THE SIMULATION OF A RESIDENTIAL SPACE-COOLING SYSTEM POWERED BY THE THERMAL OUTPUT OF A COGENERATION DEVICE

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ABSTRACT

A component model representing a thermally activated cooling (TAC) unit has been developed within the ESP-r/HOT3000 simulation engine. This model was designed to permit the simulation of cooling systems driven by the thermal output of fuel cells and other cogeneration technologies. This new model was combined with a component model of a fuel-cell cogeneration unit and others representing the balance of the HVAC plant to form a complete system that supplies a house's electricity, domestic hot water, space heating, and space cooling requirements. Extensive testing has been performed to verify the implementation of the TAC unit model into ESP-r/HOT3000 source code and the plant network representing the complete HVAC system. This work provides the platform for future work that will validate the TAC model against empirical data and that will study the energy and greenhouse gas emissions and the technical and economic feasibility of using cogeneration with thermally activated cycles to cool Canadian houses.

INTRODUCTION

Residential cogeneration is an emerging technology with a high potential to deliver energy efficiency and environmental benefits. The concurrent production of electricity and heat from a single fuel source can reduce primary energy consumption and associated greenhouse gas (GHG) emissions. The distributed generation nature of the technology also has the potential to reduce electrical transmission inefficiencies and alleviate utility peak demand problems. Leading contenders for cogeneration within single-family and low-rise multi-family residential buildings include fuel cells, internal combustion engines, and Stirling engines.

Cogeneration technologies have only modest electrical conversion efficiencies (~40% LHV[1] for fuel cells; 10% for Stirling; 30% for internal combustion engines). Consequently, the effective exploitation of their thermal output is critical if high levels of energy efficiency and the associated environmental benefits are to be achieved.

A number of companies (e.g. Fuel Cell Technologies, Global Thermoelectric, Sulzer-Hexis, Plug Power, Vaillant) are actively developing and beginning to commercialize fuel-cell-based cogeneration systems (FC-cogeneration) for the residential market. (The interested reader is referred to US-DOE, 2000, for a overview of fuel cell technologies and to Ellis and Gunes, 2002, for a discussion of FC-cogeneration.) This activity has motivated the research community to integrate the modelling of FC-cogeneration systems into whole-building simulation programs in order to study and assess the technology (e.g. Beausoleil-Morrison et al 2002; Ferguson and Ugursal 2002; Vetter and Wittwer 2002; Kelly and Beausoleil-Morrison 2002; Dorer and Weber 2004). The level of interest in modelling is demonstrated by the fact that a new annex (Annex 42) of the International Energy Agency's Programme on Energy Conservation in Buildings and Community Systems has been formed with the objective of developing and validating cogeneration models for whole-building simulation programs[2]. Such integrated simulation environments permit a detailed analysis of the energy flows between the building's thermal loads, occupant- and HVAC-driven electrical loads, the FC-cogeneration unit, and the balance of the HVAC plant (pumps, fans, thermal storage systems, fan-coils, etc).

FC-cogeneration units produce significant amounts of thermal energy, and as stated above, capturing and using this energy to offset the house's thermal requirements is key to achieving high overall system efficiency. Numerous modelling studies have examined using the FC-cogeneration unit's thermal output to offset domestic hot water (DHW) and/or space heating requirements (Kreutz and Ogden 2000; Beausoleil-Morrison et al 2002; Ferguson and Ugursal 2002; Vetter and Wittwer 2002; Braun 2002; Dorer and Weber 2004). Some of these early analyses have revealed that even in cold climates, when the fuel cell is operated to follow electric loads the FC-cogeneration system's thermal output can exceed a house's requirements for DHW and space heating in the spring and

[1] It is customary to express the efficiency of electrical generation equipment relative to the source fuel's lower heating value, LHV (also known as the net calorific value). In contrast it is customary—in Canada and many other countries—to relate a heating system's efficiency relative to the source fuel's higher heating value, HHV (also known as the gross calorific value).

[2] The Annex 42 web site, *www.cogen-sim.net*, describes the project's objectives and approach.

fall, and certainly in the summer.

