IMPLEMENTATION AND VALIDATION OF COMBUSTION ENGINE MICRO-COGENERATION ROUTINE FOR THE SIMULATION PROGRAM IDA-ICE

Kari Alanne[1], Niklas Söderholm[1], Kai Sirén[1]
[1]Helsinki University of Technology, Finland
Corresponding email: kari.alanne@tkk.fi

ABSTRACT

Combined generation of thermal and electrical energy in units with electrical power less than 10 kW provides an attractive option for the energy supply of residential buildings due to their potential to high overall efficiency and thus capability of reducing emissions. Dynamic simulations of such systems are required to performance assessments that aim at finding the most energy efficient system topologies. This paper presents the implementation of a combustion engine-based micro-cogeneration routine into IDA-ICE, which is a widely used building simulation program in the Nordic Countries. The routine utilizes specifications defined in the IEA Annex 42 and the implementation is validated by way of inter-program testing.

INTRODUCTION

The concept micro-cogeneration commonly refers to systems that simultaneously generate both electricity and utilizable heat the electrical power being less than 10 kW_e. When applied to satisfy the power demands of lighting, appliances and building services plus the thermal demands of space heating and production of hot water in residential buildings, the overall plant efficiency of up to 85 % may be reached. The most suitable micro-cogeneration technologies in the above mentioned power range are fuel cells and Stirling engines, but there are also successful experiences of the application of internal combustion engines (ICE) with 1 kW_e electrical power. The use of combustion engines in micro-cogeneration is supported by the fact that the technology is already on the market or is mature enough to be introduced in terms of mass-production of residential micro-cogeneration plants in less than five years. Moreover, combustion engines and particularly Stirling engines are capable of using various fuels (e.g. biomass) without mentionable high-cost pre-handling processes for fuel.

The efficient application of micro-cogeneration presumes that both the supply and demand of thermal and electrical power are in balance, which provides with a great challenge for design, operation and analysis of these systems. One of the major research issues at the moment is finding out cost and energy efficient system topologies through computational performance assessments. Because the interaction between a micro-cogeneration system and a building encompasses dynamic processes, the whole system must be examined in terms of building simulation, which allows the use of a short time step (e.g. Hawkes&Leach, 2005 and Onovwiona et al., 2006).

In the Annex 42 of International Energy Agency (IEA), simulation models for the above purpose were developed (Beausoleil-Morrison, 2008). The objective was to find “grey box” approaches that i) account for thermal, chemical and electrical phenomena necessary to properly depict processes in the sense of building simulation and ii) support the integration of micro-cogeneration technologies into larger context in building services. Routines for both fuel cell and combustion engine based micro-cogeneration were developed and so far they have been implemented into the simulation programs ESP-r, TRNSYS and EnergyPlus. These implementations have been validated both by several empirical runs and inter-program tests (Beausoleil-Morrison, 2008).

IDA-ICE (Indoor Climate and energy) is an advanced, widely used building simulation tool in the Nordic Countries both in commercial and research use, providing dynamic simulation for heat transfer, internal and solar gains and airflows in buildings. The program was developed in the Royal Institute of Technology and the Swedish Institute of Applied Mathematics. The details in complete have been reported by Björsell et al. (1999) and Sahlin (1996). Test studies have been accomplished by Achermann et al. (2003) and Travesi et al. (2001). The only implementation of Annex 42 models into IDA-ICE up to date is a routine for solid-oxide fuel cell (SOFC) (Vesanen et al., 2007).

This paper presents the implementation of the Annex 42 combustion engine routine into IDA-ICE. The model is validated by way of inter-program comparison with ESP-r, TRNSYS and EnergyPlus, following the principles presented by Beausoleil-Morrison & Ferguson (2007). Furthermore, the model extension to exhaust gas heat recovery is discussed.

SIMULATION

Modeling premises

The main component of a combustion engine-based micro-cogeneration power plant is a reciprocating engine. In the present study, the engine may be either a traditional internal combustion engine (ICE), where the combustion takes place in a cylinder, or a Stirling engine (SE), which incorporates a closed thermo-dynamic cycle and an external thermal source. Hence, almost any heat source may be utilized, provided that a sufficient temperature difference between hot and cold ends is achieved (e.g. Kongtragool & Wongwises, 2003). Moreover, a plant consists of a generator linked to the engine, and a gas-to-water heat exchanger and ancillaries such as water circulation pumps and control systems.

