SIMULATING FUEL-FIRED COMBINATION SPACE AND DOMESTIC WATER HEATING SYSTEMS

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ABSTRACT

HVAC systems that supply both space heating and domestic hot water are becoming more popular in residential buildings In North America. With these “combination systems” water from a fuel-fired storage tank is used to satisfy domestic hot water needs directly. Space heating is accomplished convectively, by circulating hot water from the tank through a water-to-air heat exchanger in a fan-coil system.

An explicit plant modelling approach was used to represent a combination system within the ESP-r/HOT3000 simulation program. A simplified representation of the fuel-fired storage tank was employed in this initial phase of the research in order to explore the factors that significantly affect the modelling. As such, issues such as stratification in the tank, condensation of exhaust gases, cooling from ventilation air during burner off time were not included in the model.

Measured data from a conventional power-vented water heater and from a high-efficiency condensing water heater were used to test and validate the model. Observations from these comparisons demonstrate the shortcomings and strengths of this modelling approach that utilizes the simplified representation of the fuel-fired storage tank.

The paper will detail the modelling approach employed, contrast the model and measured results, and summarize a literature survey on the modelling of boilers and fuel-fired storage tank models.

INTRODUCTION

It is now possible to buy on the North American market a system that combine the functions of space and domestic hot water (DHW) heating into a single unit. The system consists of a fuel-fired water storage tank connected to a heating coil in the air-handler that heats the supply air to the house. In addition the tank supplies the various outlets in the house for domestic hot water consumption. Such a system is referred to as a combination system. One advantage of the system is that it reduces the capital cost needed to perform space and DHW heating. This is achieved by replacing a conventional system consisting of a furnace/boiler and a DHW tank by a single fuel-fired storage tank. This in turn also leads to a reduction in the indoor space occupied by the space and DHW heating systems. In addition, the new design eliminates the frequent cycling associated with the operation of a furnace or boiler. As a result, there is the potential to improve the part-load performance with the use of a single fuel-fired tank for both space and DHW heating. The improvement in the part-load performance will be especially pronounced when a combination system is used instead of a conventional power vented furnace or boiler with an efficiency that decreases significantly at part load.

This paper deals with the implementation and validation of a simulation model for combination systems. The ESP-r/HOT3000 simulation engine is used as the platform for this work. This simulator is based upon the comprehensive and extensively validated ESP-r program developed at the University of Strathclyde (ESRU 2000), with algorithmic additions by the CANMET Energy Technology Centre (CETC) to support the modelling of Canadian (and international) housing. The first part of the paper is a summary of a literature review on the subject of simulation models for fuel-fired water storage tanks and boilers. The simulation model, developed for the present study, and its implementation in the ESP-r/HOT3000 engine are then presented. Then results obtained using the simulation model are compared against experimental data for two combination systems tested at the Canadian Centre for Housing Technology (CCHT).

MODELS FOR FUEL-FIRED WATER STORAGE TANKS AND BOILERS: LITERATURE SURVEY

According to the ASHRAE Reference Guide for Dynamic Models of HVAC Equipment (Bourdouhe et al. 1998), the boiler models found in the literature fall under either of three categories: 1) fundamental or less empirical models, 2) semi-empirical models, 3) or fully-empirical models. Fundamental or less empirical models use an analytical approach to characterize the combustion and the heat transfer inside the boiler. These models require detailed information about the boiler shape and size. Semi-empirical models require a combination of modelling the heat transfer inside the boiler and empirical data input. For fully-empirical models, the heat transfer is not modelled. Model parameters are then determined based on in-situ and/or laboratory measurements.

Examples of fundamental models for boilers are found in Chi et al. (1983), Idem et al. (1992), Marchant et al. (1980), and Yau and Rose (1993). All of these models are transient except the models by Idem and Yau and Rose. In general these models are based on using several nodes to represent different parts of the boiler or water heater. For example, nodes can be used to represent the combustion gases in the combustion chamber, the flue gases, the water side of the heat exchanger, the metal wall and the tubes, the shell, etc… Energy balance equations are then applied to each of these nodes. Then models are formulated for the various radiation and convection heat transfers taking place inside the boiler. A heat transfer model is needed for the combustion chamber to quantify the heat transfer from combustion gases to the walls of the boiler or the tubes of the heat exchanger when the burner is on. When the burner is off, a heat transfer model is needed between the ventilation air and the boiler walls and tubes of the heat exchanger. A model is also needed for the interface between the water and tube wall and between the boiler shell and surrounding air. In addition, a model is needed for the heat transfer between the flue gases and the boiler. If there is water condensation in the flue, then a heat transfer model between the flue gases and the unit is needed for the dry region and another one for the wet region (Yau and Rose 1993).

