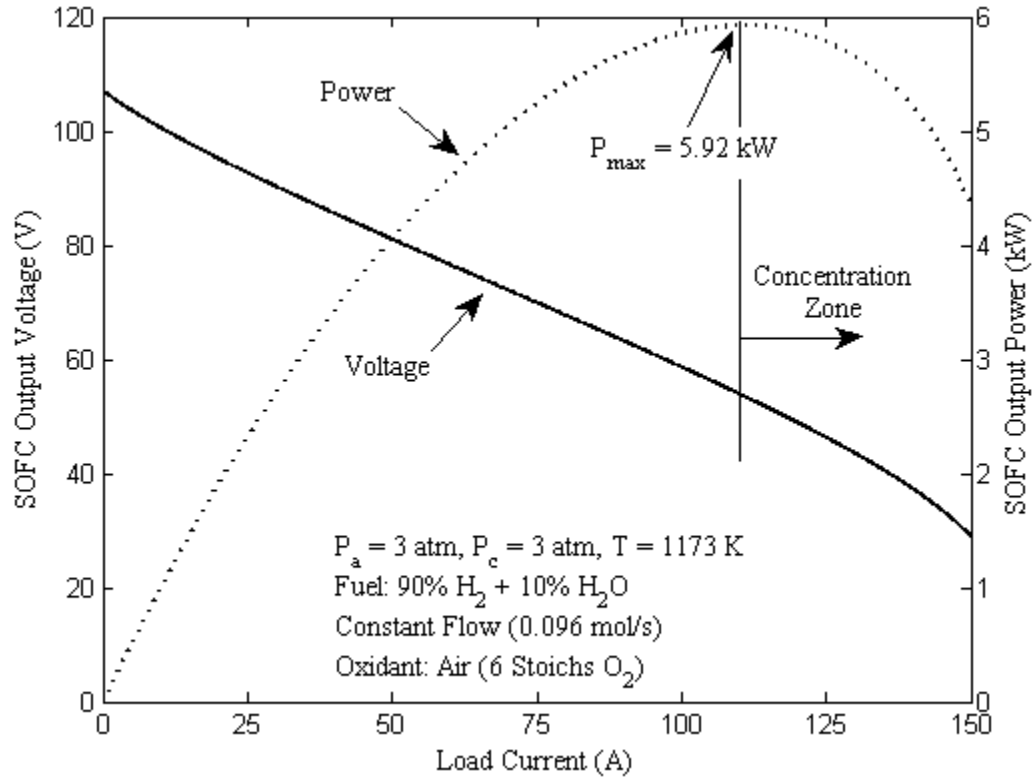


Fig. 2.5 $V-I$ and $P-I$ characteristics of SOFC model reported in [25].

The activation voltage drop dominates the voltage loss in the low-current region. As the load current increases, the Ohmic voltage drop increases quickly and is the chief contributor to the SOFC voltage reduction. When the load current exceeds a certain value (110 A for the 5 kW model under the given operating conditions), the fuel cell output voltage will decrease abruptly due to the concentration voltage drop inside the fuel cell. The concentration voltage drop is a function of current, pressure and temperature. The effective partial pressures of H_2 and O_2 are less than those in the electrode channels while the effective partial pressure of H_2O is higher than that in the anode channel. Thus, the internal voltage of the fuel cell is less than the calculated value. The voltage difference is referred to as the concentration voltage drop [4].

Model Configuration

The 5 kW stack model consists of 96 individual fuel cells used to generate the nominal rating. For the purposes of the research in this thesis, the SOFC has a nominal voltage of 220 V and rated power of 480 kW. To achieve the desired power rating, the 5 kW stack can be interconnected and combined to increase the power. Since the maximum power is the preferred operating point of each 5 kW stack, it is noted from Fig. 2.3 that the current and voltage corresponding to the maximum power is approximately 110 A at 54 Vdc. However, in order to leave a safe margin so the SOFC stack does not enter the concentration zone, the operating point is chosen to be 100 A at an output voltage of 55 Vdc.

A 20 kW module of four 5 kW stacks in series is formed with a nominal output voltage of 220 Vdc, found by Eq. (2.8) [4].

$$N_s = \frac{V_{\text{SOFC array}}}{V_{\text{SOFC stack}}} = \frac{220 \text{ Vdc}}{55 \text{ Vdc}} = 4 \quad (2.8)$$

The 20 kW module, consisting of four 5 kW SOFC stacks, can be connected in two parallel units to form a 40 kW array, found by Eq. (2.9) [4].

$$N_p = \frac{P_{\text{array}}}{N_s \times P_{\text{stack}}} = \frac{40 \text{ kW}}{4 \cdot 5 \text{ kW}} = 2 \quad (2.9)$$

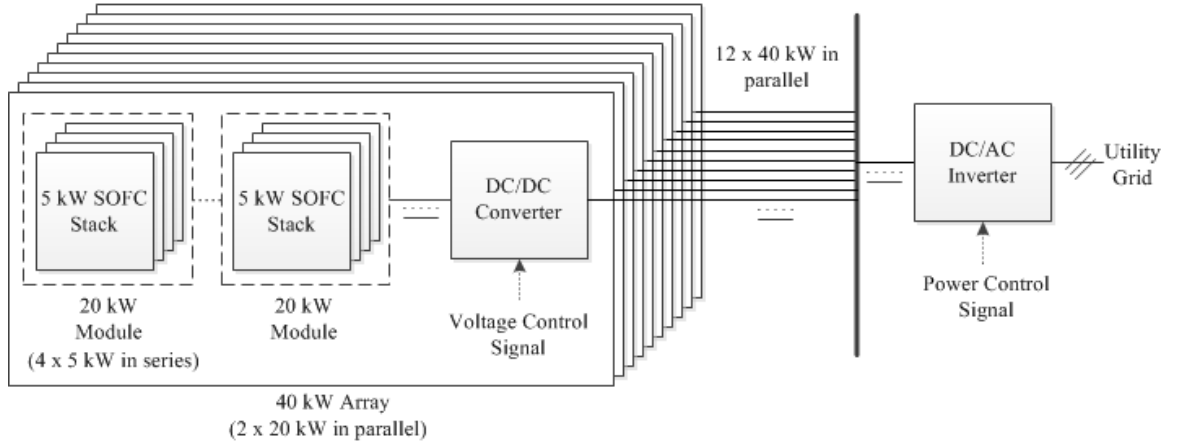


Fig. 2.6 SOFC system configuration diagram.

Each 40 kW SOFC array is connected to a common dc bus through individual boost dc/dc converters based on a 5.0 kHz switching frequency for insulated gate bipolar transistor (IGBT) electronic switches. The 40 kW array output voltage of 220 Vdc is boosted to a dc bus voltage of 480 V through the use of the dc/dc converters. In order to achieve a nominal power rating of 480 kW, a total of twelve 40 kW SOFC modules are placed in parallel. The complete SOFC system configuration is shown in Fig. 2.6.

Additional details on the 5 kW SOFC model development can be found in [4]. Further details on the module system configuration can be found in [26].

Anode & Cathode Channel Fuel Flow

The steady-state response of the dynamic SOFC model is similar to that reported in [4] and has been validated by operating data from industry sources. In order to account for the dynamic behavior of a CC system, the quantity calculations of unused fuel from the anode channel of the SOFC needed to be implemented. Assuming full availability of hydrogen fuel and large stoichiometric quantities of water vapor and oxygen at the cathode, the mass flow rate for H_2 , H_2O and O_2 at the inlet and outlet of the anode and cathode flow channels can be expressed as follows [4]:

$$M_{H_2}^{in} = M_a \cdot \chi_{H_2}^{in} = M_a \cdot \frac{p_{H_2}^{in}}{P_a^{ch}} \quad (2.10a)$$

$$M_{H_2}^{out} = M_a \cdot \chi_{H_2}^{out} = M_a \cdot \frac{p_{H_2}^{out}}{P_a^{ch}} \quad (2.10b)$$

$$M_{H_2O}^{in} = M_a \cdot \chi_{H_2O}^{in} = M_a \cdot \frac{p_{H_2O}^{in}}{P_a^{ch}} \quad (2.11a)$$

$$M_{H_2O}^{out} = M_a \cdot \chi_{H_2O}^{out} = M_a \cdot \frac{p_{H_2O}^{out}}{P_a^{ch}} \quad (2.11b)$$

$$M_{O_2}^{in} = M_c \cdot \chi_{O_2}^{in} = M_c \cdot \frac{p_{O_2}^{in}}{P_c^{ch}} \quad (2.12a)$$

$$M_{O_2}^{out} = M_c \cdot \chi_{O_2}^{out} = M_c \cdot \frac{p_{O_2}^{out}}{P_c^{ch}} \quad (2.12b)$$

Where M_i is the molar flow rate of species i (mol/s), M_a and M_c are the molar flow rates of the anode and cathode channels, respectively (mol/s), χ_i is the mole fraction of species i , p_i is the partial pressure of species i (Pa), and P_a and P_c are the overall pressures of the gas mixture in the anode and cathode channels, respectively. The differential equations for the partial pressure of H_2 and H_2O can be written as follows [4]:

$$\frac{dp_{H_2}^{ch}}{dt} = \left(\frac{2n_a RT}{V_a P_a^{ch}} \right) (p_{H_2}^{in} - p_{H_2}^{ch}) - \frac{RT}{2FV_a} i \quad (2.13a)$$

$$\frac{dp_{\text{H}_2\text{O}}^{\text{ch}}}{dt} = \left(\frac{2n_a RT}{V_a p_a^{\text{ch}}} \right) (p_{\text{H}_2\text{O}}^{\text{in}} - p_{\text{H}_2\text{O}}^{\text{ch}}) + \frac{RT}{2FV_a} i \quad (2.13b)$$

Where R is the gas constant 8.3143 J/(mol K), T the gas temperature (K), V_a the volume of the anode channel (m^3), and F is Faraday's constant (96487 C/mol).

The molar flow rate of H_2 and H_2O out of the anode channel, Eq. (2.10b) and (2.11b), were implemented in Matlab/Simulink within the existing SOFC model and is a part of the research presented within this thesis.

For further details concerning the development of the thermodynamic modeling of the SOFC, the reader is referred to [4].

SOFC Simulation

A simulation of the SOFC model was conducted for the purpose of verifying the implemented equations of unused fuel from the anode channel. A simulation of the 5 kW SOFC stack model was conducted over the range of rated input load currents from 0 to 150 A. The simulation parameters for a 5 kW stack model are shown in Table 2.1.

Table 2.1 Simulation Parameters of 5 kW SOFC Stack Model [4]

Parameter	Value
Fuel flow	0.096 mol/s (90% H_2 + 10% H_2O)
Air flow	0.012 mol/s
Pressures (anode & cathode)	3 atm
Initial fuel & air temperature	1173 K

The mass flow rate and the partial pressure of H_2 at the outlet of the anode channel, found via Eq. (2.10a) and (2.13a), is shown to exhibit a correlation. With a constant mass flow rate of H_2 at the inlet of the anode channel over the range of operating

currents of the SOFC model, the partial pressure of H_2 decreases causing a decrease in unused H_2 , as shown in Fig. 2.7.

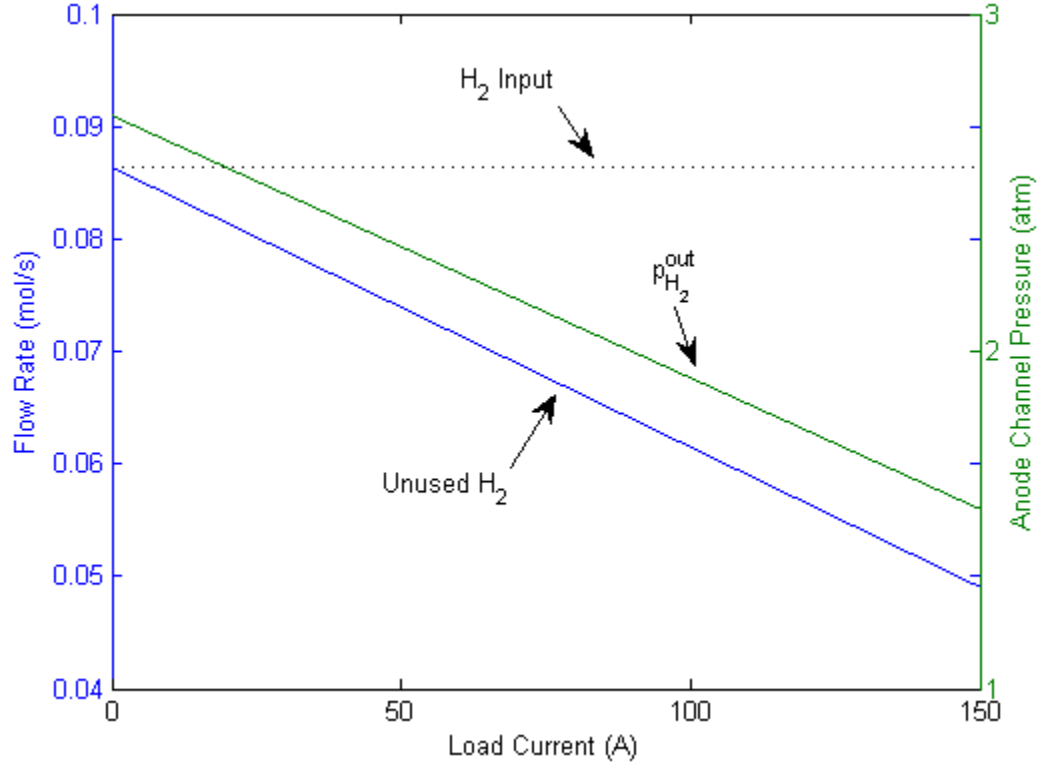


Fig. 2.7 Unused and partial pressure of H_2 in anode channel with constant rate of input fuel, H_2 .

As the load current increases within the SOFC the partial pressure of H_2 at the outlet of the anode channel decreases in a linear fashion. The decrease in pressure is due to the definition of the rate of change of partial pressure of H_2 with respect to axis x , given as follows [4]:

$$\frac{dp_{H_2}}{dx} = -\frac{RT}{D_{H_2,H_2O}} \frac{i_{den}}{2F} \quad (2.14)$$

Where $D_{i,j}$ is the effective binary diffusivity of i - j pair of gas species (m^2/s) and i_{den} the current density (A/m^2). For any current density, or load current, greater than

zero, the rate of change of partial pressure of H_2 will be negative. As the load current increases up to the maximum value of 150 A the partial pressure of H_2 will continue to decrease, indicating a decrease in unused H_2 , as shown in Fig. 2.7.

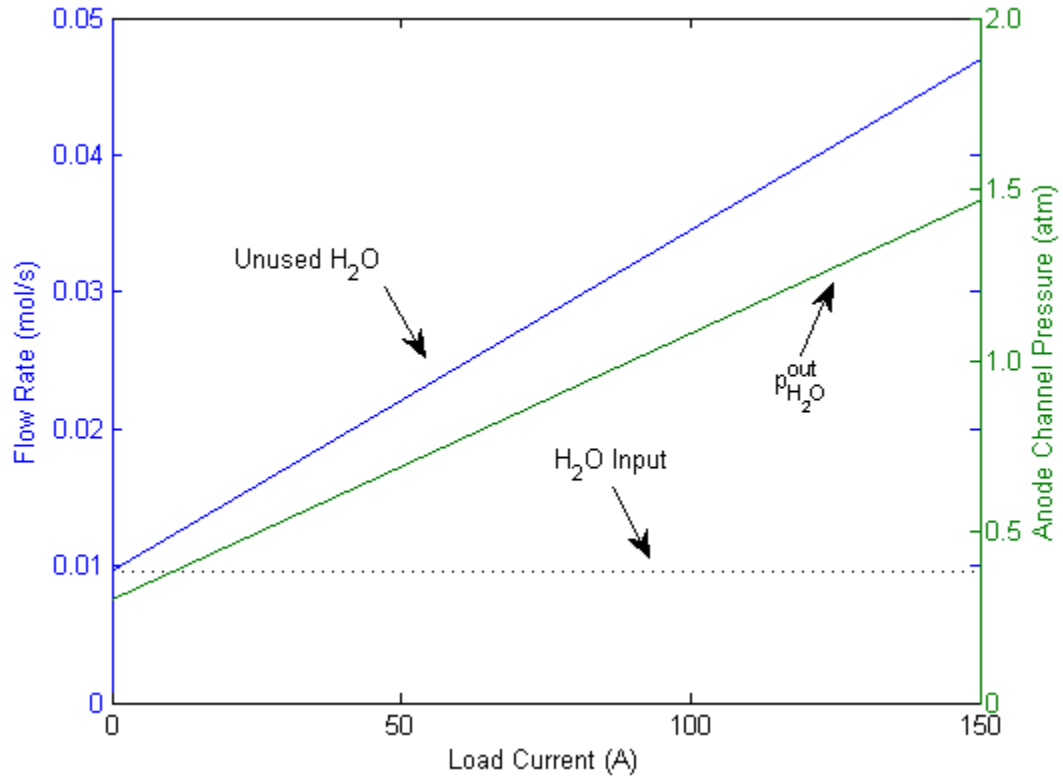


Fig. 2.8 Unused and partial pressure of H_2O in anode channel with constant rate of input water vapor, H_2O .

The gas at the anode channel is a mixture of H_2 and H_2O , as indicated by Fig. 2.1; hence, the partial pressure of H_2O is examined as well. The mass flow rate and the partial pressure of H_2O at the outlet of the anode channel exhibits a correlation with a constant mass flow rate of H_2O at the inlet of the anode channel over the range of operating currents of the SOFC model, as shown in Fig. 2.8.

In a similar fashion as H_2 , during the increase of load current within the SOFC the partial pressure of H_2O at the outlet of the anode channel increases in a linear fashion. The increase in pressure is due to the definition of the rate of change of partial pressure of H_2O with respect to axis x , given as follows [4]:

$$\frac{dp_{H_2O}}{dx} = \frac{RT}{D_{H_2,H_2O}} \frac{i_{den}}{2F} \quad (2.15)$$

For a current density greater than zero, the rate of change of partial pressure of H_2O will be positive, indicating more unused H_2O as the load current increases, as verified by simulation.

The SOFC was also considered for a steady state operation case at a nominal current of 100 A and rated power of 5 kW; therefore the load current was considered to ramp up to a constant operating point set to the nominal rating of 100 A at 55 Vdc. The results of the simulation are shown in the following figures.

Exhibiting the same startup characteristics as previously shown, the amount of unused H_2 along with the partial pressure of H_2 in the anode channel decrease as the load current is increased. Once the SOFC reaches the operating point of 100 A at 55 Vdc, the mass flow rate and the partial pressure of H_2 at the outlet of the anode channel remain constant, as shown in Fig. 2.9. The molar flow rate and the overall pressure of the gas mixture in the anode channel, P_a , remain constant throughout the simulation, regardless of load current. When the load current is constant, the partial pressure of H_2 of the anode channel is constant due to the rate of change equal to zero, as found by Eq. (2.13a).

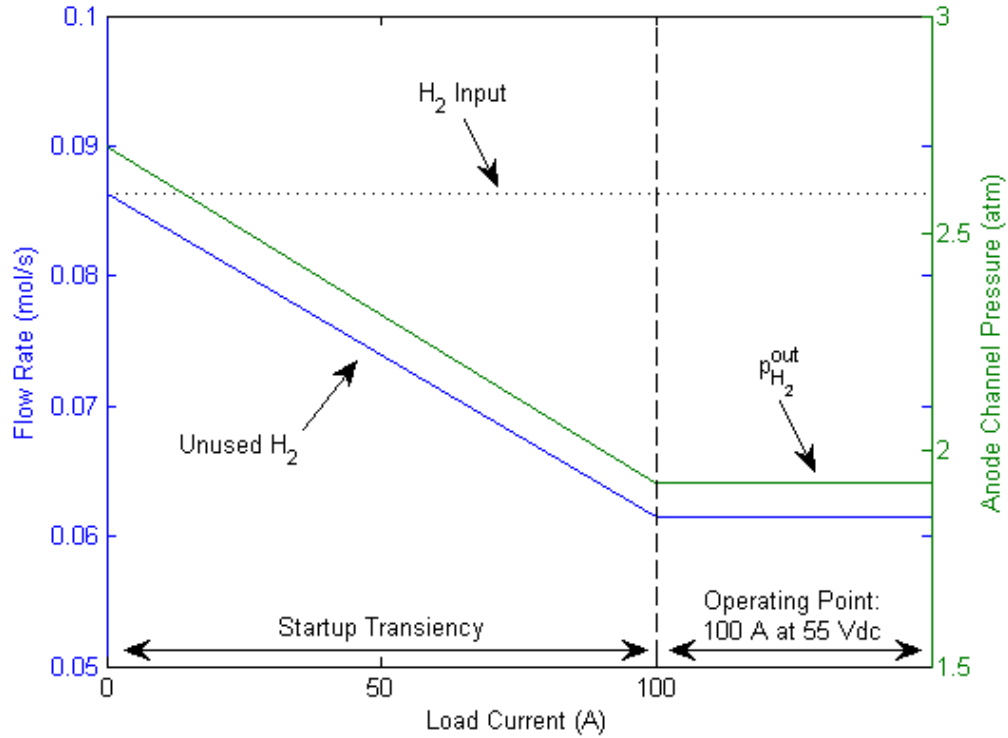


Fig. 2.9 Unused and partial pressure of H₂ in anode channel with constant rate of input fuel, H₂ at nominal rating of 100 A at 55 Vdc.

In a similar fashion, the same startup characteristics of the SOFC are demonstrated when examining the usage of H₂O; namely, the amount of unused H₂O and the partial pressure of H₂O in the anode channel increase as the load current increases. Based on the results of unused H₂ and the implementation of Eq. (2.13a), once the SOFC reaches the operating point of 100 A at 55 Vdc, the mass flow rate and the partial pressure of H₂O at the outlet of the anode channel remain constant, as expected and as shown in Fig. 2.10.

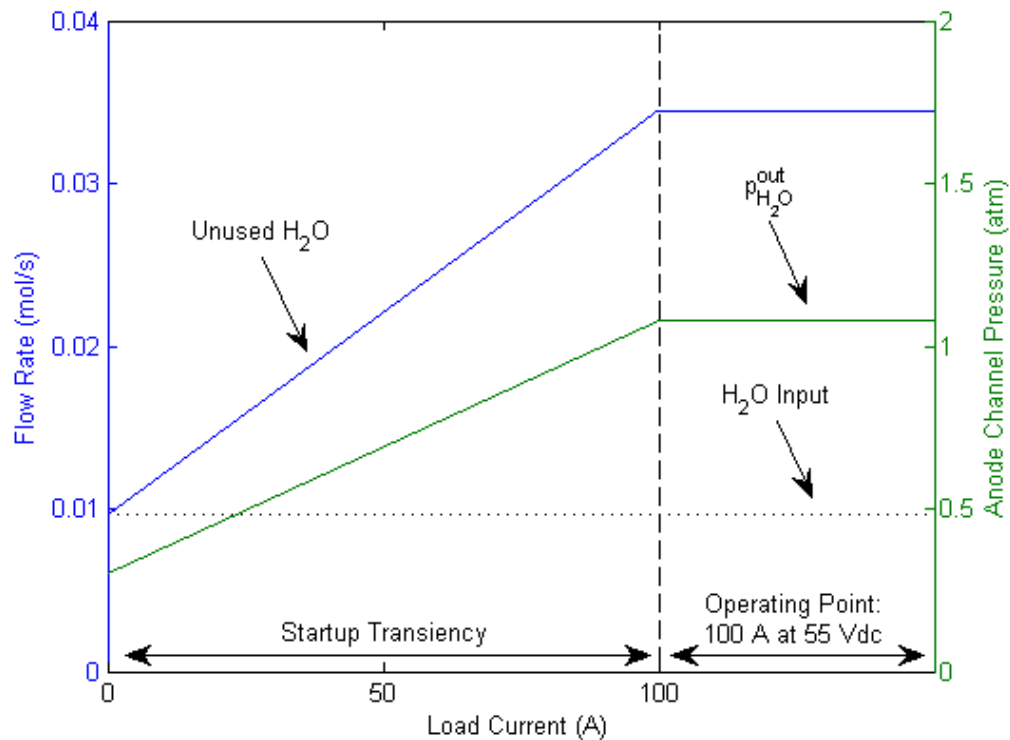


Fig. 2.10 Unused and partial pressure of H₂O in anode channel with constant rate of input at nominal rating of 100 A at 55 Vdc.

Microturbine

Microturbines (MT) are small gas turbines that have the ability to function with a variety of fuels at high and low pressures, including, but not limited to natural gas, propane and biogas, at nameplate capacities between 10 and 400 kW of electricity [27].

Microturbine Operation

A basic gas turbine driving a generator is shown in Fig. 2.11. Fresh air is drawn into a compressor where spinning rotor blades compress the air, increasing the temperature and pressure. The hot pressurized air is mixed with fuel and burned in the combustor. The combustion exhaust are expanded in a turbine and released to the

atmosphere. The compressor and expander are connected by a single shaft so that a portion of the rotational energy of the turbine powers the compressor [27].

Microturbines operate based on the thermodynamic process known as the Brayton cycle, shown in Fig. 2.12 [7], [28].

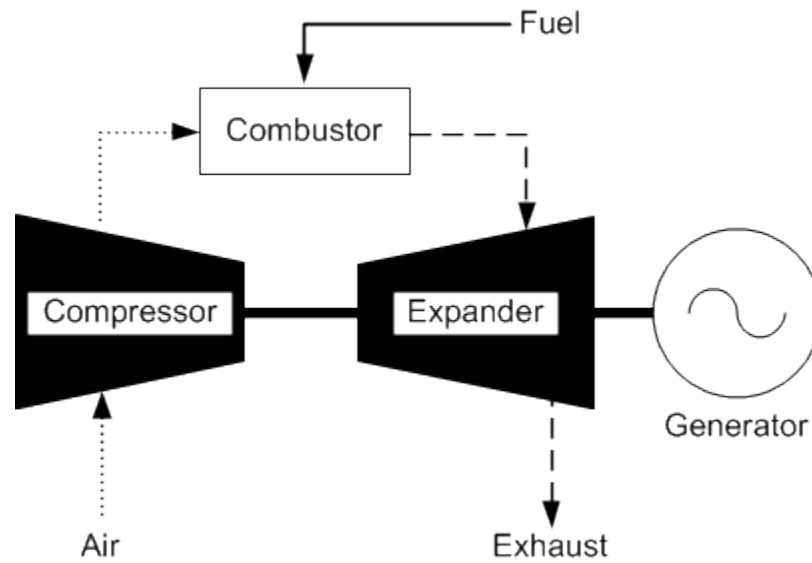


Fig. 2.11 Basic operation flow diagram of a gas turbine.

In this cycle, the inlet air is pressurized in a compressor, from state 1 to state 2, to which fuel is mixed in the combustor and burned, from state 2 to state 3. The heated combustion gas is expanded in the turbine section, from state 3 to state 4. The cycle is completed when the energy is released from the system, from state 4 to state 1, in the case of this research, utilizing the heated combustion gas to produce rotating mechanical power to drive the compressor and an electric generator, mounted on the same shaft. The Brayton cycle is also depicted with flow streams and components to describe the basic operation of a gas turbine, shown in Fig. 2.12.

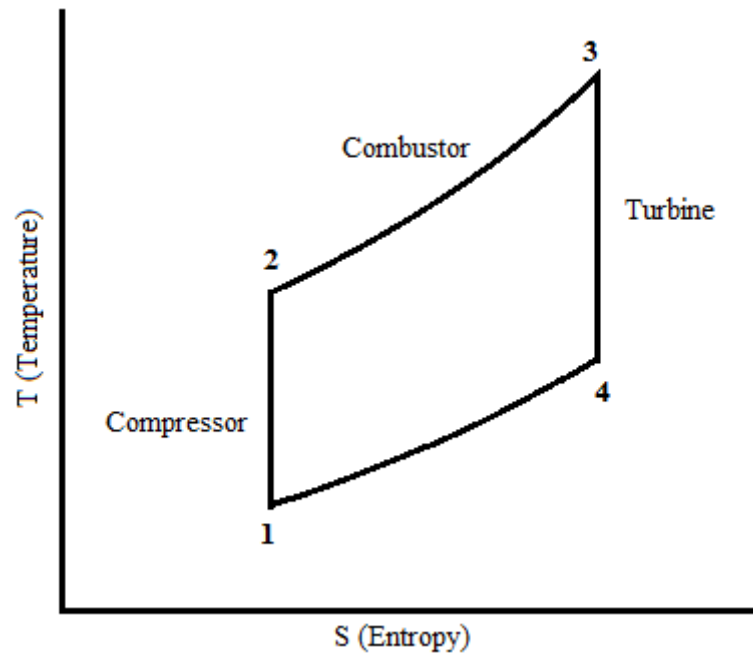


Fig. 2.12 Ideal Brayton cycle T-s diagram.

The stand-alone efficiency of a MT is 15 to 17% [7]. Such a low efficiency can be increased to between 33 and 37% utilizing a heat exchanger, or recuperator within the system [6].

Types of Microturbines

There are multiple designs and implementations of microturbines, as one would expect to find. The two most common are a single-shaft design, as shown in Fig. 2.9 and used for research in this thesis, and a double-shaft design. The double-shaft design operates under the same processes of a single-shaft design, except the generator and power turbine are mounted on different shafts, as shown in Fig. 2.13.

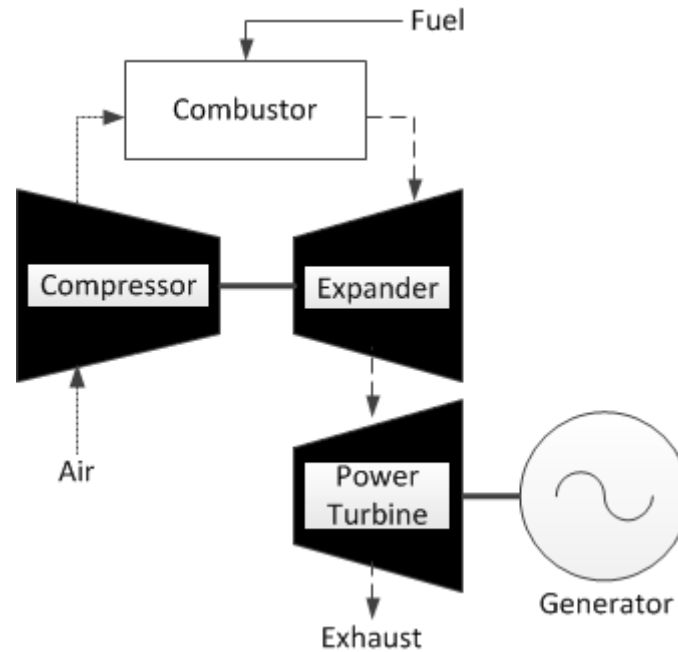


Fig. 2.13 Basic operation flow diagram of a double shaft gas turbine.

The advantages to using a multiple shaft gas turbine may include higher thermal efficiency and operating pressure ratios, and a greater operating speed range. However, some disadvantages include, but are not limited to, being less tolerant of transients, an increase of noise and vibration and potential for mechanical failure [29].

The research presented in this thesis focuses on the usage of a single-shaft MT model. Single-shaft MTs are generally used for generator applications due to their fast response for load following applications. Double-shaft MTs are largely used for mechanical drive and compressor applications due to their ability to simultaneously operate with different pressure stages in the expander and power turbine, which allows for different rotational shaft speeds. The single-shaft design of the MT was selected due to the application of the research presented in this thesis, along with the potential to not require gearboxes, lubricants, coolants, or pumps, the potential for simpler installation,

higher reliability, reduced noise and vibration, lower maintenance requirements, lower emissions, and continuous combustion [6]. In conjunction with the advantages listed, the single-shaft design was chosen due to the prior validation and usage of the single shaft model for distributed generation applications [6], [22].

Operational Speed and Frequency of MT

Depending on the output capacity of the MT, the range of rotational speeds varies from 50,000 to 120,000 rpm, and as a result high operational frequency, which must be rectified to DC and inverted to 50 or 60 Hz, contingent to the usage [30]. The power electronics necessary for the rectification and inversion of frequency within the system have been previously modeled and verified [6], [31]. The implementation of the power electronics in [6] and [31] have been modified for the purpose of the research presented in this thesis to include islanded and grid-connected operations within a microgrid environment.

Microturbine Model Overview

A transfer function model for a per unit gas turbine, developed by [32], along with a thermodynamic model developed by the author has been used for this research. The necessity of integrating the two model types is explained in further detail in the following sections.

Transfer Function Model

The mathematical representation of a microturbine consists of a single-shaft design, generator driven gas turbine model that includes speed control, temperature

control, and a fuel system. This model has been successfully implemented for DG simulation applications [14], [33].

The microturbine is controlled by speed control under partial load conditions and temperature control functions. There exists an inherent acceleration control within the speed control function to prevent runaway speeds of the shaft. The output of these control functions are inputs to a least value gate (LVG), such that the lowest value of shaft speed, acceleration, and the MT's internal temperature is output. The output of the LVG is input to the fuel system block, resulting in the minimal usage of fuel provided to the compressor/combustor/turbine block, shown in Fig. 2.14 [6], [32].