The engine model developed in the IEA Annex 42 has its roots in the work of McCrorie et al. (1996), Kelly (1998) and Pearce et al. (2001). The model was formulated keeping in mind the trade-off between simplicity and resolution. Here, the objective is to depict the plant operation in terms of limited number of control volumes, still capturing extensively the factors that affect the interaction between the building and the plant. For example, the sensitivity of plant efficiencies to cooling water temperature and standby, start-up and cool-down characteristics is taken into account. Particular effort has been made to adapt the models to dynamic time steps from a few seconds to a couple of minutes. Instead, exact analysis of physical and chemical phenomena such as combustion process and thermo-dynamic cycles has been omitted and substituted by engine specific, empirical correlations that rely on measured data. Those correlations are converted to parametric equations, which are linked to a common structure applicable to all combustion-based micro-cogeneration engines. (Beausoleil-Morrison et al., 2007a)

Model topology

The device model consists of three main control volumes, namely i) the energy conversion control volume that expresses the conversion of fuel to electricity and heat in the engine under steady-state conditions, ii) the thermal mass control volume that represents the thermal mass or the thermal capacitance of the engine as a whole, and iii) the cooling water control volume that characterizes the heat exchange between the exhaust gas and the cooling water and all engine parts in immediate thermal contact. The control volumes are shown in Figure 1.

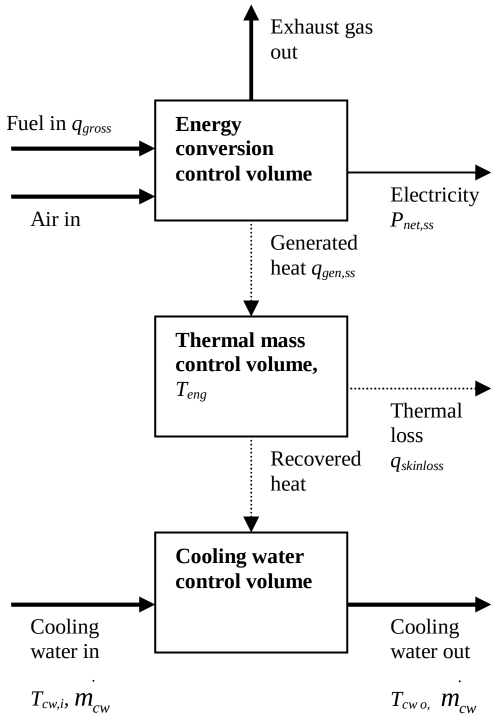

Figure 1 The control volumes associated with Annex 42 model

The incomplete steady-state combustion is given as:

$$q_{gross} = \dot{m}_{fuel}\, LHV_{fuel} \tag{1}$$

$$P_{net,ss} = \eta_e q_{gross} \tag{2}$$

$$q_{gen,ss} = \eta_q q_{gross} \tag{3}$$

where q_{gross} is the gross thermal input to the system, $\dot{m}_{fuel}$ is fuel flow, LHV_{fuel} is the lower heating value of the fuel, η_q and η_e are the net thermal and electrical efficiencies, respectively, $P_{net,ss}$ is the steady-state electrical output and $q_{gen,ss}$ is the steady-state heat generation. The model applies to any fuel when an empirically measured heating value is available.

The original Annex 42 model presents 27-term tri-variate polynomials to express the efficiencies η_q and η_e (Beausoleil-Morrison et al., 2007a). Many of these reduce to zero during calibration, anyway (Beausoleil-Morrison et al., 2007b). The first seven terms for both efficiencies are given in Eqs (4) and (5).

$$\begin{aligned}\eta_e = a_0 + a_1 P_{net,ss}^{\;2} + a_2 P_{net,ss} \\ + a_3 \dot{m}_{cw}^{\;2} + a_4 \dot{m}_{cw} \\ + a_5 T_{cw,i}^{\;2} + a_6 T_{cw,i}\end{aligned} \tag{4}$$

$$\eta_q = b_0 + b_1 P_{net,ss}{}^2 + b_2 P_{net,ss} + b_3 \dot{m}_{cw}{}^2 + b_4 \dot{m}_{cw} + b_5 T_{cw,i}{}^2 + b_6 T_{cw,i} \quad (5)$$

where $a_0 \ldots a_6$ and $b_0 \ldots b_6$ are empirical constants, $\dot{m}_{cw}$ and $T_{cw,i}$, are the mass flow and temperature of cooling water, respectively.