Examples of Semi-Empirical boiler models can be found in Clarke (2001), Naeslund (1995), Ojanen (1985), and Park and Kelly (1989). All of these models are transient. In this case, energy balance equations are used for the various sections of the unit. In addition models are used to characterize the heat transfer processes between the different sections. As stated previously, these models need also empirical data input. For example in the model by Naeslund, steady-state experimental data is needed to determine the only constant in the steady-state temperature expressions of the flue gases. The model by Ojanen estimates most of the heat transfer coefficients based on test data. In the model by Park and Kelly the boiler heat exchanger model uses test data as input. This model also uses empirical constants to characterize the rise, during on periods, and fall, during off periods, of the flue gas temperature.

The literature review yielded many more references for Empirical than for both Semi-empirical and Fundamental boiler models. Examples of these models are given by Bonne (1976), Chi and Kelly (1978), Claus and Stephane (1985), Laundry et al. (1994), Laret (1989), and Lebrun et al. (1993). In the case of these models, the heat transfer processes between the various components are not modelled analytically starting from fundamental principles. These models rely heavily on input from test data.

For example, in the model by Bonne, the flue temperature variation is measured experimentally at various furnace loads. For this model also empirical constants are needed to characterize the airflow through the flue and stack during the period when the burner is off. This off-time airflow is used to estimate the ventilation losses of the boiler. For the model by Chi and Kelly, the flue gas temperature and CO_2 concentration at steady-state are measured experimentally. Also for this model the flue temperature variation is measured during cool-down and warm-up tests from steady-state conditions. Empirical constants are also needed to determine the ventilation flow during burner off-time.

All of the references cited previously deal with models for fuel-fired boilers. Some of these models also apply for furnaces such is the case for example for the model by Bonne (1985). One reference by Paul et al. (1994) deals with the validation of a computer program (TANK) used for the analysis of gas-fired water-storage tanks. The reference only contains simulation results from the TANK computer program, but no details are given about the fundamentals of the simulation model. It seems then that simulation tools for the analysis of fuel-fired storage tanks are limited. The present work is then carried to implement a simulation model for fuel-fired storage tanks into a building simulation program (ESP-r/HOT3000). This tool can then be used to analyse the performance of combination systems.

FUEL-FIRED STORAGE TANK MODEL

A three-node model will be used, similar to the design used by Kelly (2001). Node 1 represents the water and the tank casing. Node 2 represents the combustion chamber. Node 3 represents the flue gases. This is illustrated in Figure 1. A three-node approach was chosen as this will allow the model to be easily expanded in the future to explicitly consider non-steady combustion efficiency and the heat transfer processes between the flue and the water tank.

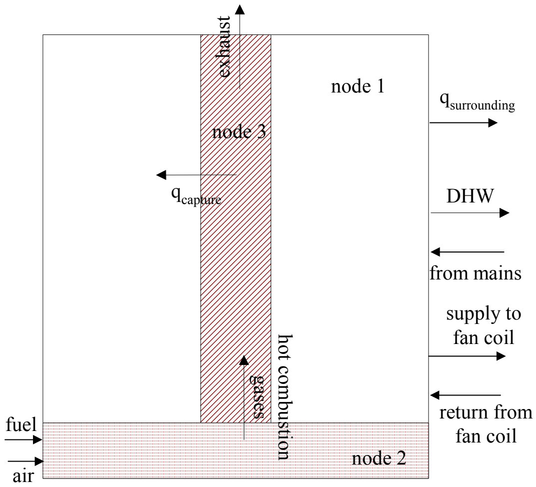

Figure 1: Three-node representation of gas-fired storage tank

Energy Balance on Node 1

Energy balance for node1 is given by,

$$(Mc_P)_1 \frac{\partial T_1}{\partial t} = q_{capture} - q_{surrounding} - (\dot{m}c_P)_{fan-coil}(T_1 - T_r) - (\dot{m}c_P)_{DHW}(T_1 - T_m) \quad (1)$$

$q_{capture}$ is the heat transferred to the water and $q_{surrounding}$ is the heat loss to the surrounding through the tank shell. The 3rd and 4th terms in equation 1 represent the net energy flows to the fan coil and for DHW consumption, respectively. The DHW mass flow and main temperature are input to the model.

