Experimental and modeling investigation of an ICE based
micro-cogeneration device considering overheat protection controls

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Abstract

A dynamic simulation model for the micro-cogeneration unit based on internal combustion engine (ICE) is built as a new component on TRNSYS. This model considers dynamic processes including start-up, cool-down, and overheat protection controls. Through an experiment study, the steady and dynamic performances of the unit are presented and analyzed for calibrating the parameters of the dynamic model. Then the model is validated by the experiment data during steady and dynamic processes. The improvement of prediction accuracy of the model by considering dynamic processes is analyzed. The validation results show that the dynamic model can well predict the characteristics of the unit during start-up, cool-down, and overheat protection control processes. The cumulative prediction errors of the dynamic model decrease with the increase of the start-stop interval. By considering the dynamic processes, 5.18% and 3.2% of Q_{th} and η_{total} cumulative prediction errors are decreased respectively when the start-stop interval is 0.5h. The primary energy saving ratio and CO₂ emission reduction ratio increases with the increase of the start-stop interval. The start-stop interval for the unit in this paper should be longer than 0.5 h, or the energy consumption of micro-cogeneration unit would be larger than that of conventional system.

Keywords: micro-cogeneration; dynamic simulation model; internal combustion engine; Overhear protection control;

Nomenclature

CCHP	Combined cooling, heating and power
CO ₂ E	CO ₂ emission
CO ₂ ERR	CO ₂ emission reduction ratio
CS	Conventional system
GA	Genetic algorithm
HETS	Hybrid electric-thermal load strategy
ICE	Internal combustion engine
LHV	Low heat value
LP	Linear programming
LPG	Liquefied petroleum
MCHP	Micro combined heating and power
MD	Operation strategy based on minimum distance
MILP	Mixed-integer linear programming
MINLP	Mixed-integer nonlinear programming
NG	Natural gas
PESR	Primary energy saving ratio
PSO	Particle swarm optimization
PtoT	Power to thermal ratio

Symbols

χ	Molar fraction of constituent (%)
a	Coefficient of electric efficiency
b	Coefficient of heat generation efficiency of jacket water
C_p	Specific heat at constant pressure (kJ/kg K)
E	Electricity (kW)
e	Excessive
$Error$	Error (%)
H	Enthalpy (kW)
m	Mass flow rate (kg/s)
MC	Thermal mass (J/K)
Q	Thermal energy (kW)
P	Pump
T	Temperature (°C)
UA	Overall thermal conductance (W/K)
V	Volumetric flow rate (m ³ /s)
α	Power related to tube bank arrangement
β	Power related to thermo-physical properties
ε	Control Coefficient
η	Efficiency (%)
θ	Heat exchanger structure factor
λ	Fuel consumption coefficient

μ	CO ₂ emission conversion factor (g/kWh)
τ	Time (s)
<i>Subscripts</i>	
<i>air</i>	Air
<i>boiler</i>	Boiler
<i>bypass</i>	Bypass
<i>cool-down</i>	Cool-down phase
<i>cw</i>	Cooling water
<i>dynamic</i>	Dynamic model
<i>el</i>	Electricity
<i>eng</i>	Engine
<i>flue</i>	Flue gas
<i>fuel</i>	Fuel
<i>gen</i>	Generation
<i>gross</i>	Gross
<i>hx</i>	Heat exchanger
<i>in</i>	Inlet
<i>jacket</i>	Jacket
<i>jw</i>	Jacket water
<i>lasttime</i>	Last time step
<i>loss</i>	Loss
<i>max</i>	Maximum
<i>net</i>	Net
<i>normal</i>	Normal operation phase
<i>out</i>	Outlet
<i>overheat</i>	Overheat
<i>pro</i>	Protection
<i>q</i>	Quality
<i>room</i>	Room
<i>set</i>	The setting point
<i>skin</i>	Skin
<i>standby</i>	Standby phase
<i>start-up</i>	Start-up phase
<i>steady</i>	Steady model
<i>stop</i>	Stop
<i>th</i>	Thermal
<i>thp</i>	Thermal priority mode
<i>total</i>	Total

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1. Introduction

With the advance of technology, CCHP (Combine Cooling, Heating and Power) systems have been generalized as an important technology for reducing pollutant emissions and energy consumption in many countries. In the process of generalization, some key technical issues should be solved, which can be concluded as follows:

- ◆ optimal sizing and design of the system;
- ◆ optimal operation strategy and control;
- ◆ mismatch between energy demand and supply.