Heat transfer in the engine

As seen in Figure 1, the model basically contains two major heat transfer processes, namely i) heat transfer from exhaust gases to the cooling water flow and ii) from the engine to the ambient. Here, the engine is considered one thermal capacity MC_{eng} with an average temperature T_{eng} and the encapsulated cooling water another with thermal capacity MC_{cw}, inlet temperature $T_{cw,i}$ and bulk temperature of the cooling water control volume (= outlet temperature) $T_{cw,o}$.

The governing energy balance for the engine control volume can be written as

$$MC_{eng} \frac{dT_{eng}}{dt} = UA_{HX} \left(T_{cw,o} - T_{eng} \right) + UA_{loss} \left(T_{amb} - T_{eng} \right) + q_{gen,ss} \quad (6)$$

where UA_{HX} and UA_{loss} are engine specific, empirically determined heat transfer coefficients. The first term on the right side of Eq.(6) depicts the heat transfer between the engine and the cooling water and the second term equals to heat flow from the engine to the ambient (in temperature T_{amb}), i.e. skin losses ($q_{skinloss}$).

Correspondingly, the energy balance for the cooling water control volume is

$$MC_{cw} \frac{dT_{cw,o}}{dt} = \dot{m}_{cw} c_{p,cw} \left(T_{cw,i} - T_{cw,o} \right) + UA_{HX} \left(T_{eng} - T_{cw,o} \right) \quad (7)$$

where the first term on the right side indicates the heat flow recovered to the water and the second term equals the heat transfer between the engine and the cooling water.

Operational modes

The engine model encompasses three modes of operation: i) steady-state function, ii) engine warm-up and iii) shutdown. Additionally a cool-down period of a user-specified length is initiated after the shut down order. During warm-up, the engine generates a limited amount of electricity and thermal power. Under cool-down and standby operation, the device does not generate heat or electricity, but consumes some electricity to maintain activation functions or complete the shutdown.

Internal combustion engines respond to activation order promptly, whereas Stirling engines require a warm-up period where the power output is sensitive to operational temperatures. In the Annex 42 model, the power output during this period is approximated as

$$P_{net,warm-up} = P_{\max} k_p \left(\frac{T_{eng} - T_{amb}}{T_{eng,nom} - T_{amb}} \right) \quad (8)$$

where P_{max} is the steady-state maximum power for the engine, k_p is an empirical coefficient and $T_{eng,nom}$ is nominal operating temperature. A peak occurs then in the fuel flow and is calculated in the model from

$$\dot{m}_{fuel} = m_{fuel,ss} + k_f m_{fuel,ss} \frac{T_{eng,nom} - T_{amb}}{T_{eng} - T_{amb}} \quad (9)$$

where $\dot{m}_{fuel,ss}$ is the fuel flow in steady-state conditions, k_f is empirical parameter.

Model integration

The current implementation (IDA-ICE) presumes that the plant is connected to a hydronic heating system via a buffer storage that allows the parallel application of more than one thermal source. The engine is situated in a separate zone in the building model, which provides a proper data on the skin-losses that can be utilized in heating up neigh-bouring rooms and preheating the combustion air. The system's control ensures that when thermal surplus occurs, the device is deactivated. In the case of thermal shortage, an auxiliary burner is employed to ensure sufficient heat generation. Three operational strategies are introduced. The first mode follows electrical load primarily, but also thermal load to some extent. The others either follow the electrical or thermal load. Threshold values for temperatures and electrical/thermal power are given as set points to determine whether the device is activated or deactivated. The integration of the engine model into the heat distribution system and the engine's operational strategies are illustrated in Figures 1 and 2.