Given this, the exploitation of the thermal output for space cooling using thermally activated cycles may be an effective strategy for maximizing overall system efficiency. Additionally, this could result in significant GHG emission savings relative to cooling systems based on conventional mechanical vapour compression cycles. During periods of high cooling demand, a FC-cogeneration unit may not be able to produce sufficient electricity to power a conventional air-conditioning system as well as the house's demands for lighting, appliances, and fans for air circulation. This would necessitate the importation of electricity from the grid at times of peak grid demands, when incremental generation is likely coming from coal and other high-GHG-emitting fossil sources.

A research project is underway to investigate the energy and GHG implications and the technical and economic feasibility of coupling a thermally activated cooling (TAC) unit to a residential FC-cogeneration system. Building upon previous work that involved the development of a model for FC-cogeneration systems (Beausoleil-Morrison et al 2002), an explicit plant model of a TAC unit was developed within the ESP-r/HOT3000 simulation engine. This paper begins with a brief overview of the ESP-r/HOT3000 simulation engine, then describes the TAC model and its integration with the FC-cogeneration model to create a coherent HVAC system for supplying a house's electrical, DHW, space heating, and space cooling needs. The testing performed to verify the implementation of the TAC model and the model of the overall HVAC system is then presented. The validation of the TAC model against empirical data and the assessment of the FC-cogeneration / TAC system will be treated in subsequent papers. Finally, conclusions are drawn and recommendations made for future work.

ESP-r/HOT3000 SIMULATION ENGINE

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 2002), with algorithmic additions by the CANMET Energy Technology Centre (CETC) to support the modelling of Canadian (and international) housing. This section briefly summarizes ESP-r/HOT3000's simulation methodology.

Partitioned solution approach

While most building analysis tools exclusively simulate thermal processes, ESP-r/HOT3000, in contrast, strives to model all relevant physical processes in an integrated and rigorous fashion. The following modelling domains are treated: building thermal; inter-zone air flow; intra-zone air flow; heat, air, water, and moisture flow within the HVAC system; electric power flow; and illumination.

ESP-r/HOT3000 employs a partitioned solution approach, applying customized solvers to each model domain. This permits an optimized treatment of each of the disparate equation sets. In this manner, one solver processes the building thermal domain, another treats the HVAC domain, while yet another handles network air flow (to resolve inter-zone flow). Interdependencies are handled by passing information (handshaking) between the solution domains on a time-step basis, this allowing the global solution to evolve in a coupled manner (Clarke 2001a).

Building thermal domain

ESP-r/HOT3000 simulates the thermal state of the building using a control-volume heat-balance methodology, which is elaborated in detail by Clarke (2001b). This encompasses three principle steps. The building is discretized by representing rooms and fabric components (walls, windows, roofs, floors) with control volumes. A heat balance considering the relevant energy flow paths (convection, radiation, infiltration, ground heat transfer, etc.) is written for each control volume. A simultaneous solution is performed on the equation set to predict the thermal state of each control volume and the heat flows between control volumes for a given point in time. This process is repeated for each time-step of the simulation.

Explicit HVAC domain

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 of 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. The energy balances are expressed in the following form,

$$(Mc_p)\frac{\partial T}{\partial t} = \sum_{i=1}^{i=n} q_i \tag{1}$$

Where M is the mass of the control volume, c_p its heat capacity, T its temperature, t is time, and q_i is an energy flow into the control volume.

The left side of this equation represents the rate of change of energy storage in the control volume. The right side represents all the energy flows which affect the control volume's thermal state. Depending upon the component under consideration, these energy flows might be a convective flux from the skin of the component to the containing room, an energy release due to combustion, or advection resulting from water or air flow through the control volume. These energy flows can be expressed with simple or complex models and can be based upon first-principle or empirical approaches, as the situation dictates. Similar equations are written to represent the water and air mass balances on each control volume.

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 (refer to Hensen 1991 for details).

The simulation of the building thermal and HVAC domains evolves by handshaking solution variables between the domains each time-step. For example, the building thermal domain's room air temperature solution is passed to the HVAC domain. These data are used to calculate containment losses in the energy balances for certain HVAC components (e.g. a boiler) and for controlling components in the HVAC plant network. The energy injected to rooms by the HVAC system is then communicated to the building thermal domain where it is used in the formation and solution of the energy balances for the rooms. This process is repeated each time-step of the simulation.