For the optimal sizing and design aspect, many mathematical methods have been developed. They can be divided into two genres. The first genre estimates the size of prime mover based on the duration curves of electricity demand or thermal demand. This approach is called maximum rectangle method. Ref. [1], [2], [3], [4] and [5] have carried out this approach in some cases. A detailed analysis and improvement of this approach have also been made. However, electricity and thermal demands could not be simultaneously considered by this kind of approach. Therefore, there may be a big error between the optimal result and the real situation when using this approach. Another genre achieves the optimal result by using the optimization algorithm based on the simulation results of CCHP system. The optimization algorithms are diverse, including LP (linear programming) [6], MILP (mixed-integer linear programming) [7], MINLP (mixed-integer nonlinear programming) [8], GA (genetic algorithm) [9], PSO (particle swarm optimization) [12], etc. On the other hand, many complex optimal operation strategies were proposed, such as Hybrid electric-thermal load strategy (HETS) [10] and operation strategy based on minimum distance (MD) [11].

The accuracy of system simulation model directly determines the credibility of the optimal results. Hourly energy demand data are used in most of the simulation models in these lectures [6-11]. It has been revealed that the simulation results are impacted significantly by the simulation time-step of the model [13]. Moreover, the models used in these studies are based on the steady-state performance of the micro-cogeneration unit, which called steady model in this paper. The dynamic processes, such as start-up, cool-down, and overheat protection operation, are ignored by these models. So it is with the energy quality of the recovery heat. In actual fact, the thermal efficiency of micro-cogeneration units is

low during start-up period [14]. The outlet water temperature from the unit would violently fluctuate during overheat protection process, which would influence the performance of heat recovery devices. Some optimal operation strategies may make the unit start and stop frequently. However, the negative effect of start and stop operation are not considered by the steady model. Therefore, the optimal result may be very different with the fact when using the steady model. Although the current literature had developed a large amount of theoretical optimization methods, little work has been done for implementing these methods by the control methods based on physical parameters in actual system [15]. CCHP system is one kind of complex system. In reality, the experiment for verifying the performance of the optimization method is both time consuming and expensive. So an accuracy dynamic simulation model is needed to complete this work.

Through the analysis above, it can be found that a dynamic model, which makes a good agreement with the actual performance, is very necessary for the generalization and application of CCHP system. This model should also be suitable with annual dynamic building energy simulations [16]. Several authors have focused on this research area in recent years. Ferguson et al. [17] proposed a combustion cogeneration model for building simulation which was a part of IEA (International Energy Agency) Annex 42 [23]. Rosato et al. [18] calibrated and validated a dynamic model based on Annex 42 for simulating thermal and electric performance of a 6 kW internal combustion engine-based micro-cogeneration device in the TRNSYS environment [22]. This model was proved to accurately simulate the unit's dynamic and steady state performance. They also evaluated the dynamic performance of a building-integrated cogeneration system by using this model [19]. Bouvenot et al. [16] implemented a dynamic model based on experimental investigations of a wood pellet steam engine micro CHP for building energy simulation. In this model, the combustion reactions model of wood pellet was included. Rey et al. [20] implemented a model based on the experimental data of an internal combustion engine micro-cogeneration system by using TRNSYS software. The performance of system operating in different modes was analyzed.

The cooling, heating and power demands of the building are complex and changeable. And selling electricity to the grid for micro-CCHP system is not allowed in many areas of the world. As a result, the following electricity load operation strategy is used generally in these areas. Under this operation strategy, exhausting the excess heat is unavoidable. In this case, the overheat protection controls would

be triggered frequently. Besides, the outlet temperature of cooling water would fluctuate acutely when the overheat protection control is switched between on and off. The fluctuation of cooling water temperature would influence the coefficient of performance of the thermal driven chiller in CCHP system [25]. However, the overheat protection controls are seldom considered by the dynamic model in present studies [17-20].