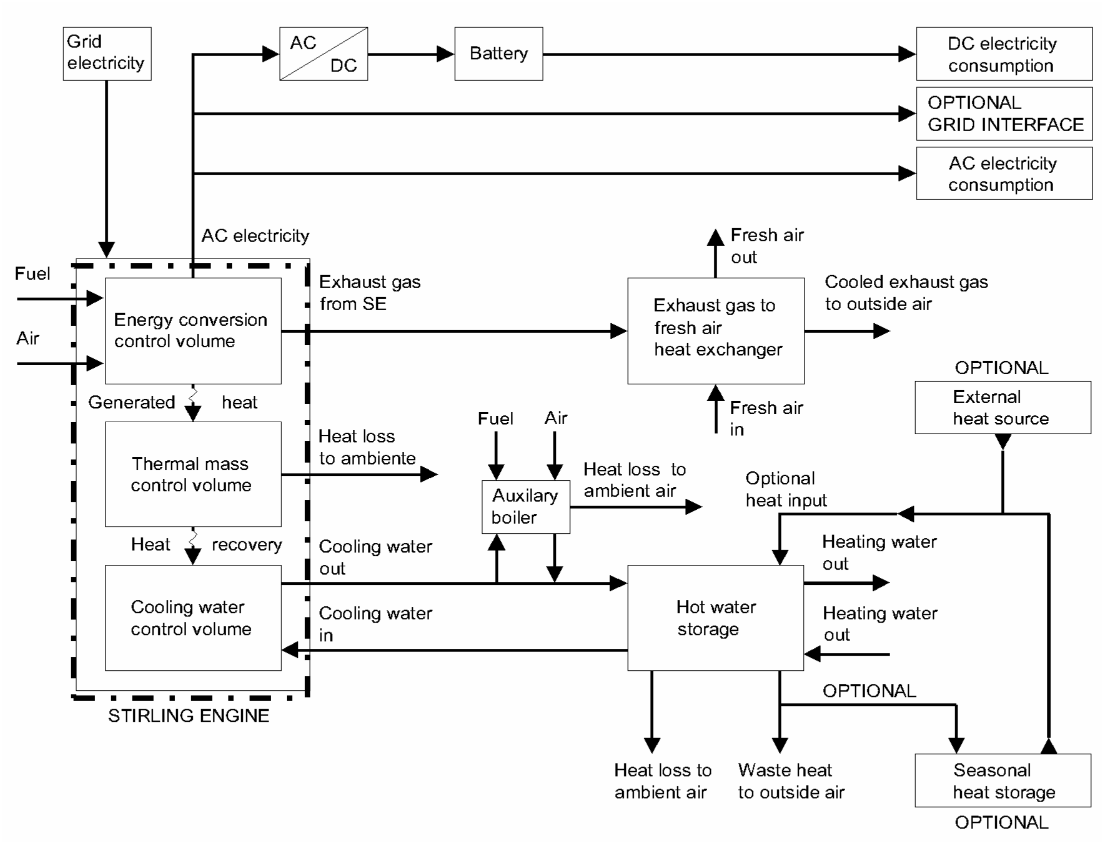

Figure 2 The integration of the engine model

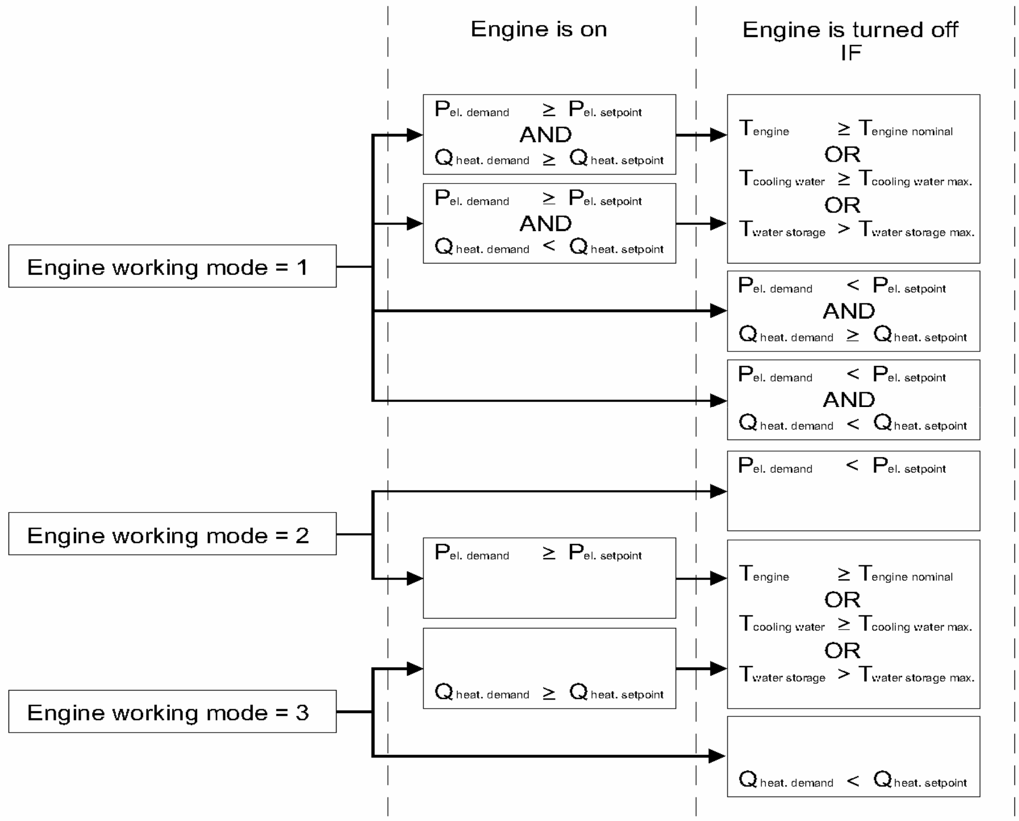

Figure 3 Operational strategies in the present implementation

A couple of new schemes for the utilization of waste heat are introduced in terms of the present implementation of Annex 42 combustion engine model. Here, warm exhaust gases can be used either to pre-heat i) the supply air or ii) combustion air. A user can also select whether he/she wants to recover the heat of exhaust gases before or after the water cooling loop.

The enthalpy flow of exhaust outlet is first determined from the steady state energy balance for the conversion control volume and is defined as

$$\dot{H}_{exh} = \left(1-\eta_q\right) q_{gross} - P_{net,ss} \qquad (10)$$