HVAC SYSTEM CONFIGURATION

Many possible configurations are feasible for constructing a FC-cogeneration system that provides electricity, DHW, space heating, and space cooling to a house. The system conceived for the current study is illustrated schematically in Figure 1. The HVAC system is composed of a FC-cogeneration unit, a TAC unit, a hot water storage tank, a cold water storage tank, an air handling unit (AHU) with two water-to-air heat exchangers, and associated pumps and piping. The two water tanks are used to buffer the thermal output of the FC-cogeneration unit and the cooling output of the TAC unit.

The FC-cogeneration unit's exhaust-to-water heat exchanger supplies hot water to the TAC unit's generator which extracts energy from this hot water stream. The somewhat cooled water stream is then delivered

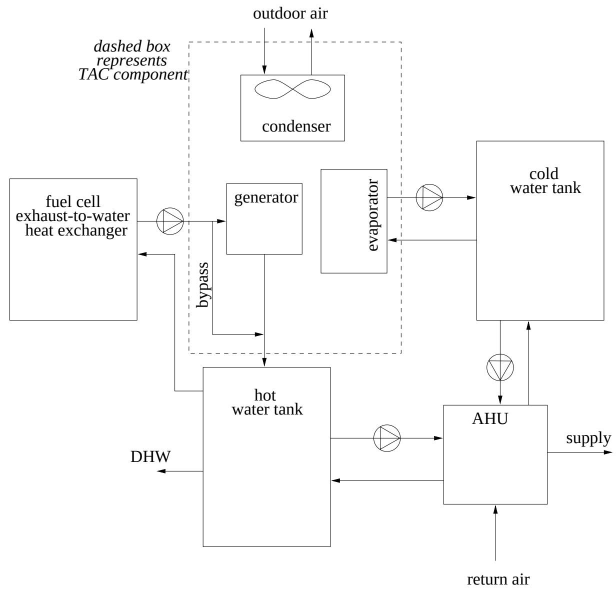

Figure 1: FC-cogeneration / TAC HVAC system

to the hot water storage tank. Water is returned from the hot water tank to the inlet of the FC-cogeneration's exhaust-to-water heat exchanger. The bypass loop upstream of the generator is activated when the TAC unit is inoperative. The TAC unit's operation is governed by the temperature of the cold water tank: the TAC unit is inoperative when this temperature is below a set-point; and operative when this temperature is above another set-point.

The hot water tank directly supplies DHW requirements, whereas the pump connecting the hot water tank and the air handling unit (AHU) is activated when the house has a demand for space heating. A burner within the tank is activated when insufficient heat is supplied by the FC-cogeneration unit to supply the DHW and space-heating needs of the house.

The TAC unit's thermodynamic cycle is not explicitly modelled. Rather, its components which interact with other components in the plant network or with the environment are represented in this model. Consequently, Figure 1 presents three of the TAC unit's components (enclosed by the dashed box). The generator interacts with the fuel cell's heat recovery device and the hot water tank, as previously discussed. The condenser rejects heat to the outdoor environment: outdoor air is circulated over the condenser's heat exchanger by the condenser fan, which operates when the TAC unit is operative.

A pump circulates water from the cold water tank over the TAC unit's evaporator to extract energy from the tank. This pump is activated only when the TAC unit is operating. The pump connecting the cold water tank and the AHU is activated when there is a demand for space cooling.

The system is also configured to use the TAC unit as a heat rejection device to prevent the hot water tank from overheating. This scenario can occur when the fuel cell's thermal output exceeds DHW, space heating, and space cooling requirements. When the tank temperature reaches the 'heat dump' set-point, the pump circulating water through the fuel cell's heat exchanger, the TAC unit's generator, and the hot water tank is activated. Similarly, the pump circulating water between the evaporator and the cold water tank is shut off, and the condenser's fan is activated. The heat rejection mode of operation takes precedence over the cooling mode. Because of this, when the TAC unit is used to reject excess heat from the hot water tank, it cannot extract energy from the cold water tank. Since the thermodynamics of the TAC unit's cycle are not explicitly treated, there is an inherent assumption that the TAC unit can operate in this mode. Future experimental work will explore the validity of this assumption.