$q_{capture}$ is given by,

$$q_{capture} = \mu_{combustion+flue} \cdot HHV \cdot \dot{m}_{fuel} \quad (2)$$

$\mu_{combustion\ +\ flue}$ is the overall efficiency at converting the fuel's chemical energy content to thermal energy in the water. This efficiency is assumed to be constant in the current model and given as a user input. The fuel flow rate (kmol/s) is determined using equation 5. HHV in equation 1 is the energy content of the fuel expressed with the higher heating value (MJ/kmol fuel). It is calculated using,

$$HHV_{fuel} = \sum_{i=1}^{i=3} n_i \cdot W_i \cdot HHV_i \quad (3)$$

Where n_i, W_i, and HHV_i are the mole fraction, molecular weight, and HHV of constituent i. There are four possible fuel constituents: methane CH_4, ethane C_2H_6, and propane C_3H_8. The user can control the mole fractions of these constituents, as well as the inerts nitrogen and carbon dioxide. As such, the user can model any composition of natural gas.

The heat lost from the warm tank to the surrounding zone is calculated with,

$$q_{surrounding} = (UA)_{tank} \cdot (T_1 - T_{room}) \quad (4)$$

The tank's *(UA)* value is a user input. T_{room} is the temperature of the room containing the tank.

After replacing equations 2, 3, and 4 into equation 1, it is then possible to derive the fully implicit and fully explicit forms of the energy balance equation for node 1. These forms are needed to derive the proper parameters needed for implementing the model as an ESP-r/HOT3000 plant component.

Energy Balance on Node 2

The energy balance for node 2 can be given by,

$$(\dot{m}h)_{fuel,in} + (\dot{m}h)_{air,in} + q_{burner} = (\dot{m}h)_{hot-comb-gases} \quad (5)$$

The enthalpy of the fuel and air streams entering the combustion chamber is calculated as described by Beausoleil-Morrison (2001). The mass flow of fuel (kmol/s) flowing into the combustion chamber is a controlled variable. It is calculated based upon the heat output from the burner and the HHV,

$$\dot{m}_{fuel} = \frac{q_{burner}}{HHV} \cdot \frac{1MJ}{10^6 J} \quad (6)$$

It is to be noted that an efficiency does not appear in Equation 5. This combustion efficiency is accounted for in the evaluation of the $q_{capture}$ term in Equation 2. The implicit assumption here is that all the fuel's chemical energy content is converted in node 2. This is clearly a modelling artifact. However, due to the form of equations 1 and 5, this modelling artifact will have no impact upon the energy transferred to the water or the fuel consumption. If the modelling of the combustion and flue-to-water heat exchanger processes are refined in the future, a more realistic treatment of the $q_{capture}$ and q_{burner} terms can result.

The air flowing into the combustion chamber is calculated based upon the stoichiometric air requirement, the fuel flow rate calculated with equation 6, and the user-supplied excess air requirement,

$$\dot{m}_{air-in} = (\dot{m}_{fuel})(S_{air/fuel})(E_{air}) \quad (7)$$

The last term in equation 7 is the excess air ratio (a fraction). The second term in equation 7 is the ratio of the molar flow rate of air to the molar flow rate of fuel to react the fuel stoichiometrically. It is determined from the flow rate of the fuel constituents and the following chemical reactions. Note that this assumes the combustion process is complete, a reasonable approximation for the purposes of calculating the incoming air flow rate.

$$\begin{aligned} &CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O \\ &C_2H_6 + \frac{7}{2}O_2 \leftrightarrow 2CO_2 + 3H_2O \\ &C_3H_8 + 5O_2 \leftrightarrow 3CO_2 + 4H_2O \end{aligned} \qquad (8)$$

The last term in equation 5 is evaluated at node 2's temperature. The chemical reactions described in equation 7 and equations 18 to 22 of Beausoleil-Morrison (2001) establish the flow rates of the exhaust gases. The enthalpy of the exhaust gases is evaluated as described in Beausoleil-Morrison (2001).

