MODELLING BUILDING-INTEGRATED STIRLING CHP SYSTEMS

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ABSTRACT

A model suitable for studying building-integrated Stirling combined heat and power (CHP) systems has been developed and integrated in to the ESP-r simulation program. The model relies on empirical data to predict the thermal and electrical output of the CHP unit in response to conditions in the building. In this paper, the development and calibration of the model are discussed, and the model's predictions are compared experimental data collected at the Canadian Centre for Housing Technology.

INTRODUCTION

In recent years, decentralized combined heat and power (CHP) systems have received considerable attention as an alternative to traditional centralized electrical supply. By exploiting the simultaneous electric and thermal output of CHP devices, overall efficiencies greater than 90% (based on the lower heating value, LHV) can be achieved.

While much of this interest has focused on fuel-cell technologies, combustion-based CHP technologies continue to mature. Among these, CHP systems based on the Stirling-cycle offer efficient and reliable operation. Recently Entchev et al. [2] installed a small, 750 W (electric) Stirling generator in an experimental test-house, and observed combined LHV efficiencies of 77–88%. Henckes and Stripf [4] have tested a larger 9.5 kW (electric) unit on a laboratory bench, and reported overall LHV efficiencies ranging from 84–92%.

While these studies illustrate the energy savings potential of Stirling CHP systems, experiments can only investigate a handful of different system configurations and load profiles. Building simulation complements these efforts by permitting researchers to investigate optimal system configurations, and to extrapolate empirical results to different climates, load profiles, and building types.

Recognizing the importance in modelling these systems, the International Energy Agency approved the formulation of a new research annex (Annex 42) under the Energy Conservation in Buildings and Community Systems implementing agreement. Key objectives of Annex 42 include the development of simulation models for building-integrated fuel cell, Stirling, and internal combustion based CHP systems, and the comparison of these systems in different case studies.

In support of Annex 42's research goals, an empirical model suitable for modelling building-integrated Stirling CHP systems is being developed and integrated into the ESP-r whole-building simulation program. [1] The model will be capable of predicting the performance of a Stirling CHP unit in response to conditions in the building and mechanical heating plant.

The experimental characterization of Stirling CHP systems is the subject of ongoing research by other Annex 42 participants. When these studies are complete, comprehensive calibration and validation of the Stirling CHP model will be possible. Results from these validation efforts and detailed case studies applying the model will be the subject of a future paper.

The present paper discusses the challenges associated with modelling building-integrated Stirling CHP systems, and describes a preliminary model for representing Stirling-based CHP systems in residential applications. Calibration of the model is discussed, and the results are compared to data collected by Entchev et al. [2] at the Canadian Centre for Housing Technology (CCHT).

Future work will explore merging the Annex 42 Stirling model specification with a similar specification developed by Kelly [6] for internal combustion based CHP technology, as well as further calibration and validation work using data collected in Annex 42.

STIRLING COGENERATION SYSTEMS

Figure 1 depicts the energy and mass flows in a simple, fossil-fueled Stirling CHP system. Fuel and air are burned in the heater, transferring high-grade heat to the Stirling engine. The Stirling engine comprises a sealed piston-cylinder device filled with an inert gas such as helium. The Stirling engine converts a fraction of the high-grade heat into mechanical work, and rejects the remainder of the heat at low temperature to the cooling water. In CHP applications, the surplus heat recovered in the cooling water is used to meet heating and domestic hot water loads. A generator converts the engine's mechanical work output into electricity, some of which is used to meet building loads. Urieli and Berchowitz [10] provide a complete discussion of the operation and analysis of Stirling engines.

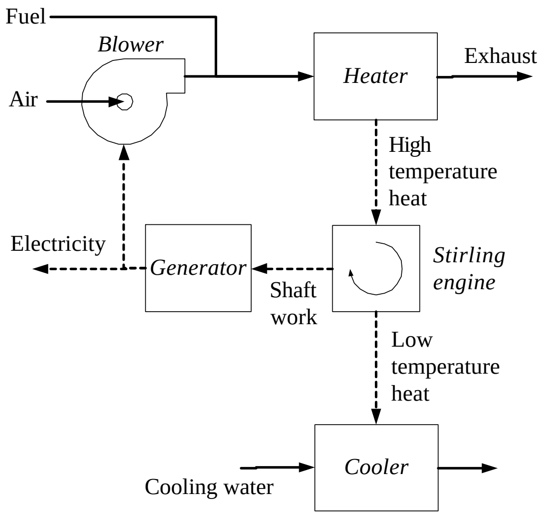

Figure 1: Mass flows (solid lines) and energy flows (dashed lines) in simple, combustion-based Stirling CHP system