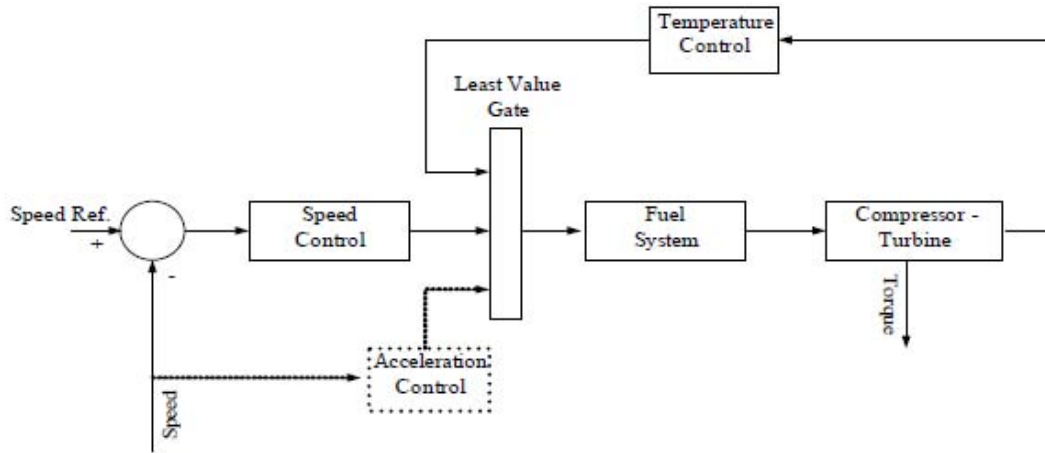


Fig. 2.14 Block diagram of microturbine model with control systems [6].

Speed Control. The speed control block calculates the error between the reference speed of one per-unit and the MTG shaft speed. The modeling of speed control is typically achieved through the implementation of a lead-lag transfer function [32], or by a PID controller [15]. For the purposes of this research, a lead-lag transfer function

was chosen to model the speed controller, shown in Eq. (2.16). The complete speed controller is shown in Fig. 2.15.

$$\text{Lead/Lag} = \frac{K(T_1s + 1)}{T_2s + Z} \quad (2.16)$$

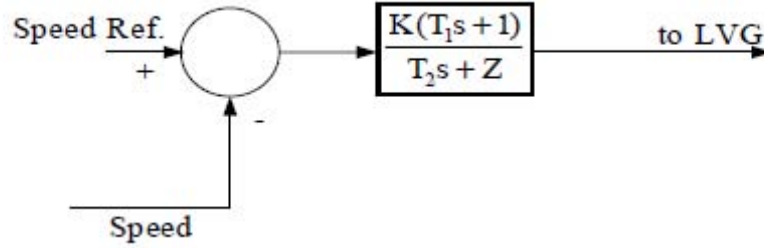


Fig. 2.15 Speed controller for MT model [6].

Where K is the controller gain, T_1 (T_2) is the governor lead (lag) time constant, and Z is a constant representing the governor mode, either droop or isochronous.

A droop governor is a proportional speed controller where the output is proportional to the error in speed. An isochronous speed controller is a proportional-plus-reset speed controller where the rate of change of the output is proportional to the speed error. The primary reason for utilizing acceleration control is to limit the rate of the shaft acceleration during startup. If the operating speed of the system is shown to be close to its rated speed without runaway occurrences, the acceleration control would remain unused and thereby could be removed from the model [32]. In order to decrease simulation overhead, acceleration control was excluded for this research.

Fuel System. The fuel system is comprised of a fuel valve, valve positioner, and, an actuator. The fuel system output fuel stream flow rate is calculated from the inertia of the valve positioner and actuator, Eq. (2.17).

$$E_1 = \frac{K_v}{T_v s + c} F_d \quad (2.17)$$

Where K_v is the valve positioner gain, T_v is the valve positioner time constant, c is a constant, F_d is the input of the valve positioner, and E_1 is the output of the valve positioner. The actuator transfer function is:

$$Wf = \frac{K_f}{T_f s + c} E_1 \quad (2.18)$$

Where K_f is the actuator gain, T_f is the actuator time constant, c is a constant, and Wf is the fuel demand signal in per-unit.

The per-unit gas turbine shaft speed N is multiplied by the output of the LVG, which is the least amount of fuel necessary to produce a given shaft speed. The resulting value is scaled by a gain, $K3$, delayed, and added to the minimum amount of fuel flow necessary to ensure combustion at no-load, rated speed, defined as $K6$. The fuel signal of minimum fuel and additional fuel (F_d) is the input to the fuel system block consisting of the valve positioner and actuator (Fig. 2.16). To limit rapid changes within the fuel system, the minimum amount of fuel necessary for continuous combustion and additional required fuel for temperature control scaled by a gain, defined with an upper limit of one per-unit. Specifically,

$$(LVG * N)K3 + K6 = 1 \quad (2.19)$$

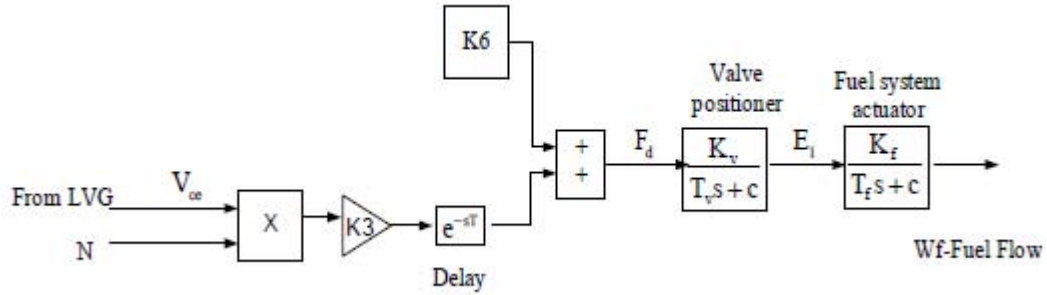


Fig. 2.16 Fuel control system for MT model [6].

Compressor/Combustor/Turbine. The compressor, combustor, and turbine blocks comprise the basis of a microturbine and can be modeled as a linear mechanism, with the exception of the rotor time constant [32]. Each stage of fuel/exhaust flow was modeled by a transfer function representing time and transport delays, shown in Fig. 2.17.

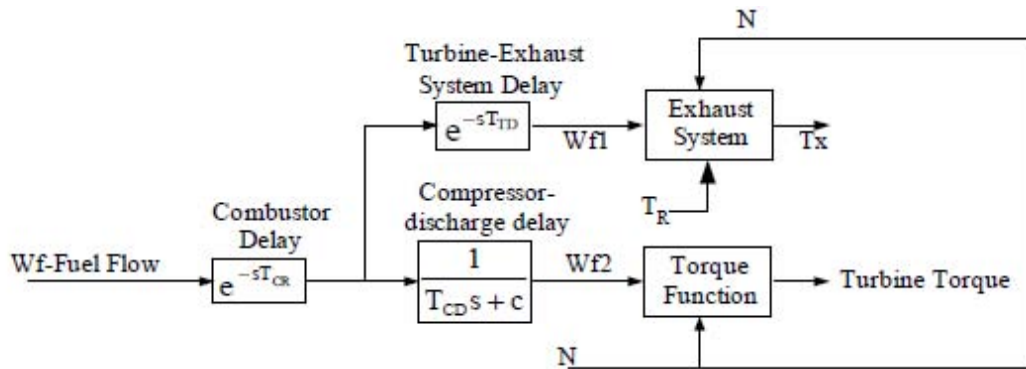


Fig. 2.17 Compressor, combustor, and turbine representations in MT model [6].

The fuel system output signal, or fuel flow demand signal, is input to the combustor delay block representing the combustion reaction time, T_{CR} , within the combustion chamber. The delayed combustion stream is split to calculate the exhaust stream temperature and torque of the turbine. Before the stream signal reaches the exhaust system block, a transport delay, T_{TD} , was implemented to represent the transport

of a stream from the combustion chamber to the expander. In the same manner, prior to the input of the stream signal to the torque block, a time delay associated with the compressor discharge, T_{CD} , was implemented.

The exhaust system block calculates the temperature of the exhaust stream, T_X (°F), by the following equation [32]:

$$T_X = T_R - 700(1 - Wf_1) + 550(1 - N) \quad (2.20)$$

Where T_R is the reference temperature (°F), N is the rotational per-unit shaft speed of the gas turbine, and Wf_1 is the per-unit fuel flow demand signal after passing through the combustor and transport delay blocks.

The torque block calculates the torque of the gas turbine shaft in per-unit by the following equation [32]:

$$\text{Torque} = K_{HHV}(Wf_2 - 0.23) + 0.5(1 - N) \quad (2.21)$$

Where K_{HHV} is the higher heating value coefficient of the gas stream in the combustion chamber and Wf_2 is the fuel flow demand signal after passing through the combustion and compressor discharge delay blocks. The value of K_{HHV} and the constant 0.23 originate from the linearity of the typical power/fuel flow rate characteristics of a gas turbine; namely, at zero power the fuel rate is 23% and at rated power the fuel rate is 100% [32], [34].

Temperature Control. The temperature control block limits the gas turbine output power at a given operating setpoint (or reference) temperature of the gas turbine, independent of ambient temperature, as shown in Fig. 2.18. An air/fuel mixture is burned in the combustor, resulting in an exhaust stream at increased pressure and temperature to

produce turbine torque. The exhaust stream temperature is measured by a series of thermocouples with radiation shields.

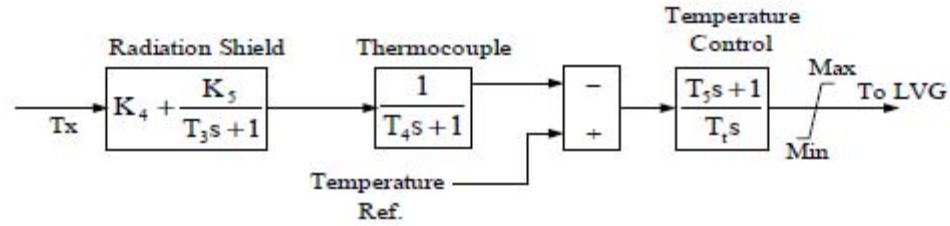


Fig. 2.18 Temperature controller for MT model [6].

Where T_t is the temperature controller integration rate, T_3 is a time constant associated with the radiation shield, T_4 is a time constant associated with the thermocouple, T_5 is a time constant associated with the temperature controller, and K_4 and K_5 are constants associated with the radiation shield.

The difference between the given operating setpoint temperature of the gas turbine and the output from the thermocouple block is calculated. When the operating temperature of the gas turbine is less than the given operating setpoint temperature, the difference is positive. The difference, or the exhaust temperature of the gas turbine, is input into the temperature control block. When the input to the temperature control block is positive, meaning the operating temperature of the gas turbine is less than the reference temperature, the output of the temperature control is positive, indicating an increase in temperature of the gas turbine. Likewise, when the input to the temperature control block is negative, meaning the operating temperature of the gas turbine is more than the reference temperature, the output of the temperature control is negative, indicating a decrease in operating temperature of the gas turbine. The output of the temperature

control block is the temperature control signal and the input to the LVG. When the temperature control signal becomes less than the speed control signal, the gas turbine will operate on temperature control instead of speed control.

Since the rest of the work completed by the author was developed with actual, not per-unit, units and temperature in units of Kelvin, the gas turbine model temperatures were converted to Kelvin for all outputs from the gas turbine block and to Fahrenheit for all inputs to the gas turbine block. The outputs of the transfer function model are turbine torque (pu), fuel demand (pu) and exhaust temperature (°F and K). As the SOFC, heat exchanger, and combustor models are based on thermodynamic principles, the fuel and air flow rates of the MT were required. In order to calculate the amount of fuel and air necessary for a given rated power, a thermodynamic model was created and integrated into the existing model.

Thermodynamic Model

Since the transfer function model of the MT lacks thermodynamic calculations, as present elsewhere in the system, a limited thermodynamic model was constructed. The model constructed is limited in that it merely acts as a bridge between the transfer function model of the MT and the thermodynamic energy flows necessary for the remaining system.

In order to maintain an accurate yet non-complex model, it is assumed that the mass flow into the MT is equivalent to the mass flow out of the MT. The mass flow stream in was assumed to consist of a fuel stream and air stream, shown in Eq. (2.22).

$$\dot{m}_{out} = \dot{m}_{in} = \dot{m}_{air} + \dot{m}_{fuel} \quad (2.22)$$

Since air is comprised of approximately 78% N₂ and 20.9% O₂ with the remaining percentage consisting of trace amounts of other gases, the trace gases are neglected from the calculations. For the purposes of this research, air is assumed to consist of 79% N₂ and 21% O₂.

The air to fuel ratio (AFR) must be calculated in order to maintain continuous combustion. Although air consists of N₂ and O₂, the reactants of the combustion reaction only rely on O₂. For this reason, the AFR is based on mass ratios of the O₂ to fuel. The AFR ratio is calculated to be 3.989, shown in Eq. (2.23).

$$\frac{m_{air}}{m_{fuel}} = \frac{2m_{O_2}}{m_{CH_4}} = \frac{2 \cdot 32 \frac{g}{mol}}{16.0425 \frac{g}{mol}} = 3.989 \quad (2.23)$$

This result indicates a necessary ratio of O₂ to CH₄ at 3.989. As atmospheric air is comprised of 21% O₂ for the purposes of this research, it follows that the AFR needs to account for the composition of air. The modification for atmospheric air flow was calculated to be 18.9952:1 AFR, as shown in Eq. (2.24).

$$\frac{2m_{O_2}}{m_{CH_4}} = 3.989 \Rightarrow \frac{m_{air}}{m_{fuel}} = \frac{3.989}{0.21} = 18.9952 \approx 19 \quad (2.24)$$

For the research within this thesis, the calculated AFR of 18.9952:1 is defined as 19:1. This slightly increased AFR indicates the mixture will be fuel lean, or contain more air than fuel.

The interconnection between the transfer function model of the MT and the remainder of the model is required in terms of fuel flow rate due to the lack of other available rates within the transfer function model. The other flows, namely the air flow, are then extrapolated based on the previously calculated AFR. Since the air flow has

been written in terms of fuel flow as defined by the AFR, it remains to define the oxidant in terms of fuel as well. The mass of the oxidant, atmospheric air, has been defined as 79% N₂ and 21% O₂ and can be written as

$$m_{air} = 0.79 m_{N_2} + 0.21 m_{O_2} \quad \left(\frac{g}{mol} \right) \quad (2.25)$$

Where m_{N_2} is 28 g/mol and m_{O_2} is 32 g/mol. In order to get the mass of air in terms of oxidant only, Eq. (2.25) was multiplied by one in terms of the mass of O₂, shown in Eq. (2.26).

$$\begin{aligned} m_{air} &= (0.79 m_{N_2} + 0.21 m_{O_2}) \frac{0.21 m_{O_2}}{0.21 m_{O_2}} \\ &= 0.79 m_{N_2} \left(\frac{0.21 m_{O_2}}{0.21 m_{O_2}} \right) + 0.21 m_{O_2} \\ &= \left(\frac{0.79 m_{N_2}}{0.21 m_{O_2}} \right) 0.21 m_{O_2} + 0.21 m_{O_2} \quad \left(\frac{g}{mol} \right) \end{aligned} \quad (2.26)$$

By substituting the molar masses of N₂ and O₂, rearranging and combining the mass of O₂ terms, Eq. (2.27) is obtained.

$$\begin{aligned} m_{air} &= \left(\frac{0.79 \times 28 \frac{g}{mol}}{0.21 \times 32 \frac{g}{mol}} \right) 0.21 m_{O_2} + 0.21 m_{O_2} \\ &= (3.2917) 0.21 m_{O_2} + 0.21 m_{O_2} \\ &= (4.2917)(0.21 m_{O_2}) \end{aligned} \quad (2.27)$$

Based on this result the mass of atmospheric air was found to be approximately 28.8402 g/mol when using the rounded value of 4.2917, shown in Eq. (2.28).

$$\begin{aligned} m_{air} &= (4.2917)(0.21 m_{O_2}) \\ &= 4.2917 \times 0.21 \times 32 \frac{g}{mol} \\ &= 28.8402 \frac{g}{mol} \end{aligned} \quad (2.28)$$

To verify this result, the molecular mass of air was calculated by using the definition of the composition of air, Eq. (2.25), as shown in Eq. (2.29).

$$\begin{aligned}
 m_{air} &= 0.79 m_{N_2} + 0.21 m_{O_2} \\
 &= 0.79 \times 28 \frac{g}{mol} + 0.21 \times 32 \frac{g}{mol} \\
 &= 28.8400 \frac{g}{mol}
 \end{aligned} \tag{2.29}$$

The error of 0.0002 g/mol is acceptable for the purposes of the research presented in this thesis and stems from rounding errors. Combining the results of Eq. (2.27) and the assumed AFR of 19:1, the oxidant of the combustion reaction, O_2 , can be written in terms of proportional fuel consumption by setting the two equations equal to each other based on their equivalence to the value of molecular mass of atmospheric air. The derivation can be shown in the following manner [35]:

$$\begin{aligned}
 \text{AFR: } m_{air} &= 19 m_{fuel} & \text{Eq. (2.27): } m_{air} &= 4.2917 m_{O_2} \\
 \Rightarrow m_{air} &= 19 m_{fuel} = 4.2917 m_{O_2} & & \tag{2.30a}
 \end{aligned}$$

$$\Rightarrow m_{O_2} = \frac{19}{4.2917} m_{fuel} \tag{2.30b}$$

The AFR of 19:1 can be substituted into Eq. (2.22) to attain a value of mass flow in and mass flow out in terms of fuel flow, shown in Eq. (2.31).

$$\begin{aligned}
 \dot{m}_{out} &= \dot{m}_{in} = \dot{m}_{air} + \dot{m}_{fuel} \\
 &= 19 \dot{m}_{fuel} + \dot{m}_{fuel} \\
 &= 20 \dot{m}_{fuel}
 \end{aligned} \tag{2.31}$$

The relationship defined by Eq. (2.31) states that the mass flow in and out of the combustion chamber of the MT can be written as the fuel flow multiplied by 20.

In order to ultimately determine the amount of fuel necessary to produce a given amount of power within the MT combustor, the energy flow of the MT combustor must

be found. The heat energy produced from a chemical reaction can be described in terms of enthalpy by the following general thermodynamic relationship [28], [35]:

$$\text{Heat Energy} = \left(\sum H_{products} \right) \dot{m}_{out} - \left(\sum H_{reactants} \right) \dot{m}_{in} \quad (2.32)$$

The energy from heat of the products (CO_2 , H_2O) and reactants (CH_4 , O_2) when CH_4 is used as a primary fuel for combustion can be written based on their stoichiometric quantities

$$\sum H_{products} = \Delta H_{\text{CO}_2} + 2 \cdot \Delta H_{\text{H}_2\text{O}} \quad (2.33)$$

$$\sum H_{reactants} = \Delta H_{\text{CH}_4} + 2 \cdot \Delta H_{\text{O}_2} \quad (2.34)$$

The generic enthalpy equation, shown in Eq. (2.32), can be written in terms of actual products (CO_2 , H_2O) and reactants (CH_4 , O_2), shown in Eq. (2.35).

$$\begin{aligned} \text{Heat Energy} &= (\Delta H_{\text{CO}_2} + 2 \cdot \Delta H_{\text{H}_2\text{O}}) \dot{m}_{out} - (\Delta H_{\text{CH}_4} + 2 \cdot \Delta H_{\text{O}_2}) \dot{m}_{in} \\ &= (\Delta H_{\text{CO}_2} + 2 \cdot \Delta H_{\text{H}_2\text{O}}) \dot{m}_{out} - \Delta H_{\text{CH}_4} \dot{m}_{fuel} - 2 \cdot \Delta H_{\text{O}_2} \dot{m}_{\text{O}_2} \end{aligned} \quad (2.35)$$

Since the mass flow out of the MT combustor and the mass flow of oxidant have been derived in terms of mass flow of fuel, the energy within the combustion reaction can be written in terms of fuel flow as well, shown in Eq. (2.36).

$$\begin{aligned} \text{Heat Energy} &= (\Delta H_{\text{CO}_2} + 2 \cdot \Delta H_{\text{H}_2\text{O}}) (20 \dot{m}_{fuel}) - \Delta H_{\text{CH}_4} \dot{m}_{fuel} \\ &\quad - 2 \cdot \Delta H_{\text{O}_2} \left(\frac{19}{4.2917} \dot{m}_{fuel} \right) \end{aligned} \quad (2.36)$$

The fuel flow term can then be factored:

$$\text{Heat Energy} = \dot{m}_{fuel} \left[20 \cdot \Delta H_{\text{CO}_2} + 40 \cdot \Delta H_{\text{H}_2\text{O}} - \Delta H_{\text{CH}_4} - \left(\frac{38}{4.2917} \right) \cdot \Delta H_{\text{O}_2} \right] \quad (2.37)$$

Rearranging and solving for fuel flow:

$$\dot{m}_{fuel} = \frac{\text{Heat Energy}}{20 \cdot \Delta H_{CO_2} + 40 \cdot \Delta H_{H_2O} - \Delta H_{CH_4} - \left(\frac{38}{4.2917}\right) \cdot \Delta H_{O_2}} \quad (2.38)$$

Based on the result found in Eq. (2.38), given an amount of heat energy necessary for a desired amount of power, and by calculating the enthalpy of the combustion reaction for the products and reactants, the fuel flow rate corresponding to the desired amount of power can be calculated.

There are many methods to calculate the enthalpy of a reaction [28]; however, the method used for the research within this thesis utilized the Shomate equation for standard enthalpy, as defined by [36]. In order to account for the varying temperatures, the standard enthalpy equation is defined as

$$H^o - H_{298.15}^o = At + \frac{Bt^2}{2} + \frac{Ct^3}{3} + \frac{Dt^4}{4} - \frac{E}{t} + F - H \quad (2.39)$$

Where H^o is the standard enthalpy, $H_{298.15}^o$ is the standard enthalpy at 298.15 K, t is the operating temperature of the species (K), A , B , C , D , E , F , H are the Shomate equation parameters for thermochemical functions and differ by chemical species and temperature ranges [36].

MT Simulation

The thermodynamic model of the MT was implemented within the Matlab/Simulink environment. A simulation of the MT at a rated power of 250 kW was conducted for the purpose of verifying the implemented equations of operating temperature, thermodynamic heat flow, fuel flow, and power calculations. The configuration of the simulation is shown in Fig. 2.19. The simulation parameters for a MT per-unit and thermodynamic models are shown in Table 2.2.

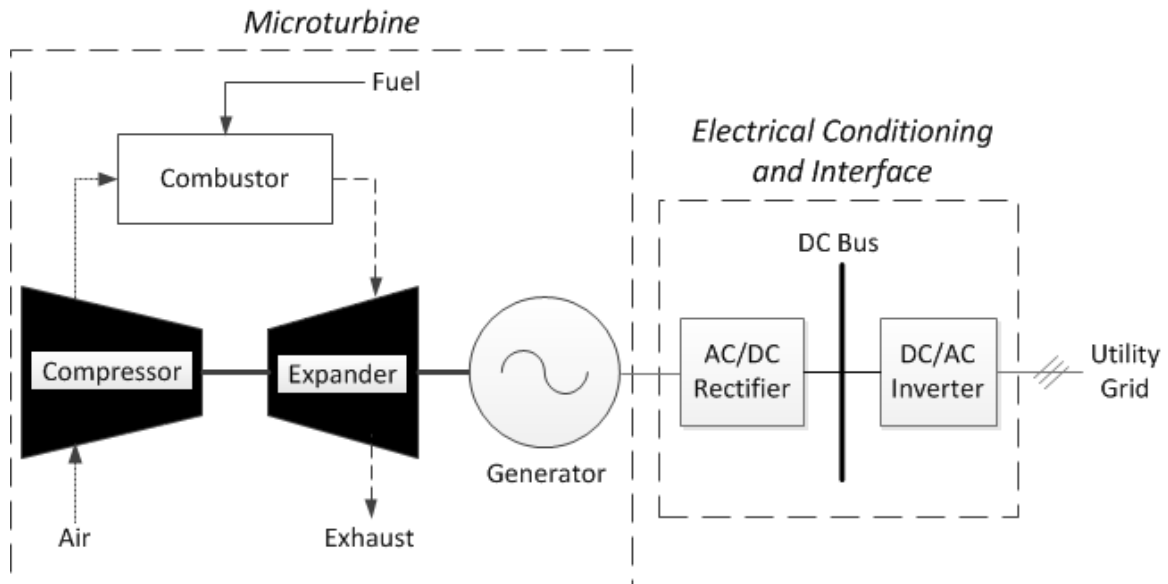


Fig. 2.19 Configuration of MT simulation interconnections.

Table 2.2 Simulation Parameters for MT

Parameter	Value
Rated Power	250 kW
Operating temperature	750 K
Ambient temperature	298 K
Initial fuel & air temperature	298 K
Overall efficiency of MT	35 %

The amount of energy necessary to produce 250 kW with an overall efficiency of 35 percent can be expressed through the definition of thermal efficiency for a heat engine [37], written as

$$Q_{in} = \frac{W_{out}}{\eta_{thermal}} = \frac{250 \text{ kW}}{0.35} = 714.286 \text{ kW} \left(\text{or } \frac{kJ}{s} \right) \quad (2.40)$$

Where Q_{in} is the heat energy input, W_{out} is the work, or mechanical energy, output, and $\eta_{thermal}$ is the overall thermal efficiency of the system. The efficiency rating of 35 percent was chosen to be a representative efficiency of a small scale gas turbine for

DG applications [38]. The necessary energy from heat, or change in enthalpy for a 250 kW rated MT system and the operating temperature can be applied to Eq. (2.38) to calculate the required quantity of fuel and air flow to produce a combustion reaction of the same rating. In particular, for a rated power of 250 kW at an operating temperature of 750 K, the amount of fuel and air flow necessary to provide continuous combustion can be derived. For clarity, the denominator of Eq. (2.38) is calculated separately by the Shomate equation, then substituted into the original equation, shown as follows

$$\begin{aligned}\Delta H &= 20 \cdot \Delta H_{CO_2} + 40 \cdot \Delta H_{H_2O} - \Delta H_{CH_4} - \left(\frac{38}{4.2917}\right) \cdot \Delta H_{O_2} \\ &= 20 \times 19.89 \frac{kJ}{mol} + 40 \times 16.30 \frac{kJ}{mol} - 23.40 \frac{kJ}{mol} - \left(\frac{38}{4.2917}\right) \times 12.18 \frac{kJ}{mol} \\ &= 918.63 \frac{kJ}{mol}\end{aligned}\tag{2.41}$$

The change in enthalpy, $\Delta H = 918.63 \text{ kJ/mol}$, is then substituted into Eq. (2.38)

$$\dot{m}_{fuel} = \frac{\text{Heat Energy}}{\Delta H} = \frac{714.286 \frac{kJ}{s}}{918.63 \frac{kJ}{mol}} = 0.776 \frac{mol}{s} \times 16.04 \frac{g}{mol} = 12.47 \frac{g}{s}\tag{2.42}$$

With a known AFR of 19:1, the mass flow rate of the oxidant can then be found

$$\dot{m}_{air} = 12.47 \frac{g}{s} CH_4 \cdot AFR = 12.47 \frac{g}{s} \cdot 19 = 237.02 \frac{g}{s}\tag{2.43}$$

The electrical power generated by the MT transfer function model and thermodynamic interconnection model at a rated power of 250 kW is shown in Fig. 2.20. The power measured by the transfer function model closely follows that calculated by the thermodynamic model once the rated power is achieved. During startup, both models show similar transient behavior with the thermodynamic model exhibiting a more gradual increase to the rated power.

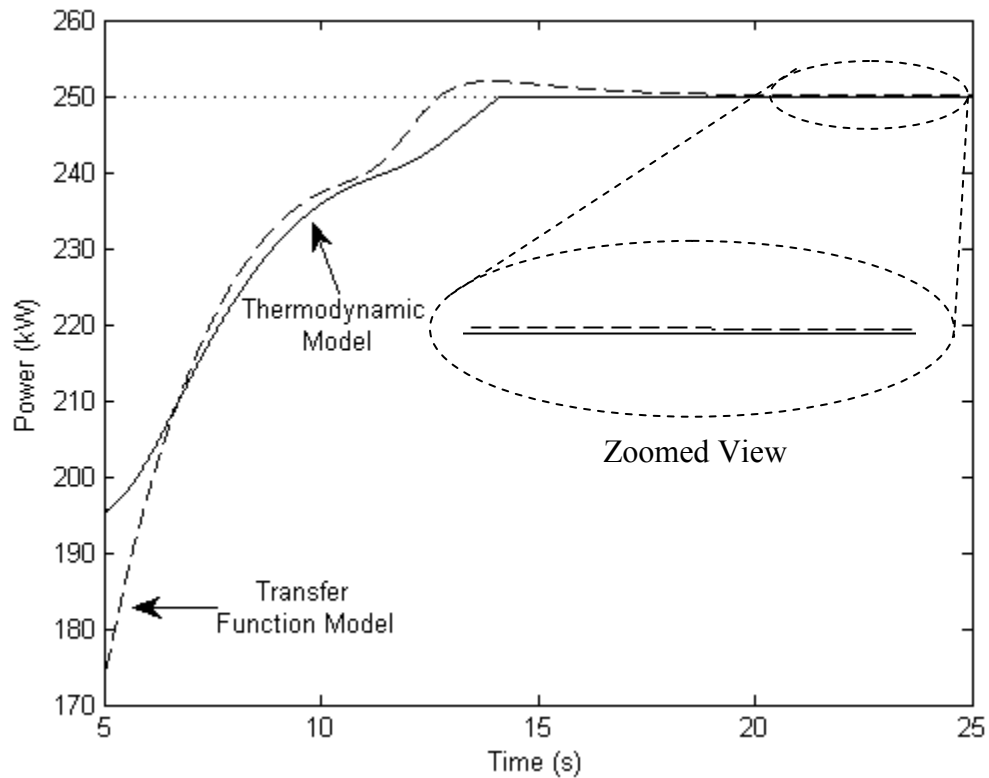


Fig. 2.20 Power generated by thermodynamic and transfer function model of MT at rated power of 250 kW.

Once the thermodynamic model reaches the desired power, the fuel and air quantities are switched to a steady state operation, shown as the abrupt change at 14 s. The transfer function model demonstrates a rapid increase of power from 5 to 10 s, followed by the power leveling around 240 kW from 10 to 12 s, an increased to 253 kW by 14 s, and finally a slow convergence to rated power for the remainder of the simulation. The power from the transfer function model is measured at the output of the DC/AC inverter, whereas the power calculated from the thermodynamic model is found from the thermal energy required to produce a rated power of 250 kW from a combustion reaction, shown in Fig. 2.21.

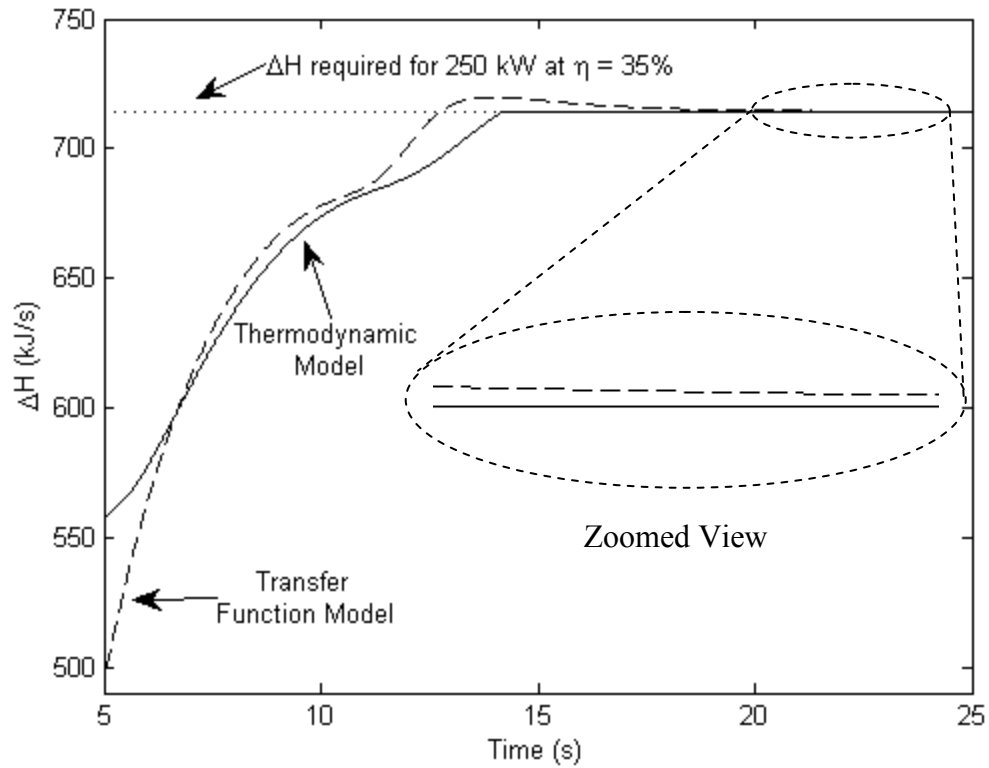


Fig. 2.21 Energy from ΔH of thermodynamic and transfer function model of MT at rated power of 250 kW.

As expected, the change in enthalpy of the system calculated by the transfer function and thermodynamic models demonstrate the same operational characteristics. As previously discussed, and shown in Eq. (2.40), the amount of energy from heat required to produce a desired amount of power with an associated efficiency are related by a scaling factor of efficiency. For this reason, the power produced and change in enthalpy of the system possess the same operational characteristics.