The approach outlined in equations 26 to 26b in Beausoleil-Morrison (2001) is used to linearize the last term of equation 5. With this and the above discussion, the energy balance on node 2 can be written as,

$$\begin{aligned} (\dot{m}c_P)_2 T_2 = (\dot{m}h)_{fuel,in} + (\dot{m}h)_{air,in} + q_{burner} + \\ (\dot{m}c_P)_2 (25^o C) \end{aligned} \qquad (9)$$

Energy Balance on Node 3

The energy balance on node 3 can be written as,

$$\begin{aligned} &(\dot{m}h)_{hot-comb-gases} = q_{capture} + (\dot{m}h)_{exhaust} \\ &(\dot{m}h)_2 = q_{capture} + (\dot{m}h)_3 \end{aligned} \qquad (10)$$

Again using the linearizing approach discussed above, the energy balance on node 2 can be written as,

$$\begin{aligned} (\dot{m}c_P)_3 T_3 - (\dot{m}c_P)_2 T_2 = (\dot{m}c_P)_3 \left(25^o C\right) - \\ (\dot{m}c_P)_2 \left(25^o C\right) - q_{capture} \end{aligned} \qquad (11)$$

Implementing Model into ESP-r/HOT3000

The model for fuel-fired storage tank described previously is then implemented as a plant component in the ESP-r/HOT3000 engine. ESP-r/HOT3000's explicit HVAC modelling domain is based upon a component-level approach whereby users assemble components into a coherent HVAC system. Data must be provided to define each component (e.g. a boiler) and the arrangement of the components. Users must also specify how components are controlled, indicating what variables are sensed (e.g. air temperature in a room), and how components are actuated (e.g. water flow through a coil) in response to the sensor signals.

Each component in the HVAC network is represented by one or more control volumes, and each control volume is characterized by mathematical models that describe the control volume's energy and mass exchanges with connected components and the environment. Writing energy and mass balances for each control volume leads to the formation of three matrices of equations that describe the HVAC plant network's thermal and mass flow state. A direct solution approach is used to solve these three matrices. As the equation set is highly non-linear, iteration is used to reform and resolve the matrices until convergence is achieved.

The energy balance equations (equations 1, 9, and 11) for the three nodes representing the fuel-fired storage tank, and all the other relevant equations, are used to derive the appropriate energy balance matrix coefficients and constants for all the unknowns, which are the temperatures of the three nodes. These matrix coefficients and constants are deduced based on a set procedure for ESP-r/HOT3000 to properly describe the energy balance equations for a plant component. These coefficients and constants are incorporated within ESP-r/HOT3000 into the energy balance matrix for all the plant components before being passed to the plant matrix solver.

COMPARISON OF MODEL PREDICTIONS TO EXPERIMENTAL DATA

Two fuel-fired water storage tank systems (Combo1 and Combo2) were tested at the Canadian Centre for Housing Technology (CCHT). Combo1 is a high efficiency-condensing water heater, whereas Combo2 is a conventional power-vented water heater. Fuel consumption measured during these tests for the two combo systems is compared against predictions of the simulation model implemented in ESP-r/HOT3000. These comparisons are carried for the period of February 7th to 13th 2000 for Combo1 and February 1st to 6th 2000 for Combo2.

Experimental Setup

As indicated previously the two systems are tested at the Test House of the Canadian Centre for Housing Technology. The house has two storeys, a basement, an attic, and a garage. The two storeys and the basement are heated by a forced-air heating system. The two combination systems are installed in the basement. During the testing of either of the two systems (Combo1 and Combo2), hot water to the heating coil of the forced-air system of the house is circulated from the fuel-fired storage tank when there is a need for heating in the house. The water storage tank also supplies water to the various outlets in the house for domestic hot water consumption (DHW). In this case, water from the tank is mixed with cold water through a mixing valve to maintain a water supply temperature of 50 ^oC to the taps and showers. The hourly water supply to the house for DHW is shown in Figure 2. A schematic of the experimental layout is shown in Figure 3.