MODELLING STRATEGY

A wide range of models are available for characterizing the subcomponents of a Stirling CHP system [10]. However, most of these models require i) very short time steps (≈1/100 s) that are incompatible with those used in building simulation programs (1-3600s), and ii) detailed data describing the engine geometry and operating characteristics, which are unlikely to be available for the Stirling CHP systems being developed for cogeneration use.

Pearce et al. [8, 9] contributed early models of building-integrated Stirling CHP systems. They assumed the Stirling CHP unit and balance-of-plant performance can be characterized by constant efficiencies, and predicted the CHP unit and balance of plant response to i) measured load profiles for residential houses in the UK, and ii) a lumped capacitance building model. In their investigation of optimal control of Stirling engines, Peacock and Newborough [7] further resolved the Stirling engine's operating regime into three part-load operating points, and assumed constant electrical and thermal efficiencies for these operating points. Most recently, Hawkes and Leach [3] used a polynomial equation to describe the ratio of a Stirling engine's heat to power output as a function of loading.

While these models describe the sensitivity of the Stirling CHP's performance to the unit's loading at varying levels of resolution, no attempt has been made to characterize the CHP unit's thermal response to other conditions in the mechanical plant, such as cooling water temperature and flow rate. These conditions may have a significant affect on the unit's thermal performance — especially if the unit is fitted with a condensing exhaust-gas heat exchanger.

In 2004, Kelly [5] provided a set of principles guiding development of CHP system models in Annex 42. These guidelines recommend that, in the context of building simulation, CHP systems be represented as an assembly of subsystem models. Each subsystem component is characterized using parametric equations or empirical data describing its part-load and transient behaviour. This *grey-box* modelling approach allows components of the model to be adapted and reused when modelling different systems, without requiring the detailed formulation of a fully mechanistic model based on first-principles.

While conceptually elegant, this approach poses significant challenges for model calibration and validation. Some of the energy flows, such as the reactant and product gas flows, enter and leave the system "box" and thus can be easily measured. Others, such as the shaft power produced by the engine and the electrical power required by the blower motor, flow between components within the system, and require invasive measurements to quantify accurately.

The independent tests of Stirling CHP units undertaken by Entchev et al. [2] and Henckes and Stripf [4] have not included the invasive measurements required to collect these data. Future testing of Stirling CHP units in support of Annex 42 is planned at Forschungsstelle für Energiewirtschaft (FfE) of Germany, but this study will not include invasive subsystem measurements either.

Since data characterizing individual control volumes will not be available, the Stirling CHP system model aggregates the performance of the blower, heater Stirling engine and generator subsystems into correlations describing its steady-state electrical and thermal efficiencies. The system's dynamic characteristics are accounted for by (a) coupling the steady-state model to a lumped-parameter thermal model, and (b) constraining the model's electrical output using empirical data.

MODEL TOPOLOGY

The Stirling power system is divided into three control volumes:

The **energy conversion control volume** is used to characterize the steady-state rate of energy conversion in the device.

The **thermal mass control volume** aggregates the thermal mass of the engine and heater into a single control volume.

The **cooling water control volume** aggregates the engine cooling water, and encapsulating engine cooler and exhaust gas heat exchangers into a single control volume.

The energy flows between these control volumes are depicted in Figure 2.

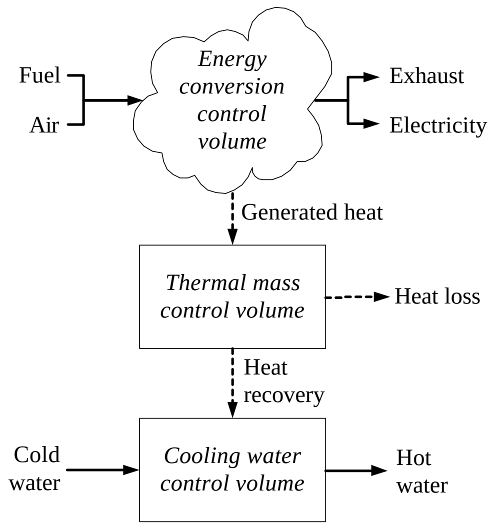

Figure 2: Control volume topology

Energy conversion control volume

The energy conversion control volume characterizes the steady-state conversion of energy from fuel to heat and electricity. The model does not attempt to solve the energy balance for the energy conversion control volume. Instead, the rates of heat generation and power production are calculated using overall electrical and thermal energy conversion efficiencies that aggregate the effects of incomplete combustion, friction and vibration, and energy use by auxiliary systems:

$$q_{gross} = \dot{m}_{fuel} \cdot LHV_{fuel} \quad (1)$$

$$P_{net,ss} = \eta_p q_{gross} \quad (2)$$

$$q_{gen} = \eta_q q_{gross} \quad (3)$$

where q_{gross} is the gross heat input into the system, $P_{net,ss}$ is the steady-state net electrical output, q_{gen} is the rate of steady-state heat generation, and η_p and η_q are the steady-state electrical and thermal conversion efficiencies of the engine, respectively. Finally, $\dot{m}_{fuel}$ is the system fuel flow rate, and LHV_{fuel} is the lower heating value of the fuel used by the system.

The system's steady-state electrical and thermal efficiencies are determined using empirical correlations relating the conversion efficiencies to the flow rate and inlet temperature of cooling water ($\dot{m}_{cw}$ and $T_{cw,i}$, respectively), and the gross heat input (q_{gross}) to the system:

$$\begin{aligned}\eta_p = a_0 &+ a_1(q_{gross}) + a_2(q_{gross})^2 \\ &+ a_3(\dot{m}_{cw}) + a_4(\dot{m}_{cw})^2 \\ &+ a_5(T_{cw,i}) + a_6(T_{cw,i})^2\end{aligned} \quad (4)$$

$$\begin{aligned}\eta_q = b_0 &+ b_1(q_{gross}) + b_2(q_{gross})^2 \\ &+ b_3(\dot{m}_{cw}) + b_4(\dot{m}_{cw})^2 \\ &+ b_5(T_{cw,i}) + b_6(T_{cw,i})^2\end{aligned} \quad (5)$$

where a_0–a_6 and b_0–b_6 empirical coefficients.

The heat exchange between the exhaust gas and cooling water in the condensing heat exchanger is not explicitly modelled. Instead, these effects are aggregated into the overall thermal efficiency coefficient (η_q). This approach clearly reduces the complexity of the model and introduces some error, but the authors deemed this simplification necessary given the lack of invasive measurements necessary to calibrate a higher-resolution model.

Thermal mass control volume

The model assumes that the thermal mass associated with the heater and engine can be lumped into a single, homogeneous control volume. The thermal energy stored within the thermal mass control volume is quantified using an aggregate thermal mass, $[MC]_{chp}$ and an average temperature T_{chp}.

The model assumes that heat transfer between the thermal mass and cooling water control volumes is proportional to the temperature difference between these control volumes:

$$q_{HX} = UA_{HX}(T_{chp} - T_{cw,o}) \quad (6)$$

where q_{HX} is the rate of heat recovery, UA_{HX} is the overall heat transfer coefficient between the control volumes, and T_{chp} and $T_{cw,o}$ are the average temperatures within the control volumes.

The rate of heat loss to the room is also assumed to be governed by Newton's Law of Cooling:

$$q_{loss} = UA_{loss}(T_{room} - T_{chp}) \quad (7)$$

where q_{loss} is the rate of heat loss,

The energy balance for the Stirling CHP system, UA_{loss} is the coefficient of heat loss, and T_{room} is the temperature in the surrounding enclosure.

Finally, the energy balance in the thermal mass control volume is:

$$\begin{aligned}[MC]_{chp}\frac{dT_{chp}}{dt} &= q_{gen} - q_{HX} - q_{loss} \\ &= q_{gen} - UA_{HX}(T_{chp} - T_{cw,o}) \\ &\quad -UA_{loss}(T_{chp} - T_{loss})\end{aligned} \quad (8)$$

where $[MC]_{chp}$ and T_{chp} are the effective heat capacitance of the Stirling and the average temperature of the thermal mass control volume, respectively.