MODEL OF TAC UNIT

This section describes the component model developed for ESP-r/HOT3000's explicit HVAC domain to represent the TAC unit.

The model was configured to represent two TAC technologies of interest: an ammonia absorption cycle and an ejector refrigeration cycle (Sun and Eames, 1996). As mentioned earlier, the TAC unit's thermodynamic cycle is not explicitly modelled. Rather the model was configured in such a fashion that all its relevant input parameters could be derived from empirical testing of a coherent TAC unit. The three-node component model used to represent the TAC unit is illustrated in Figure 2 (the variables listed in the figure are defined in the development that follows).

The energy balances elaborated below employ a quasi steady-state modelling approach. Thermal transients within the TAC unit are not considered whereas a steady-state solution is performed each time-step in response to time-varying conditions within the plant network. It is important to note that although thermal transients within the TAC unit are not considered, the thermal transients within the hot and cold water storage tanks are.

The thermal state within the TAC unit must be known at one point within its thermodynamic cycle. This is necessary in order to determine the temperatures of the water streams leaving the generator and the evaporator. In the analysis presented below the temperature of the refrigerant at the entry of the generator (T_{gen-r})

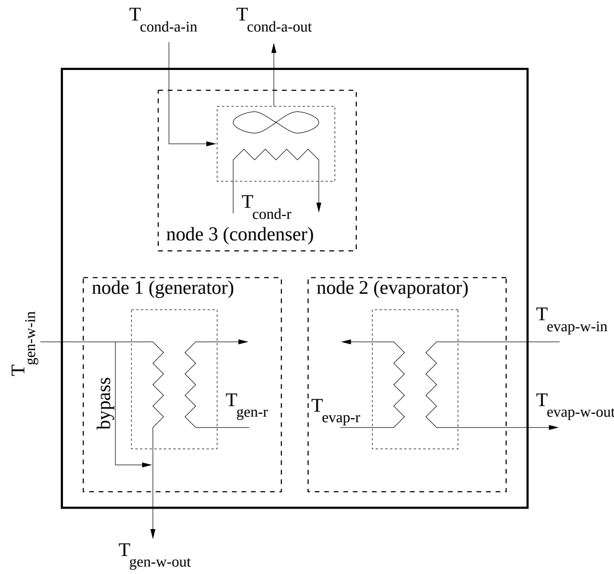

Figure 2: Three-node representation of TAC unit

is assumed to be fixed and known (user defined)[3]. The validity of this assumption for non-steady-state modes of operation will be assessed through future experimental work, and model enhancements will be made if warranted.

The energy balances employed within the model for the three nodes are presented below.

Node 1 energy balance

The energy balance on node 1 is given by,

$$q_{gen} = (\dot{m}c_p)_{gen-w} \cdot (T_{gen-w-in} - T_{gen-w-out}) \tag{2}$$

Where q_{gen} is the heat transferred from the hot water stream to the refrigerant (W). $\dot{m}_{gen-w}$ is the mass flow rate (kg/s) and $c_{p,gen-w}$ is the heat capacity (J/kgK) of the water circulating from the fuel cell's exhaust gas heat exchanger to the TAC unit's generator.

The generator is represented using a constant effectiveness approach, which is well suited to the objectives of this project. This could be replaced by a more detailed NTU approach in the future should increased modelling resolution be required. The maximum possible heat exchange would occur if the water were cooled to the refrigerant temperature,

$$q_{gen,max} = (\dot{m}c_p)_{gen-w} \cdot (T_{gen-w-in} - T_{gen-r}) \tag{3}$$

The generator's heat exchanger effectiveness is supplied by the user. It is defined as,

$$\varepsilon = \frac{q_{gen}}{q_{gen,max}} \tag{4}$$

[3] The refrigerant temperatures at the evaporator (T_{evap-r}) and condenser (T_{cond-r}) are not required as inputs, nor are they solved within the model.