Experimental Data

Table 1 lists some of the variables that are monitored during the experiments. Average hourly values are collected for each of the temperatures listed in Table 1. Temperatures of the make up water to the tank, water after the mixing valve, and hot water from the tank to the mixing valve are recorded only when there is a call to supply water for DHW consumption. Similarly, the temperatures of supply and return air to the house and supply and return water to the heating coil are recorded only when there is call for heating in the house.

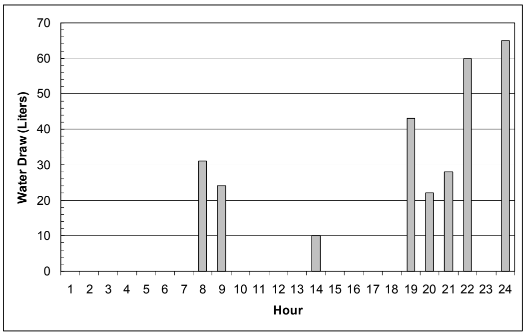

Figure 2 Hourly water draw for DHW consumption

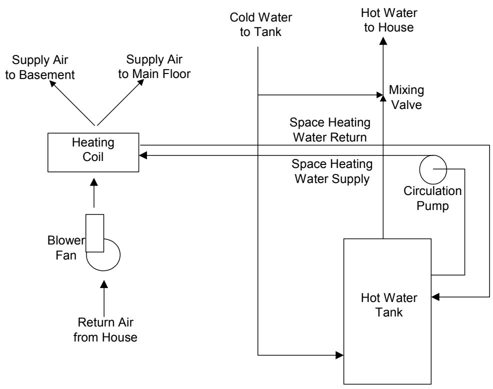

Figure 3 Layout of the experimental setup

Using the hourly measurements for the water temperature to and from the heating coil and water mass flow rate, it is possible to determine the hourly load at the coil. This load will be imposed directly on the storage tank as explained in the section on the ESP-r/HOT3000 Simulation Model. In addition, as indicated previously, the hourly water flow out of the mixing valve, at a temperature of 50 ^oC, is given in Figure 2. This information with hourly measurements of the make up water temperature and the temperature of the water from the tank can be used to determine the hourly water flow rate out of the tank to satisfy DHW consumption.

Table 1
List of some of the monitored variables during tests

Variable Description
Temperature of main floor (^oC)
Temperature of second floor (^oC)
Temperature of basement (^oC)
Temperature of make up water to the tank (^oC)
Temperature of mixed water after the mixing valve (^oC)
Temperature of hot water from tank to mixing valve (^oC)
Temperature of supply air to house (^oC)
Temperature of return air from house before blower (^oC)
Temperature of supply water to heating coil (^oC)
Temperature of return water from heating coil (^oC)
DHW draw to the house (L)
Water flow to heating coil (L)
On-time of circulation pump (sec)
On-time of blower (min)
Gas consumption (m^3)

Characteristics of Combo1 and Combo2 systems

Tables 2 and 3 list the characteristics of the two Combo systems based on equipment specifications from the manufacturers. As indicated previously, Combo1 is a high efficiency-condensing water heater and Combo2 is a conventional power-vented water heater.

Table 2
Characteristics of Combo1 system

Capacity	29,300 W
Steady-state efficiency	94%
Tank Volume	129 Liters
Total Mass	145 kg
Set Point	59 ^oC
Shell Insulation	0.65 W/K

Table 3
Characteristics of Combo2 system

Capacity	23,400 W
Steady-state efficiency	83.5%
Tank Volume	189 Liters
Total Mass	210 kg
Set Point	59 ^oC
Shell Insulation	0.98 W/K

ESP-r/HOT3000 Simulation Model

Since the purpose of the this validation is to assess the accuracy of the fuel-fired water storage tank model, the ESP-r/HOT3000 simulation model of the test house and the mechanical system is set up so as to duplicate as closely as possible the loads imposed on the water tank during the test. The house model consists of four thermal zones: one for the main floor and the second floor combined, one for the basement, one for the attic, and one for the garage. The basement is controlled to the average temperature measured during the experiment. The model for the mechanical system consists of the fuel-fired tank, the circulation pump from the tank to the heating coil, the heating coil, and the circulation fan as shown in Figure 3.