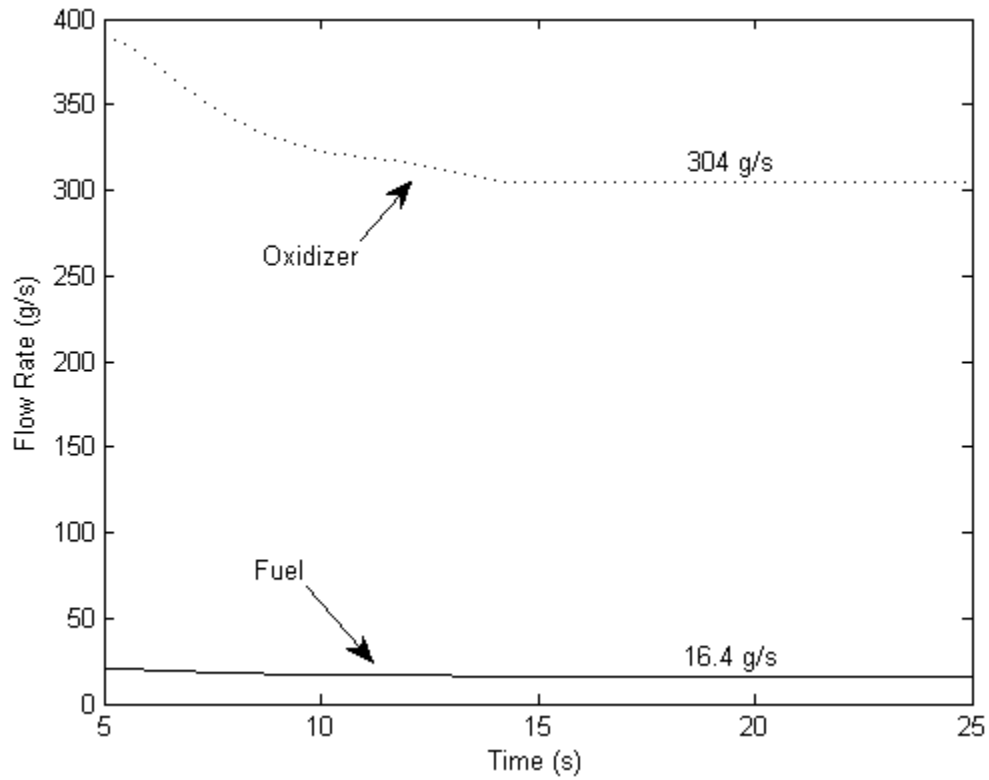


Fig. 2.22 Mass flow rate (g/s) of fuel and oxidizer in MT at rated power of 250 kW, from simulation.

The fuel and air flow rates were calculated, by Eq. (2.38), to be 12.47 g/s, or 0.776 mol/s, and 237.02 g/s, or 7.4 mol/s, respectively, to generate 250 kW with an efficiency of 35 percent and an AFR of 19. As Fig. 2.22 and Fig. 2.23 demonstrate, the fuel and air flow rates are higher than previously calculated. The fuel and air flow rates from simulation are 16.4 g/s, or 1.0 mol/s, and 304 g/s, or 9.53 mol/s, respectively. Once the rated power is attained, the fuel and air flow rates are switched to a steady state operation. The higher values attained through simulation are due to the convergence of the thermodynamic model to rated power prior to reaching optimal operating temperature. The amount of fuel and air required to produce a given amount of power is

decreased as the operating temperature increases. The heat energy of the reactants increases with temperature, thus requiring a lower flow rate.

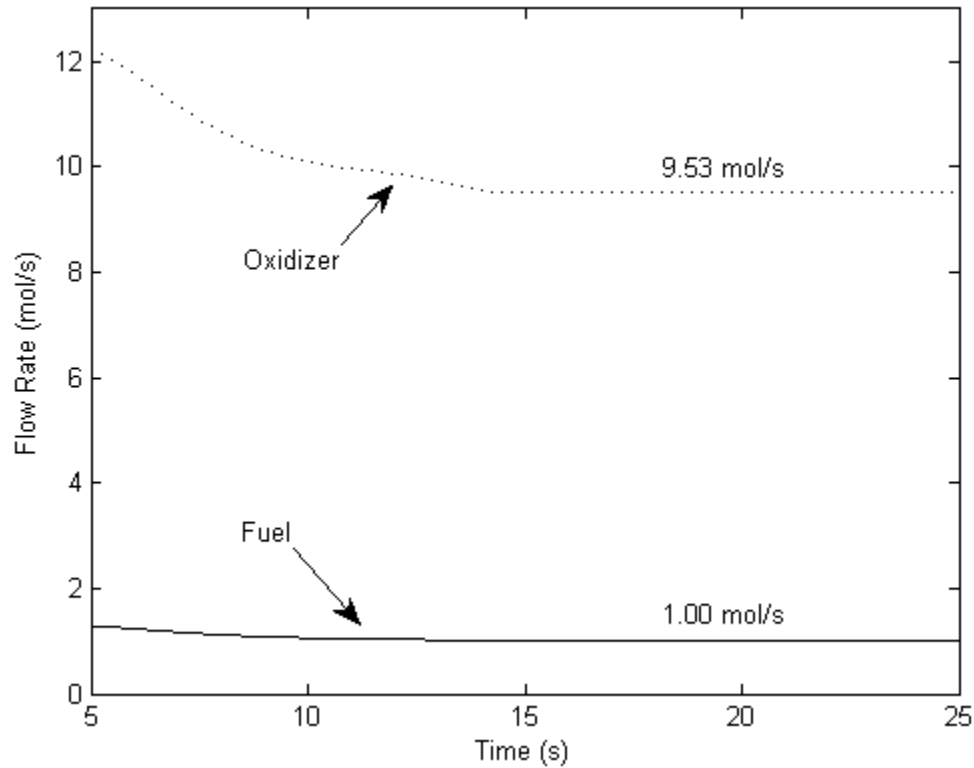


Fig. 2.23 Molar flow rate (mol/s) of fuel and oxidizer in MT at rated power of 250 kW, from simulation.

The operating temperature of the MT is not achieved during the time scale of the simulation, as shown in Fig. 2.24, and continues to slowly increase to 686 K within the 25 s simulation time. Since the calculated values of fuel and air flow used the operating temperature of 750 K, the amount of fuel to generate the rated power at a lesser temperature would produce a less efficient combustion reaction and thereby require more fuel and air flow. Hence, the simulation results of fuel and air flow exceed the calculated values.

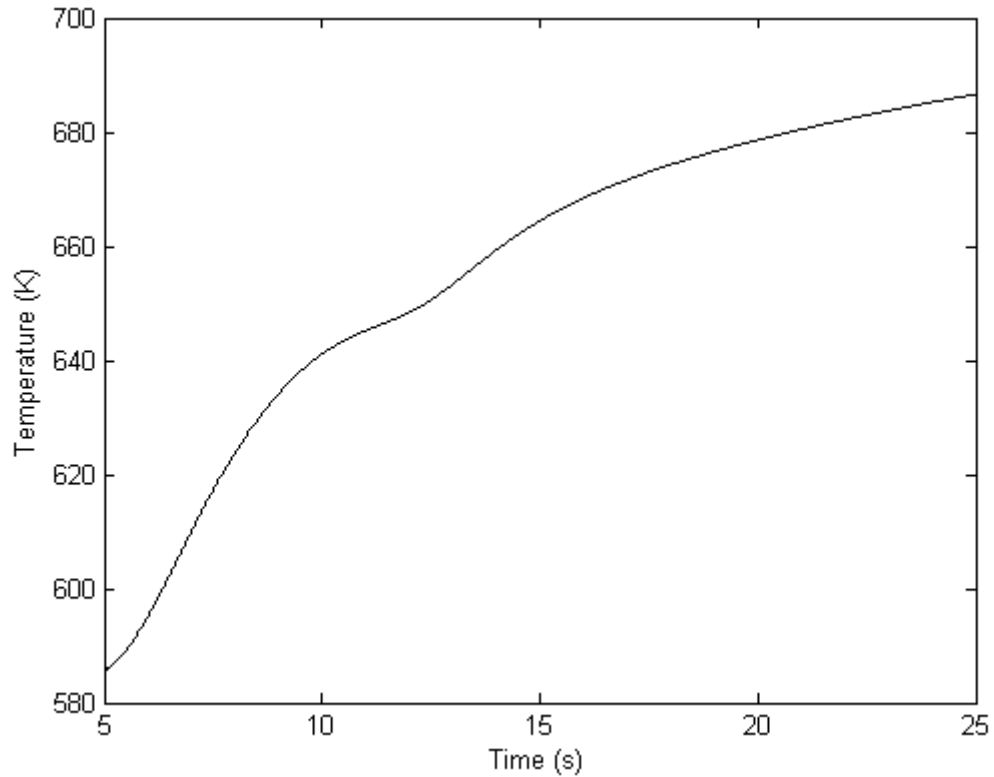
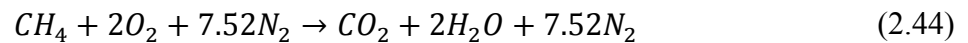


Fig. 2.24 Temperature of exhaust of MT at rated power of 250 kW and operating temperature of 750 K.

Combustor

Combustor Operation

Combustion is a series of exothermic reactions within an air/fuel mixture producing heat [35]. The fuel for the combustor model is assumed to be methane, or CH_4 , and the oxidizer is assumed to be air. The balanced combustion reaction with these reactants is shown in Eq. (2.44).



It is worth noting that it is possible to burn the unused fuel, hydrogen, from the SOFC anode channel resulting in the same amount of H_2 and H_2O as in (2.44), as demonstrated by Eq. (2.45).

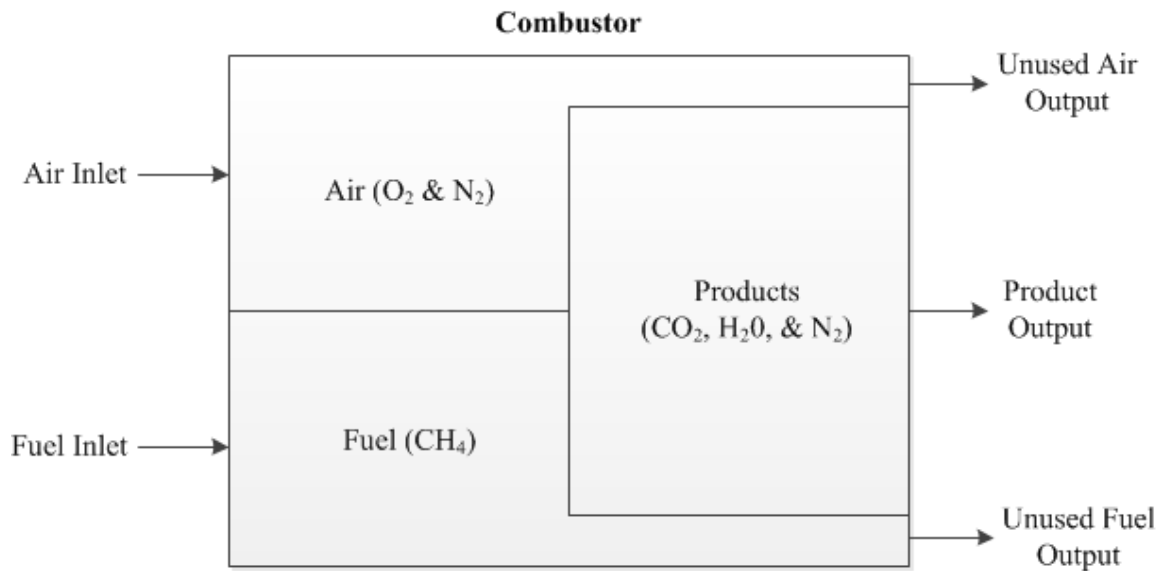
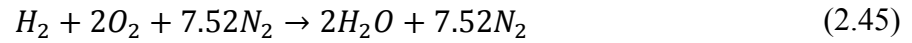


Fig. 2.25 Conservation of species diagram of combustor.

The basic operation of a combustor is shown in Fig. 2.25. The air/fuel mixture enters the chamber whereupon exothermic reactions occur. The resulting products in the form of a heated exhaust stream, along with any unused air or fuel, exit the combustor. The heat from the reaction increases the temperature of the exhaust stream compared to the temperature of the input air/fuel mixture.

Combustor Model

The fuel for the combustor model is assumed to be methane, or CH_4 , and the oxidizer is assumed to be air. When the system is in CC mode the unused fuel from the

SOFC anode channel is a mixture of H_2 and H_2O . The model accounts for this change in fuel, from CH_4 to H_2 , by calculating the enthalpy of an H-O combustion reaction defined in Eq. (2.45).

Conservation of Chemical Species. Due to the scope of this research a simplified, or algebraic, but still valid, approach to defining the conservation of chemical species within the combustion reaction was taken. The ideal mixture equations can be defined by the mass fractions of each species within the combustion reaction. Specifically, the ideal mass fraction of the oxidizer (O_2 and N_2), fuel (CH_4), and products (CO_2 , H_2O , and N_2) are defined by the ratios of their respective masses to the total mass of all species, shown in Eq. (2.46)-(2.48).

$$w_{oxidizer} = \frac{M_{oxidizer}}{M_{oxidizer} + M_{fuel} + M_{products}} \quad (2.46)$$

$$w_{fuel} = \frac{M_{fuel}}{M_{oxidizer} + M_{fuel} + M_{products}} \quad (2.47)$$

$$w_{products} = \frac{M_{products}}{M_{oxidizer} + M_{fuel} + M_{products}} \quad (2.48)$$

Where w is the mass fraction of each compound and M is the molar mass of each compound (g/mol). This conservation can also be demonstrated graphically, as shown in Fig. 2.25.

Based on the assumption of an ideal mixture of gas, the mass fraction of the products can be written

$$w_{products} = 1 - w_{oxidizer} - w_{fuel} \quad (2.49)$$

As Fig. 2.25 demonstrates, the oxidizer and fuel enter the combustion chamber, whereupon combustion of the species occurs, resulting in the creation of the product species. The unused oxidizer and fuel exit the combustor along with the product species. The conservation of chemical species equations are supplemented by conservation of mass equations described in the following section.

Conservation of Mass. The conservation of mass equations describe the consumption and production of chemical species based on collision theory of combustion reactions presented by [35]. During the combustion reaction both the oxidizer and fuel are consumed. However, the rate at which oxidizer and fuel are consumed depends on the stoichiometry of the reaction. For this reason, the mass conservation equations have been written in terms of the oxidizer and fuel, shown in Eq. (2.50), (2.51).

$$V \frac{dw_{oxid}(t)}{dt} = \dot{m}_{in} w_{oxid_{in}}(t) - \dot{m}_{out} w_{oxid_{out}}(t) + M_{fuel} k(t) V \frac{w_{oxid}}{w_{fuel}} \quad (2.50)$$

$$V \frac{dw_{fuel}(t)}{dt} = \dot{m}_{in} w_{fuel_{in}}(t) - \dot{m}_{out} w_{fuel_{out}}(t) + M_{fuel} k(t) V \quad (2.51)$$

Where V is the volume of the combustor (assumed to be 1 m^3), $w_{oxid_{in}}$ and $w_{fuel_{in}}$ are the mass fraction of oxidizer and fuel into the combustor, $w_{oxid_{out}}$ and $w_{fuel_{out}}$ are the mass fraction of oxidizer and fuel out of the combustor, $\dot{m}_{in}$ is the mass flow rate into the combustor, $\dot{m}_{out}$ is the mass flow rate out of the combustor, and k is the Arrhenius rate term for consumption of species due to the combustion reaction.

The Arrhenius term is defined for a fuel of CH_4 as [35]:

$$k(t) = \left(-24100 \frac{\text{kmol}}{\text{kg} \cdot \text{s}} \right) w_{fuel}(t)^{-0.3} (0.233 w_{oxid}(t))^{1.3} e^{\frac{-15908}{T(t)}} \quad (2.52)$$

For further detail the Arrhenius form and subsequent calculations of coefficients for a given species, the reader is directed to [35].

Conservation of Energy. The first law of thermodynamics for an open system, such as a fuel cell or gas turbine, specifies that the rate of change of energy produced by the system is equivalent to the difference in flow rates in and out of the system. For a system under constant pressure, the first law can be written, in terms of enthalpy and heat flow, in the following manner [28]:

$$dQ - dW = \sum_{products} \dot{n}_i \Delta H_i - \sum_{reactants} \dot{n}_i \Delta H_i \quad (2.53)$$

Where dQ is the rate of heat produced within the system (J/s), dW is the rate of work within the system (J/s), $\Delta H_{reaction}$ is the change in enthalpy within the system (J/s), $\dot{n}_i$ is the molar flow rate of the input or output stream (mol/s), and H_i is the specific enthalpy of the input or output stream (J/mol). Eq. (2.54) is implemented under the assumption of constant pressure operation written as:

$$\Delta H = Q = \dot{n}_i c_p \Delta T \quad (2.54)$$

Where c_p is the specific heat capacity of a given compound (J/mol K) and ΔT is the change in temperature of the input and output streams (K). The specific gas heat capacity was calculated using the Shomate equation, as defined by [36]. By definition, specific heat capacity is the amount of energy required to increase the temperature of a compound by 1°C, leading to the fact that the specific heat capacity will vary for different temperatures [28]. In order to account for the varying temperatures, the specific gas heat capacity equation is defined as [36]

$$C_p = A + Bt + Ct^2 + Dt^3 + \frac{E}{t^2} \quad (2.55)$$

Where t is the operating temperature of the species (K), A , B , C , D , E are the Shomate equation parameters for thermochemical functions and differ by chemical species and temperature ranges [36].

Combustor Simulation

The thermodynamic combustor model was implemented within the Matlab/Simulink environment. A simulation was conducted for the purpose of verifying the implemented equations of combustion. The simulation was considered for steady state operation and transient occurrences of the CC system for input fuels of H_2 and CH_4 over a variety of input fuel temperatures. The simulation parameters for H_2 are shown in Table 2.3, Table 2.4 and parameters for CH_4 are shown in and Table 2.5.

Table 2.3 Simulation Parameters for Input Fuel of H_2 with Various Input Fuel Temperatures

Parameter	Value
Fuel flow	0.0013 g/s
Oxidant flow	0.0430 g/s
Input fuel temperatures	1073K, 1173 K, 1273K
Initial combustor temperature	1173 K

The fuel flow chosen was determined to be the approximate amount of unused H_2 at the outlet of the anode channel of the 5 kW SOFC model. The air flow rate was found by the AFR of H_2 , calculated to be 33:1, shown in Eq. (2.56)

$$AFR_{H_2} = \frac{m_{oxidant}}{m_{H_2}} = \frac{132 \frac{g}{mol}}{2 \cdot 2.01588 \frac{g}{mol}} = 32.74 \cong 33 \quad (2.56)$$

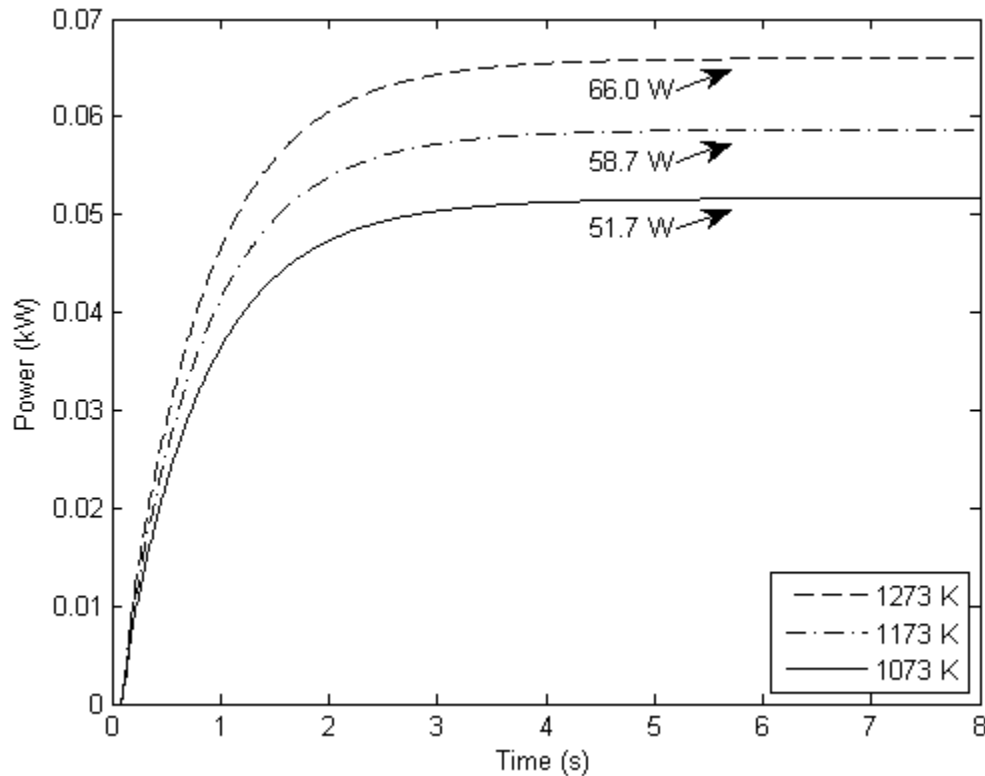


Fig. 2.26 Power produced from combustion of H₂ for input fuel temperatures of 1073 K (bottom), 1173 K (middle), 1273 K (top).

When the input fuel temperatures into the combustor are increased, the heat energy generated shows the same trend, shown in Fig. 2.26. For a fuel temperature of 1073 K, 51.7 W are produced, while for a fuel temperature of 1273 K, 66 W are produced. The increase of power produced is due to the increase of thermal energy at higher temperatures.

For a flow rate producing a small (< 1 kW) amount of thermal energy a corresponding small increase of output temperature is produced, shown in Fig. 2.27 [39]. It follows that as the volume of input fuel is increased, there is a equivalent increase in combustion temperature, as demonstrated by [40]. Since the time for combustion to take place is primarily dependent on the volume of fuel/oxidant flow into the combustor and

the volume of the combustor, a medium time scale combustion rate on the order of seconds was selected as an average representative combustion model.

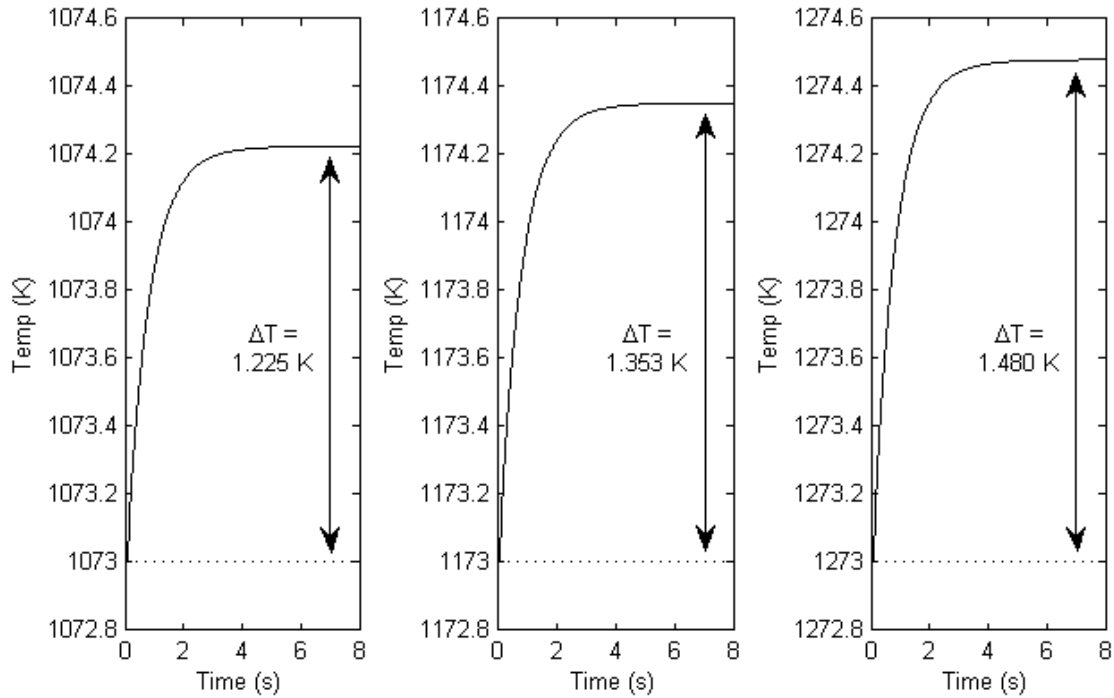


Fig. 2.27 Output temperatures due to combustion of H₂ for input fuel temperatures of 1073 K (left), 1173 K (middle), 1273 K (right), from simulation.

As Fig. 2.27 demonstrates, the combustion reaction increases the temperature of the products from 1173 K to approximately 1174.35 K for an input fuel temperature of 1173K. Likewise, a similar change in temperature is shown for input fuel temperatures of 1073K and 1273K. It should be noted the slow rate of complete combustion, on the order of seconds, is due to a small value of fuel flow. The increase of ΔT for increased initial temperatures is caused by the increase of heat capacity and subsequent enthalpy for higher temperatures.

Another simulation was conducted to observe the reaction of the combustor to a continual increase of input fuel temperature, based on the behavior of the SOFC operating temperature on the medium time scale. The simulation parameters are given in Table 2.4.

Table 2.4 Simulation Parameters for Input Fuel of H_2 with Various Increasing Input Fuel Temperatures

Parameter	Value
Fuel flow	0.0013 g/s
Oxidant flow	0.0430 g/s
Input fuel temperatures	1073K, 1173 K, 1273K
Initial combustor temperature	1173K
ΔT of input fuel temperature	0.4 K/s

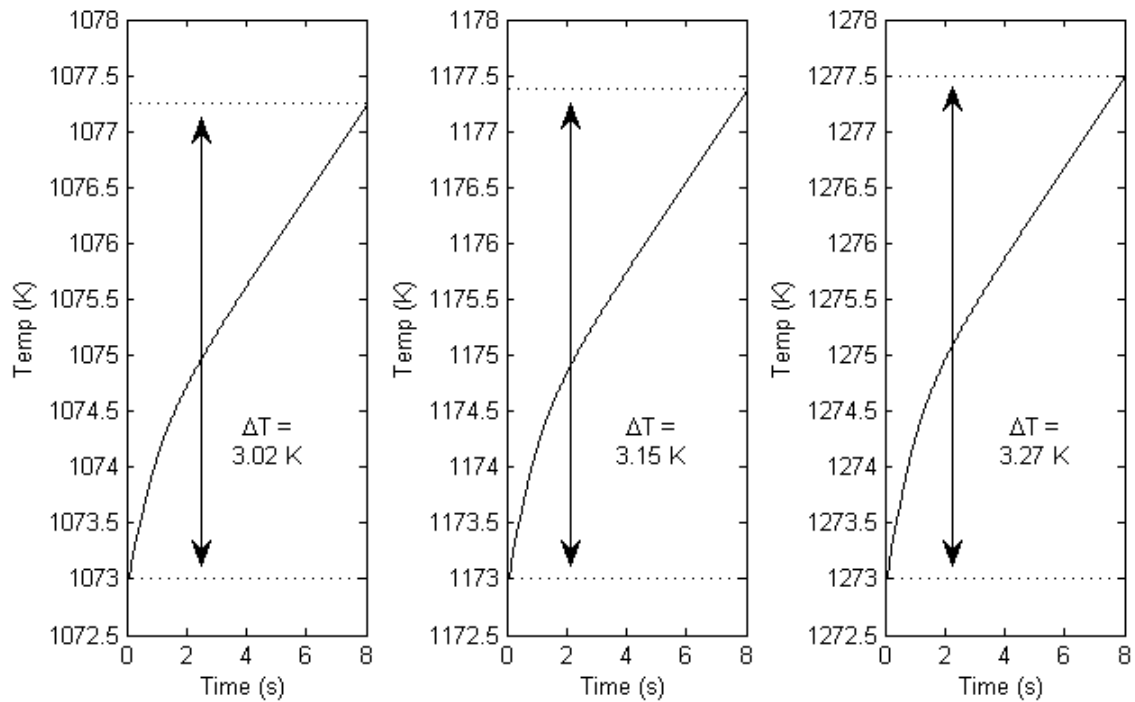


Fig. 2.28 Output temperatures due to combustion of H_2 for input fuel temperatures of 1073 K (left), 1173 K (middle), 1273 K (right) with input temperature increase rate of 0.4 K/s, from simulation.

The rate of increase of temperature was selected as a best linear fit of the SOFC operating temperature on the medium time scale.

When the temperature of input fuel is increased at a rate of 0.4 K/s, a similar rate of increase is observed in the temperature of the products, as shown in Fig. 2.28. The same trend of increased ΔT for increased input fuel temperature is demonstrated, indicating an increase of efficiency at higher temperatures.

Another simulation was conducted with an input fuel of CH_4 at a constant rate with various initial temperatures to observe the model on a larger combustion scale. The simulation parameters are given in Table 2.5.

Table 2.5 Simulation Parameters for Input Fuel of CH_4 with Various Increasing Input Fuel Temperatures

Parameter	Value
Fuel flow	1 g/s
Oxidant flow	19 g/s
Input fuel temperatures	1073K, 1173 K, 1273K
Initial combustor temperature	1173K

The fuel flow was chosen to be 1 g/s with an air flow of 19 g/s to demonstrate the behavior of the combustor model on a larger combustion reaction.

Fig. 2.29 shows the power produced from the combustion reaction of CH_4 as fuel. The amount of power produced is greater than that of H_2 , in the previous simulations, due to the increased fuel flow rate. Numerically, the power produced is 12.14 kW with an input fuel temperature of 1073 K, 13.66 kW with a fuel temperature of 1173 K, and 15.22 kW with a fuel temperature of 1273 K, as shown in Fig. 2.29. The increase of temperature increases the enthalpy of the system, as previously demonstrated with H_2 .

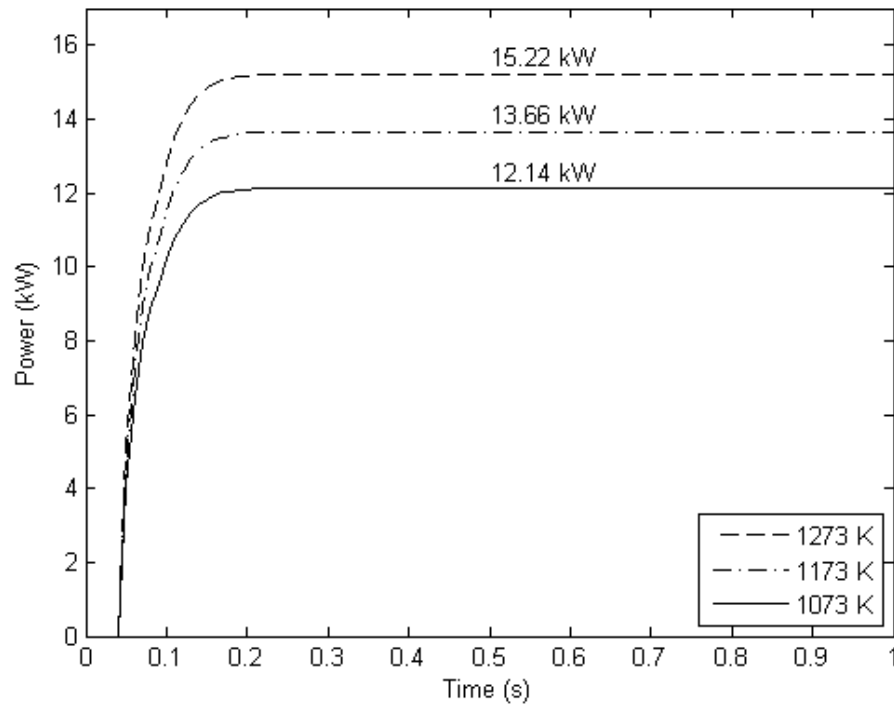


Fig. 2.29 Power produced due to combustion of CH_4 for input fuel temperatures of 1073 K (bottom), 1173 K (middle), 1273 K (top), from simulation.

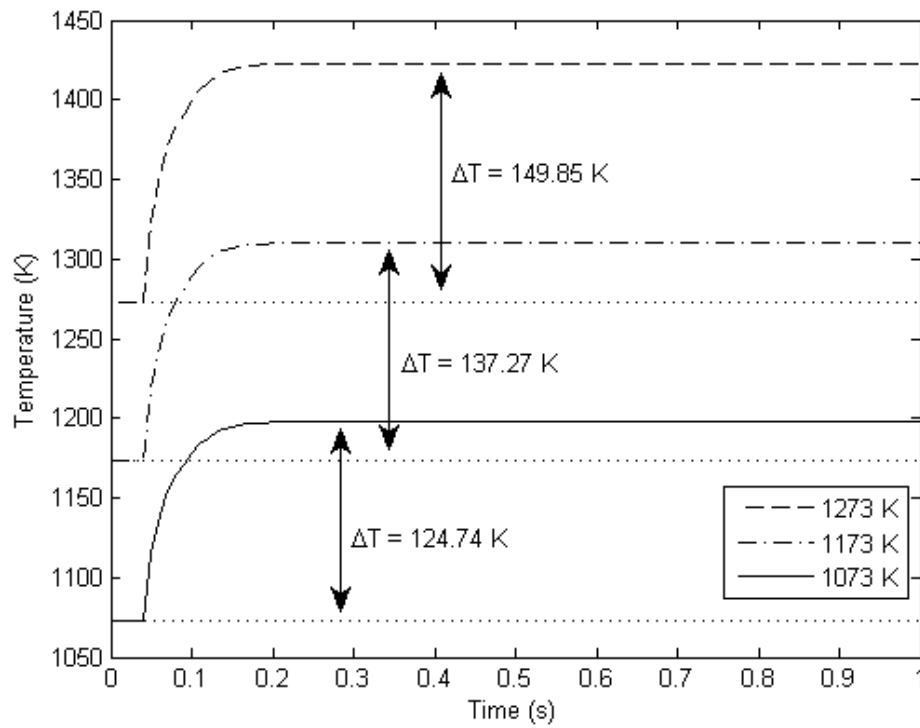


Fig. 2.30 Output temperature due to combustion of CH_4 for input fuel temperatures of 1073 K (bottom), 1173 K (middle), 1273 K (top), from simulation.