Substituting equations 3 and 4 into equation 2 and arranging the terms containing solution variables (the temperatures of the fluid streams entering and leaving nodes) on the left, leads to the form of the heat balance that is used to represent node 1,

$$T_{gen-w-out} - (1-\varepsilon) \cdot T_{gen-w-in} = \varepsilon \cdot T_{gen-r} \tag{5}$$

Equations 2 through 5 are valid only when water is flowing through the generator. When water is diverted through the bypass (ie. when the TAC unit is de-activated in response to the cold water tank's temperature) q_{gen} will equal zero and $T_{gen-w-out}$ will be made to equal to $T_{gen-w-in}$.

Node 1 energy balance when TAC modulated

Equations 2 through 5 hold true only when the bypass valve upstream of the generator diverts the entire flow to the generator. This valve modulates to divert some of the hot water past the generator if the water's temperature and flow rate would result in excessive cooling output from the TAC unit. This is controlled using a quantity supplied by the user which specifies the maximum allowable heat transfer from the water to the generator ($q_{gen-limit}$). The energy balance employed for node 1 when the valve modulates to divert some of the flow around the generator is presented in this section.

The energy balance on node 1 is given by,

$$q_{gen} = (\dot{m}_{gen-w} - \dot{m}_{divert}) \cdot c_{p_{gen-w}} \tag{6}$$

$$\cdot (T_{gen-w-in} - T_{gen-w-out-before-mix})$$

Where $\dot{m}_{gen-w}$ is the flow rate of the water upstream of the diverting valve and $\dot{m}_{divert}$ is the flow rate of the water diverted around the generator. Therefore, $(\dot{m}_{gen-w} - \dot{m}_{divert})$ is the flow rate of the water flowing through the generator. $T_{gen-w-out-before-mix}$ is the temperature of the water exiting the generator, but upstream of the valve that mixes this flow with the flow that was diverted around the generator.

From equations 3 and 4 and the above, the heat exchange at the generator can be given by,

$$q_{gen} = \varepsilon \cdot (\dot{m}_{gen-w} - \dot{m}_{divert}) \cdot c_{p_{gen-w}} \cdot (T_{gen-w-in} - T_{gen-r}) \tag{7}$$

Combining equations 6 and 7 leads to the solution of temperature of the water exiting the generator, but upstream of the mixing valve,

$$T_{gen-w-out-before-mix} = \varepsilon \cdot T_{gen-r} + (1-\varepsilon) \cdot T_{gen-w-in} \tag{8}$$

The bypass valve is controlled to constrain q_{gen} to $q_{gen-limit}$. This fact and the result of equation 8 are used to solve for the position of the bypass value by re-arranging equation 6,

$$\dot{m}_{divert} = \dot{m}_{gen-w} - \frac{q_{gen-limit}}{c_{p_{gen-w}}(T_{gen-w-in} - T_{gen-w-out-before-mix})} \tag{9}$$

The state of the water exiting the generator node ($T_{gen-w-out}$) is determined by solving the heat balance of the two streams flowing into the mixing valve downstream of the generator,

$$\dot{m}_{gen-w} \cdot T_{gen-w-out} = \tag{10}$$

$$(\dot{m}_{gen-w} - \dot{m}_{divert}) \cdot T_{gen-w-out-before-mix} + \dot{m}_{divert} \cdot T_{gen-w-in}$$

Substituting equation 8 into equation 10 and re-arranging, leads to the final form of the heat balance when the bypass valve is modulating,

$$T_{gen-w-out} - \left[\left(\frac{\dot{m}_{gen-w} - \dot{m}_{divert}}{\dot{m}_{gen-w}}\right)(1-\varepsilon)\right] \cdot T_{gen-w-in} \tag{11}$$

$$= \left(\frac{\dot{m}_{gen-w} - \dot{m}_{divert}}{\dot{m}_{gen-w}}\right) \cdot \varepsilon \cdot T_{gen-r}$$

Node 2 energy balance

The energy balance on node 2 is given by,

$$q_{evap} = (\dot{m} c_p)_{evap-w} \cdot (T_{evap-w-in} - T_{evap-w-out}) \tag{12}$$

Where q_{evap} is the heat transferred from the water to the refrigerant. The flow rate of water through the evaporator, $\dot{m}_{evap-w}$, is a user input.