In recent years, the mixed effect absorption chiller [25] is introduced to the internal combustion engine based micro-CCHP system, which can improve the total energy and exergy efficiency of the system. This chiller is powered by jacket water and exhaust gas waste heat respectively. Besides, the exhaust gas deep-recovery is help to enhance the primary energy efficiency of the CCHP system [26]. What's more, the temperature and flow of exhaust gas change a lot when the internal combustion engine operates from no-load operation to full load operation [27]. Therefore, the status parameters (temperature and flow) of exhaust gas should be considered in the dynamic model of CCHP system. In present studies [17-20], the recovered heat from the engine is used to generate hot water. So the status parameters are seldom considered by the ICE based simulation model in these studies.

In this paper, an ICE based dynamic simulation model, which considers the overheat protection control and status parameters of exhaust gas, is built as a new component on TRNSYS software platform. This model is calibrated and validated by the experimental data including the characteristics of the unit in overheat protection processes.

The experimental study methods including test bench set-up and evaluation equations are showed in the subsequent section. Also, the governing equations of the dynamic simulation model are showed. The experimental results are presented and analyzed for parameter calibration in section 4. In section 5, the validation and analysis of the dynamic model are showed. The improvement of prediction accuracy of the model considering dynamic processes is analyzed in section 6. Finally, some conclusions are summarized.

2. Experimental set-up

A schematic diagram of the test bench is shown in **Fig. 1**. It consists of one micro-cogeneration unit, two heat recovery exchangers, one thermal storage tank, two cooling coils, several water pumps and some sensors. The micro-cogeneration unit has a nominal power of 25kWe with the nominal generating efficiency of 28 %. And the continuous operation power is 22kWe. The power can step change from

2kWe to 25kWe. The dimensions of the device module are 3150×1050×1850 mm. For this test bench, the unit is connected to the electric grid with power injection. Photographs of the experimental facilities are show in **Fig.2**.

The micro-cogeneration unit can operate in three modes: manual mode, thermal priority mode and electricity priority mode. When the unit operates in manual mode, the generation can be set by the user. When the unit operates in thermal priority mode, the outlet temperature of the cooling water can be set by the user. When the unit operates in electricity priority mode, it will operate following the electric load.

A part of waste heat from exhaust gas and engine jacket can be recovered by the unit. The maximum thermal power and thermal efficiency of the unit are 60kW and 60%, respectively. The function of the overheat protection control is to keep the temperature of engine (the maximum temperature of water-glycol mixture in the primary circulation) and cooling water in the secondary circulation lower than the setting values. The pump P1 would be turned on to partially bypass the water-glycol mixture into the cooler when its temperature is higher than the setting point. In thermal priority mode, the exhaust gas would be partially bypassed to ambient when the cooling water temperature higher than the setting value. When the cooling water temperature is higher than the threshold, the unit would be shut down in thermal priority mode. The heat absorbed by cooling water can be stored in a 2.0 m³ thermal storage tank or sent to the cooling coils directly.

The positions of the experimental plants in the test bench are shown in **Fig. 1**. The measured parameters, the correspondent metrological and uncertainties are summarized in **Table 1**. The interval of data acquisition is 5 s. For the limitation of natural gas supply, the fuel gas used in this test is liquefied petroleum gas (LPG). The composition of the fuel gas is achieved through the test of a Gas Chromatograph-Mass Spectrometer. It consists of 96.8% C₄H₁₀, 2.98% N₂ and 0.22% CH₄.

Three kinds of experimental procedures are included in this paper, as show in **Table 2**. During procedure A, generation of the unit and blast capacities of fan coils are fixed. This procedure is use to test the performances of steady state in different generation capacities. Procedure B is used to test the characteristics of the overheat protection control in thermal priority mode. The blast capacities of fan coils remain in a fixed value before the outlet cooling water temperature stabilizes at the setting value (45 °C in this study). Then the blast capacities of fan coils are changed to adjust the temperature of cooling water, as show in **Fig. 7**. Procedure C is use to test the characteristics of the overheat protection

control in manual and electricity priority mode. During this procedure, pump P4 is turned off. And the generation of the unit is fixed to charge the storage tank and then trigger the overheat protection control. The threshold of the engine temperature is set to 88 °C in this study.