Fig. 2.30 shows the output temperature from the combustion of CH_4 with input fuel temperatures of 1073 K, 1173 K, and 1273 K. The same trend of increasing ΔT with increased fuel temperature is exhibited, with $\Delta T = 124.74$ K for a fuel temperature of 1073 K, $\Delta T = 137.27$ K for a fuel temperature of 1173 K, and $\Delta T = 149.85$ K for a fuel temperature of 1273 K. The time scale of combustion has also changed from seconds, as shown with the combustion of H_2 , to 10^{-1} seconds due to the increased volume of input fuel flow while maintaining the same volume of the combustor.

A simulation was conducted to exhibit the reaction of the combustor model to varying flow rates. The input fuel of CH_4 was chosen with the oxidant flow decreased from 19 g/s to 9 g/s at 1 s and then increased to 29 g/s at 2 s. The input fuel flow was maintained at 1 g/s.

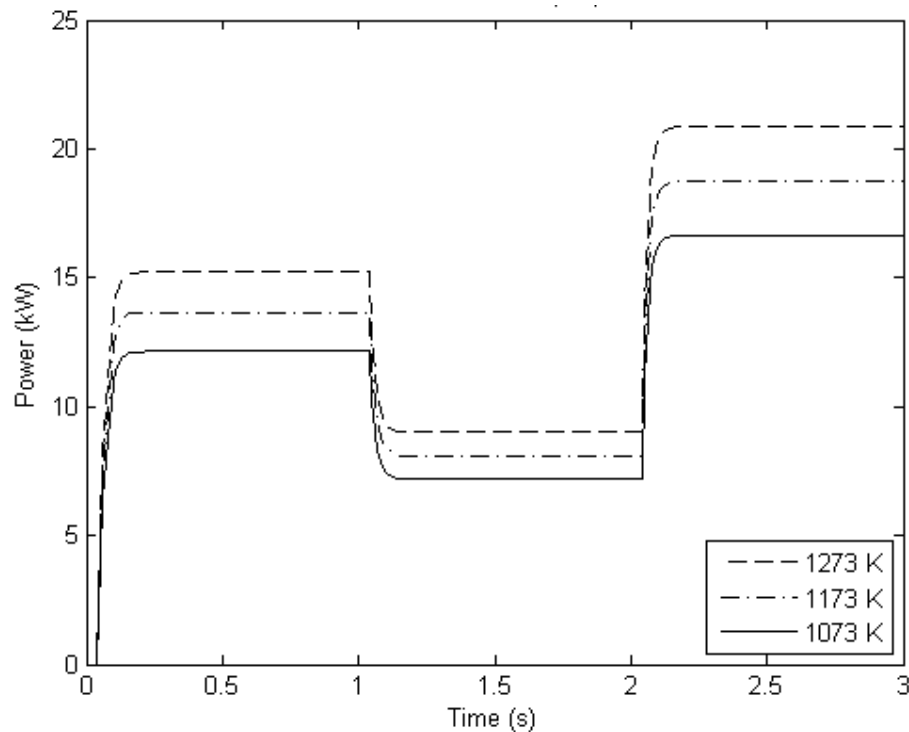


Fig. 2.31 Power produced from combustion of CH_4 for input fuel temperatures of 1073 K (bottom), 1173 K (middle), 1273 K (top) with varying oxidizer input flow rates.

As Fig. 2.31 demonstrates, when the oxidizer flow is decreased, the power produced decreases and increases when oxidizer flow is increased. The changes in power corresponding to oxidizer flow rates are due to creating a fuel rich and fuel lean AFR when the oxidizer is decreased and increased, respectively. In the same fashion, the temperature response of the combustor model to changes in oxidizer flow rates are observed in Fig. 2.32. The output temperature is decreased when the oxidizer flow rate is decreased and increased as a result of an increase in oxidizer flow. In a fuel rich mixture the combustion efficiency will decrease, meaning an increase in unused fuel. Similarly, in a fuel lean mixture, the amount of fuel reacting with the oxidizer is increased along with the efficiency of combustion [35].

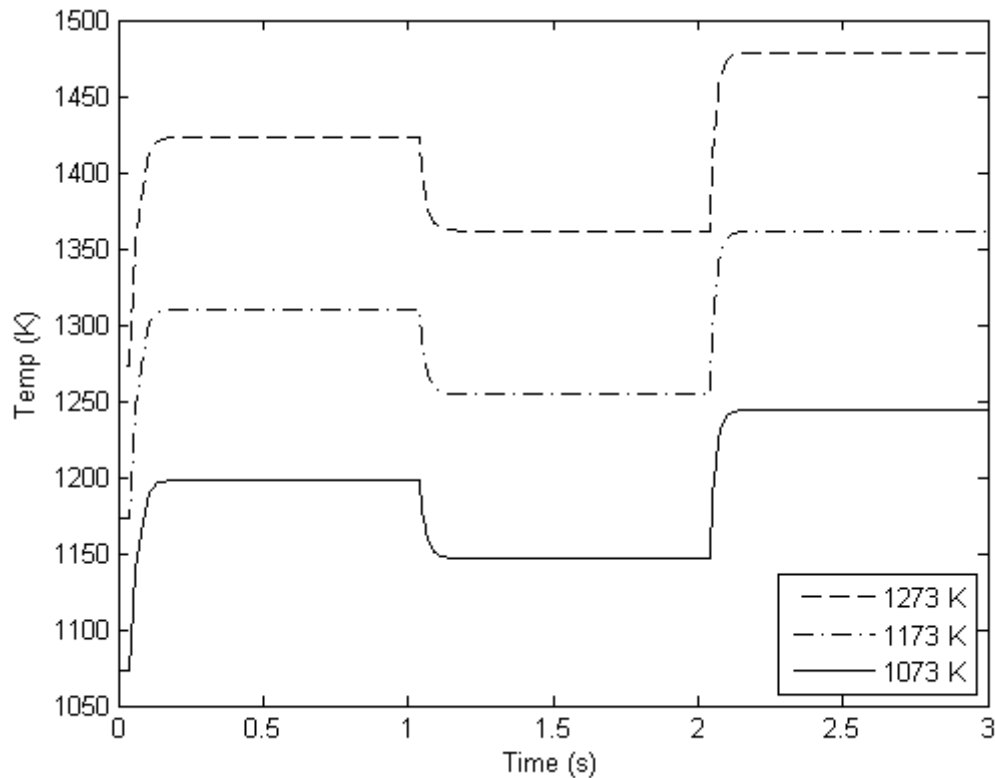


Fig. 2.32 Temperature change due to combustion of CH_4 for input fuel temperatures of 1073 K (bottom), 1173 K (middle), 1273 K (top) with varying oxidizer input flow rates.

Heat Exchanger

Heat Exchanger Operation

A heat exchanger transfers the energy of two cross-streams for the benefit of increasing the energy, or temperature, of one of the streams. The two streams can be classified as a hot and cold stream, as shown in the configuration of Fig. 1.1. As the hot stream and cold stream flow through the heat exchanger, the energy of the hot stream is transferred to the cold stream. The energy flow through the heat exchanger is shown in Fig. 2.33.



Fig. 2.33 General operation of a heat exchanger with air and fuel as inputs.

As Fig. 2.33 demonstrates, the hot stream (air) enters one channel of the heat exchanger (1) while the cold stream (fuel) enters another channel (2). The energy from the hot stream is transferred, via a solid medium, to the cold stream. This results in the loss of energy from the hot stream and gain of energy within the cold stream in the form of heat loss and gain, respectively. For this research, an indirect, or countercurrent, heat exchange method was assumed, indicating a medium exists between the two streams.

When mixture, or direct contact, occurs between the two streams, the heat exchanger is said to have a direct method of transferring heat. For this system, the two streams consist of fuel and air in gaseous form.

Heat Exchanger Model of the SOFC-MT CC System

The thermodynamic mathematical model of the heat exchanger consists of a fuel stream and an exhaust/air stream, as shown in Fig. 2.31. The fuel stream consists of the unused fuel from the anode channel of the SOFC, and the exhaust stream consists of half the exhaust stream from the compressor of the MT. The gas heat capacity, C_p , of the fuel stream is calculated by (2.55). The heat capacity of the fuel stream is calculated for each element of the fuel and multiplied by the temperature and flow rate of that particular element to calculate the molar enthalpy of the stream, using (2.54).

Due to the operational temperatures of the SOFC and MT at approximately 1200 K and 750 K, respectively; the unused fuel from the SOFC anode channel is considered to be the hot stream, while the exhaust from the MT is considered to be the cold stream. The temperatures of the air and fuel outlets are the only concern for the purpose of this research. The temperatures of the air and fuel outlets are assumed to be the temperatures of the metal at the outlets, and are given by (2.57) and (2.58), respectively.

$$\frac{dT_{cold_{out}}}{dt} = \frac{1}{m_{metal}C_{p_{metal}}} [\dot{m}_{cold}C_{p_{cold}}(T_{cold_{in}} - T_{cold_{out}}) + q] \quad (2.57)$$

$$\frac{dT_{hot_{out}}}{dt} = \frac{1}{m_{metal}C_{p_{metal}}} [\dot{m}_{hot}C_{p_{hot}}(T_{hot_{in}} - T_{hot_{out}}) - q] \quad (2.58)$$

Where $T_{i_{out}}$ is the temperature of the metal at the i outlet, $T_{i_{in}}$ is the temperature of the metal at the i inlet, m_{metal} is the mass of the heat exchanger plate, C_{p_i} is the

specific heat capacity of the i material or stream, $\dot{m}_i$ is the mass flow rate of the i stream, and q is the rate of heat transfer found by Eq. (2.59) [42].

$$q = \frac{e^\chi}{\frac{e^\chi}{\dot{m}_{hot}C_{p_{hot}}} - \frac{1}{\dot{m}_{cold}C_{p_{cold}}}} (T_{hot_{in}} - T_{cold_{in}}) \quad (2.59)$$

Where χ is a dimensionless parameter defined as

$$\chi = k_{cond} \cdot l_{HX} \cdot \left(\frac{1}{\dot{m}_{hot}C_{p_{hot}}} - \frac{1}{\dot{m}_{cold}C_{p_{cold}}} \right) \quad (2.60)$$

Where k_{cond} is the thermal conductivity of the streams (W/m K) and l_{HX} is the thickness of the heat exchanger plate (mm).

The high operating temperature of the system led to the assumption that the metal used in the heat exchanger requires a relatively large specific heat capacity. A metal of carbon steel was selected to represent a common heat exchanger material for the specified temperature range of the system, with a specific heat capacity of approximately 0.7 J/g K [43]. The specific heat capacity of the metal only affects the rate of heat transfer and not the final temperatures of the streams. The steady state temperatures of the streams will be reached on a large time scale (10^2 - 10^3 s) as a result of increasing the specific heat capacity of the metal.

The inputs for the heat exchanger block are: mass flow rate of unused fuel from the anode channel of the SOFC (mol/s) and associated temperature of stream (K), along with half of the mass flow rate of exhaust from the compressor of the MT (mol/s) and associated temperature of the stream (K). The outputs of the block are the temperatures of the fuel and air streams at the outlets of the heat exchanger (K).

Heat Exchanger Simulation

The thermodynamic heat exchanger model was implemented within the Matlab/Simulink environment. A simulation was considered for steady state operation with a cold stream of exhaust from the MT (CO₂ and H₂O) and a hot stream of unused fuel from the anode channel of the SOFC (90% H₂ and 10% H₂O) with a variation of temperature. The simulation parameters are shown in Table 2.6.

Table 2.6 Simulation Parameters for Heat Exchanger with Various Input Cold Stream Temperatures

Parameter	Value
Cold stream flow (MT exhaust)	140 g/s (46.667 g/s CO ₂ , 93.333 g/s H ₂ O)
Hot stream flow (SOFC unused H ₂ & H ₂ O)	0.106 g/s H ₂ , 0.773 g/s H ₂ O
Input cold stream temperatures	700K, 800 K, 900K
Input hot stream temperature	1173 K

The hot stream, SOFC unused H₂, flow rate was determined to be the approximate amount of unused H₂ at the outlet of the anode channel of the 5 kW SOFC model. The cold stream, MT exhaust, flow rate was determined to be approximately half of the exhaust flow of the MT rated at 250 kW.

The ideal heat transferred between the countercurrent streams is governed by conservation of energy and can be described by the enthalpy balance written as [41]

$$Q = \dot{m}_{hot} C_{p_{hot}} (T_{hot_{in}} - T_{hot_{out}}) = \dot{m}_{cold} C_{p_{cold}} (T_{cold_{out}} - T_{cold_{in}}) \quad (2.61)$$

As Eq. (2.61) suggests, even though the flow rate of the cold stream is much greater than the hot stream, the energy from heat transfer will be equivalent in both directions due to conservation of energy. Furthermore, for similar specific heat capacities of the hot and cold streams, a large difference in flow rates implies a large difference in

temperature changes. Specifically, the large flow rate of the cold stream suggests a small change in temperature when compared to large change in temperature suggested by the small flow rate of the hot stream.

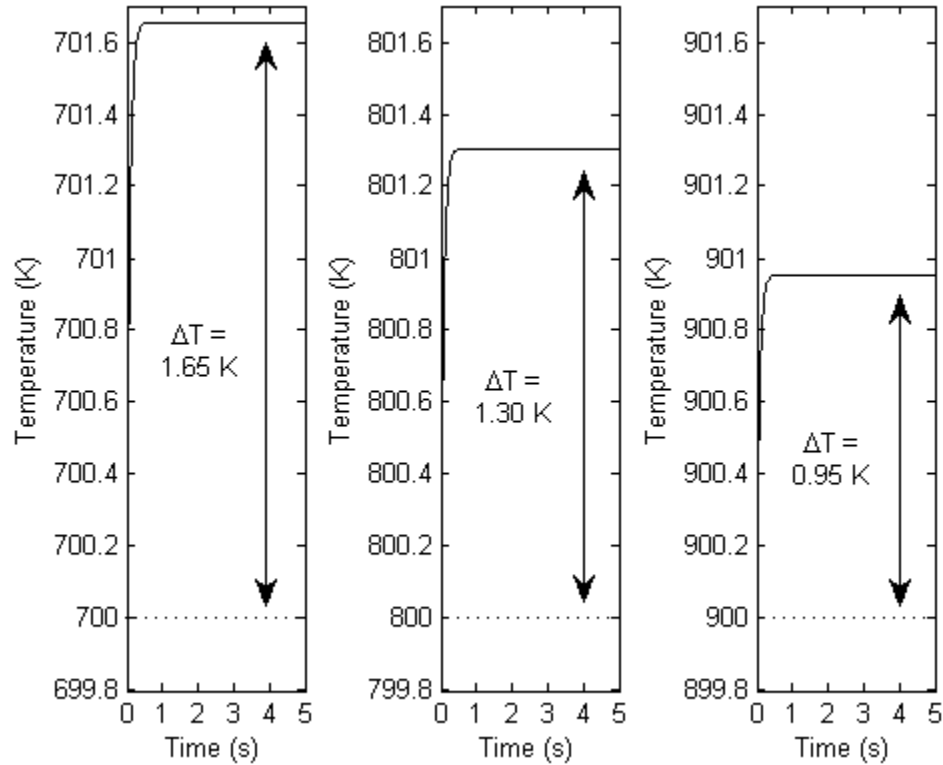


Fig. 2.34 Outlet temperature of cold stream for input temperatures of 700 K (left), 800 K (middle), 900 K (right).

As Fig. 2.34 demonstrates, the cold stream temperature is increased 1.65 K when the input cold stream is 700 K, 1.30 K for an input cold stream temperature of 800 K and 0.95 K for an input cold stream temperature of 900 K. The decrease of temperature change as the input cold stream temperature increases is caused by the decrease of heat transfer between the streams.

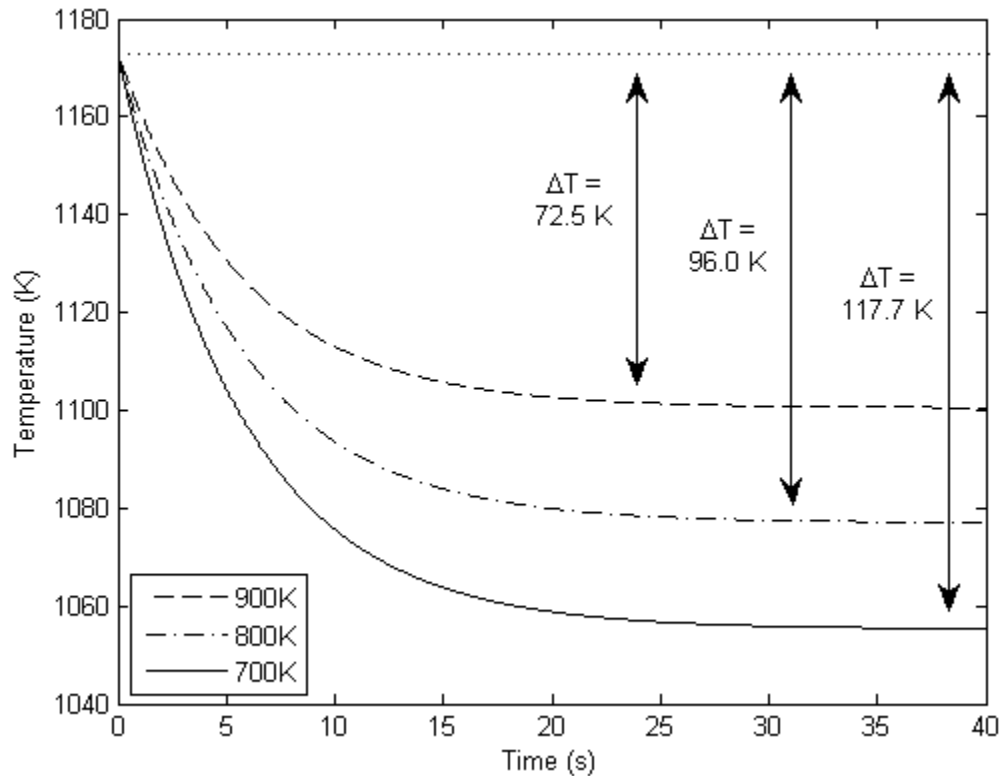


Fig. 2.35 Outlet temperature of hot stream for input cold stream temperatures of 700 K (bottom), 800 K (middle), 900 K (top).

In the same manner, Fig. 2.35 depicts the change in temperature of the hot stream with an initial input temperature of 1173 K and initial cold stream temperatures of 700 K, 800 K, and 900 K. The heat transferred to the cold stream is lost from the hot stream, thus a drop of temperature. As the input cold stream temperature increases the change in temperature of the hot stream decreases, as less heat is transferred due to the smaller temperature differential. In order to verify these findings, the energy conservation of the system must be examined.

Based on the conservation of energy the enthalpy balance, when the input cold and hot stream temperatures are 800 K and 1173 K, respectively, can be shown in the

following manner through the use of Eq. (2.61), data provided by simulation, and the specific heat capacities of the species, listed in Table 2.7.

Table 2.7 Specific Heat Capacities of Species in Heat Exchanger at Various Temperatures

Species	Temperature (K)	Specific Heat Capacity (J/g K)
H ₂	1173	15.317
H ₂ O	1173	2.410
CO ₂	800	1.169
H ₂ O	800	2.150

$$\begin{aligned}
 Q_{hot} &= \dot{m}_{hot} C_{p_{hot}} (T_{hot_{in}} - T_{hot_{out}}) \\
 &= \left[(\dot{m} C_p)_{H_2} + (\dot{m} C_p)_{H_2O} \right] (T_{hot_{in}} - T_{hot_{out}}) \\
 &= \left[\left(0.106 \frac{g}{s} \times 15.317 \frac{J}{g \cdot K} \right) + \left(0.773 \frac{g}{s} \times 2.410 \frac{J}{g \cdot K} \right) \right] \\
 &\quad \times (1173 K - 1077.016 K) \\
 &= \left(3.496 \frac{J}{s \cdot K} \right) 95.984 K \\
 &= 335.6107 \frac{J}{s}
 \end{aligned} \tag{2.62}$$

$$\begin{aligned}
 Q_{cold} &= \dot{m}_{cold} C_{p_{cold}} (T_{cold_{out}} - T_{cold_{in}}) \\
 &= \left[(\dot{m} C_p)_{CO_2} + (\dot{m} C_p)_{H_2O} \right] (T_{cold_{out}} - T_{cold_{in}}) \\
 &= \left(47.055 \frac{g}{s} \times 1.169 \frac{J}{g \cdot K} \right) + \left(94.116 \frac{g}{s} \times 2.150 \frac{J}{g \cdot K} \right) \\
 &\quad \times (801.304 K - 800 K) \\
 &= \left(257.372 \frac{J}{s \cdot K} \right) 1.304 K \\
 &= 335.6644 \frac{J}{s}
 \end{aligned} \tag{2.63}$$

As Eq. (2.62) and (2.63) show, the calculated energy from heat transferred is nearly equivalent for the hot and cold stream, verifying the conservation of energy based on enthalpy balance. The calculated difference of heat transfer of the countercurrent

streams is 0.0537 J/s. The small scale of fuel flow in the hot stream can account for this discrepancy of 53.7 mW and within the acceptable range of error due to the overall scale of heat transfer greater than 335 J/s, or 0.335 kW.

For all cold stream input temperatures, the final temperatures were attained on the order of seconds, whereas the hot stream steady state temperatures were achieved after tens of seconds. The difference between the volumetric flow rates of both streams and the principals of molecular diffusion led to the convergence to steady state on different time scales.

Another simulation was conducted to observe the reaction of the heat exchanger to similar mass flow rates of the countercurrent streams. The simulation parameters are given in Table 2.8.

Table 2.8 Simulation Parameters for Heat Exchanger with Similar Flow Rates and Various Input Cold Stream Temperatures

Parameter	Value
Cold stream flow (MT exhaust)	140 g/s
Hot stream flow (SOFC unused H ₂ & H ₂ O)	10.18 g/s of H ₂ , 74.21 g/s of H ₂ O
Input cold stream temperatures	700K, 800 K, 900K
Input hot stream temperature	1173 K

When the hot stream flow rate is increased by a factor of 96 to represent a 480 kW SOFC power plant, the cold stream temperature is increased 165.4 K when the input cold stream is 700 K, 130.4 K for an input cold stream temperature of 800 K and 95.5 K for an input cold stream temperature of 900 K, as shown in Fig. 2.34. It should be noted that there exists a correlation between the flow rate of a stream and the change in temperature during heat transfer. The decrease of temperature change as the input cold

stream temperature increases is caused by the decrease of heat transfer between the streams.

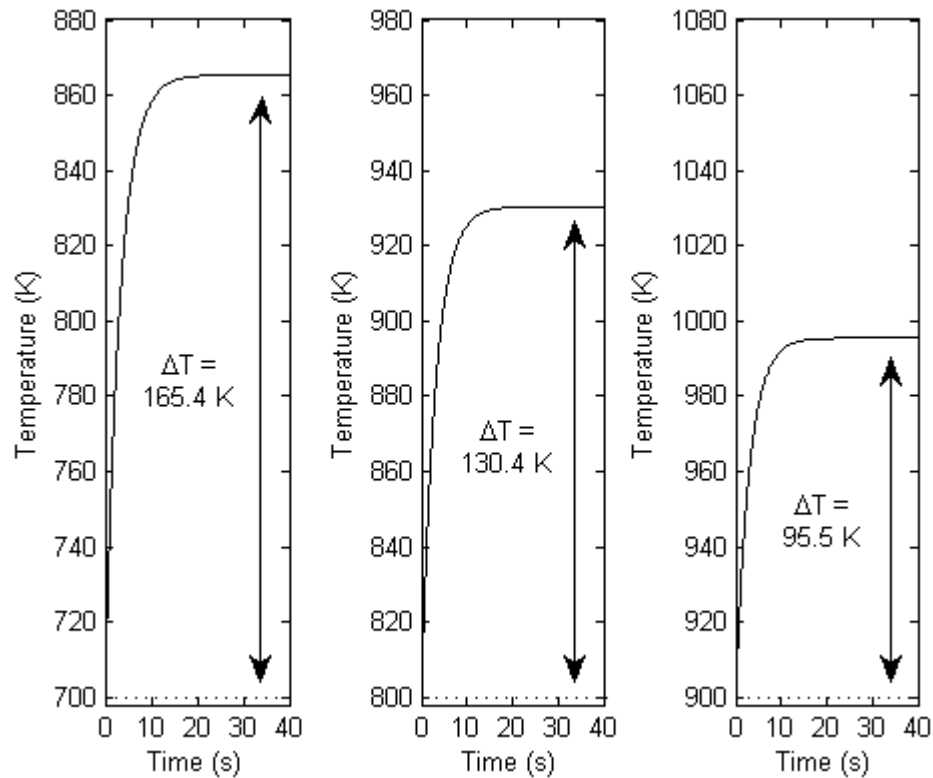


Fig. 2.36 Outlet temperature of cold stream for input temperatures of 700 K (left), 800 K (middle), 900 K (right) with similar hot stream flow rate.

In the similar manner, Fig. 2.37 shows the change in temperature of the hot stream with an initial input temperature of 1173 K and initial cold stream temperatures of 700 K, 800 K, and 900 K. The heat transferred to the cold stream is lost from the hot stream, thus a drop of temperature. The increase of hot stream flow rate increases the amount of heat transfer, but not the steady state temperature of the hot stream. The time scale for steady state convergence is also lessened for the hot stream and is similar to that of the cold stream.

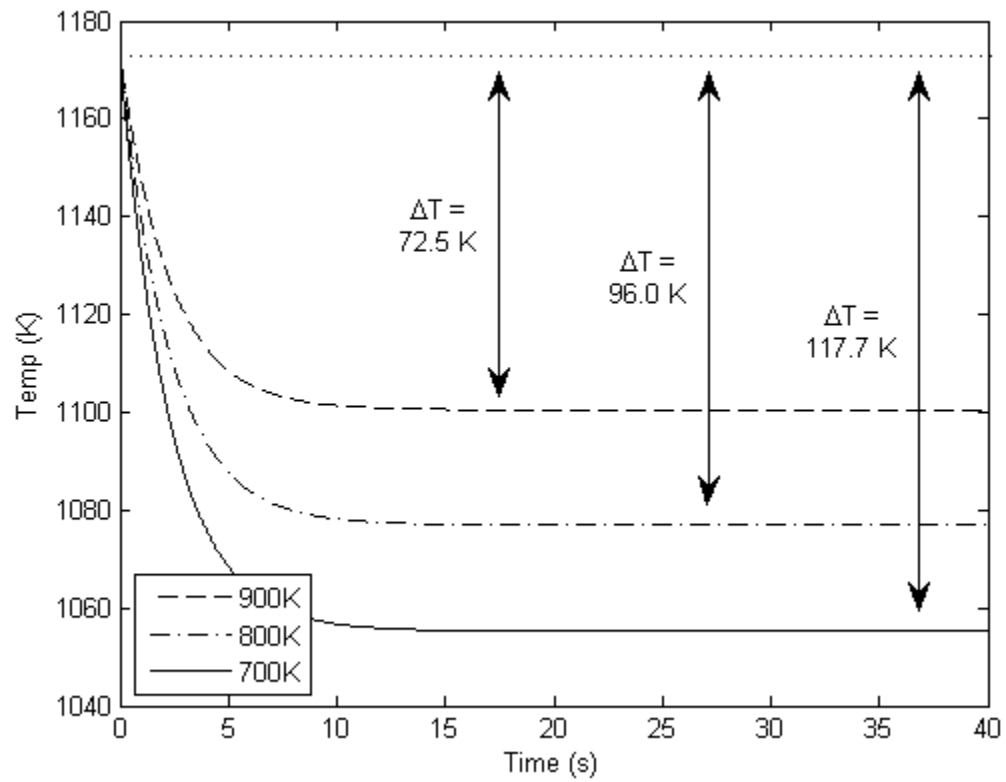


Fig. 2.37 Outlet temperature of hot stream for input cold stream temperatures of 700 K (bottom), 800 K (middle), 900 K (top) with increased hot stream flow rate.

EFFICIENCY EVALUATION OF SOFC-MT-CC SYSTEM

Combined-Cycle

Combined-Cycle Operation

Combined cycle (CC) operation refers to the coupling of heat streams for purposes within the system itself. For the research of this thesis, CC operations refer to the coupling of the MT exhaust gas stream and the unused SOFC fuel stream from the anode channel. Such an implementation of CC operation is typical in SOFC-MT systems, utilizing the unused exhaust from the MT in a heat recovery operation to preheat the fuel into the MT to increase efficiency.

Combined-Cycle Model

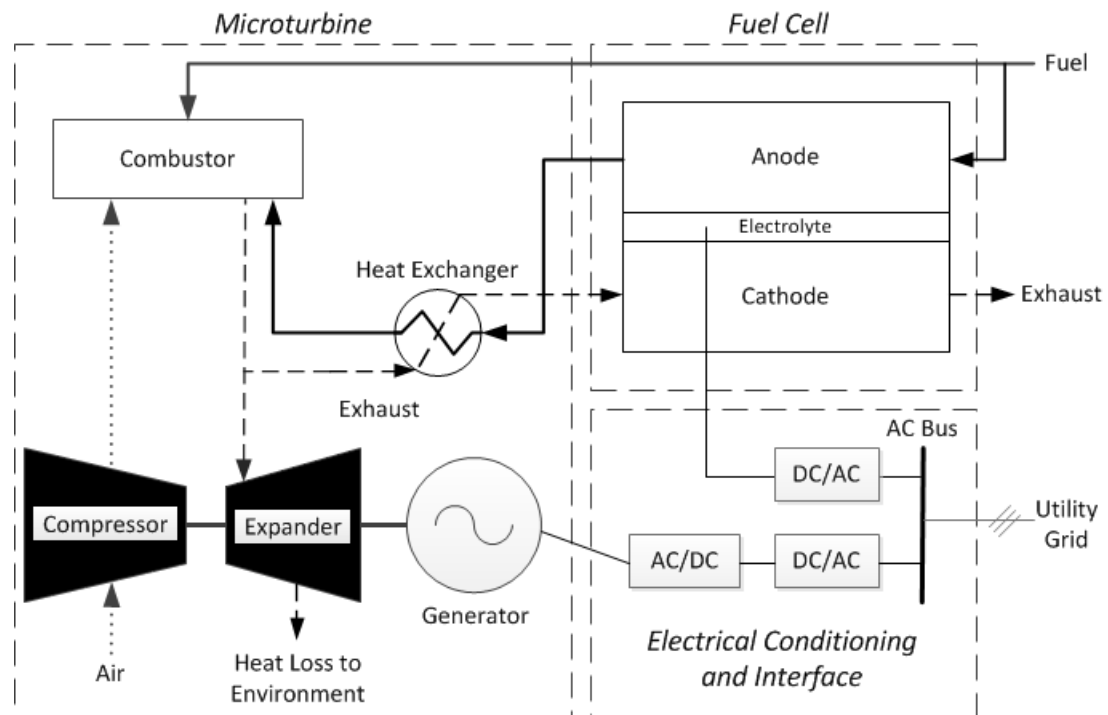


Fig. 3.1 Combined cycle operation of SOFC/MT system.

In term of energy flow, CC can be described as using the same heat source for multiple purposes, namely for mechanical energy, which in turn provides additional electrical energy through an electrical generator. The exhaust from one heat engine can be used as the heat source in another heat engine. In the case of this thesis, the exhaust from the combustor flows through one channel of a heat exchanger while the unused fuel from the SOFC flows in the other channel. The cross streams of air and fuel allow for the heat energy within the fuel entering the MT combustor to transfer to the exhaust stream into the cathode of the SOFC, instead of ambient air, as shown in Fig. 3.1.

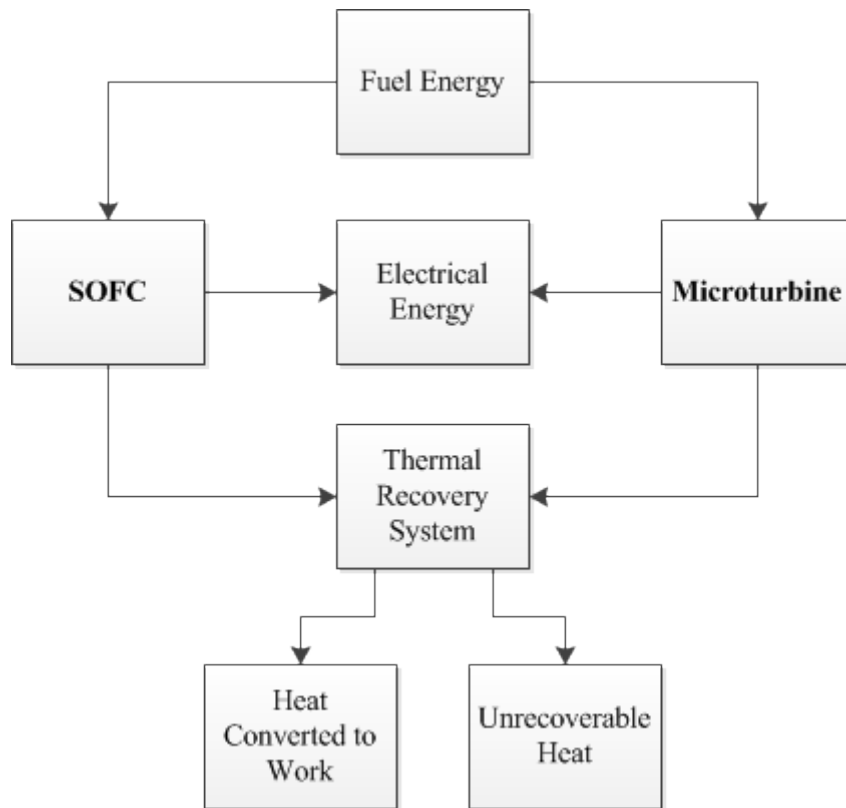


Fig. 3.2 Energy flow diagram of combined cycle SOFC/MT system.