Cooling water control volume

Heat is transferred to the cooling water via the exhaust gas heat exchanger and the Stirling engine cooler heat exchanger. The energy balance of the cooling water control volume is modelled dynamically using the following differential equation:

$$[MC]_{HX}\frac{dT_{cw,o}}{dt} = [\dot{m}C_p]_{cw}(T_{cw,i} - T_{cw,o}) - UA_{HX}(T_{cw,o} - T_{chp}) \quad (9)$$

where $[MC]_{HX}$ is the total thermal capacitance of the cooling water control volume, including the water-side of the heat exchanger and encapsulated water, and $\dot{m}_{cw}$ and $C_{p,cw}$ are the cooling water flow rate and specific heat capacity, respectively. $T_{cw,i}$ and $T_{cw,o}$ are the cooling water temperatures at the inlet and outlet of the cooling water control volume, respectively.

Combustion air flow

The air flow into the unit does not affect the model's performance predictions. However, if the unit draws its combustion air from its surrounding enclosure, the induced air flow will have a significant effect on the building infiltration and must be accounted for by the building simulation program.

The air stoichiometry is regulated to manage the heater's combustion efficiency, operating temperature and emissions. Since the single control volume used to model the Stirling CHP system does not provide adequate resolution to quantify these effects, the stoichiometry is correlated to the fuel flow:

$$\frac{\dot{N}_{air}}{\dot{N}_{fuel}} = c_0 + c_1(\dot{N}_{fuel}) + c_2(\dot{N}_{fuel})^2 \quad (10)$$

where $\dot{N}_{air}$ is the molar air flow, and c_0–c_6 are empirical coefficients.

Transient power output

Henckes and Stripf [4] observed that the response times of a Stirling CHP unit to rapid changes in system operating point are on the order of several minutes, suggesting that Stirling CHP devices cannot follow the rapidly-changing electric loads in residential buildings.

The dynamic response of the system's power delivery (P_{net}) depends on the constraints on the system's fuel flow rate, its thermal mass, and the energy use of the heater's blower and other auxiliary electrical equipment. In the absence of detailed data describing these effects, the dynamic behaviour of the system's electrical output is described using a linear derivative:

$$\frac{dP_{net}}{dt} = \frac{\left|P^{t+\Delta t}_{net,ss} - P^{t}_{net}\right|}{\Delta t} \quad (11)$$

The rate of change in the system's power output is compared to the maximum rate of change derived from empirical data $(dP_{net}/dt)_{max}$, and adjusted to ensure that the maximum rate of change is not exceeded.

Table 1: Stirling CHP unit standby, startup and cool-down mode parameters

Parameter	Standby	Startup	Cool-down
Duration	–	Δt_{start}	Δt_{cool}
Fuel flow	$\dot{m}_{fuel,off}$	$\dot{m}_{fuel,start}$	$\dot{m}_{fuel,cool}$
Power gen.	$P_{net,off}$	$P_{net,start}$	$P_{net,cool}$
Heat gen.	–	†	–

† Evaluated using Equation 3.

Standby, startup and cool-down operating modes

Equations 1–11 are used to characterize the CHP unit's behaviour during normal operation. When operating in standby, start-up and cool-down modes, the fuel use, electrical and thermal output are described using constant, average values estimated from experimental data. The duration of the start-up and cool down periods are also specified.

The parameters characterizing the Stirling CHP unit's start-up and cool-down behaviour are summarized in Table 1.

MODEL CALIBRATION

Entchev et al. [2] installed a small, Stirling CHP unit an experimental test house at the Canadian Centre for Housing Technology (CCHT), and measured its performance when subjected to thermal loads. The test house is one of two identical houses built to assess the performance of innovative residential energy technologies.

Stirling engine and balance of plant configuration

The Stirling CHP unit used in this study exhibits relatively low electrical efficiency (8-9% LHV), and a high heat-to-power ratio (≈9:1). Despite its modest 750 W_e electrical output, the system is capable of producing nearly 7 kW of heat.

The Stirling CHP unit is regulated using an on-off controller. Once operational, it exhibits nearly constant fuel flow and electrical output. During its start-up phase, the unit consumes fuel at a slightly higher rate than in its steady-state operation phase. The unit's electrical output is also lower during the startup phase.