On the basis of Shomate equation, the enthalpy flow can be also stated as

$$\dot{H}_{exh} = \frac{AT_{exh}}{1000} + \frac{B}{2}\left(\frac{T_{exh}}{1000}\right)^2 + \frac{C}{3}\left(\frac{T_{exh}}{1000}\right)^3 + \frac{D}{4}\left(\frac{T_{exh}}{1000}\right)^4 + E\frac{1000}{T_{exh}} \qquad (11)$$

where $A,B,\ldots,E$ are derived from experimental coefficients for each constituent of the exhaust gas. Hence, they must be calculated separately for each fuel. The temperature T_{exh} can be solved from Eq. (11) and applied as an input parameter for the heat recovery model of IDA-ICE, which is illustrated as a screenshot in Figure 3.

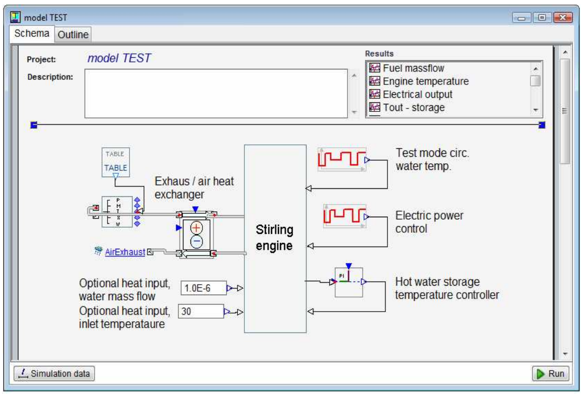

Figure 4. The simulation environment in IDA-ICE.