The energy flow of the system begins with the fuel energy provided to the SOFC and MT, which, in turn produce electrical energy. The heat losses of each device are utilized with a thermal recovery system in the form of a combined cycle. The thermal recovery system either converts the heat to work or is lost to the environment. The energy flow of the system is visualized in Fig. 3.2.

By coupling the exhaust stream of the MT to the combustor and heat exchanger, along with the unused fuel from the SOFC anode channel to the heat exchanger, the system is said to be in CC operations. For the system to not operate in CC the heat exchanger would be disconnected, causing the unused fuel from the SOFC anode channel to not be preheated and utilized within the MT combustor and the exhaust stream from the MT to succumb to heat loss to the environment; ultimately decreasing the overall efficiency of the system through more losses.

Combined-Cycle Simulation

Table 3.1 Simulation Parameters of SOFC-MT-CC Model

Parameter	Value
Fuel cell	
SOFC power plant	220 V/480 kW Twelve 40 kW SOFC arrays connected in parallel
SOFC array	220 V/40 kW, consisting of 4 (series) x 2(parallel) = eight 5 kW SOFC stacks
Fuel molar flow rate	9.216 mol/s (90% H ₂ + 10% H ₂ O)
Air molar flow rate	110.592 mol/s
Pressures (anode & cathode)	3 atm
Initial fuel & air temperature	1173 K
Microturbine	
Rated power	250 kW
Operation efficiency	35 %
Operation temperature	750 K

The SOFC-MT-CC model was implemented within the Matlab/Simulink environment. A simulation was conducted for the purpose of demonstrating the efficiency increase of CC operation for a SOFC-MT system. The simulation was considered for steady state operation over a variety of input fuel temperatures. The model was simulated in non-CC and CC operation in order to determine an increase of efficiency. The simulation was run for the medium time scale (120 s). The simulation parameters are shown in Table 3.1.

The additional energy generated from the unused fuel from the SOFC anode channel decreases the amount of power required and subsequent the fuel consumption of the MT. The amount of unused fuel and water vapor exiting the SOFC anode channel under full load operating condition, from Eq. (2.10b) and (2.11b) is shown in Fig. 3.3.

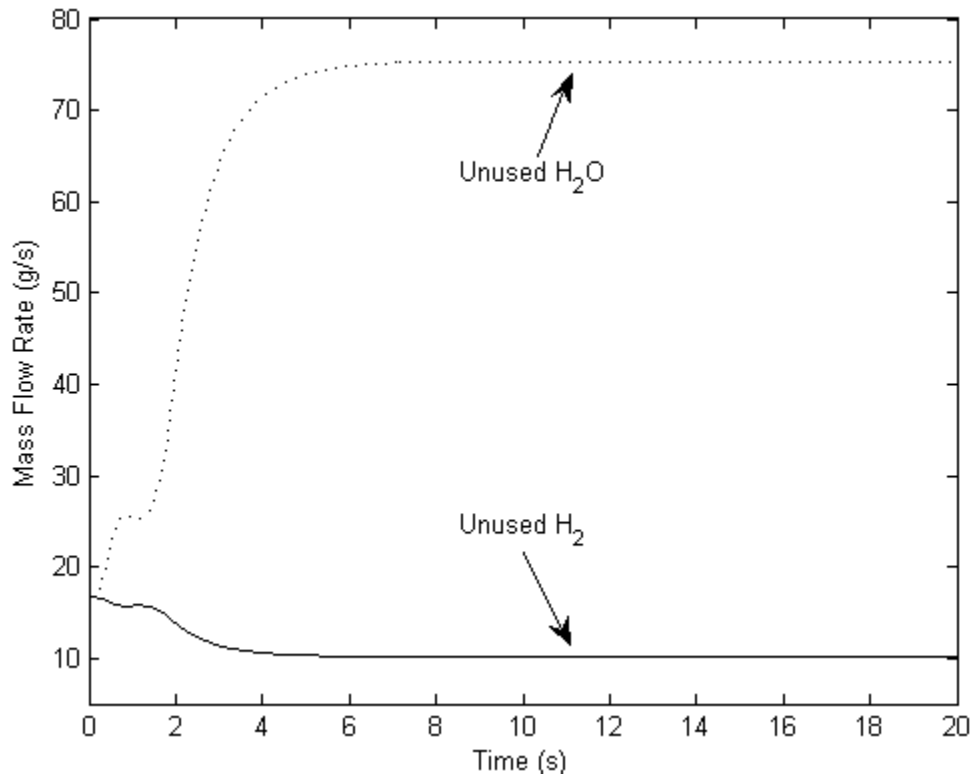


Fig. 3.3 Unused H₂ and H₂O from SOFC anode channel in CC operation.

The unused fuel reaches a steady state value in eight seconds. During startup of the SOFC the amount of unused H_2 decreases from 16.7 g/s to a steady state value of 10.2 g/s. Inversely the amount of unused H_2O increases during startup from 16.6 g/s to a steady state value of 75.5 g/s. When the SOFC reaches its operating point the efficiency of fuel utilization increases, demonstrated by a decrease of unused H_2 . Although the simulation was conducted over 120 s, the unused H_2 and H_2O is shown over 20 s to properly show the startup characteristics of the SOFC. The amount of H_2 and H_2O remained constant for the remainder of the simulation at 10.2 g/s and 75.5 g/s, respectively.

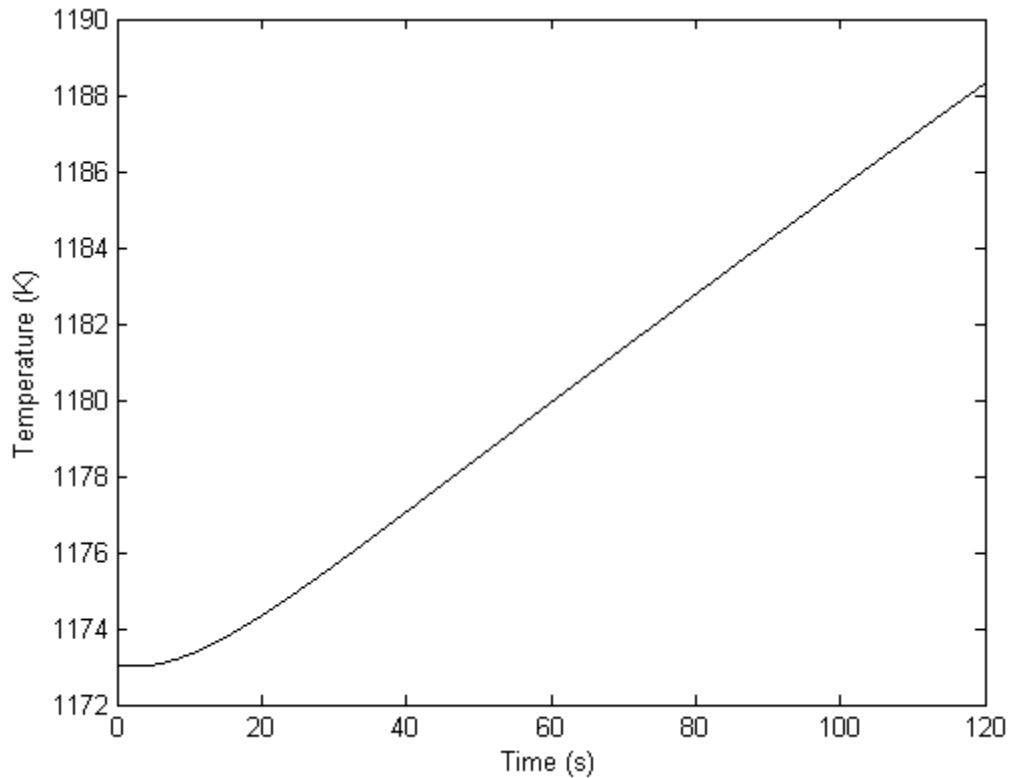


Fig. 3.4 Temperature of species at outlet of SOFC anode channel over medium time scale.

Even though the output flow rate of the unused species achieves a steady state during the medium time scale, the large time scale temperature characteristics of the SOFC still exist. The temperature of the unused fuel and water vapor exiting the SOFC anode channel over the medium time scale simulation is shown in Fig. 3.4.

As Fig. 3.4 demonstrates, the initial temperature of unused H_2 and H_2O exiting the SOFC anode channel is equivalent to the initial fuel input temperature of 1173 K. The cell temperature increases during continued operation of the SOFC causing a temperature increase at the outlet of the anode channel. The rate of increase is near linear for the medium time scale. However, the temperature of the outlet species of the anode channel arrives at a steady state value over the large time scale, as shown in Fig. 3.5.

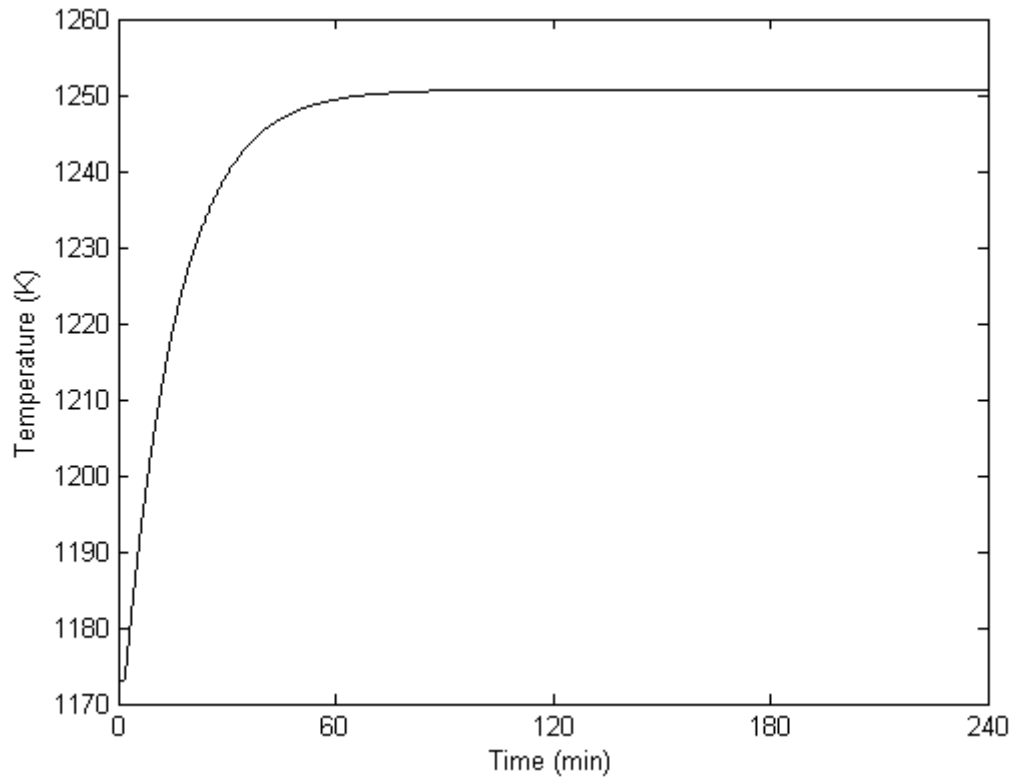


Fig. 3.5 Temperature of species at outlet of SOFC anode channel over large time scale [4].

The steady state temperature of the unused species at the outlet of the SOFC anode channel arrives near 1250 K after approximately 100 min. of operation. The slow thermodynamic response of the SOFC is expected and has been validated by [4].

The amount of power, active and reactive, provided by the SOFC is shown in Fig. 3.6. After start up transients, the active power delivered by the SOFC is around 480 kW and the reactive power is approximately 30 kVAR. The fluctuations of active and reactive power observed in combined cycle are due the frequency deviations within the SOFC power electronics. Consequently, the active and reactive power observed from the MT exhibits increased fluctuations, as compared to the results obtained for the isolated MT case given in Chapter 2.

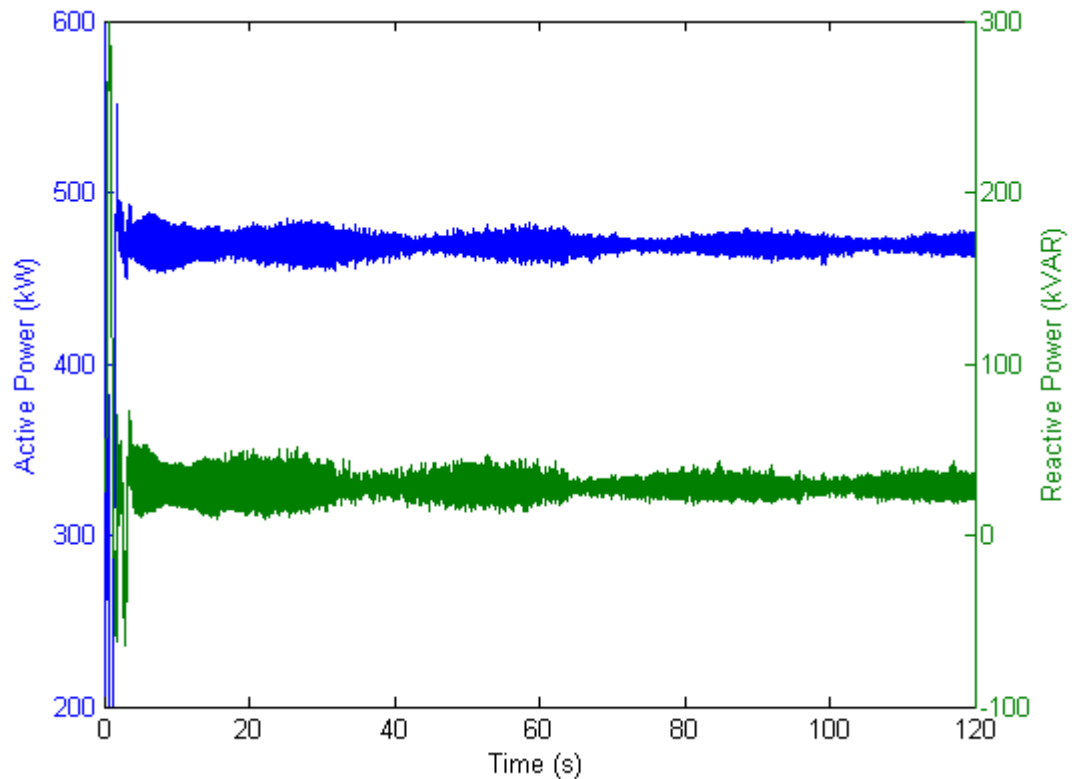


Fig. 3.6 Active and reactive power delivered by the SOFC.

The unused fuel exiting the anode channel is transported to the combustor to provide an increase of efficiency by decreasing the fuel necessary to operate the MT at rated power. The amount of energy added from heat due to combustion of the unused fuel of the SOFC anode channel is shown in Fig. 3.7.

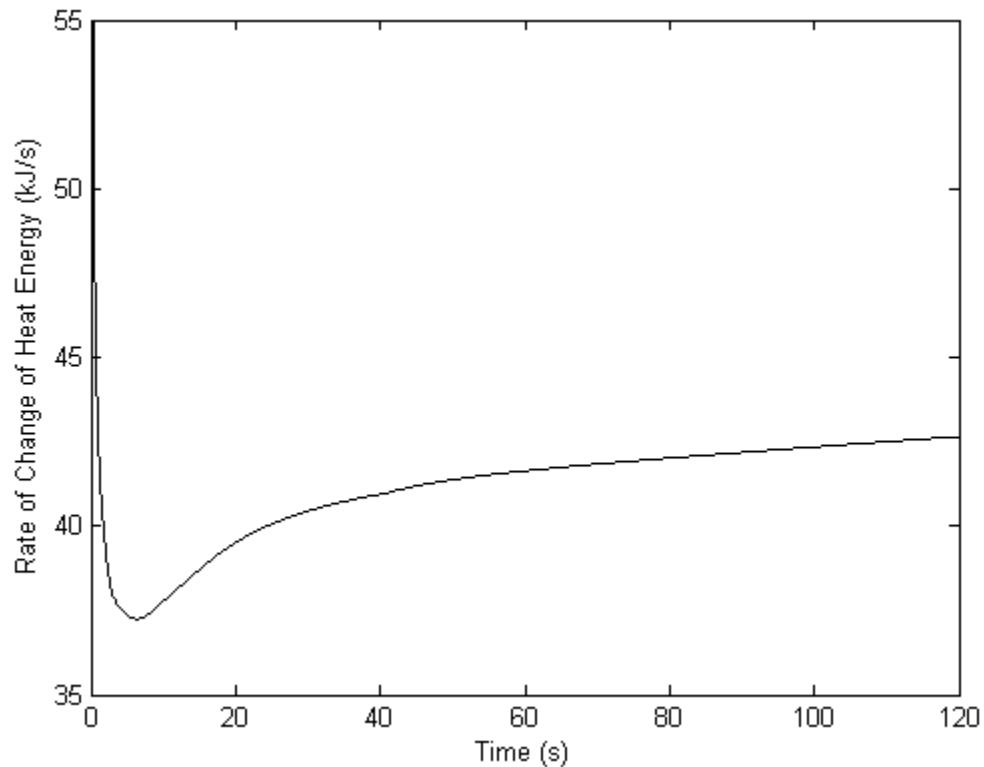


Fig. 3.7 Rate of change of heat energy flow rate from combustion of unused species from SOFC anode channel.

The amount of heat energy produced from the combustion of unused species exhibits a dramatic decrease during startup of the SOFC due to the change of amount of unused species over the same time period. After the amount of unused species exiting the anode channel reaches steady state, the rate of change of heat energy at the outlet of the combustor increases. The rate of increase lessens during the simulation to a final value

around 43 kJ/s. The rate of energy from heat can also be regarded as the amount of power produced due to combustion of the unused species. Although the amount of unused species achieves a near constant value the energy from heat continues to increase. The continued increase of heat energy can then be attributed to the increasing temperature of the unused species. Given the flow rate of a given species remains constant, an increase in temperature or specific heat capacity, which also increases with temperature, will cause an increase in energy from heat. In the case of this simulation, the flow rate remains near constant by 20 seconds while the temperature and subsequent specific heat capacity increase, causing an increase in energy from heat. Likewise, the change in enthalpy of the system also exhibits an increase of the same nature, shown in Fig. 3.8.

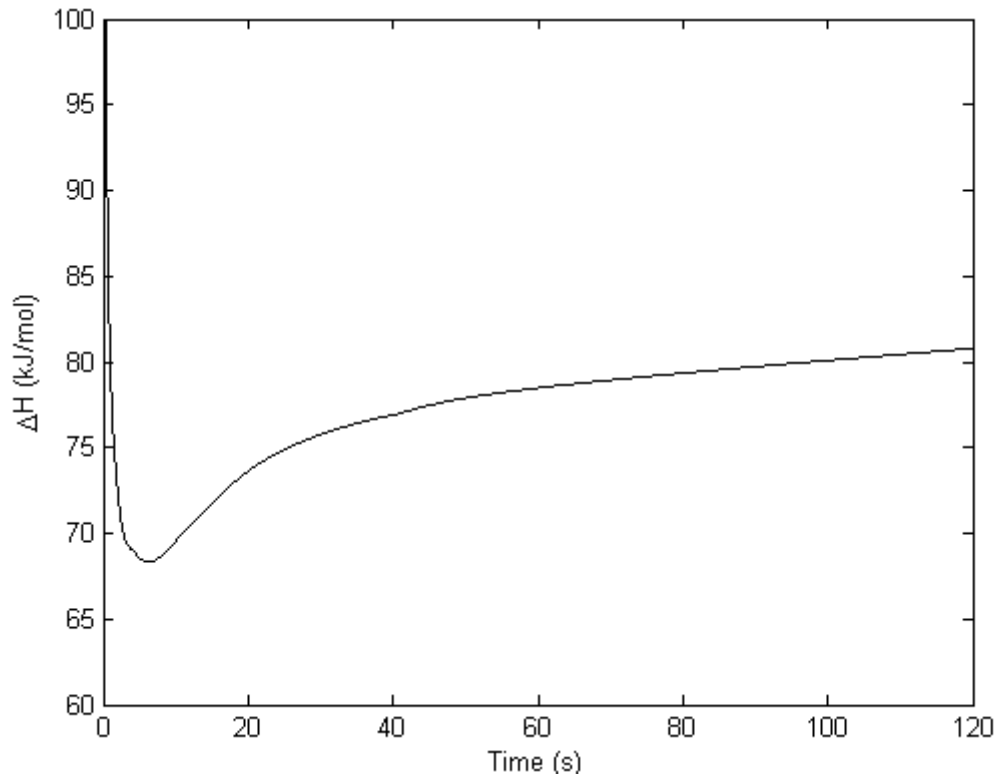


Fig. 3.8 Change in enthalpy, ΔH , of combustion products from unused species of SOFC anode channel.

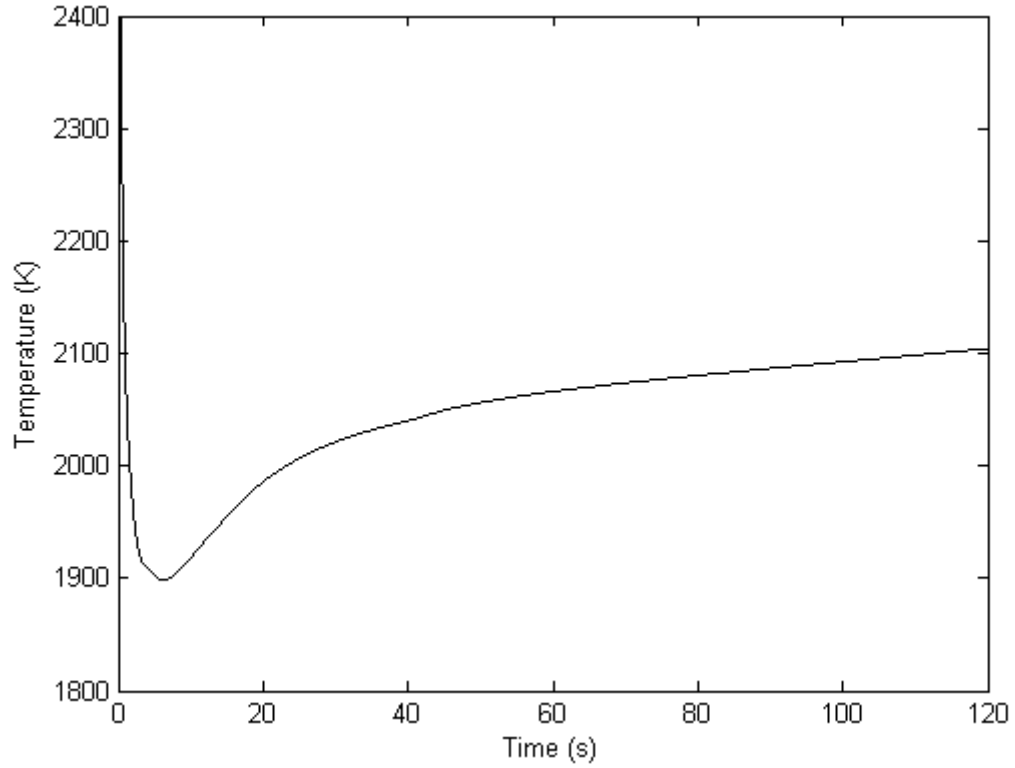


Fig. 3.9 Adiabatic flame temperature from combustion of unused species of SOFC anode channel.

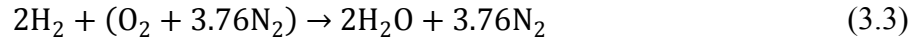
The temperature that results from a complete combustion reaction, or the adiabatic flame temperature, is shown in Fig. 3.9. The temperature resulting from combustion is found by calculating the change of temperature of the system based on Eq. (2.54). The change in temperature is summed with the initial temperature to find the final temperature, shown in Eq. (3.1) and (3.2).

$$Q = \dot{n}_i c_p \Delta T \Rightarrow \Delta T = \frac{Q}{\dot{n}_i c_p} \quad (3.1)$$

$$T_f = T_i + \Delta T = T_i + \frac{Q}{\dot{n}_i c_p} \quad (3.2)$$

The adiabatic flame temperature for the combustion of a stoichiometric CH_4 -air mixture, shown in Eq. (3.3), can be alternatively calculated by classical constant-pressure

estimation techniques based on enthalpy balance of the reactants and products, shown as follows [35].



For a constant-pressure system of 1 atm and an initial reactant temperature, T_i , of 298 K, the following assumptions are used:

- Complete combustion occurs, i.e. the product mixture consists of only H_2O and N_2 .
- The product mixture enthalpy is estimated using specific heat capacities at 1200 K, or the average of the initial temperature and the adiabatic flame temperature, T_{ad} , assumed to be 2100 K, i.e. $T_{avg} = \frac{1}{2}(T_i + T_{ad})$.

The species involved with the reaction, along with their respective values of enthalpy of formation and specific heat capacities necessary for estimating the adiabatic flame temperature are given in Table 3.2, with data provided by [36].

Table 3.2 Properties of Combustion Species for Adiabatic Flame Temperature Estimation

Species	Enthalpy of Formation at Standard Conditions $\Delta_f H_{\text{gas}}^o$ (kJ/kmol)	Specific Heat Capacity at $T_{\text{avg}} = 1200$ K C_p (J/mol K)
H_2	0	--
O_2	0	--
N_2	0	33.72
H_2O	-241 826	43.75

The enthalpy balance of the system can be represented as

$$H_{\text{react}} = \sum_{\text{react}} N_i \Delta_f H_i^o = H_{\text{prod}} = \sum_{\text{prod}} N_i \Delta_f H_i^o \quad (3.4)$$

Where N_i is the stoichiometric coefficient of species i and $\Delta_f H_i^o$ is the enthalpy of formation of species i . The enthalpy of the reactants can be written as

$$\begin{aligned} H_{react} &= N_{H_2} \Delta_f H_{H_2}^o + N_{O_2} \Delta_f H_{O_2}^o + N_{N_2} \Delta_f H_{N_2}^o \\ &= (2)(0) + (1)(0) + (3.76)(0) \\ &= 0 \end{aligned} \quad (3.5)$$

Since the adiabatic flame temperature is desired, the enthalpy of the products can be written as

$$\begin{aligned} H_{prod} &= \sum N_i [\Delta_f H_i^o + C_{p_i}(T_{ad} - 298)] \\ &= N_{H_2O} [\Delta_f H_{H_2O}^o + C_{p_{H_2O}}(T_{ad} - 298)] + N_{N_2} [\Delta_f H_{N_2}^o + C_{p_{N_2}}(T_{ad} - 298)] \\ &= (2)[-241\,826 + 43.75(T_{ad} - 298)] + (3.76)[0 + 33.72(T_{ad} - 298)] \\ &= (87.5 + 126.7872)T_{ad} - 509\,728 - 37\,784 \\ &= 212.2872 T_{ad} - 547\,512 \end{aligned} \quad (3.6)$$

Equating H_{react} and H_{prod} to solve for T_{ad} can then be written as

$$T_{ad} = \frac{547\,512}{212.2872} \cong 2555\,K \quad (3.7)$$

Comparing the result of the simplified estimation in Eq. (3.7), given the crudeness of the assumptions, with the equilibrium based computation of a hydrogen-air combustion of 2382 K [44], the result of 2555 K is a rather close estimation with a difference less than 200 K.

Following the same trend as the energy flow rate and enthalpy of the stream, the temperature of the products starts at a high value and decreases during the startup transiency of the system. Once the temperature has decreased to approximately 1900 K, the temperature exhibits the same increasing characteristics as energy and enthalpy to arrive at a temperature around 2100 K at 120 s. The temperature values reported are

consistent with the known adiabatic flame temperatures of common fuels, as shown in [35], [44].

The power from the microturbine generator originates from the combined energy from heat due to the combustion of unused fuel from the SOFC anode channel and the combustion of an input fuel of CH_4 . The combination of the heat energy streams flow over the turbine, turning the shaft, generating electricity through the implementation of an induction generator and subsequent power electronics. Since the amount of heat energy from the combustion of unused species from the SOFC anode channel has been discussed, the power, or heat energy flow rate, from the combustion of CH_4 in the MT is shown in Fig. 3.10

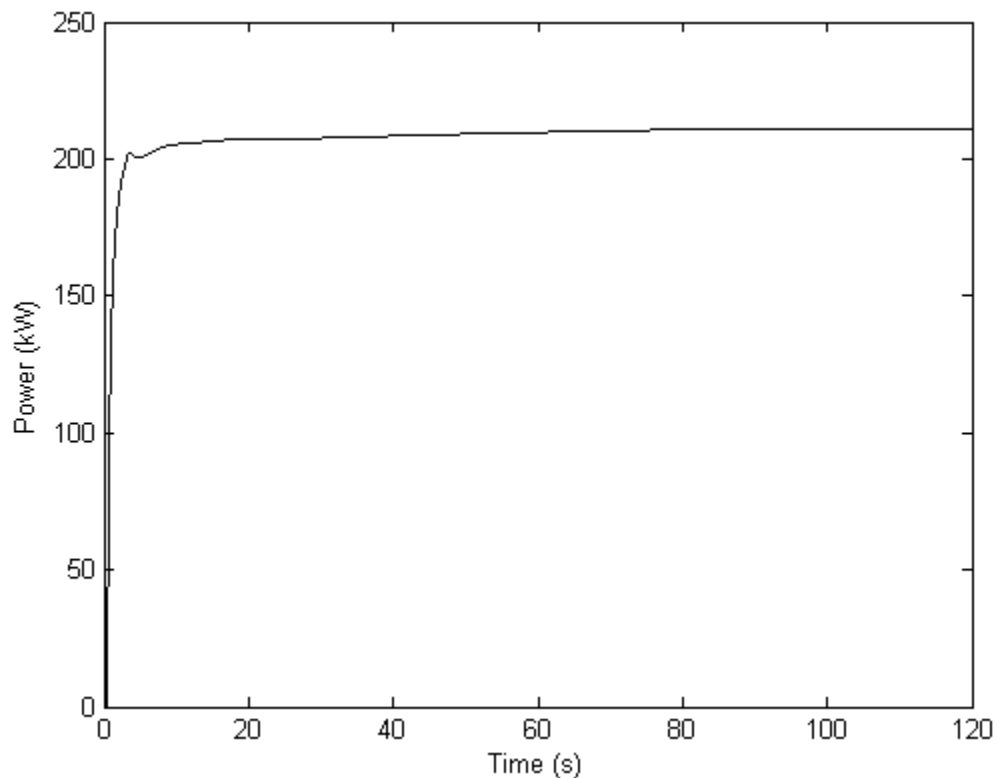


Fig. 3.10 MT power generated due to combustion of CH_4 as fuel.

As Fig. 3.10 shows, the contribution of power generated due to the combustion of CH_4 rapidly increases during the startup of the system and arrives at a near constant value around 210 kW after 120 s.

The amount of fuel and air flow to produce a continuous combustion reaction capable of generating the necessary power in non-CC and CC operation is shown in Fig.

3.11

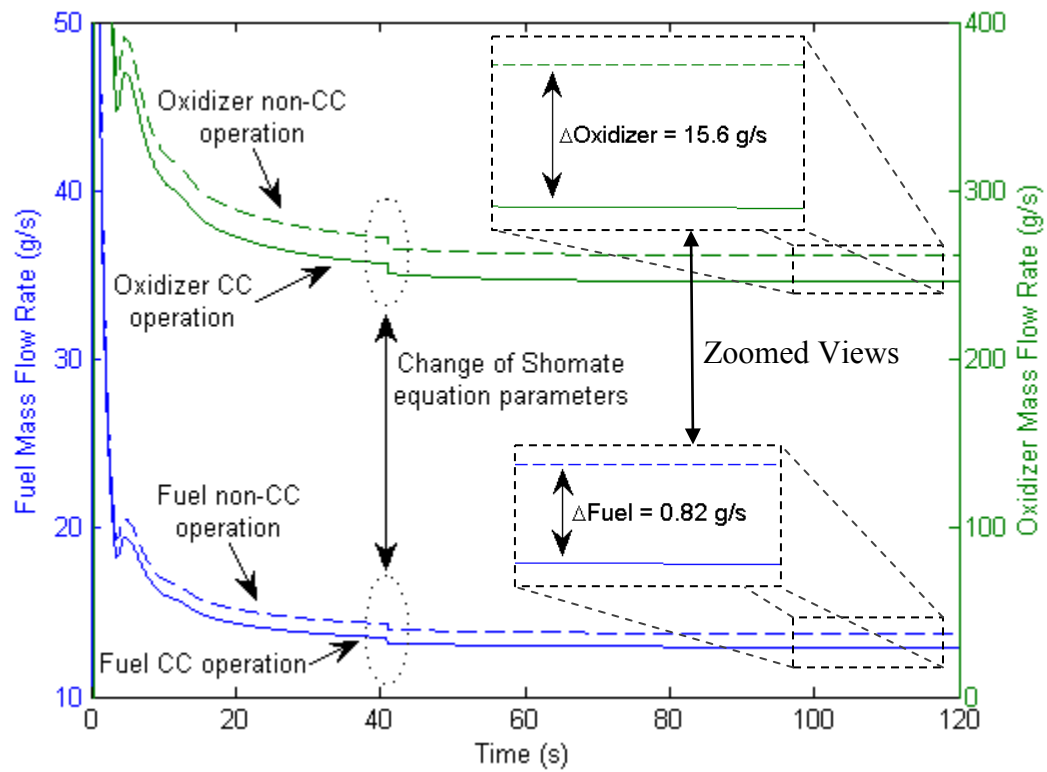


Fig. 3.11 Fuel and oxidizer mass flow rates of the MT corresponding to power required from the combustion of CH_4 .