The TAC unit's performance is characterized with a coefficient of performance which relates the cooling power to the heat input,

$$COP = \frac{q_{evap}}{q_{gen}} \tag{13}$$

A parametric-empirical approach is used to relate the COP to the TAC unit's operating conditions. Specifically, it determines the performance in response to the water inlet temperatures at the generator and evaporator, and to the air inlet temperature at the condenser. The equation was structured to facilitate the determination of the coefficients from laboratory experiments,

$$COP = a + b_1 \cdot (T_{gen-w-in} - T_{gen-ref}) \tag{14}$$

$$+ b_2 \cdot (T_{cond-a-in} - T_{cond-ref}) + b_3 \cdot (T_{evap-w-in} - T_{evap-ref})$$

$$+ c_1 \cdot (T_{gen-w-in} - T_{gen-ref})^2 + c_2 \cdot (T_{cond-a-in} - T_{cond-ref})^2$$

$$+ c_3 \cdot (T_{evap-w-in} - T_{evap-ref})^2$$

$$+ d_1 \cdot (T_{gen-w-in} - T_{gen-ref}) \cdot (T_{cond-a-in} - T_{cond-ref})$$

$$+ d_2 \cdot (T_{gen-w-in} - T_{gen-ref}) \cdot (T_{evap-w-in} - T_{evap-ref})$$

$$+ d_3 \cdot (T_{cond-a-in} - T_{cond-ref}) \cdot (T_{evap-w-in} - T_{evap-ref})$$

Where a, b_i, c_i, and d_i are the parametric coefficients supplied by the user. $T_{gen-ref}$, $T_{cond-ref}$, and $T_{evap-ref}$ are reference temperatures and represent the "standard" operating conditions for the TAC unit. The structure of equation 14 can be altered in the future should it be found that it cannot well represent the TAC unit's performance.

Substituting equations 13 and 2 into equation 12 and arranging the terms containing solution variables on the left, leads to the form of the energy balance that is used to represent node 2,

$$T_{evap-w-out} - COP \cdot \frac{(\dot{m}c_p)_{gen-w}}{(\dot{m}c_p)_{evap-w}} \cdot T_{gen-w-out} \qquad (15)$$

$$- T_{evap-w-in} = -COP \cdot \frac{(\dot{m}c_p)_{gen-w}}{(\dot{m}c_p)_{evap-w}} \cdot T_{gen-w-in}$$

Where COP is calculated using equation 14.

Equations 12 through 15 are valid only when the pump circulating water from cold tank through the evaporator is active. When this pump is shut off q_{evap} will equal zero and $T_{evap-w-out}$ will be made to equal $T_{evap-w-in}$.

Node 3 energy balance

The energy rejected from the refrigerant at the condenser's heat exchanger is given by,

$$q_{cond} = q_{gen} + q_{evap} + q_{pump} \qquad (16)$$

Where q_{pump} is the electrical power (W) of the pump used to circulate refrigerant within the TAC unit. This is a user-defined quantity. It is assumed that the pump is located such that all the electrical power it consumes is added to the refrigerant as thermal energy.

Node 3 represents the air side of the condenser heat exchanger. An energy balance on this node is given by,

$$q_{cond} + q_{fan} = (\dot{m}c_p)_{cond-a} \cdot (T_{cond-a-out} - T_{cond-a-in}) \qquad (17)$$

Where q_{fan} is the electrical power of the fan used to circulate air over the condenser's heat exchanger. It is assumed that the fan is located such that all the electrical power consumed by its motor is added to the air stream as thermal energy.

The temperature of the air at the condenser inlet is determined by,

$$T_{cond-a-in} = T_{outdoor} + \Delta_{cond} \qquad (18)$$

Where $T_{outdoor}$ is the outdoor dry-bulb temperature from the weather file. Δ_{cond} is a constant user-defined value which represents local heating effects in the vicinity of the condenser air inlet. It is used to account for such phenomena as: solar gains warming the air in the vicinity of the condenser; poor air circulation caused by locating the condenser in an area which is sheltered by the building and vegetation; and 'heat island' effects which often lead to warmer summer temperatures in urban and suburban areas relative to the airport (where weather data is usually recorded). Although the current form of equation 18 recognizes the importance of local heating effects, further research is required to develop an algorithm to calculate Δ_{cond} dynamically during a simulation as a function of prevailing conditions. This is felt to be an important consideration because TAC cycles are more sensitive to condenser temperatures than are conventional air-conditioning systems.