Entchev et al. coupled the Stirling CHP unit to a water storage tank, as depicted in Figure 3. The unit's operation was regulated to meet the thermal loads on the tank using an on-off controller. The CHP unit was activated when the temperature in the storage tank dropped below 60 °C, and deactivated when the temperature of the water leaving the CHP unit rose above 80 °C. When the CHP unit was active, cooling water was supplied at a constant rate of 0.208 L/s. Once the system was deactivated, the cooling water pump remained active for another 25 minutes while the CHP unit continued to transfer heat to the cooling water.

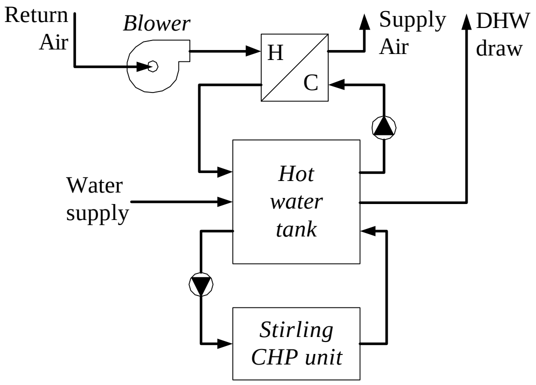

Figure 3: Arrangement of Stirling CHP unit in CCHT mechanical plant.

Uncertainty and calibration strategy

The experimental work conducted by Entchev et al. was completed in 2003, prior to commencement of the Annex 42 working phase. Thus, the experiments were not designed to support Annex 42's validation activities. Use of these data to calibrate and validate the Stirling CHP unit model introduces uncertainties—some of these uncertainties might have been mitigated if the testing plan could have been extended to accommodate Annex 42's needs, while others arise from the real-world limitations of the instrumentation and data acquisition equipment. Nevertheless, the CCHT data set provide a rich description of Stirling engine performance and have been used for the calibration of the Stirling CHP model.

Significant sources of uncertainty associated with using the data collected during the CCHT tests to calibrate the Stirling CHP unit model include:

Air flow and casing temperature measurements: The experiments conducted at CCHT did not include supply air or exhaust flow rates or casing temperature, which were of limited importance to the CCHT study. Without these data, calibration of the model's heat loss and air flow correlations is not possible.

Differing time resolutions: Measurements of electrical output were taken at 5 and 15 minute intervals, while the thermal data were collected at 1 minute intervals. As a result, it is impossible to compare the electrical output to the instantaneous cooling water temperature.

Steady state measurements: The CCHT test facility is designed to replicate real-world conditions inside a residence, and cannot impose the "artificial" steady-state conditions. All of the experiments conducted were dynamic tests in which the temperature of the cooling water varied continuously according to conditions in the water tank. Since none of the measurements describe the system under steady-state conditions, it is difficult to determine the model's steady-state and dynamic parameters from a single, dynamic response.

The Stirling CHP unit tested by Entchev et al. was designed to operate at a constant fuel flow rate once its startup period is completed. Therefore, there is no need to characterize the sensitivity of the unit's electrical and thermal efficiencies to the fuel flow rate, and coefficients a_1, a_2, b_1 and b_2 in Equations 4 and 5 were assumed to be zero.

Entchev et al. configured the cooling water pump to provide a constant flow rate. Therefore the CCHT data does not explore the unit's sensitivity to the cooling water flow rate, and the coefficients a_3, a_4, b_3 and b_4 in Equations 4 and 5 were assumed to be zero. Thus the model's predictions, when calibrated using the CCHT data, are only valid if the same cooling water flow rate (0.208 L/s) is specified.

It's possible the fluctuations observed in the electrical data can be attributed to the effects of variations in the cooling water temperature on the unit's electrical efficiency. However, the 15-minute averaged electrical data collected in the CCHT project does not provide sufficient resolution to discern these effects. Since the observed fluctuations in the electrical output are small, it was assumed that i) variation in the unit's electrical output was not significant and ii) its electrical efficiency is not sensitive to the cooling water temperature (ie. a_5 and a_6 are zero).

The unit's steady-state cogeneration efficiency (η_q) is also sensitive to the cooling water temperature. Since the CCHT data does not describe the unit's steady-state response, the coefficients b_5 and b_6 that characterize the sensitivity of the steady-state cogeneration efficiency can only be calibrated using the dynamic thermal data. This task would be challenging, as the dynamic thermal model parameters also significantly effect the model's dynamic predictions, and must be estimated simultaneously using the same dataset.