During startup the amount of fuel and oxidizer is decreased due to the initial energy required for combustion. As the amount of power from the combustion of CH_4 reaches a near constant value, the amount of fuel and oxidizer required to maintain a

constant combustion reaction near steady state values accordingly. The abrupt drop of fuel and oxidizer seen at approximately 41 seconds is due to change of Shomate equation parameters [36]. The parameters provided by [36] are for a specific temperature range, if the temperature of a particular species exceeds the range, a new set of parameters are used. Also, the specified AFR was maintained throughout the simulation.

When the system is in CC operation the amount of fuel and oxidizer required to produce rated power of the MT is reduced by 5.96 %, or 0.82 g/s and 15.6 g/s, respectively. The additional energy provided by the combustion of unused species from the SOFC anode channel reduces the amount of CH_4 required by the MT during CC operation.

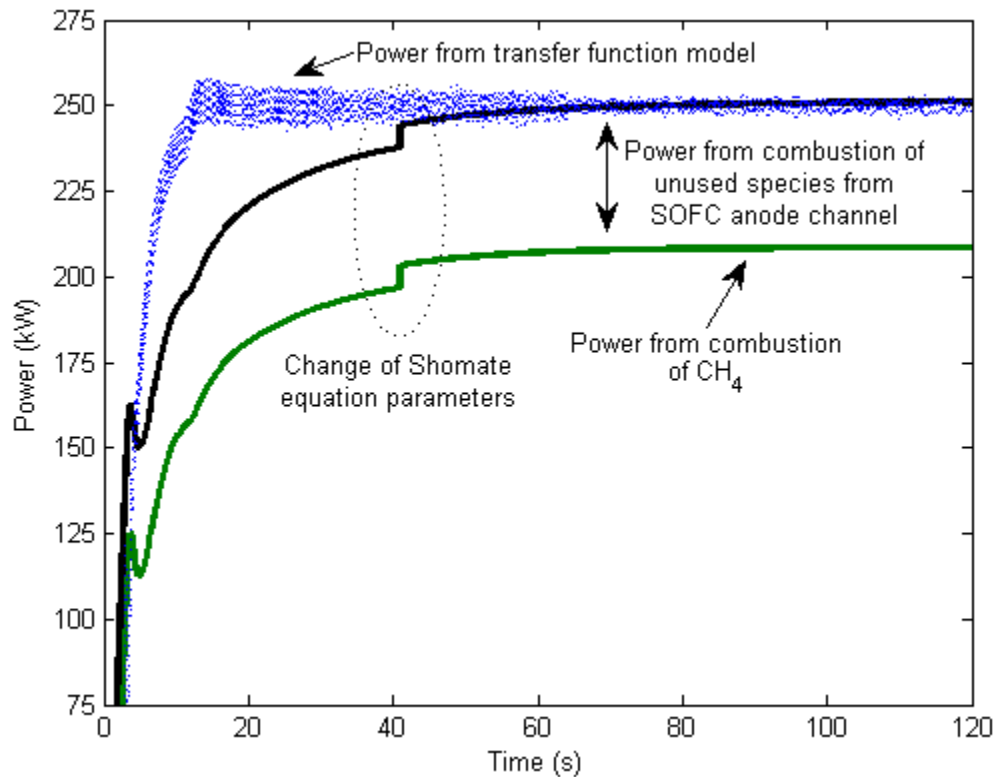


Fig. 3.12 MT power in CC mode based on the thermodynamic and transfer function models.

In order to further demonstrate the interconnection between the thermodynamic model and transfer function model, the amount of total power from the thermodynamic model in CC operation and the amount of power generated by the induction generator connected to the MT transfer function model are shown in Fig. 3.12.

The total power of the combustion of CH_4 and the unused species from the SOFC anode channel have a slower rise time compared to the power measured by the transfer function model. Consequently, the transfer function model reaches steady state quicker. This can be due to measurements based on mechanical properties, such as shaft speed and torque. The high rotational speed of the MT allows for a quick response to the voltage controller. The thermodynamic model, however, relies on the slow temperature response of the SOFC and therefore a slow convergence is observed. The step response near 41 seconds represents a change of Shomate equation parameters.

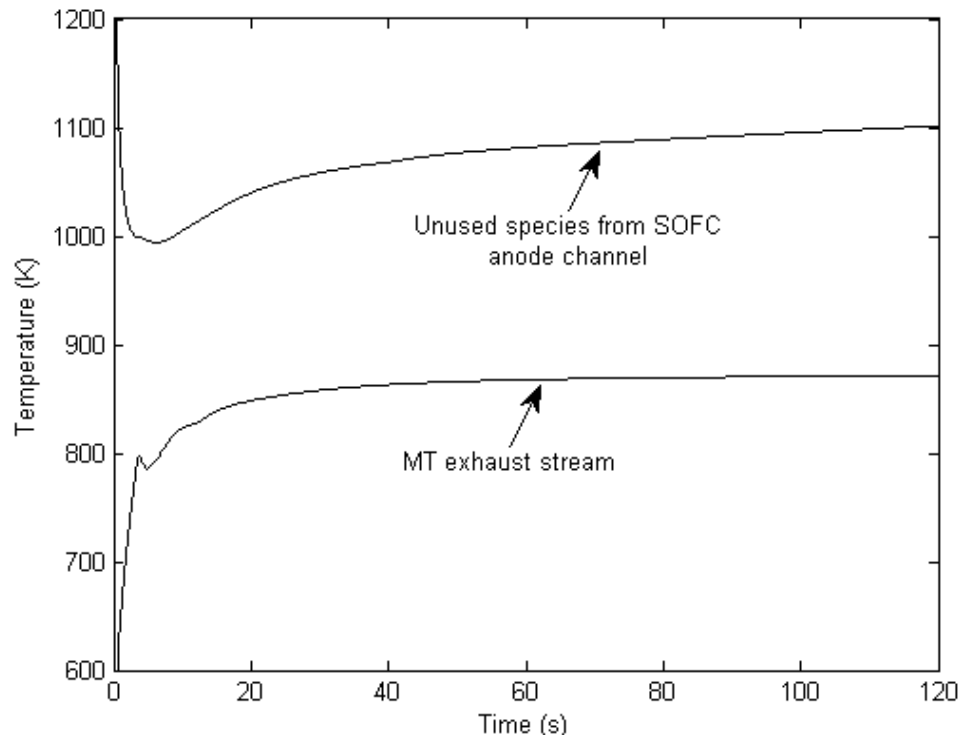


Fig. 3.13 Temperature of MT exhaust stream at outlet of heat exchanger.

Likewise, the thermodynamic modeling of the heat exchanger is affected by the slow temperature response of the SOFC, as shown by the countercurrent streams in the heat exchanger in Fig. 3.13.

As shown in Fig. 3.13, the temperature of the MT exhaust stream at the outlet of the heat exchanger is rapidly increased during startup of the system. This increase is due to the startup characteristics of the MT and the heat transferred from the countercurrent stream of the unused species from the SOFC anode channel. A rapid decrease of temperature in the stream of unused species during startup is due to the loss of energy due to transference to the MT exhaust stream. When the temperature of the MT exhaust at the inlet of the heat exchanger approaches a near constant value, after 60 seconds, the outlet of the stream temperature exhibits a slower increase of temperature through heat transference. The increase of temperature of the unused species from the SOFC anode channel stems from the nearly linear increase in temperature, previously shown in Fig. 3.4, over the medium time scale.

In addition to decreased fuel usage, the unused exhaust from the SOFC cathode channel can be utilized for heating purposes to further increase the system efficiency. This operation is called combined heat and power and is addressed in the next section.

Combined Heat & Power

CHP Operation

Combined heat and power (CHP), or cogeneration, operation refers to the repurposing of unused heat streams, typically lost to the environment, to aid in the

improvement of operational efficiency for simultaneous electricity and useful heat production [28]. The additional utilization of unused exhaust heat further decreases the emissions of the system, namely a decrease in CO₂ produced per kilowatt-hour of electrical energy generated [13].

While the benefits of employing CHP for residential water heating for a SOFC power plant model has been previously demonstrated by [13], the research presented within this thesis presents the advantages of CHP for a hybrid SOFC/MT-CC system.

CHP Model

For the system presented within this research, the CHP operation utilizes the exhaust gas from the SOFC to aid in residential or commercial heating, in particular, residential hot water production, while generating electricity, shown in Fig. 3.14.

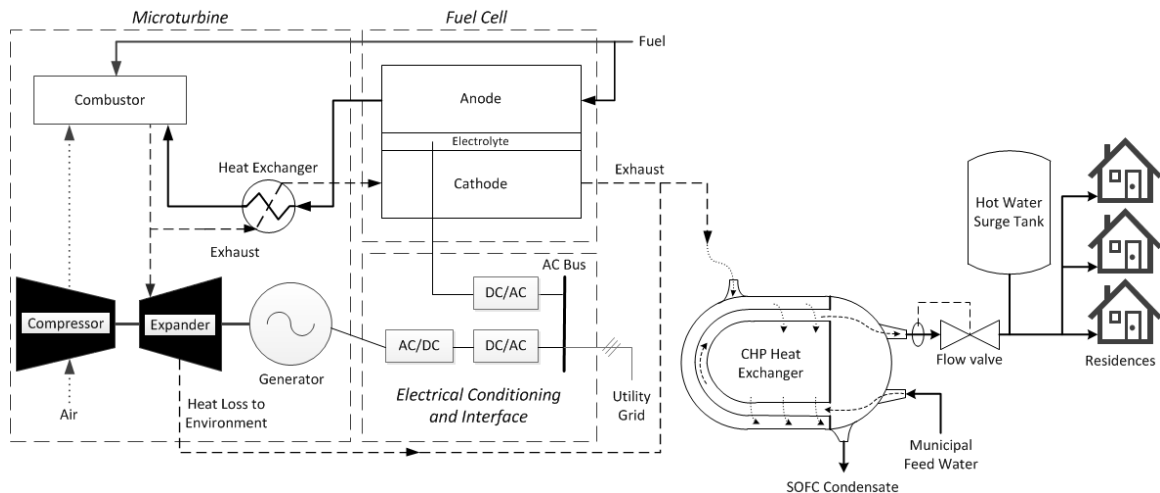


Fig. 3.14 Configuration of SOFC-MT-CC system in CHP operation for residential hot water production.

The unused exhaust of the SOFC stacks is collected in a single exhaust header. The exhaust then passes through a conventional cross-flow, shell-and-tube heat

exchanger. As the exhaust stream passes through the shell, heat is transferred to the cooler inner tubing containing the residential hot water, causing an increase of temperature in the water. The municipal water supply provides feed water to the cold inlet of the heat exchanger, assumed to be a constant 13°C. The feed water circulates around the heat exchanger tubing by the pressure maintained by the water utility, which is between 40 and 80 psi. Exploiting the existing pressure of the feed water eliminates the need for additional pumps. Once the SOFC exhaust has passed through the heat exchanger and transferred heat to the residential hot water, it is released to the environment at 200°C [13].

Although the SOFC/MT-CC system in CHP operation system is operated with a utility grid interconnection, the research presented in this thesis focused on the ability of the system to meet the power and hot water demands of a 250-home residential neighborhood in the Pacific Northwest region of the United States.

Additional details regarding the CHP model can be found in [13].

Load Data

When appropriately sizing a system for a specific load, aggregate data directly obtained from precise, real-time monitoring of the individual components that comprise the system would ideally be used. However, this approach is not widely used for residential load profiles. Consequently, the majority of residential load data currently employed in research is found through an interpolation of a typical residence configuration conjectured to suitably represent regional data, as opposed to direct monitoring [45]-[47]. Recent research on optimal modeling for the residential sector has

aided the development of better techniques for obtaining residential end-use data [48]. Unfortunately, because much of the modeling methodology cannot yet be employed, an alternative method to obtain relevant and accurate end-use profiles, developed by [49], was utilized in this study. The data attained has been derived from the End-Use Load and Consumer Assessment Program (ELCAP) studies completed in the 1990s, undertaken by the Pacific Northwest Laboratory (PNNL) for Bonneville Power Administration (BPA) and has been updated based on the most recent data available, specifically, 2010 EIA base cases [50]. For this reason, it can be considered to be representative of the end use curves for Pacific Northwest homes in the current era.

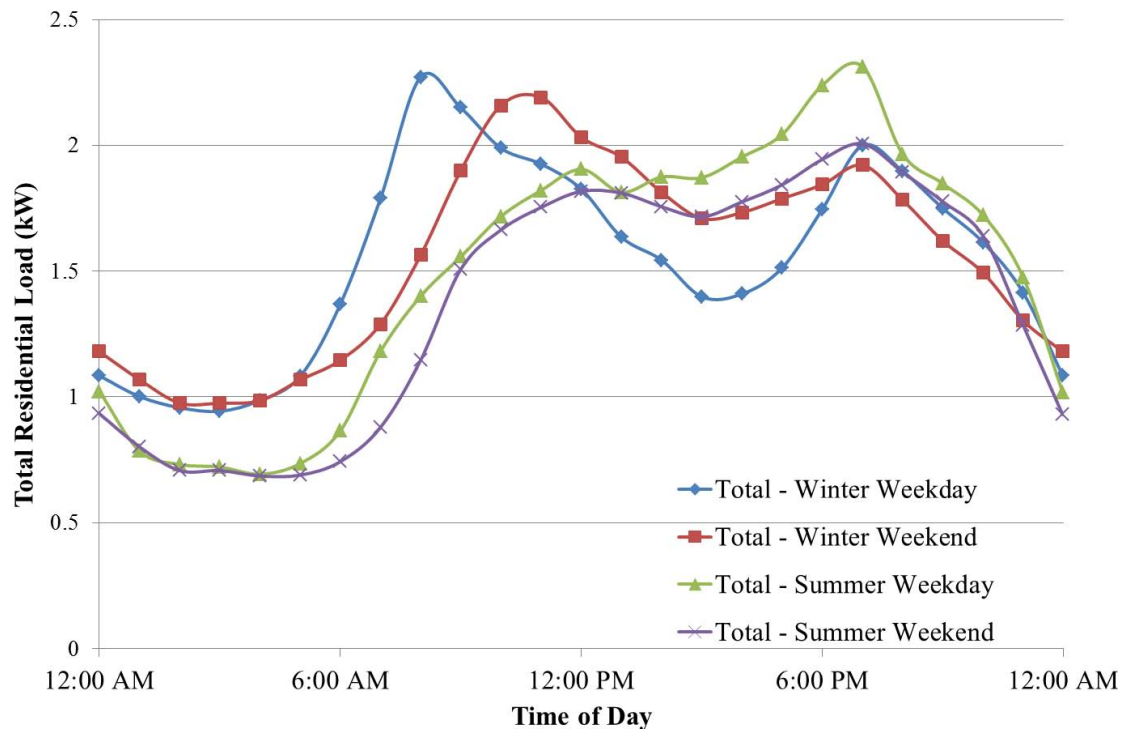


Fig. 3.15 Aggregated total residential electrical use seasonal profiles [49].

The aggregate end-use, time-of-day load curves based on seasonal usage patterns in the Pacific Northwest region of the United States are shown in Fig. 3.15. The majority

of homes in the region observed utilized electric appliances, heating and ventilation, and water heaters [49]. Fig. 3.15 shows the total aggregated end-use profiles for all-electric residences, including an electric water heater (EWH). By assuming all-electric homes for this study, the sizing of the generation system will be sufficient for a greatest electrical load scenario.

While Fig. 3.15 provides the total electrical load profiles, it lacks EWH data specific to water consumption. Therefore, the real-world residential hot water consumption data representing the time-of-use average compiled by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) is utilized and shown in Fig. 3.16 [51].

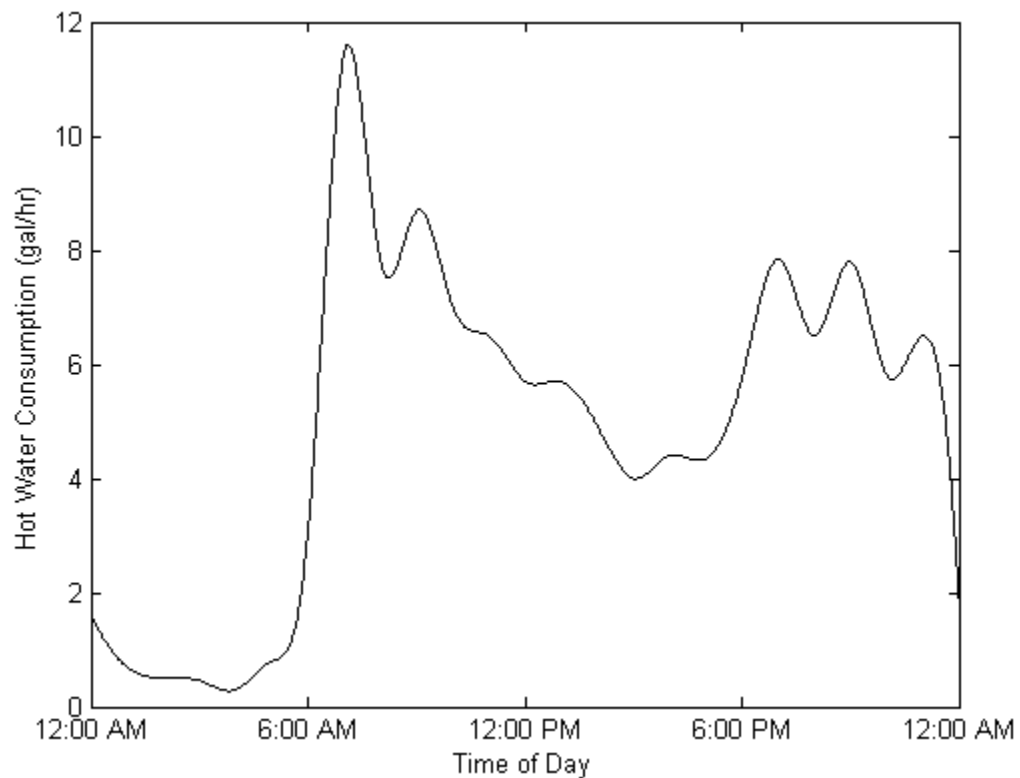


Fig. 3.16 ASHRAE residential hot water average consumption profile [51].

CHP Simulation

The SOFC-MT-CC system in CHP operation model was implemented within the Matlab/Simulink environment. The power profiles in Fig. 3.15 were implemented with the residential power factor set at 0.95 lagging [13]. The winter weekday and summer weekend profiles were selected for further analysis due to their distinctive differences of peak demand throughout the day. The simulations do not take into account previous or following day conditions. The surge tank volume is set to 8500 gal, or the approximate volumetric capacity of one tank truck, and the level is set at an initial value of 10 percent full for all simulations [13].

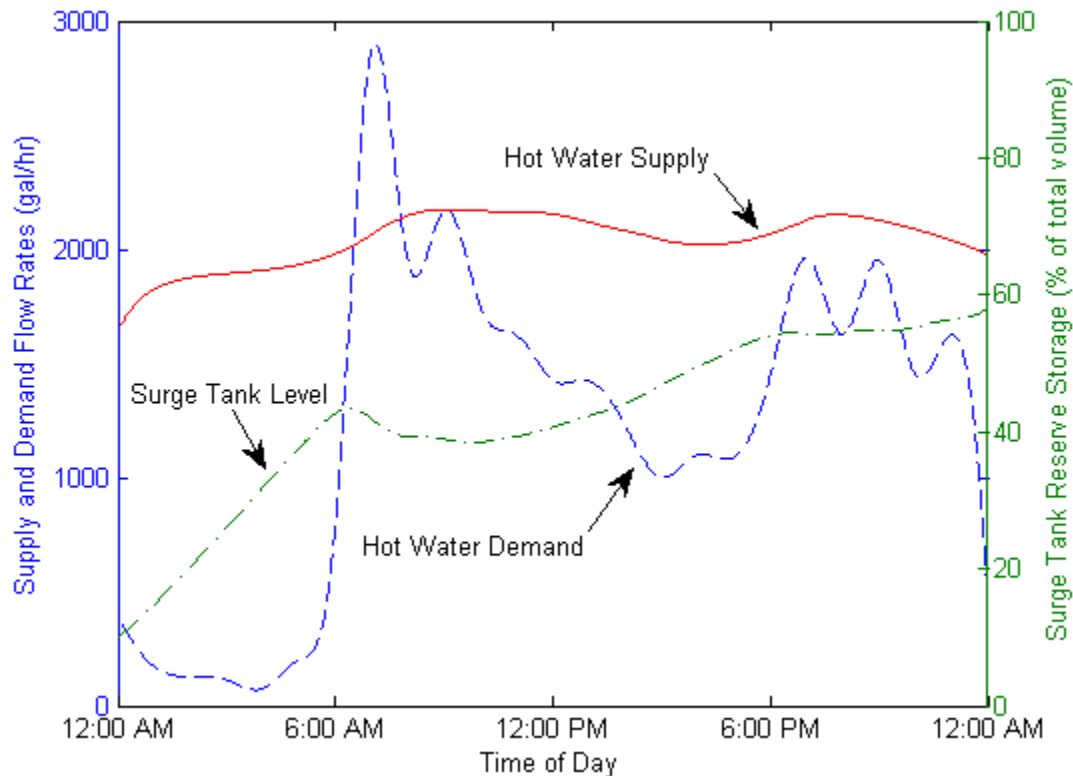


Fig. 3.17 Residential hot water supply and demand for SOFC-MT-CC-CHP system in Pacific Northwest 250-home winter weekday demand scenario.

As Fig. 3.17 demonstrates, the surge tank level increases from the initial value of 10 percent to approximately 40 percent between 12:00 and 6:00 AM due to the low demand of hot water while the amount of hot water supplied remains under 2000 gal/hr. When the demand spikes after 6:00 AM to almost 3000 gal/hr, the surge tank level decreases while the amount of available hot water, or hot water supply, increases. Throughout the remainder of the 24 hour period the surge tank level increases to a final value of 58 percent. At all times throughout the simulation the residential hot water demands were met by the CHP plant. The overall increase in surge tank storage levels indicate the CHP system contains more available energy from exhaust heat than is being utilized by the residential hot water demand.

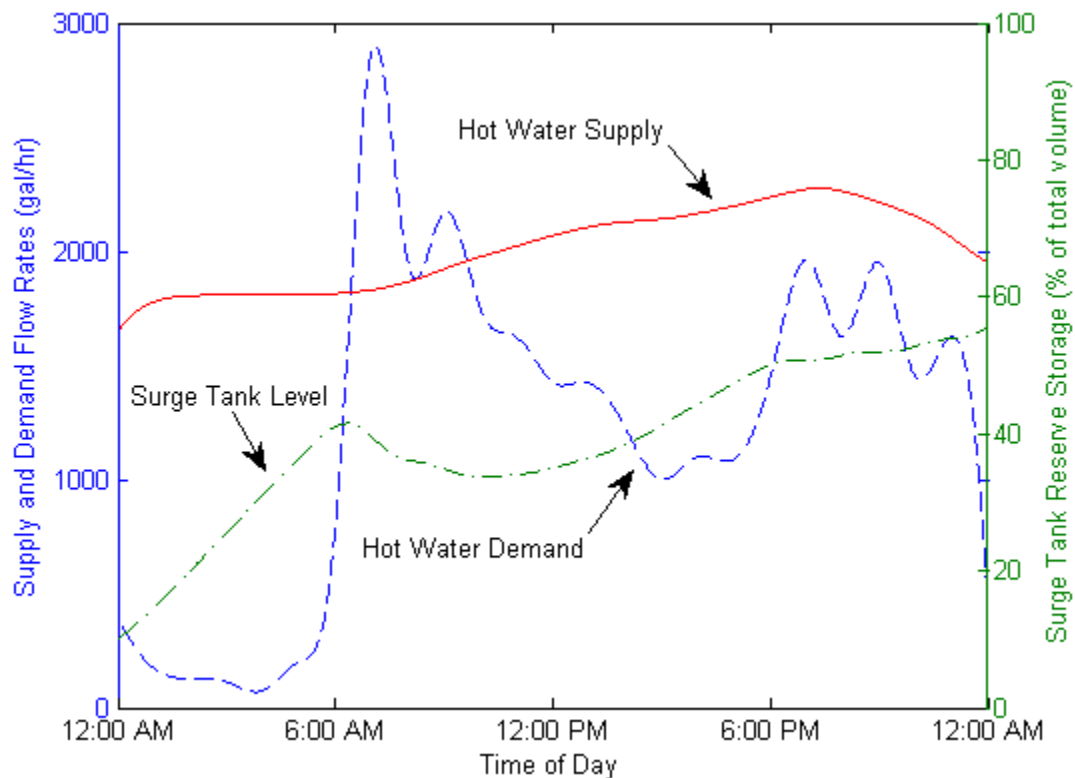


Fig. 3.18 Residential hot water supply and demand for SOFC-MT-CC-CHP system in Pacific Northwest 250-home summer weekend demand scenario.

Fig. 3.18 demonstrates the summer weekend demand scenario, with a spike of demand around 6:00 PM. Similar to the winter weekday scenario, the surge tank level increases from the initial value of 10 percent to approximately 45 percent between 12:00 and 6:00 AM due to the low demand of hot water while the amount of hot water supplied remains under 2000 gal/hr. The surge tank level decreases while the hot water supplied increases when the demand spikes after 6:00 AM. Throughout the remainder of the 24 hour period the surge tank level increases to a final value of 56 percent. The difference of tank levels for both scenarios stems from the increased demand later in the day for the summer weekend scenario. At all times throughout the simulation the residential hot water demands were again met by the CHP plant. The overall increase in surge tank storage level for both scenarios indicate the CHP system could be utilized for other applications than the residential hot water demand for further cost savings [13].

Discussion

The CC chosen for this research is a variant of a Brayton cycle interconnection. However, there are many other integration methods currently available when employing a fuel cell and gas turbine [52]. Depending on the purpose and usage of the systems, the interconnections can be reconfigured to produce greater efficiencies than those presented in this research. The CC configuration within this research was deemed a common coupling technique and thereby allows for future research to further improve the efficiency of the system.

POWER MANAGEMENT OF SOFC-MT-CC SYSTEM

The power management of the hybrid SOFC-MT-CC system is presented in this chapter. A brief description of the power electronic interfaces, which connect to a common bus, is also discussed.

The operation of the SOFC-MT-CC system is desired to be stable during transient occurrences. Beside the advantage of increased efficiency, the coupling of a fuel cell with a gas turbine also benefits in terms of transient stability and power management. The electrochemical reactions that occur in a SOFC are slow in comparison to the mechanical response of the MT. For this reason, during a transient period the fast response of the MT is utilized for voltage and frequency stabilization and to prevent damage occurring to the SOFC. For instance, rapid load fluctuations on a SOFC can result in cell degradation of the SOFC, which can lead to thermal fatigue. In order to prevent damage to the SOFC, the MT is given the responsibility of responding to transient events to maintain the voltage and frequency of the system.

System Overview

The hybrid SOFC-MT-CC system presented in Chapter 3 (see Fig. 3.1) consists of models for the SOFC, MT, heat exchanger, combustor, and power electronics interfacing. The power electronics interfacing is connected to an AC bus which supplies an AC load. In the following section, a brief description of the power electronics interfacing is given [53].

Power Electronics Interfacing

Power electronics (PE) interfacing are necessary for hybrid systems to maintain the output voltage and frequency of each generating component of the system. The PE interfacing used for the hybrid system consists of a DC/DC boost converter and DC/AC inverter for the SOFC and an AC/DC rectifier and DC/AC inverter for the MT. The PE interfacing is used to control the real and reactive power by controlling the inverter output voltage angle, and frequency. The SOFC and MT each contain their own dq transformed PE interface. Further details of the PE interfaces and their subsequent implementation can be found in [4], [6], [53].

Simulation Results

The simulation of a stand-alone MT, a stand-alone SOFC, and the SOFC-MT-CC system were conducted. In each case, a variable load was attached to record the load response capabilities of each generation source.

MT Response to Variable Load

The load response of the MT was first analyzed in order to determine the amount of time required to return to a steady state. A variable load was attached to the MT, as shown in Fig. 4.1. The variable load begins at 250 kW from $t = 0$ to 20 s, then the load decreases to 150 kW from $t = 20$ to 30 s; it then increases to 200 kW from $t = 30$ to 40 s.

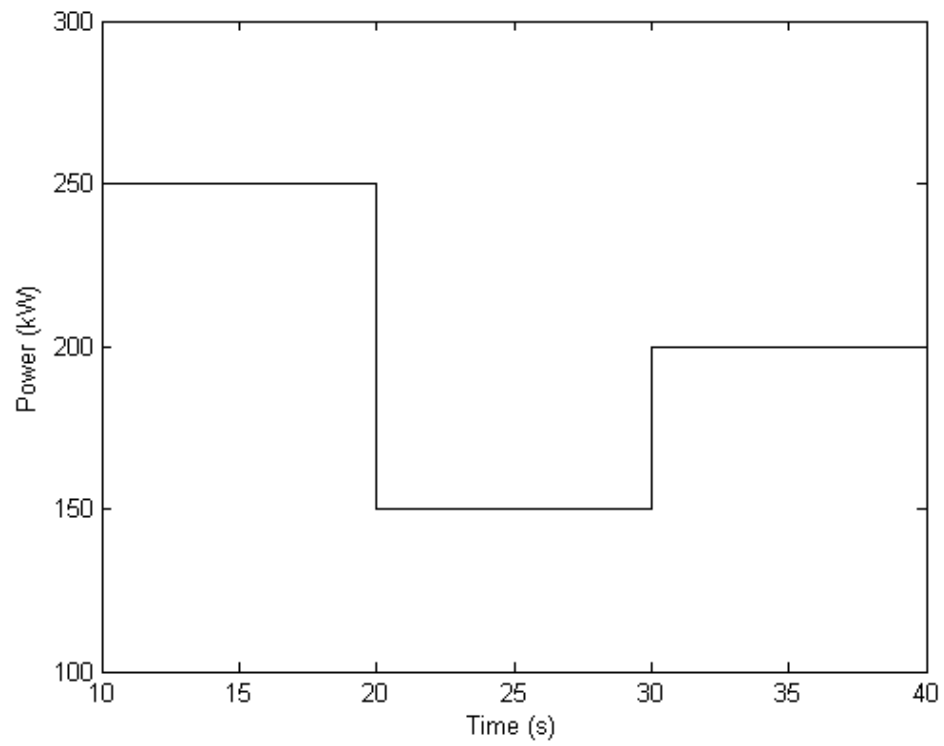


Fig. 4.1 Variable load attached to MT.

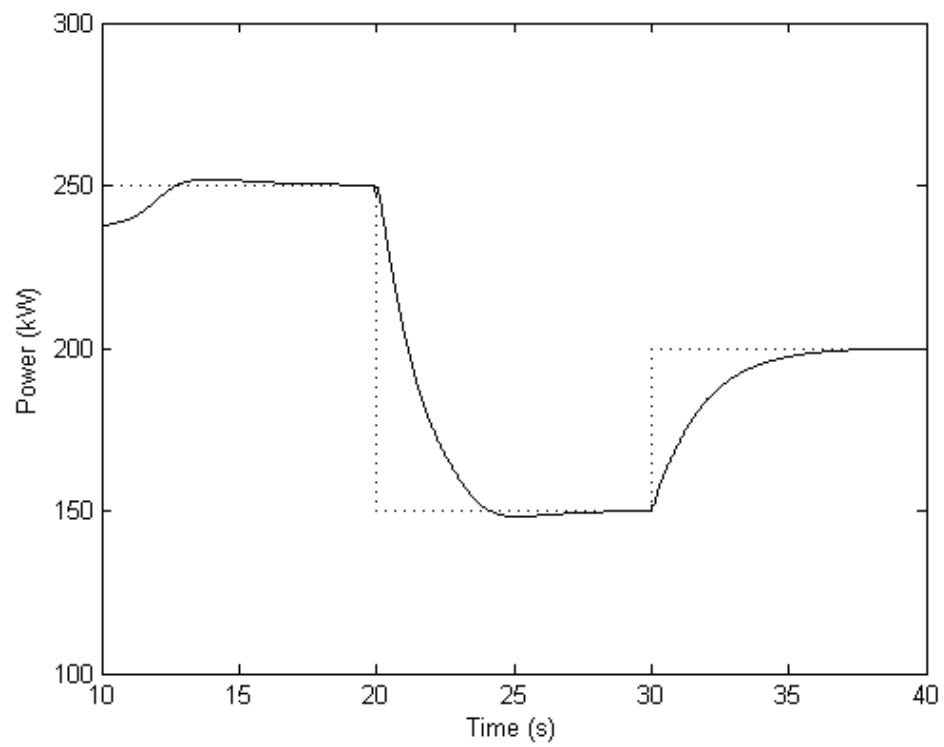


Fig. 4.2 MT active power response to variable load.