Substituting equations 2 and 12 into equation 16, then grouping equations 16 through 18 and arranging the terms containing solution variables on the left leads to the form of the heat balance that is used to represent node 3,

$$T_{cond-a-out} + \frac{(\dot{m}c_p)_{gen-w}}{(\dot{m}c_p)_{cond-a}} \cdot T_{gen-w-out} \qquad (19)$$

$$+ \frac{(\dot{m}c_p)_{evap-w}}{(\dot{m}c_p)_{cond-a}} \cdot T_{evap-w-out} = \frac{\left(q_{pump} + q_{fan}\right)}{(\dot{m}c_p)_{cond-a}} + T_{outdoor} + \Delta_{cond}$$

$$+ \frac{(\dot{m}c_p)_{gen-w}}{(\dot{m}c_p)_{cond-a}} \cdot T_{gen-w-in} + \frac{(\dot{m}c_p)_{evap-w}}{(\dot{m}c_p)_{cond-a}} \cdot T_{evap-w-in}$$

Equations 5, 11, 15, and 19 form a matrix of energy balances that relate the temperatures of the three nodes within the TAC unit to each other and to the temperatures of connected nodes within the HVAC plant network (e.g. the hot and cold water tanks). Similar matrices of equations are formed for each component in the HVAC plant network. These component matrices are assembled into a global matrix and solved using the direct-iterative technique (including handshaking with the building thermal domain) that was described earlier.

MODEL TESTING

A comprehensive series of simulation tests were performed to verify the implementation of the TAC component model into ESP-r/HOT3000 source code and the plant network representing the HVAC system depicted in Figure 1. In these tests, the response of the TAC and other components and the overall HVAC system were observed over a variety of conditions encompassing the expected range of operation. This section presents example results from this testing.

The controls on the plant network were configured as follows:

- To cool the house, the pump connecting the cold water tank and the AHU is turned on when the house's air temperature exceeds 25.5°C. To

- ensure that the system can effectively cool, this pump is only turned on if the water in the cold tank is 15°C or cooler.
- The above pump is turned off when the house's air temperature has been cooled to 24.5°C.
- To cool the cold water tank, the TAC operates in cooling mode and the pump connecting the TAC's evaporator and the cold water tank is turned on when the tank temperature rises above 10°C.
- The TAC and the above-mentioned pump are turned off when the tank has been cooled to 5°C.
- To avoid overheating the hot water tank, the TAC unit is switched into heat-rejection mode when the hot water tank's temperature exceeds 80°C. As discussed earlier, this mode of operation supersedes the cooling mode: the TAC cannot cool the cold water tank while it is rejecting excess heat from the hot water tank.
- The TAC is switched out of heat rejection mode once the hot water tank has been cooled to 70°C.

The house's interior air temperature over a typical summer day is shown in Figure 3 , while Figure 4 displays the temperature of the hot and cold tanks.

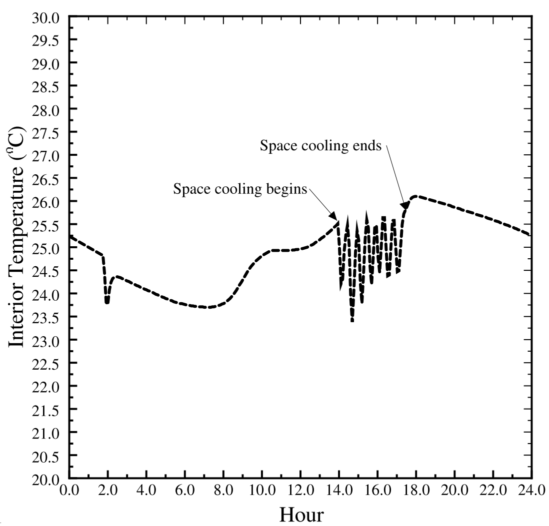

Figure 3 : House's interior air temperature

A number of operational scenarios can be seen in these figures. The TAC unit operates in cooling mode from 0h00 to 9h00 (see Figure 4): it cools the cold water tank until its temperature drops to 5°C. The TAC unit is switched out of cooling mode just after 9h00, having cooled the cold water tank to its lower set-point.