The amount of recovered heat strongly depends on the heat exchanger's efficiency and whether condensation takes place during heat transfer. In cold climates, the exhaust gas may cool down under the freezing point and cause significant problems with condensed vapour. Table 1 presents boundary temperatures for exhaust gas, when condensation occurs in various air temperatures. The example calculation contains a natural gas-fuelled, 0.7 kW_e WhisperGen Stirling Engine, plate heat exchanger with effectiveness ε = 0.8, air and exhaust gas flows of 0.078 kg/s and 0.003 kg/s, respectively. The exhaust gas constituents are 15.2% CO_2, 12.3% H_2O and 72.5% N_2, whereas the total specific heat capacity of the exhaust is 1.13 kJ $kg^{-1}K^{-1}$.

*Table 1 Dew point temperatures for exhaust gas**

Outdoor air temperature [°C]	**Exhaust gas temperature [°C]**
-26	113
-20	111
-15	107
-10	102
-5	99
0	95
5	91

* Plate surface temperature 49.9 °C

Validation

Simplifications, inaccuracies in the solutions of mathematical models and errors in programming cause differences between simulation results and actual physical processes or phenomena (Tuomaala, 2002). Therefore, every model and its implementation must be validated before computational studies. Basically three methods are available: i) empirical validation, ii) analytical validation and iii) comparative testing.

The most comprehensive validation procedure requires empirical studies, where monitored data are compared to those obtained from simulation. Experimental data has been acquired in terms of Annex 42 for at least a 700 W_e SE unit by the Canadian Centre for Housing Technology and for a 5 kW_e ICE in the "Forschungstelle für Energiewirtschaft" in Munich. The final results of these experiments were used to model calibration and a good agreement was achieved (Beausoleil-Morrison et al., 2007b). Calibration data acquired from the University of Leuven, Belgium (Mertens I. & S., 2004–2005) imply that the correlation between the primary energy demand and the electric / thermal power produced by the engine is almost linear. This correlation is especially valid when operating the engine under part-load.

Analytical validation applies best if the simulation model describes a problem governed a single mathematical expression for which there is a solution available. This is not the case in Annex 42 cogeneration model, where all the combustion phenomena related to a selected engine technology are submitted under one parametric expression. Hence, the analytical validation has been omitted.

Comparative testing refers to a procedure, where a model is implemented into various simulation programs independently and data collected from test runs are compared. Here, inputs and boundary conditions must be identical. The method is primarily used to localizing errors in programming, but it may have indirect value from the viewpoint of model accuracy, if the other implementations have been empirically tested. In Annex 42, comparative tests for combustion engine model were carried out in EnergyPlus, ESP-r and TRNSYS.

RESULTS AND DISCUSSION

Annex 42 testing program encompasses nine test series including total 44 separate cases that are discussed in detail by Beausoleil-Morrison et al. (2007c). The procedure introduces a base case which defines the initial model parameters and boundary conditions plus the system configuration comprising either external or internal cooling water pump. Table 2 summarizes selected model parameters.

Table 2 Selected initial model parameters

Parameter	**Value**	**Unit**
Fuel	100 % CH_4	-
P_{max}	1000	W
MC_{eng}	20000	JK^{-1}
MC_{cw}	20000	JK^{-1}
UA_{HX}	50	WK^{-1}
UA_{loss}	0	WK^{-1}
a_0	0.25[a]	-
b_0	0.5[a]	-
k_f	1.0[b]	-
k_p	1.0[b]	-
$T_{eng,nom}$	150[b]	°C

[a] coefficients $a_1 \ldots a_6$ and $b_1 \ldots b_6$ equal to zero
[b] only applied to Stirling engine

The validation results below encompass the test series 400, 500 and 700 in terms of six separate graphs. These three test series out of totally nine were chosen because they characterize well the engine's transient performance. In the above test series, the following transient changes are evaluated:

1. 400-series: exercise the treatment of warm-up, cool-down and standby operation. The test is valid for an internal combustion engine with an assigned mandatory cool-down period.
2. 500-series: fuel and energy flows in warm-up, cool-down and standby operation specific for a Stirling engine.
3. 700-series: the model's behaviour in case of overheating (cooling water temperature too high or cooling loop interrupted).

The 400 series test results are demonstrated in figures 4 & 5. Validation test 407 exemplifies the operation characteristics for an internal combustion engine, which typically lack a temperature dependant warm-up period contrary to a Stirling engine. The warm-up period was specified to last 900 s and the cool-down period respectively 1500 s. The engine is obliged to finish the mandatory cool-down period before it can be turned on. During the cool-down period the ancillaries within the engine are set to consume a power of 150 W and correspondingly 50 W in standby mode. The cooling water is circulated through the engine at a constant mass flow rate of 0.15 kg/s.