As Fig. 4.2 demonstrates, the MT reaches rated power of 250 kW within about 17 s with an attached load of 250 kW. When the load decreases from 250 kW to 150 kW, the electrical power output achieves steady state within about 8 s of simulation time. At $t = 30$ s, the load increases from 150 kW to 200 kW. The MT power output increases from 150 kW to 200 kW and again achieves a steady state within about 8 s of simulation time.

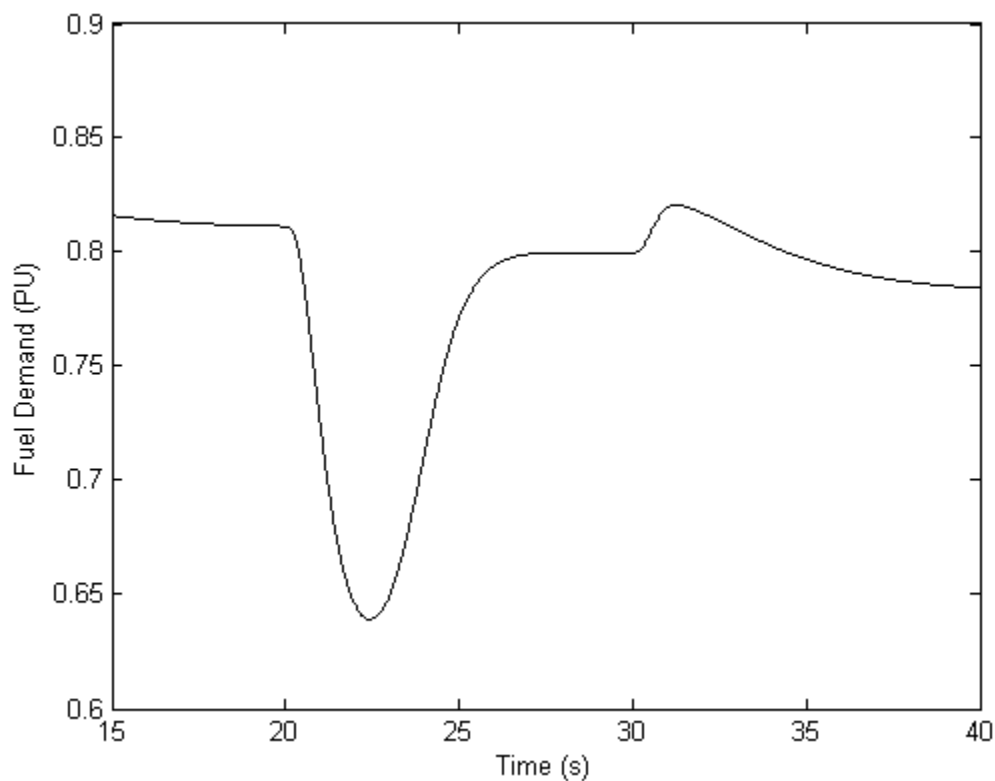


Fig. 4.3 Fuel demand signal of the MT.

Fig. 4.3 shows the fuel demand signal of the transfer function model of the MT in pu. At rated power (10 s to 20 s) the fuel demand signal is approximately 0.82 pu. The fuel demand signal decreases to about 0.63 pu when the load decreases from 250 kW to 150 kW. However, the fuel demand signal increases to about 0.8 pu when the MT has achieved a steady output power of 150 kW. While the amount of power produced is less,

the MTs temperature is still increasing; hence the fuel demand signal increases to reach the specified operating temperature of 750 K, as specified by the temperature control of the MT. When the load is increased to 200 kW at 30 s the fuel demand signal increases to provide an increase of power. As the MT achieves a steady state operation by 40 s at 200 kW the fuel demand signal decreases to around 0.78 pu during the same time period. The fuel demand signal is decreased when operating at 200 kW compared to 150 kW due to the increased operational temperature of the MT from 30 s to 40 s.

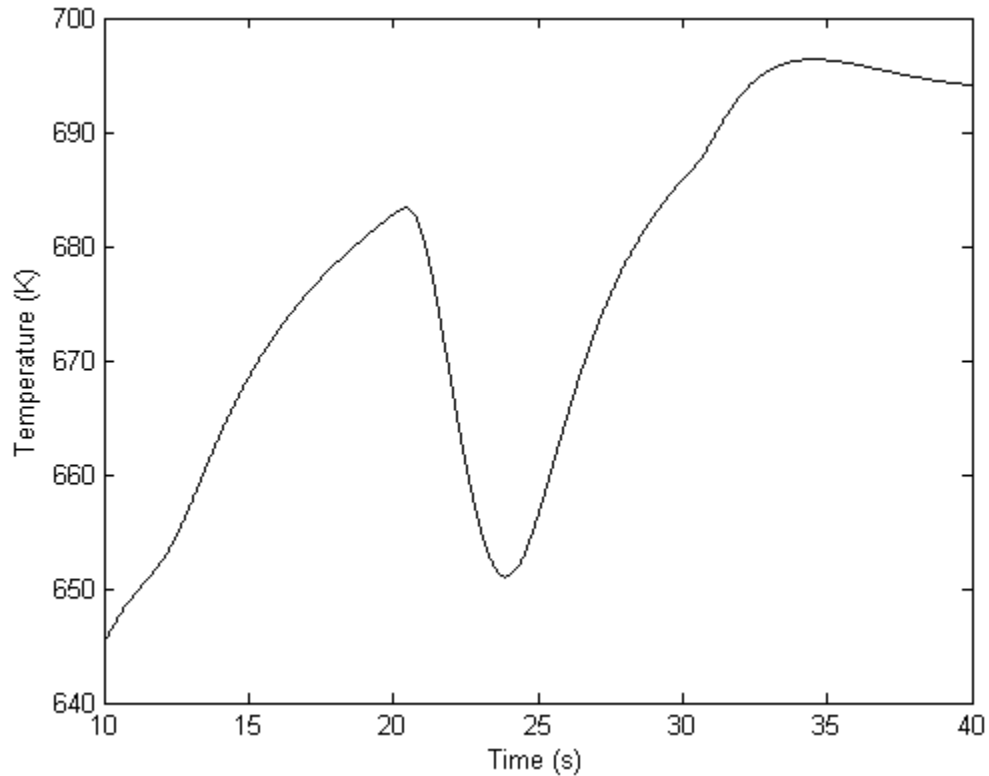


Fig. 4.4 Exhaust temperature of MT.

As Fig. 4.4 shows, the exhaust temperature of the MT increases from approximately 645 K to 683 K from 10 s to 20 s while the MT is operating at 250 kW. The temperature sharply decreases from 683 K to 650 K when the load decreases from

250 kW to 150 kW from 20 s to 24 s. The temperature then continues to increase in order to achieve the reference operating temperature of 750 K from 24 s to 30 s. The rate of change of the temperature increases slightly to reach a temperature of approximately 697 K when the load increases from 150 kW to 200 kW at 30s. The temperature then decreases slightly to reach around 694 K by the time the MT power has achieved steady state at 40 s.

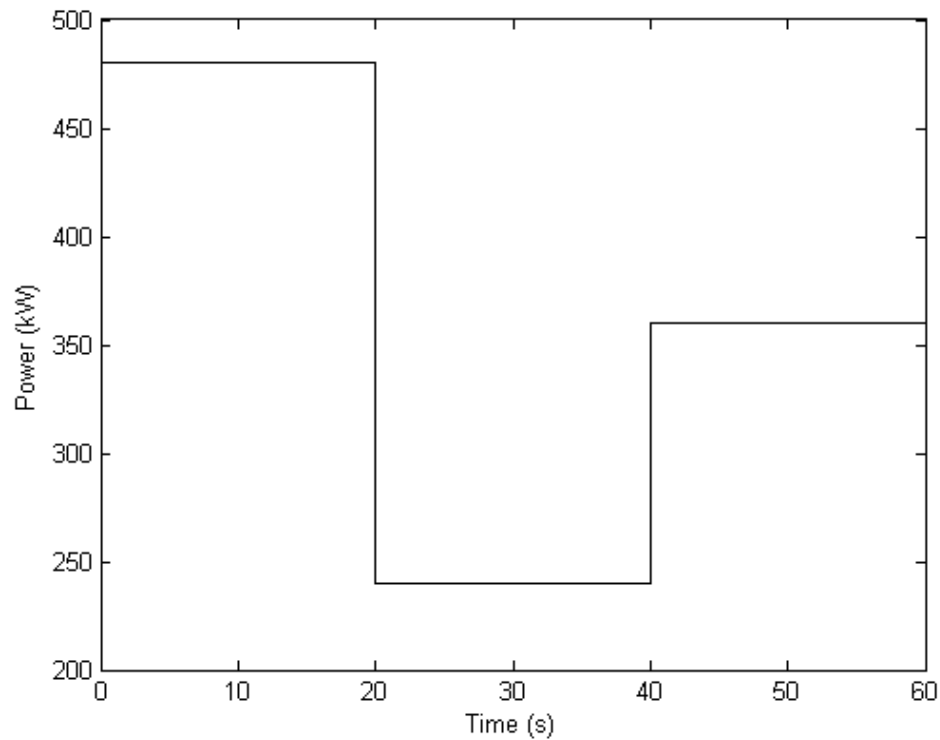


Fig. 4.5 Variable load attached to SOFC.

SOFC Response to Variable Load

The load response of the SOFC was analyzed in order to determine the amount of time required to return to a steady state. A variable load was attached to the SOFC, as shown in Fig. 4.5. The variable load begins at 480 kW from $t = 0$ to 20 s, then the load

decreases to 240 kW from $t = 20$ to 40 s, and then the load increases to 360 kW from $t = 40$ to 60 s.

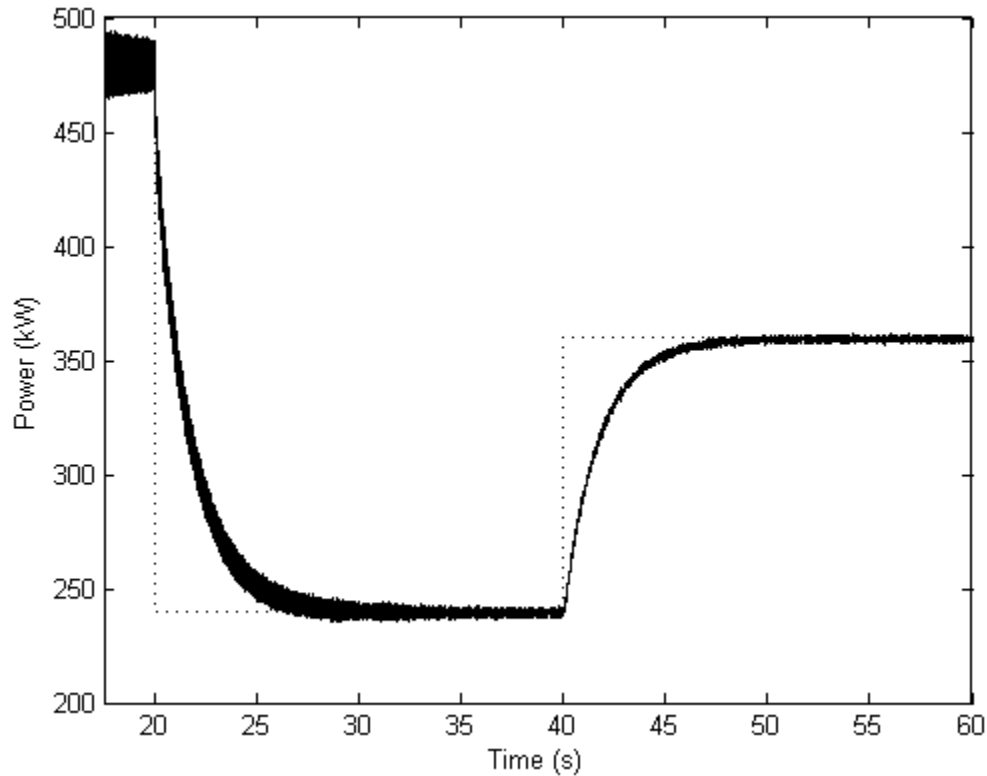


Fig. 4.6 SOFC active power response to variable load.

As Fig. 4.6 demonstrates, the SOFC reaches rated power of 480 kW at $t = 20$ s, the load decreases to 240 kW, and the SOFC follows the demanded power. At $t = 40$ s, the load increases from 240 kW to 360 kW. The SOFC active power output increases from 240 kW to 360 kW and again achieves a steady state within 15 s of simulation time.

SOFC-MT-CC Response to Variable Load

The response of the hybrid SOFC-MT-CC system to a variable load was analyzed to quantify the inability of the SOFC to adequately handle an abrupt change in load (Case

1) and the ability of the MT to quickly and effectively stabilize the system during transient occurrences (Case 2). The system is considered in grid-connected mode during a peak demand time span. During peak demand the grid is more vulnerable or susceptible to exceeding the acceptable limits of frequency, $60 \text{ Hz} \pm 0.05 \text{ Hz}$, given a sudden change in load. The grid is represented by a diesel generator in Matlab/Simulink in order to appropriately model voltage and frequency instabilities in such a scenario, as shown in Fig. 4.7. Alternatively, the system can be considered to be in stand-alone operation within a microgrid. Due to the rapid load response capability of the diesel generator, its output power was limited in order to demonstrate the MT load response without additional stabilization (Case 3).

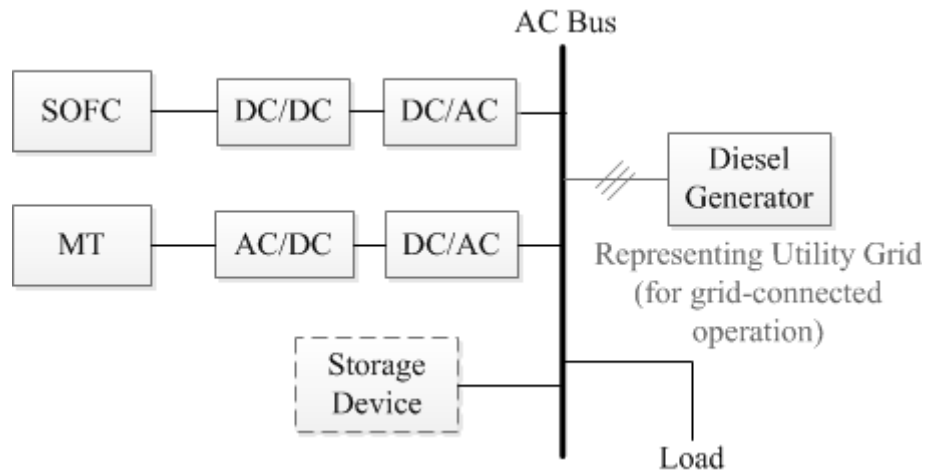


Fig. 4.7 System configuration with optional storage device and diesel generator representing the utility grid.

Fig. 4.7 shows the SOFC-MT system configuration used for simulation with an optional storage device and a diesel generator representing the utility grid. The storage

device was omitted for this research in order to properly observe the load responses of the SOFC and MT.

The load is increased in each case and one generation source is maintained at a constant output power, while the power of the other generation source is increased in response to the increased load. The active and reactive load profile is shown in Fig. 4.8.

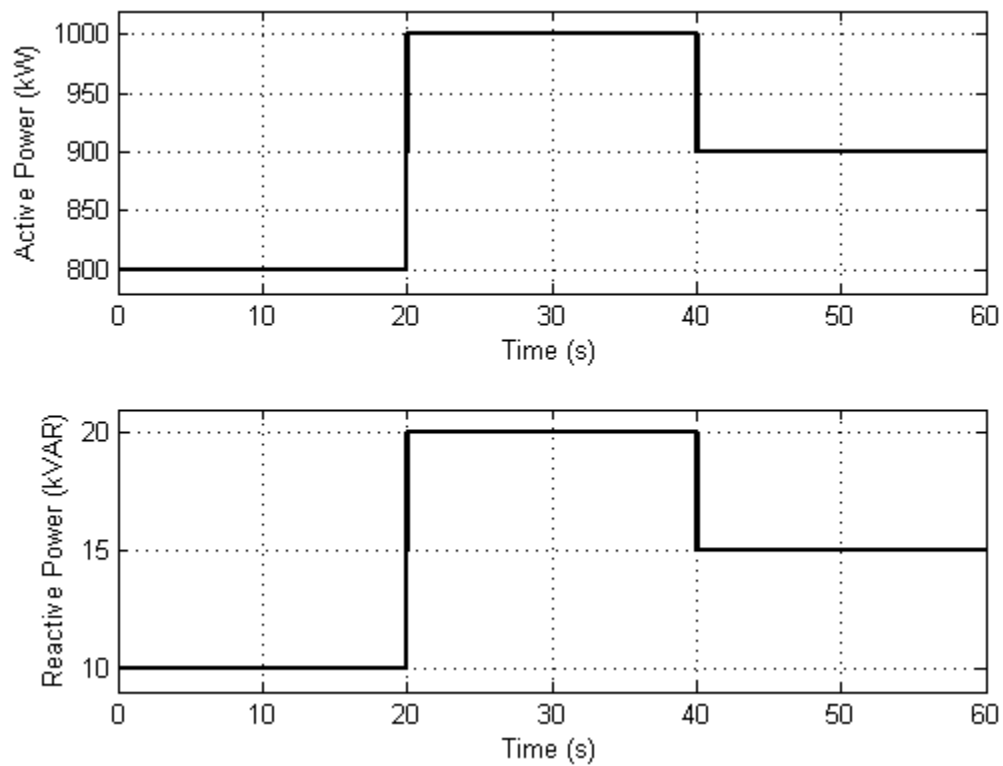


Fig. 4.8 Load profile for SOFC-MT-CC simulations in Case 1, 2 and 3.

As Fig. 4.8 shows, the load is 800 kW and 10 kVAR from 0 to 20 s. The load increases to 1.0 MW and 20 kVAR at 20 s and decreases to 900 kW and 15 kVAR at 40 s. For all cases the diesel generator is rated at 2.5 MVA.

Case 1: SOFC Response with Constant MT. The SOFC was set to an initial operating power of 380 kW. When the load is increased by 200 kW, the SOFC responds by increasing its generation up to rated power of 480 kW (Fig. 4.9). The MT power is held constant at rated power of 250 kW during the simulation (Fig. 4.10). The diesel generator increases its generating power slightly, representing the grid response of an increase in demand (Fig. 4.11).

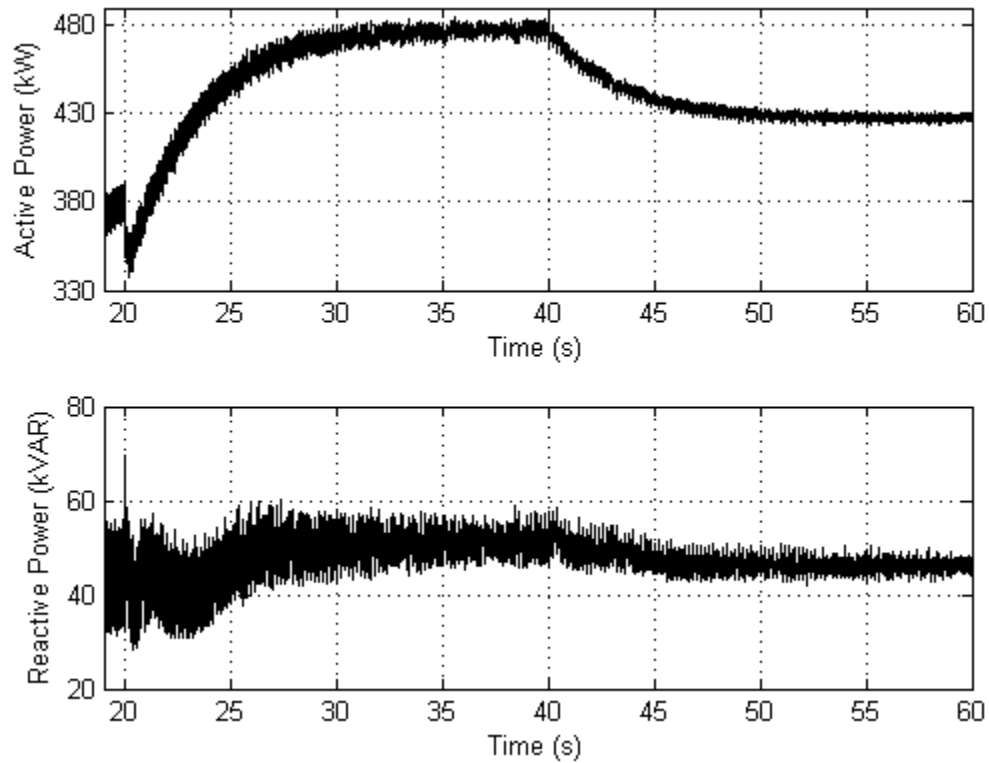


Fig. 4.9 Active (top) and reactive (bottom) power supplied by SOFC to meet load demand for Case 1.

Fig. 4.9 shows the active and reactive power supplied by the SOFC to meet the load demand. The SOFC operates at 380 kW and 45 kVAR prior to 20 s. The active power of the SOFC sharply drops when the load is increased at 20 s and then increases to

its rated power of 480 kW by 40 s. The sharp drop is due to the relative slow chemical reactions inside the SOFC. As a result, sufficient fuel will not reach the reaction site inside the SOFC; this phenomenon is referred to as fuel starvation. The reactive power decreases slightly when the load is increased at 20 s and increases slowly to around 50 kVAR. After the SOFC output active and reactive powers reach steady state, the load decreases at 40 s. The SOFC active power decreases from 480 kW to 430 kW by 55 s in response of the decreased load.

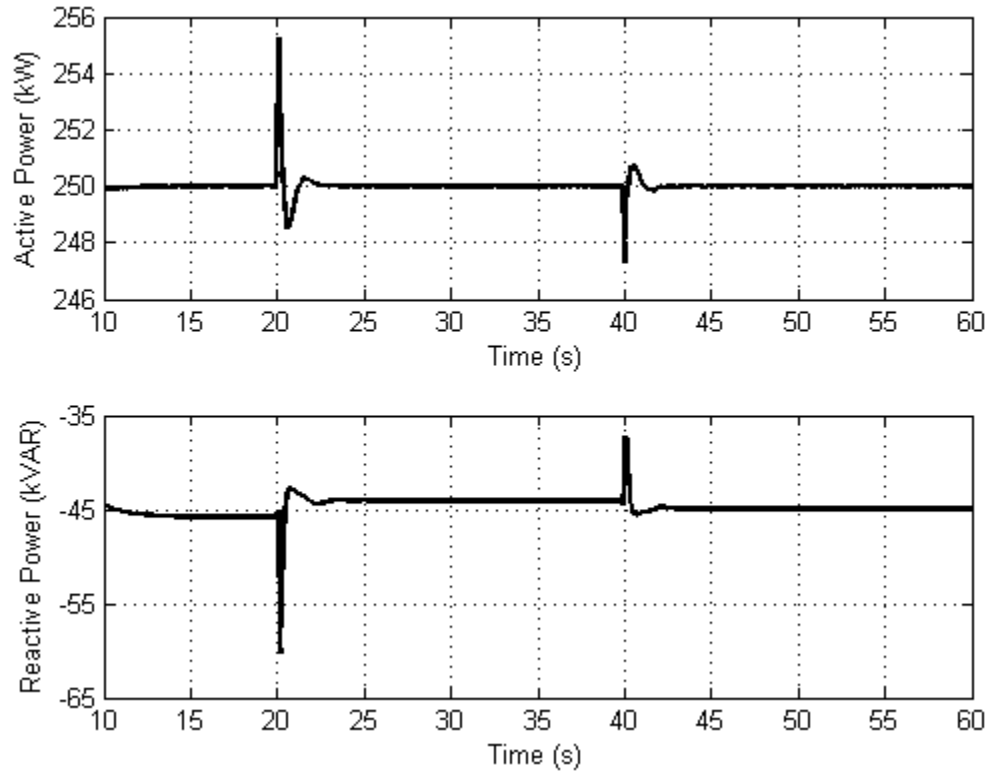


Fig. 4.10 Active (top) and reactive (bottom) rated power supplied by MT for Case 1.

Fig. 4.10 shows the active and reactive power supplied by the MT for Case 1. For this case, the MT operates at rated power of 250 kW. However, the MT is still affected by the changes in load at 20 s and 40 s due to the interconnection of the system.

Specifically, when the load is increased at 20 s the MT active power sharply spikes up to 255 kW in an attempt to increase generation to handle the increased load. Since the MT is governed to operate at rated power of 250 kW, the power decreases and arrives at 250 kW by 25 s. Likewise, the amount of reactive power absorbed by the induction generator, within the MT model, increases during the same time period to produce more active power. When the load is decreased at 40 s the active power decreases sharply in an attempt to match the decreased load. The active power of the MT returns to rated power by 45 s.

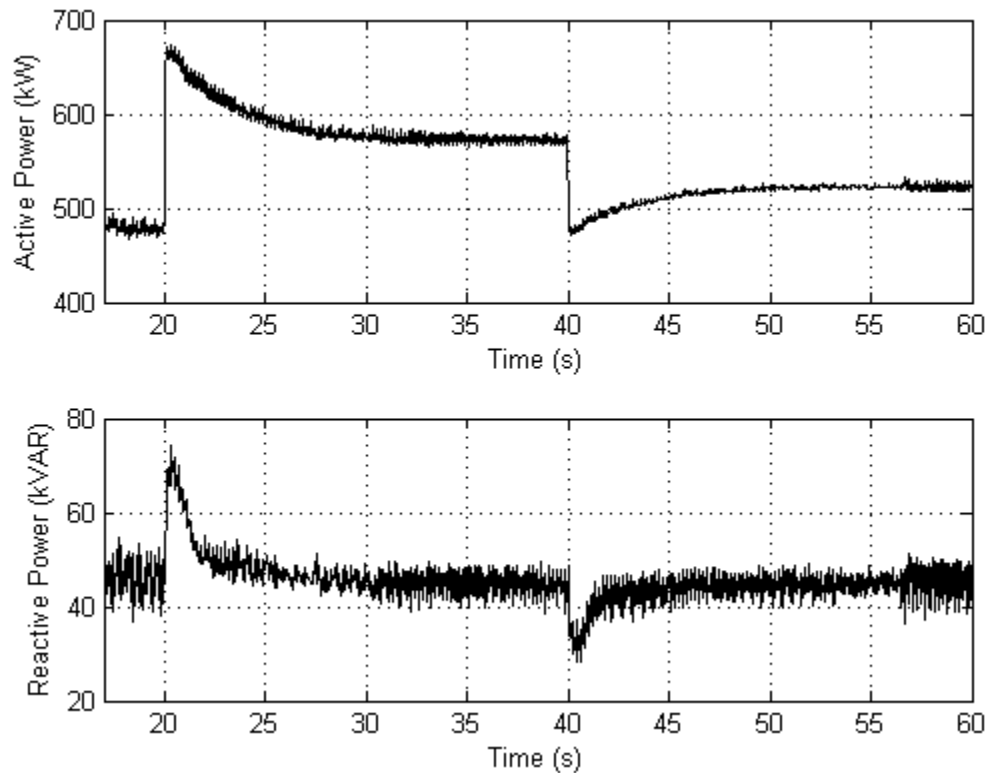


Fig. 4.11 Active (top) and reactive (bottom) power supplied by diesel generator for Case 1.

As Fig. 4.11 demonstrates the active and reactive power of the diesel generator supplied to the hybrid system for Case 1. The diesel generator supplies approximately 490 kW and 45 kVAR prior to 20 s. When the load is increased at 20 s, the active and reactive power sharply increases to about 680 kW and 75 kVAR, respectively. The active power decreases to around 580 kW while the reactive power decreases to around 45 kVAR by 40 s. When the load decreases at 40 s the active and reactive power decrease sharply and slowly arrive at steady state values of 530 kW and around 45 kVAR by 55 s.

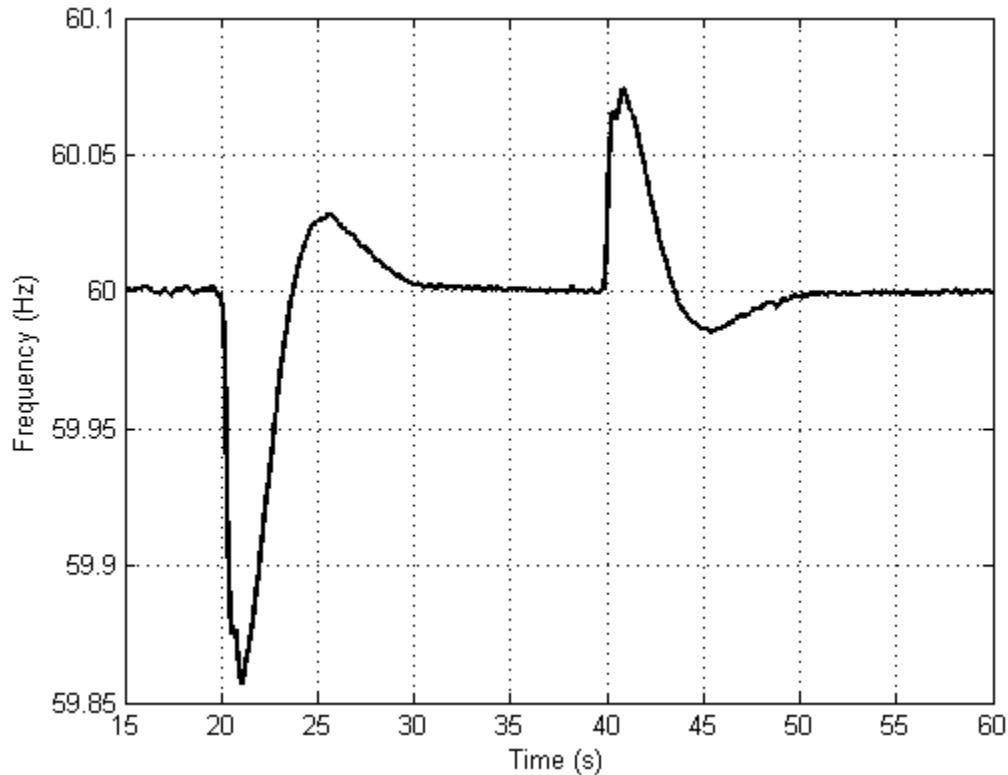


Fig. 4.12 Frequency of SOFC-MT-CC system during simulation for Case 1.

Fig. 4.12 shows the frequency of the system during the simulation for Case 1. The frequency is 60 Hz prior to 20 s when the system is at steady state. The frequency

sharply decreases to approximately 59.85 Hz when the load is increased at 20 s. The frequency then increases and reaches a steady state value of 60 Hz by 35 s. The frequency increases to around 60.07 Hz at 40 s when the load is decreased. The frequency returns to a stable 60 Hz by 55 s after the SOFC and diesel generator reach steady state. Even though the frequency exceeds the acceptable range ($60 \text{ Hz} \pm 0.05 \text{ Hz}$) of frequency for interconnection set by FERC, it is well within the performance measures currently employed that rely on 10 minute averaged data [54].

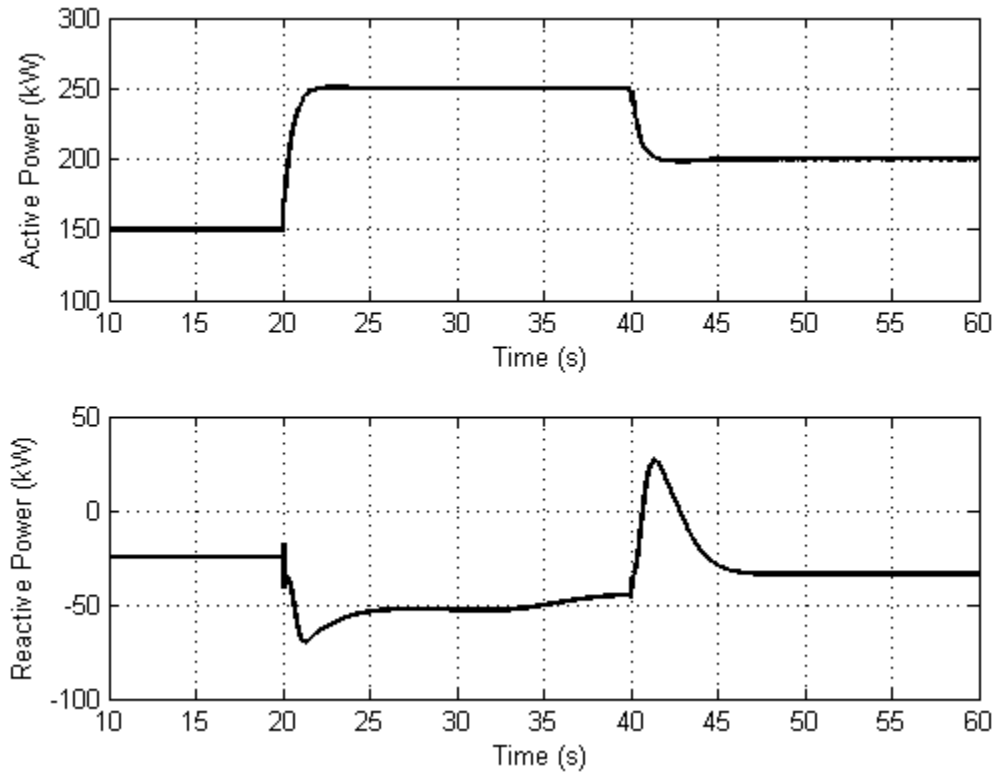


Fig. 4.13 Active (top) and reactive (bottom) power supplied by MT to meet load demand for Case 2.