To simplify calibration of the dynamic thermal model, coefficients b_5 and b_6 were assumed to be zero. This simplification likely introduces some error into the model predictions—especially when the cooling water temperature is low enough to cause condensation in the exhaust gases.

These simplifications effectively reduce the steady-state electrical and thermal efficiencies to constant coefficients ($\eta_p = a_0$ and $\eta_q = b_0$), and regrettably introduce some error into the model estimates. Future tests of Stirling CHP units designed with Annex 42 goals in mind will hopefully provide data required to estimate these parameters.

Table 2: Summary of CCHT Stirling CHP calibration data subsets

Set	Samples	Cycles	Duration (hr.)
Calibration	4033	19	67.2
Test	3802	20	63.4

Table 3: CCHT Stirling CHP unit operational characteristics.

	Time (min)	Fuel (kg/s)	Power (W)
Standby	–	0.	-20.
Startup	11.5	1.70E-04	-17.
Normal	–	1.60E-04	744.
Cool-down	25.	0.	-55.

Strategy

The CCHT data set was disaggregated into two subsets — the first set was used to calibrate the model, while the second set was used to test the model's accuracy. These subsets are summarized in Table 2.

The Stirling CHP unit model was calibrated using the CCHT data in two steps:

1. The steady state electrical efficiency, and the constant parameters used to describe the standby, start-up and cool-down modes were estimated by averaging data over extended intervals.

2. The steady-state thermal efficiency and the dynamic thermal model parameters were estimated by calibrating the model using the estimated steady state model parameters, and then comparing model estimates with the measured data.

Steady state calibration

To derive steady-state parameters for the system fuel flow, electrical efficiency and electrical output in its various modes of operation, the experimental data points were classified into groups corresponding to their corresponding operating mode (standby, startup, normal operation or cool-down).

The observed fuel flow and electrical generation data were then averaged for each of the operating modes. The resulting averaged data provide an approximate description of the CHP unit's fuel flow and electrical output during the four operating modes in each of the observed cycles. In addition, the duration of each operating mode was calculated, and these data were averaged to obtain estimates for the duration of the startup and cool-down periods. The estimated steady-state parameters used to calibrate the Stirling CHP model are presented in Table 3.

Table 4: Dynamic thermal model parameters for CCHT Stirling CHP unit

Parameter	Value	
η_q	0.8775	–
$[MC]_{HX}$	24500.	J/°C
$[MC]_{chp}$	29600.	J/°C
$[UA]_{HX}$	67.0	W/°C
$[UA]_{loss}$	3.7	W/°C

Dynamic parameter identification

To estimate the remaining parameters required by the Stirling CHP unit model, an iterative approach was used:

1. A set of input parameters was chosen.

2. A simulation was performed, during which the model was subjected to the same boundary conditions as the unit studied in the CCHT tests.

3. The model's predicted outlet temperature was compared to the measured CCHT data over the duration of the calibration data set.

4. The model inputs were adjusted, and steps 2–3 were repeated until the best-possible agreement between model outputs and empirical data was achieved.

To assist in the parameter identification procedure, the an optimization utility developed by Wetter [11] was used to identify the parameter set providing the best fit between model predictions and experimental data.

The optimized parameters the model are presented in Table 4. While these parameters provide the best agreement between the model predictions and the measurements in the CCHT calibration data, care must be taken in their use and interpretation.

The parameter optimization procedure provided estimates for five model parameters. However, a single criteria (the average absolute error between measured and predicted outlet temperatures) was used to evaluate the suitability of the parameter set. Thus, there may exist multiple sets of parameter inputs that provide similar results, and the set chosen by the optimization criteria may not be the best representation of the actual Stirling CHP unit used in the CCHT tests.