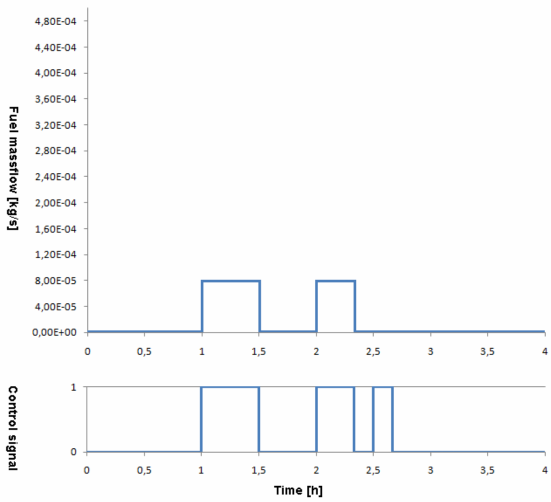

Figure 5. Test 407, ICE fuel mass flow.

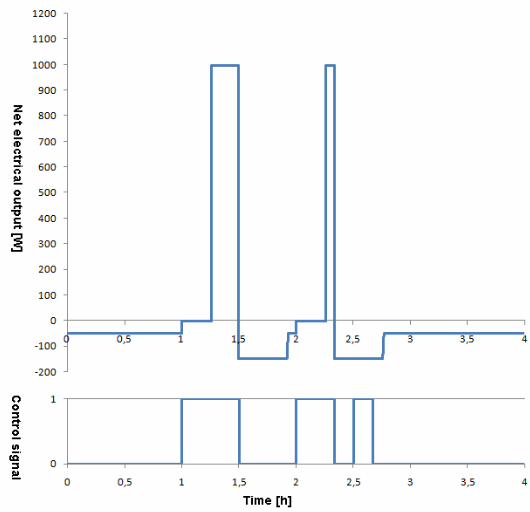

Figure 6. Test 407, ICE net electric output.

Consistent with the mandatory cool-down mode the model correctly ignores the control signal received at 02:30 h. The electricity output starts as it should when the warm-up period has been completed. Simultaneously the engine switches into steady-state fuel supply mode. The results in figure 4 & 5 agree properly with the reference case – no time dependant changes occur in the output variables during the warm-up period.

Test series 500 or more explicitly the test 501 shown in figures 6 & 7 represents the model's response to a simple step change in the input parameter (electric power demand). The test applicable to a Stirling engine should correctly predict the duration of the warm-up period according to Eq.(6) & (8). Steady-state operation should be initiated when either temperature $T_{eng,nom}$=*70*°C or $P_{net,warm-up}$= P_{max}. The cooling mass flow is kept constant at 0.15 kg/s.

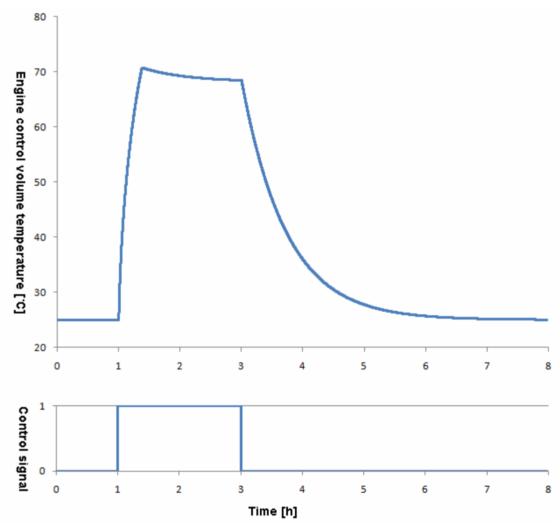

Figure 7. Test 501, SE control volume temperature.