Case 2: MT Response with Constant SOFC. The MT was set to an initial operating power of 150 kW. When the load is increased at 20 s, the MT responds by

increasing its generation up to rated power of 250 kW. The diesel generator responds by increasing supplied power slightly, as occurred in Case 1. The SOFC power is held constant at rated power of 480 kW during the simulation.

Fig. 4.13 shows the active and reactive power supplied by the MT to meet the load demand. The active power of the MT increases from 150 kW to 250 kW within 5 s after the load is increased at 20 s. The reactive power of the MT decreases during the same time period, or becomes more negative. Since the negative sign denotes absorption of power, or power flowing into the measurement reference, the reactive power of the induction generator can be equivalently described as an increase of reactive power absorbed. When the load current is increased in a system with a fixed reactive power supply, the terminal voltage of the induction generator will decrease. In order to compensate, or increase the voltage, additional reactive power is required by the induction generator [55]. In this system, the additional reactive power is supplied by the grid, represented by the diesel generator. Also, the reactive power losses are increased due to the increased current. In order for the induction generator to increase active power output, the reactive power must be increased into the device. After the MT output active and reactive power stabilizes at 250 kW and -50 kVAR, respectively, the load decreases at 40 s. The MT active power decreases from 250 kW to 200 kW from 40 to 45 s in response to the decrease of load.

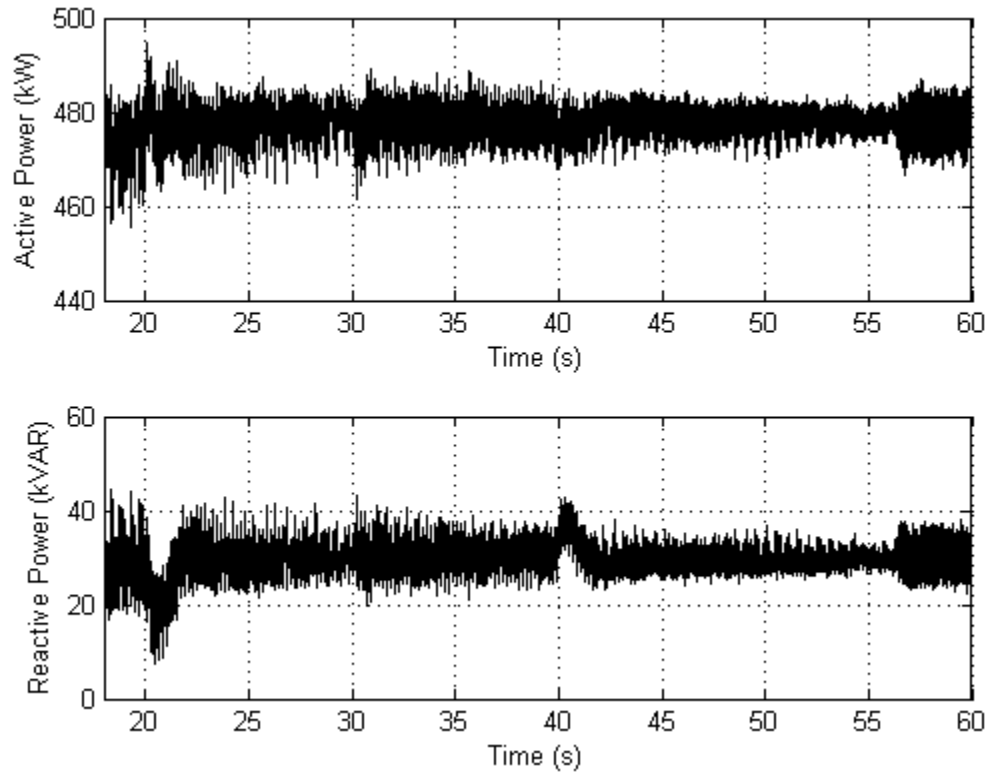


Fig. 4.14 Active (top) and reactive (bottom) rated power supplied by SOFC for Case 2.

Fig. 4.14 shows the active and reactive power supplied by the SOFC for Case 2. For this case, the SOFC operates at rated power of 480 kW. However, the SOFC is still affected by the change in load at 20 s and 40 s due to the interconnection of the system. The SOFC active power spikes up to approximately 490 kW when the load increases at 20 s. The reactive power drops from approximately 30 kVAR to 20 kVAR when the load is increased. Since the SOFC is operating at rated power of 480 kW the active and reactive power stabilize within 5 s of the disturbance. When the load is decreased at 40 s the active power barely exhibits any significant change. However, the reactive power spikes from around 30 kVAR to 40 kVAR and returns to a rated value of 30 kVAR within 5 s.

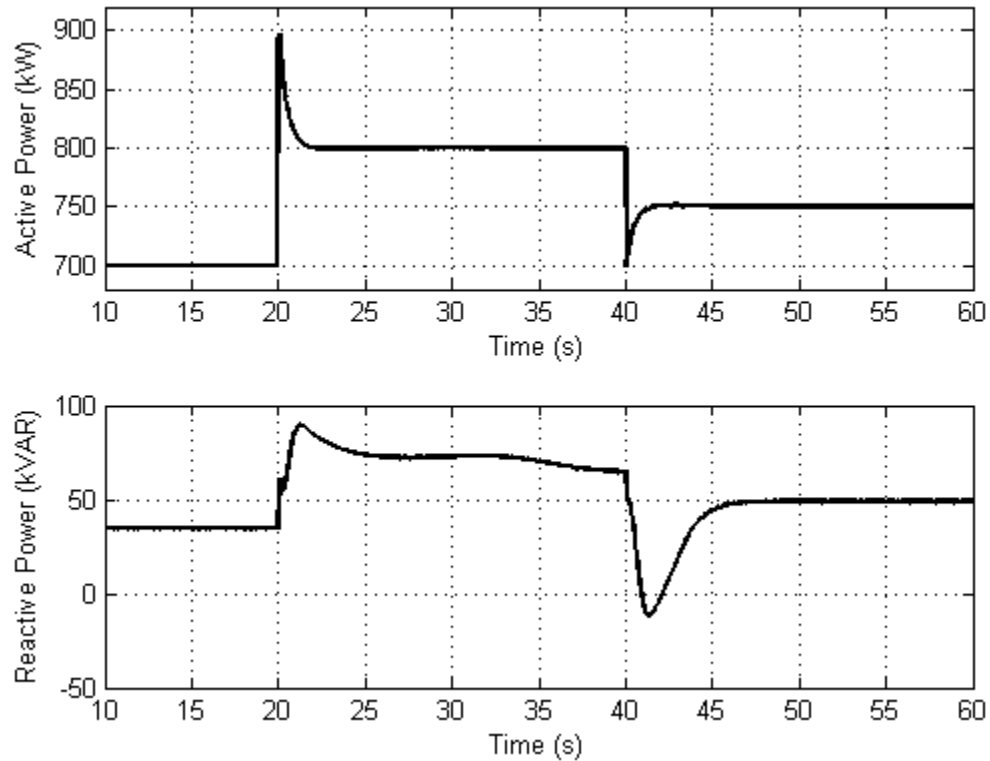


Fig. 4.15 Active (top) and reactive (bottom) power supplied by diesel generator for Case 2.

Fig. 4.15 demonstrates the active and reactive power of the diesel generator supplied to the hybrid system. The diesel generator supplies 700 kW and approximately 35 kVAR from 10 to 20 s. The active and reactive power increases to 900 kW and 80 kVAR, respectively, when the load is increased at 20 s. Even though the MT response to load changes is within a few seconds, the diesel generator response is quicker. This means that the diesel generator is picking up the abrupt load change until the MT is able to increase power. At which time the diesel generator and MT are at steady state, from 25 to 40 s. When the load decreases at 40 s, the diesel generator response is again faster than the MT. The active and reactive power decrease sharply and quickly increase to steady state values of 750 kW and 50 kVAR, respectively, by 50 s.

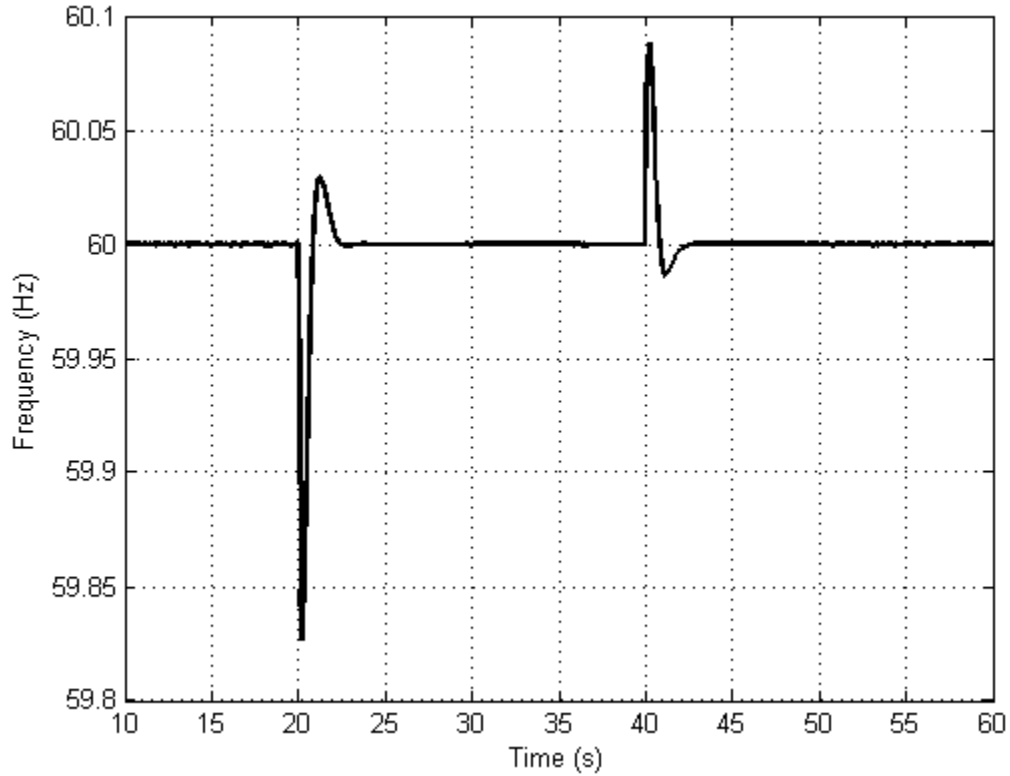


Fig. 4.16 Frequency of SOFC-MT-CC system during simulation for Case 2.

As Fig. 4.16 shows, the frequency of the system is at 60 Hz from 10 to 20 s. The frequency sharply decreases to approximately 59.825 Hz when the load is increased at 20 s. The frequency returns to the acceptable range ($60 \text{ Hz} \pm 0.05 \text{ Hz}$) of frequency for interconnection set by FERC and is within the acceptable limits of frequency stability. The frequency increases to about 60.09 Hz at 40 s when the load decreases. The frequency returns to a stable 60 Hz by 45 s after the generation sources have decreased.

Case 3: SOFC-MT-CC Response with Limited Diesel Generator. To eliminate the influence of the diesel generator on the frequency of the system and load response of the MT, the output power of the diesel generator was limited. Since the diesel generator can also be seen as a representation of a weak grid, it is equivalent to say the load

response of the MT is observed without grid support while in stand-alone operation. Without the additional generation of the diesel generator, or grid support, the system must rely on the MT for any transient occurrences.

The load attached to the system was set at 1 MW, with the MT set at 100 kW and the diesel generator at 900 kW. The load was increased at 30 s to 1.12 MW while the diesel generator was limited to 900 kW. As Fig. 4.17 shows, the MT increases generation to 220 kW within a few seconds of the load increase. The response of the diesel generator is shown in Fig. 4.18.

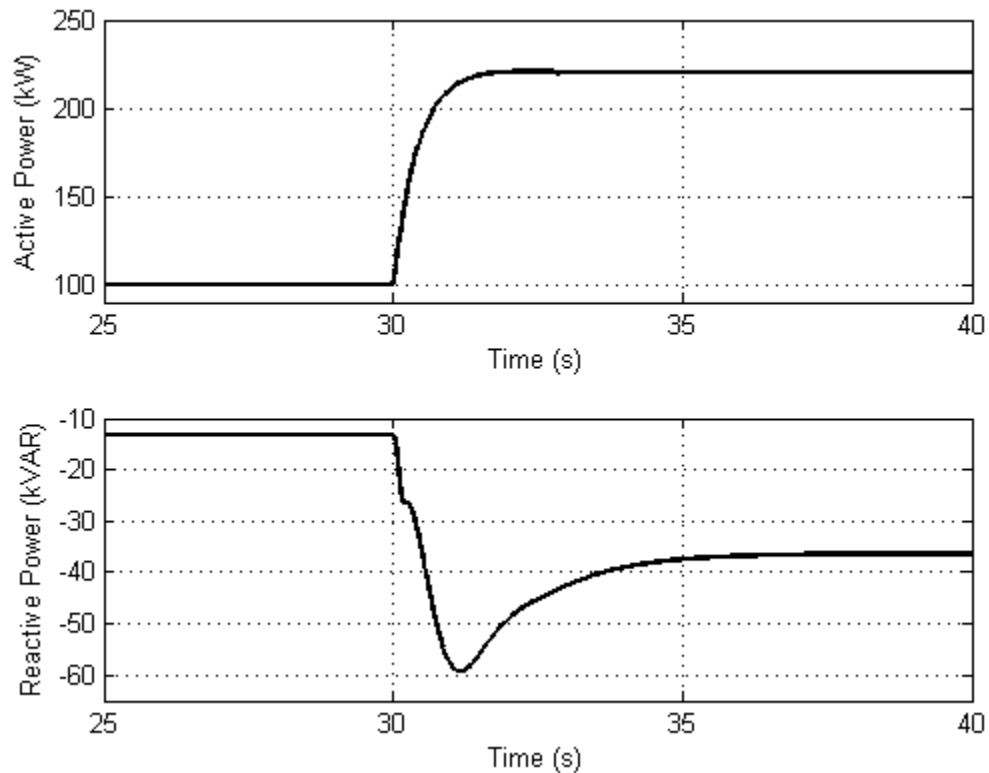


Fig. 4.17 Active (top) and reactive (bottom) power supplied by MT to meet load demand for Case 3.

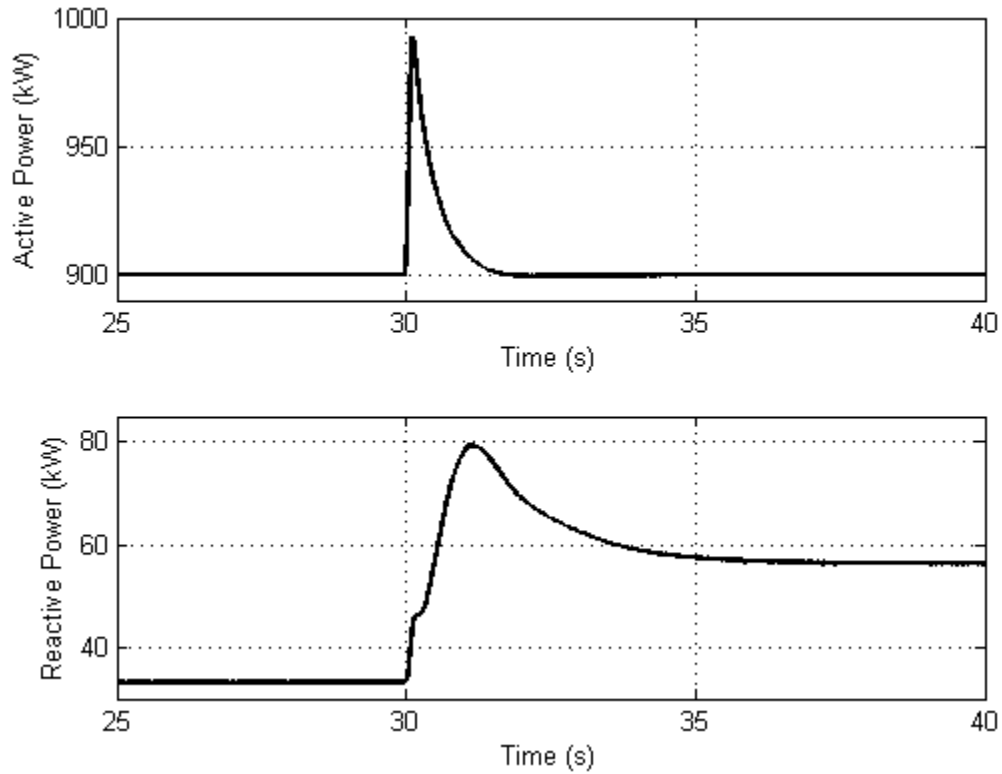


Fig. 4.18 Active (top) and reactive (bottom) power supplied by diesel generator for Case 3.

As Fig. 4.18 demonstrates, even though the diesel generator is limited to 900 kW, when the load increases at 30 s, there is a spike of power delivered. This is due to the inertia of the diesel generator reacting to the increase in load. After this transient, the active power returns to 900 kW within a couple seconds. The reactive power of the diesel generator demonstrates the additional reactive power required by the MT induction generator in order to maintain the magnetization curve. Otherwise, a major drop in generator voltage would occur [55].

The frequency of the system, meanwhile, remains stable throughout the load change, as shown in Fig. 4.19.

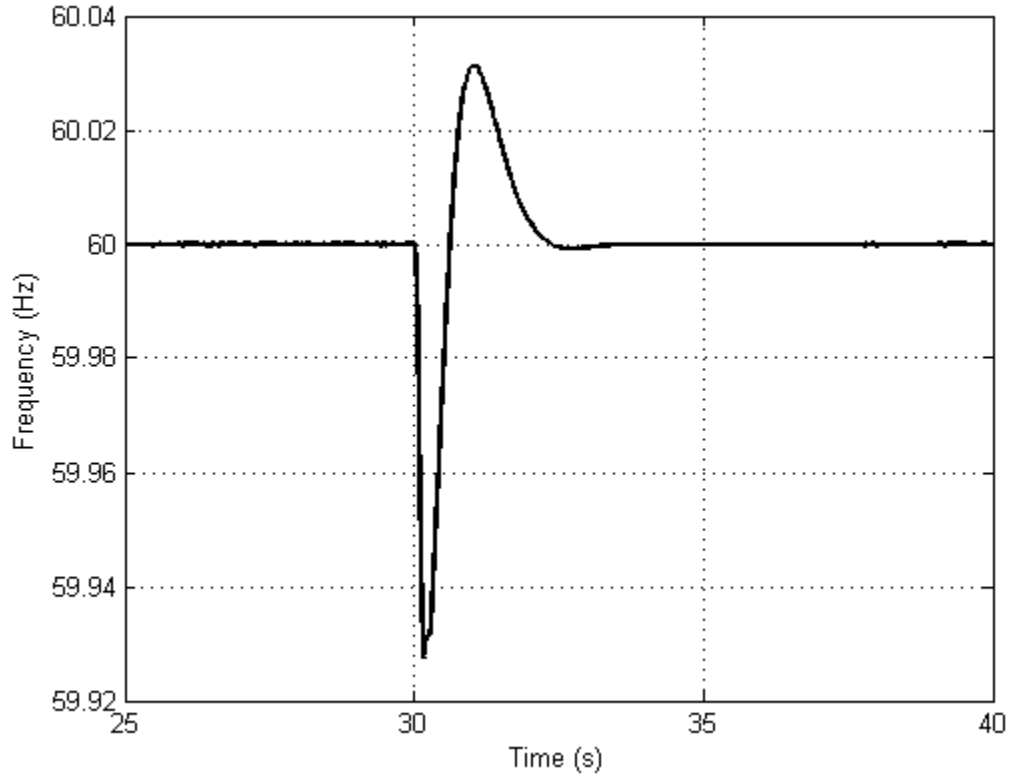


Fig. 4.19 Frequency of SOFC-MT-CC system during simulation for Case 3.

Fig. 4.19 shows the frequency of the system during the increase of load at 30 s. The frequency decreases when the load is increased and increases when the MT generation increases over the same time span. The frequency returns to a stable 60 Hz by 35 s, when the MT has achieved a steady state generation.

Discussion

The short response time of the electrical power delivered by the MT allows for an increased stability in terms of voltage and frequency regulation. Under variable load conditions, the longer response time and physical sensitivity to load changes of the SOFC suggests the necessity of coupling with a device capable of such varying load scenarios.

In this sense, a MT is capable and suited well to pair with a SOFC given its fast mechanical operation.

In order for the system to aptly respond to all transient events a fast-response energy storage device along with a proper power management strategy is required, and is recommended for fuel cell hybrid systems.

CONCLUSION

Hybrid SOFC-MT systems benefit from combined cycle operation by increasing the efficiency and transient capabilities of the system along with decreasing fuel consumption and subsequent emissions. This research considered the modeling, efficiency evaluation, and power management of a hybrid SOFC-MT system in combined cycle (CC) operation with combined heat and power (CHP) functionality for residential applications in islanded or grid-connected modes within a microgrid environment. The MT exhaust gas stream and the unused SOFC fuel stream from the anode channel were utilized in CC operation preheat the fuel in the MT for an increase of efficiency. The efficiency of the system was analyzed through the interconnection of a transfer function and thermodynamic microturbine model. The modeling of thermodynamically based combustor and heat exchanger models was also presented. The exhaust gas from the SOFC was utilized in CHP operation for residential hot water and electricity production for 250 residences in the Pacific Northwest. The load response of the SOFC and MT were also examined during transient load events. The SOFC was shown to have a slower response than the MT to load changes. The hybrid system frequency was also shown to remain stable during load transients in a microgrid environment.

Future Work

SOFC-MT hybrids have a great potential to function as effective electricity sources for the future, while more efficient fuels, i.e. hydrogen, can subsequently be integrated into the SOFC-MT installations, further improving generation efficiency and

lowering associated costs. The addition of renewable energy sources, such as a wind or PV, to the SOFC-MT-CC system can be further analyzed as a DG source. By increasing the number of generation sources, the system will be more robust, yet require further control and power management strategies to be employed. A cost benefit analysis as well as a stability analysis can also be conducted on a system with more generation sources.

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APPENDIX A

MATLAB/SIMULINK SIMULATION DIAGRAMS

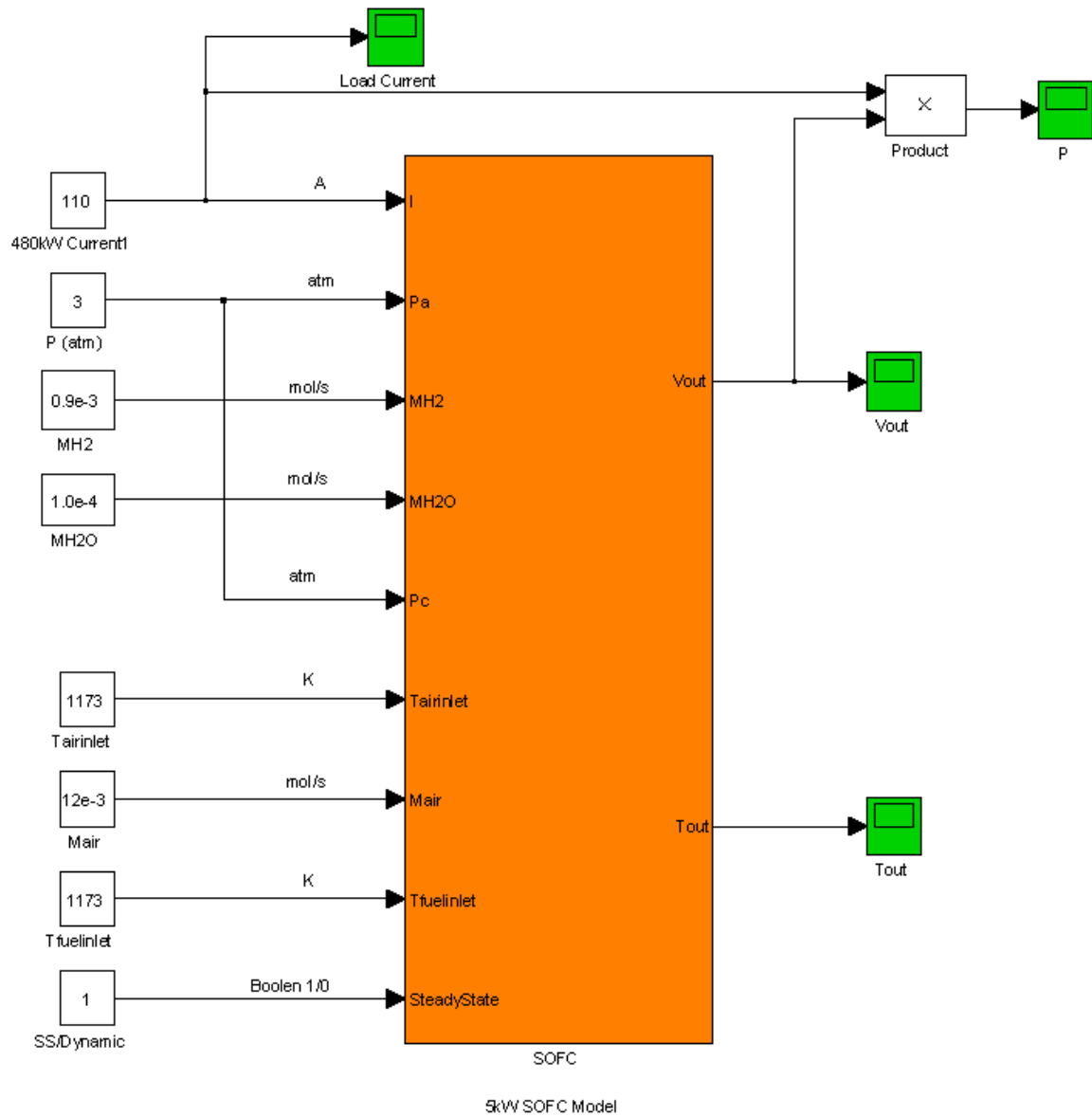


Fig. A.1 SOFC 5kW stack simulation diagram.

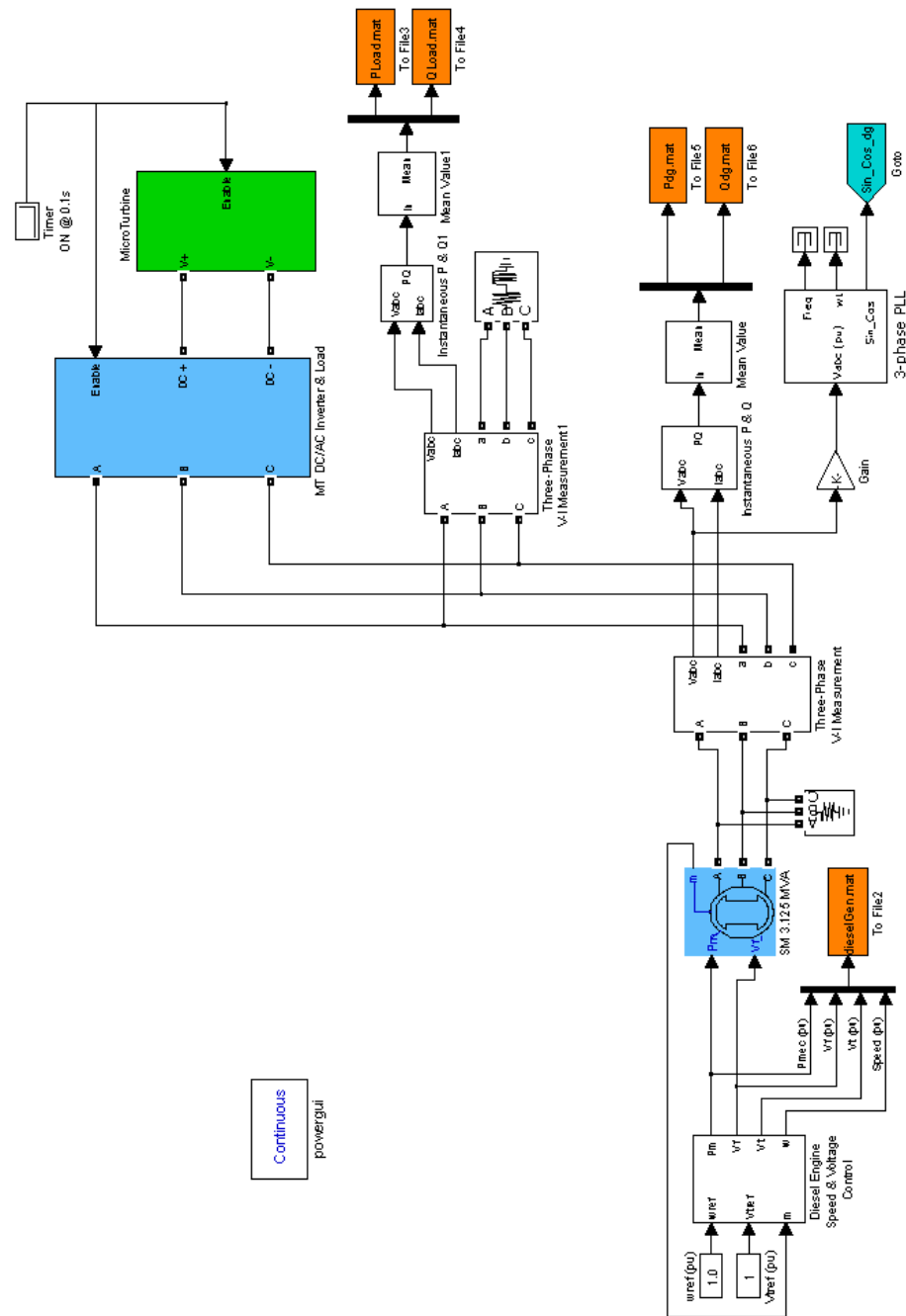


Fig. A.2 MT simulation diagram.

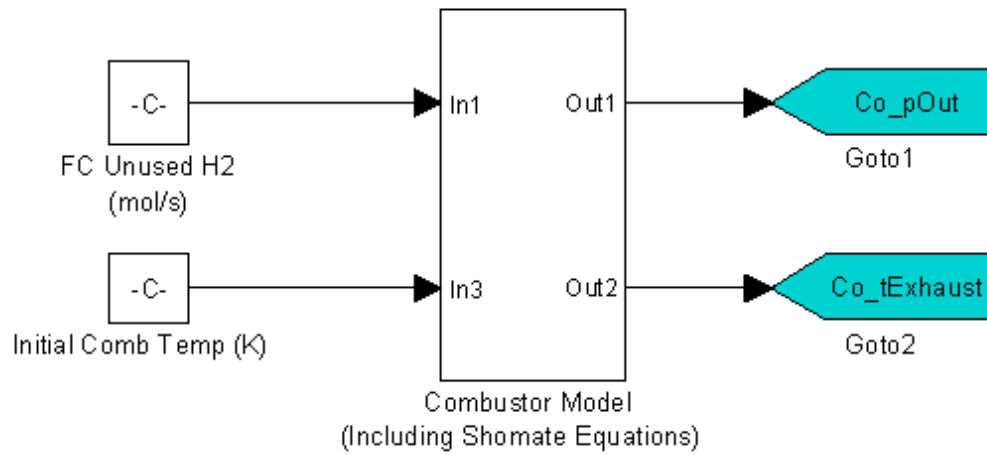


Fig. A.3 Combustor simulation diagram.

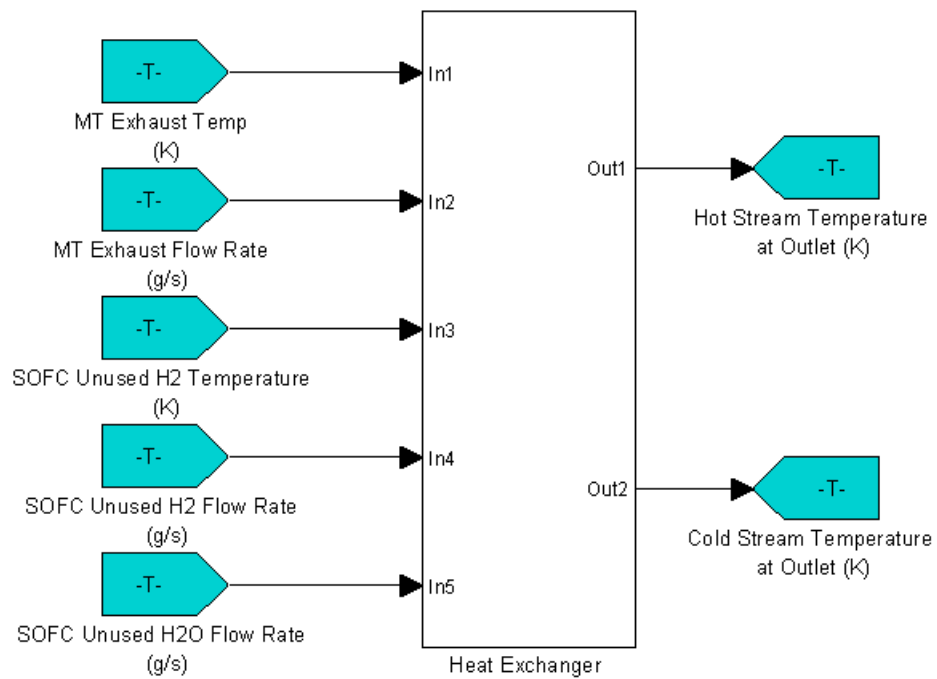


Fig. A.4 Heat exchanger simulation diagram.