Furthermore, the optimization procedure assumes that the Stirling CHP unit model (and its ESP-r implementation) is a concise and accurate representation of the test system. If significant disparities exist between the mechanics of the test system and the model's behaviour, the parameter optimization procedure will select the model inputs that reduce the effects of these disparities. While these values give the best agreement

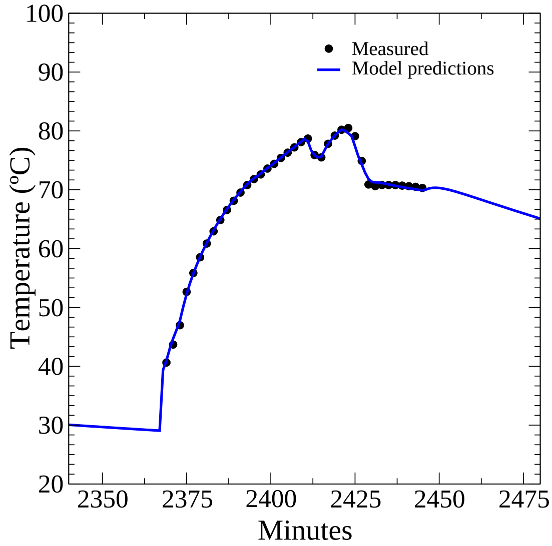

Figure 4: Comparison of measured outlet temperature and predicted values for CCHT calibration data: 2350–2475 minutes

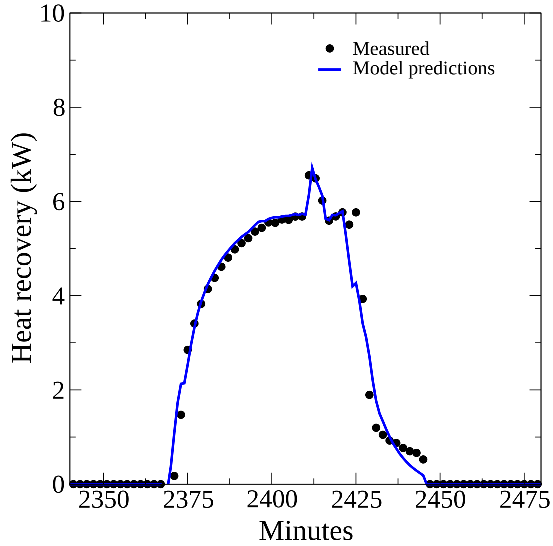

Figure 5: Comparison of measured rate of heat recovery and predicted values for CCHT calibration data: 2350–2475 minutes

with experimental results, they do not necessarily represent the physical attributes of the test system.

Figure 4 plots the experimental data long with the model predictions for a two hour period, and Figure 5 plots the predicted and observed rates of heat recovery for the same period. The experimental data was only collected during periods when cooling water was flowing through the device—thus the experimental data points correspond to periods of operation and there are gaps during periods in which the unit was inoperative. Between cycles, the thermal mass model accounts for heat loss to the surroundings, and therefore exhibits a decrease in the temperature at the heat exchanger.

COMPARISON WITH TEST DATA

The accuracy of the calibration was quantified using the remaining CCHT data. The Stirling CHP model was subjected to the cooling water temperature and flow rate, containment temperature and control signal boundary conditions observed in the test data set, and the model predictions were then compared with the experimental measurements.

Table 5 summarizes the difference between the model predictions and the experimental measurements. The model predictions show good agreement with the measured data when integrated over the test period — the heat recovery, power output and fuel use predictions differ by -2.2%, 2.8% and 1.1% respectively.

Figure 6 plots the difference between the predicted and measured rates of heat recovery over 58 hours. The thermal mass model estimates clearly continuously diverge from the experimental data.

Table 5: Comparison of Stirling CHP model predictions with CCHT test data.

Outlet temperature, absolute error	Average	°C	0.267
	Maximum	°C	4.2
	r^2	–	0.990
Heat recovery, absolute error	Average	W	233.
	Maximum	W	3630.
	r^2	–	0.958
Heat recovery, cumulative	Total heat	MJ	698.
	% error	—	-2.15
Power output, cumulative	Net output	MJ	75.0
	% error	—	2.82
Fuel use, cumulative	Fuel use	kg	18.6
	% error	—	1.11

CONCLUSIONS AND DISCUSSION

A Stirling CHP model has been developed for use in whole-building simulation programs. Based on empirical performance correlations, the model is capable of characterizing the part-load performance and transient behaviour of a Stirling CHP unit in response to conditions in the building and mechanical plant.

The model's predictions were compared with data from experimental tests conducted by Entchev et al. [2] at the Canadian Centre for Housing Technology (CCHT). Though the CCHT data is not ideally suited to validate the Stirling CHP model, the model was successfully calibrated using an parameter fitting approach.