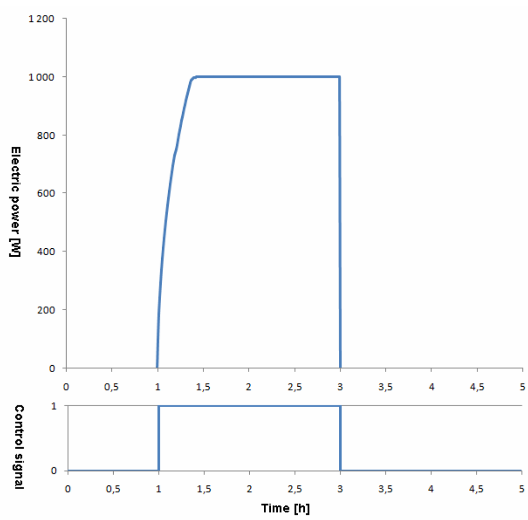

Figure 8. Test 501, SE electrical power produced.

The warm-up characteristics and the subsequent steady-state mode are accurately estimated by the model. As seen from figure 6 the transition into steady-state operation takes place at 01:23 h after which the engine's temperature slowly declines due to the constant fuel and cooling water flow. Electrical power is produced immediately after the engine has been switched on.

The third validation test, namely test 701 demonstrates the overheating protection employed by the model's low level controller if the cooling water outlet temperature exceeds the set point temperature = *70*°C. The cooling water mass flow is 0.20 kg/s throughout the test, whereas the inlet temperature is varied between 65°C and 75°C.

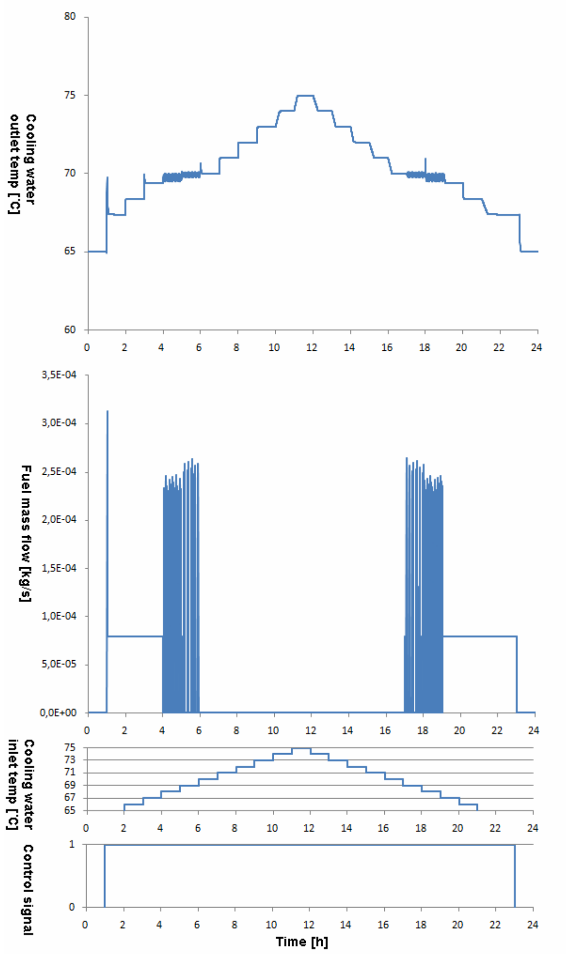

Figure 9. Test 701, overheating protection exercised by a combustion based engine.

Discontinuities in the operation mode occur between 04:00 h and 06:00 h when the controller tries to maintain the outlet temperature below *70*°C by switching between warm-up and steady-state mode. The operation characteristics are in harmony with the reference case. Uncertainties are still due to the inaccurate reference representation, which lack precision within the discontinuous operation range.

CONCLUSION

The implementation of a combustion engine-based micro-cogeneration model developed in the IEA Annex 42 was presented. The routine was installed into IDA-ICE building simulation and validated by way of inter-program comparison with EnergyPlus, ESP-r and TRNSYS. The present implementation obtained an excellent agreement with the above tools. A good accuracy is achieved thanks to the variable step size calculation used in IDA-ICE. The future research covers computational performance assessments in the sense of primary energy demand and CO_2 emissions, with the aim at finding optimally integrated and operated cost-effective solutions for residential micro-cogeneration. Both single buildings and local grid systems will be investigated and large-scale effects at national level are accounted for.

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