At this point, the temperature of the hot water tank begins to rise rapidly (see Figure 4) since the FC-cogeneration unit's thermal output is added directly to

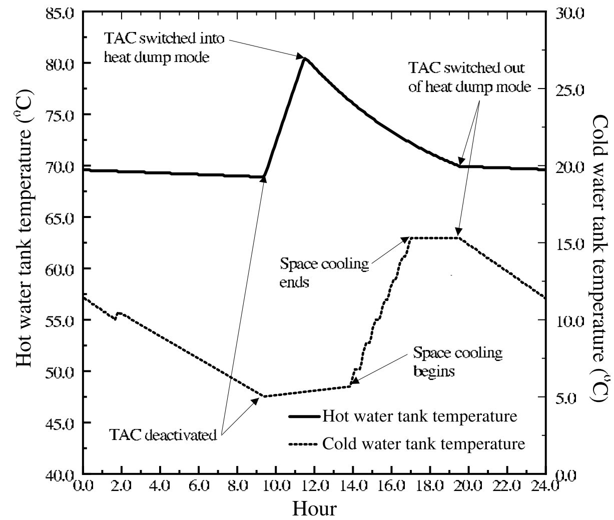

Figure 4: Hot and cold water tank temperatures

the hot water tank rather than being transferred to the TAC unit's generator. The hot water tank temperature reaches 80°C just before 12h00, triggering the system's 'heat dump' mode of operation. As the FC-cogeneration system continues to supply heat to the hot water tank, the system must operate in 'heat dump' mode until after 19h00 to cool the tank to 70°C. The temperature of the cold tank rises slowly during this same period due to heat transfer from the basement air.

The house's interior air temperature remains below the cooling set-point until about 14h00 (see Figure 3). Once this temperature rises above 25.5°C the system begins to circulate water from the cold tank to the AHU and to distribute the cooled air to the house. The system continues cooling the house until about 17h00, at which time the cold water tank temperature reaches 15°C. The system then ceases operation of the AHU as the cold water tank is too warm to effectively cool the house. As the house is no longer cooled, its air temperature rises above the system's set-point, peaking at above 26°C.

Once the system's 'heat dump' mode successfully cools the hot water tank to 70°C, it again resumes cooling the cold water tank (see 19h00 in Figure 4).

As these results show, the dual capabilities of the TAC unit to satisfy both 'heat dump' and cooling requirements make control of the system a non-trivial problem. In this example, the use of the TAC unit as a heat rejection device prevented the system from responding to the house's cooling load towards the end of the afternoon. To address this, a simulation model of a control system that maximizes the utilization of the FC-cogeneration unit's thermal output and aims to avoid the conflict between the TAC's 'heat dump' and cooling modes of operation has been developed (Ferguson 2004). This control model will be employed in the next phase of this research.

CONCLUSIONS

A component model representing a thermally activated cooling (TAC) unit has been developed within the ESP-r/HOT3000 simulation engine. This model was designed to permit the simulation of cooling systems driven by the thermal output of fuel cells and other cogeneration technologies. The model was conceived to represent two TAC technologies—ammonia absorption and ejector refrigeration cycles—although due to its generic nature it could be easily adapted to represent other TAC cycles. The model does not explicitly treat the TAC unit's thermodynamic cycle. Rather, the generator, condenser, and evaporator—the components which interact with other components in the HVAC system and the external environment—are represented and the thermal performance characterized with empirical-parametric relations.

The component model of the TAC unit was assembled with previously created component models into a network representing a complete fuel-cell-based HVAC system that supplies a house's electricity, domestic hot water, space heating, and space cooling.

A comprehensive series of simulation tests have been performed to verify the implementation of the TAC component model into ESP-r/HOT3000 source code. Significant testing has also been performed on the HVAC plant network to ensure the space heating, space cooling, DHW, and electricity supply functions operate as expected, and to verify the control behaviour of individual components.

The validation of the TAC unit model against empirical data and the derivation of the empirical coefficients describing the TAC's thermal performance from laboratory experiments will be the subject of a future paper. In the future the TAC unit model will be applied to study the energy and GHG emissions and the technical and economic feasibility of using thermally activated cycles to cool Canadian houses.

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