The experimental average hourly load at the coil can be determined as explained in the Experimental Data section. This hourly load is then read from a file and included in the tank energy balance. Given that the hourly space-heating load is read from a file, the circulation pump and the fan are modelled so that they are always off. The hourly water flow rate out of the tank for DHW consumption is also read from a file along with the measured values of the make up water temperature. The water storage tank is controlled so that the water temperature is maintained at 59±2.5 °C. In this way the boundary conditions of the model are made to agree with the measured data.

Experimental and Model Predicted Fuel Consumption

Table 4 contains values for the fuel consumption and total efficiency for Combo1 system for the period of February 7th to 13th 2000. The total efficiency is defined as the ratio of the total energy delivered for space and DHW heating to the product of the total amount of fuel consumed and the higher heating value of the fuel. The results show that when the published steady-state efficiency for the unit is used, the predicted total fuel consumption is lower than the measured value. In this case the predicted total efficiency is higher than the experimental value. A steady-state efficiency of 77% is needed to get a good agreement between predicted and measured values as shown in Table 4. The hourly variation of the predicted and measured fuel consumption for Combo1 is shown in Figure 4 for the case when the steady-state efficiency is 94%. No experimental data are available for few hours at the beginning and end of the week period. The figure indicates that the trend of the hourly variation of the fuel consumption is predicted well by the model.

The difference between predicted and measured fuel consumption and total efficiency for the condensing water heater can be due to several factors. First it is possible that the actual heat losses from the tank are higher than assumed in the simulation model. In addition, the simple model used does not account for on-cycle inefficiencies and stratification inside he tank. During testing of this unit, it was found that complete condensation did not occur all the time. It is also possible that the steady-state efficiency reported by the manufacturer over estimates the performance of the unit for the operating conditions of the experiments. However, the published efficiency, being higher than what it actually is, is very unlikely to account for the total difference between measured and predicted results.

Table 4
Experimental and predicted fuel consumption and total efficiency for Combo1

Testing Period	Case	Fuel Consumption (m^3)	Total Efficiency (%)
February 7th – February 13th	Measured	71.1	77.8
	Predicted with steady-state efficiency = 94%	58.4	93.3
	Predicted with steady-state efficiency = 77%	71.2	76.5

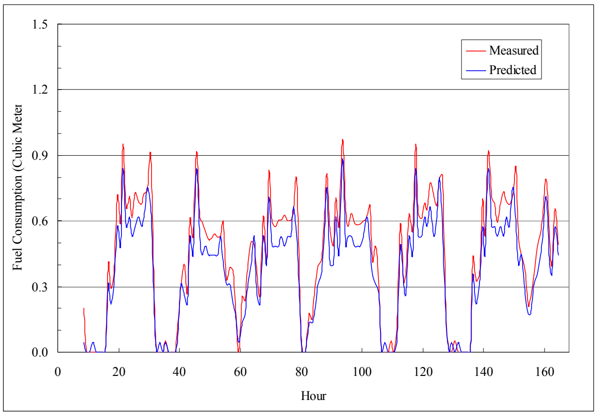

Figure 4 Variation of the experimental and predicted fuel consumption for Combo1 (steady-state burner efficiency = 94%)

Table 5 shows the total fuel consumption and the total efficiency for Combo2 system. Again in this case a steady-state burner efficiency (71.5%) less than the published value (83.5%) is needed to obtain a good agreement between the measured and predicted fuel consumption and total efficiency. The differences between measured and predicted values for this system can also be attributed to standby losses being actually higher than they are in the simulation model. The manufacturer's published steady-state efficiency can also be higher than what it is during the experiments. In addition, Combo2 is a conventional power-vented unit. When its burner is off, the system has a continuous flow of cold air through the flue. This airflow during burner off time is not currently accounted for by the simulation model. Figure 5 shows the variation of the measured and predicted fuel consumption with a steady-state efficiency of 83.5%.

The results in Figures 4 and 5 show that the model performs better at predicting the trend of the fuel consumption curve of Combo1 than Combo2. Fuel-fired condensing systems are characterized by small reductions in efficiency at part-load. For conventional power-vented units, it is expected that the efficiency substantially decreases at part-load conditions. The assumption of the present model, that the efficiency is the same for full load and part-load conditions, agrees better then with the performance of condensing systems. This might then explain why the model is better at predicting the trend of the fuel variation curve for Combo1 than Combo2.