While the estimated dynamic thermal model input parameters provide good agreement with experimental data, they must be used with caution. They were ob-

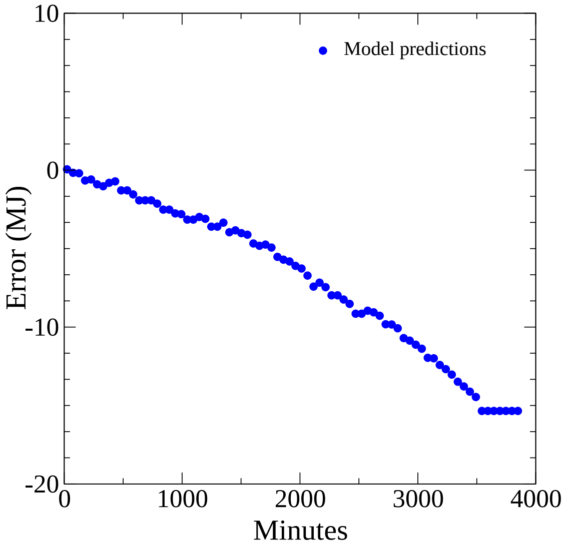

Figure 6: Cumulative difference between CCHT test data heat recovery measurements and model predictions (data-points represent every 50th measurement)

tained using an optimization tool with the agreement between model predictions and measured data as the sole fitness criteria. Thus, these solution sets may not be representative of the actual characteristics of this particular device.

Moreover, while these results show that the Stirling CHP model is capable of providing accurate predictions of the behaviour of Stirling CHP units, they are not an absolute qualification of the model's validity. Since the parameter optimization process is not constrained by the unit's physical attributes, it may adjust the parameter inputs to account for i) differences between the model's mechanics and the physical system, and ii) errors in the model implementation, phenomena which would otherwise produce disagreement between model predictions and the measured data.

Future work will explore merging the Annex 42 Stirling CHP model specification with a similar specification under development for internal combustion engine based CHP devices. Ongoing testing activities at FfE are expected to provide Annex 42 with additional data permitting more rigorous validation of these models under a broader range of conditions. These efforts will be complemented by comparative testing of different simulation platforms once the model has been implemented by other Annex 42 participants.

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REFERENCES

[1] Clarke, J. 2001 , Energy simulation in building design, second edn, Butterworth Heinemann.

[2] Entchev, E., Gusdorf, J., Swinton, M., Bell, M., Szadkowski, F., Kalbfleisch, W. and Marchand, C. 2004 , 'Micro-generation technology assessment for housing technology', Energy and Buildings **36**, 925–931.

[3] Hawkes, A. and Leach, M. 2005 , 'Impacts of temporal precision in optimization modelling of micro-combined heat and power', Energy **30**, 1759–1779.

[4] Henckes, L. and Stripf, M. 2003. , Laboratory test of the SOLO stirling CHP module, Technical report, European Institute for Energy Research.

[5] Kelly, N. 2004. , Specifications for models of cogeneration equipment: discussion document, IEA/ECBCS Annex 42 Subtask B working document, University of Strathclyde.

[6] Kelly, N. 2005. , Specifications for an internal combustion engine component model, IEA/ECBCS Annex 42 Subtask B working document, University of Strathclyde. Version 4.0.

[7] Peacock, A. and Newborough, M. 2005 , 'Impact of micro-chp systems on domestic sector co_2 emissions', Applied Thermal Engineering **25**, 2653–2676.

[8] Pearce, J., Al Zahawi, B., Aukland, D. and Starr, F. 1996 , 'Energy generation in the home: evaluation of single-house domestic combined heat and power', IEE Proceedings of Science, Measurement and Technology **143**(6), 345–350.

[9] Pearce, J., Al Zahawi, B. and Shuttleworth, R. 2001 , 'Energy generation in the home: modelling of single-house domestic combined heat and power', IEE Proceedings of Science, Measurement and Technology **148**(5), 197–203.

[10] Urieli, I. and Berchowitz, D. 1984 , Stirling cycle analysis, Adam Hilger Ltd. Bristol.

[11] Wetter, M. 2004. , GenOpt(R) Generic Optimization Program, Lawerance Berkeley National Laboratory, US Deparment of Energy.