Table 5
Experimental and predicted fuel consumption and total efficiency for Combo2

Testing Period	Case	Fuel Consumption (m^3)	Total Efficiency (%)
February 1st – February 6th	Measured	74.3	69.6
	Predicted with steady-state efficiency = 83.5%	63.7	82.7
	Predicted with steady-state efficiency = 71.5%	74.3	71.0

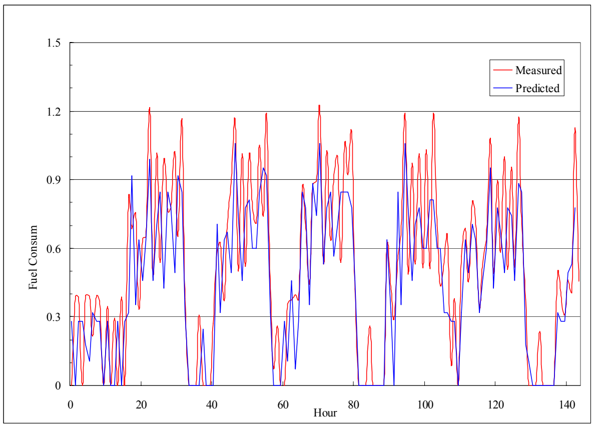

Figure 5 Variation of the experimental and predicted fuel consumption for Combo2 (steady-state burner efficiency = 83.5%)

CONCLUSIONS

A literature survey on simulation models for fuel-fired storage tanks and boilers is carried. The search indicates that there are many models for boilers and furnaces. However, it is found that there is a lack of simulation models and simulation tools to study the performance of fuel-fired water storage tanks.

A simulation model for a fuel-fired storage tank is formulated. Three nodes are used to represent the system: one for the tank wall and water, one for the flue gases, and one for the combustion chamber. The model currently assumes a constant burner thermal efficiency. The model is then implemented as a plant component in the ESP-r/HOT3000 plant-modelling environment.

Predictions from the simulation model are then compared against experimental data for a condensing water heater and a conventional power-vented water heater. The trend of the time variation curve of the fuel consumption is predicted well by the model especially in the case of the condensing unit. However, the model under predicts the total fuel consumption of the two water heaters. This can be due to several factors such as the fact that the model does not account for on-cycle inefficiencies. In the case of conventional power-vented water heater, the model does not account for off-cycle losses through the flue. There is also the possibility that the published efficiencies by the manufacturers are higher than what they actually are.

The results show then that a more detailed model is needed to better predict the performance of fuel-fired storage tanks. Many of the boiler models found in the literature survey can be beneficial in improving the current model. Results also show that it is important to have reliable data for the steady-state efficiency of combination systems. Without accurate performance data from manufacturers, it will be harder to justify the need for very detailed models for combination systems. Any improvements in the capability of the simulation model can be easily eroded by uncertainties in the model inputs such as the burner steady-state efficiency.

ACKNOWLEGMENT

The authors want to acknowledge the Director Mike Swinton and the rest of the staff at the Canadian Centre for Housing Technology for carrying the testing of the two systems. Also the authors acknowledge Dr. Evgueniy Entchev from the Advanced Combustion Technologies Laboratory of the CANMET Energy Technology Centre for collecting and providing the test data used for the validation of the model.

NOMENCLATURE

cp	Specific heat (J/kg-^{o}C)
E_{air}	Excess air ratio
h	Enthalpy (J/kg)
HHV	Higher heating value of fuel (MJ/kmol)
M	Total mass of water and tank shell (kg)
$\dot{m}$	Mass flow rate (kg/s)
n	Mole fraction
q_{burner}	Total heat input to the burner (W)
$q_{capture}$	Heat transfer from burner to tank water (W)
$q_{surrounding}$	Heat loss from tank to ambient (W)
$S_{air/fuel}$	Ratio of molar flow rate of air to molar flow rate of fuel
T_i	Temperature of node i (^{o}C)
T_m	Temperature of water from main (^{o}C)
T_r	Return water temperature from fan coil (^{o}C)
T_{room}	Temperature of surrounding air (^{o}C)
W	Molecular weight (kg/kmol)

μ Steady-state burner efficiency

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