3. Simulation models and evaluation methods

3.1 Dynamic simulation model

3.1.1 Control volumes and thermodynamic processes

As shown in **Fig.3**, four control volumes are employed to model the thermodynamic processes of the unit: combustion control volume, engine control volume, jacket water heat exchanger control volume as well as flue gas heat exchanger control volume. The governing equations of each control volume are as follows:

Control volume 1: Combustion control volume

The energy balance for combustion control volume is:

$$Cp_{flue}(T_{flue,out1}) \cdot m_{flue} \cdot T_{flue,out1} = Q_{gross} + H_{fuel} + H_{air} - Q_{gen} - E_{gen} \quad (1)$$

where Cp_{flue} is the average specific heat of the flue gas as a function of the temperature $T_{flue,out1}$; Q_{gross} is the total energy inputted into the engine; E_{gen} is the electric power output from the generator; H_{fuel} and H_{air} are the enthalpy of fuel and air put into the engine. Q_{gen} is the transfer of heat from the flue gas to the engine, which can be determined below:

$$Q_{gen} = \eta_{gen} Q_{gross} = [UA]_{eng} (T_{flue,out1} - T_{eng}) \quad (2)$$

where $[UA]_{eng}$ is the overall thermal conductance between the flue gas and the engine control volume. T_{eng} is the temperature of the engine control volume. Relationship between Q_{gross} and E_{gen} can be expressed by:

$$\eta_{el} = E_{gen} / Q_{gross} \quad (3)$$

$$Q_{gross} = LHV_{fuel} \cdot m_{fuel} \quad (4)$$

where LHV_{fuel} is the low heat value (LHV) of the fuel gas; m_{fuel} is the flow rate of the fuel gas.

The electrical efficiency of the unit η_{el} and heat efficiency of jacket water η_{gen} are defined as the function of the power generation.

$$\eta_{el} = a_0 + a_1 \cdot E_{gen} + a_2 \cdot E_{gen}^2 + a_3 \cdot E_{gen}^3 \quad (3)$$

$$\eta_{gen} = b_0 + b_1 \cdot E_{gen} + b_2 \cdot E_{gen}^2 + b_3 \cdot E_{gen}^3 \quad (4)$$

The LHV of the fuel can be estimated by the following equation:

$$LHV_{fuel} = \sum \chi_i \cdot LHV_i \quad (5)$$

where χ_i is the molar fraction of constituent i and LHV_i is the LHV of constituent i .

The flow rate of flue gas and air can be calculated through the chemical reaction of fuel combustion.

$$V_{air} = 4.76 \cdot e_{air} \cdot \frac{m_{fuel}}{\rho_{fuel}} \cdot \left[0.5\chi_{H_2} + 0.5\chi_{CO} + 1.5\chi_{H_2S} + \sum (n+k/4)\chi_{C_nH_k} - \chi_{O_2} \right] \quad (6)$$

$$V_{flue} = \frac{m_{fuel}}{\rho_{fuel}} \left\{ \begin{aligned} &\left[\chi_{CO_2} + \chi_{CO} + 2\chi_{H_2S} + \sum (n+k/2)\chi_{C_nH_k} + \chi_{H_2} + \chi_{H_2O} + \chi_{N_2} \right] \\ &+ 1.0161 \cdot \frac{(e_{air} - 1)}{e_{air}} \dot{V}_{air} \end{aligned} \right\} \quad (7)$$

$$m_{air} = \rho_{air} \cdot V_{air} \quad (8)$$

$$m_{flue} = \rho_{flue} \cdot V_{flue} \quad (9)$$

where e_{air} is the excessive air coefficient.

Control volume 2: Engine control volume

The governing equation of the engine control volume is as follows:

$$[MC]_{eng} \frac{dT_{eng}}{d\tau} = Q_{gen} - Q_{jacket,hx} - Q_{skin,loss1} - \varepsilon_{pro} \cdot Q_{overheat} \quad (10)$$

where $[MC]_{eng}$ is the thermal capacitance of the engine control volume. Q_{hx} is the transfer of heat from

the engine control volume to cooling water. $Q_{skin,loss1}$ is the transfer of heat from the engine control

volume to surroundings. $Q_{overheat}$ is transfer of heat from the engine control volume to the surroundings

through the cooling radiator. ε_{pro} represents the on(1)/off(0) state of the overheat protection control.

During the cool-down and overheat protection phase, ε_{pro} is equal to 1. Otherwise, it is equal to 0.

$$Q_{jacket,hx} = [UA]_{jacket,hx} (T_{eng} - T_{cw,out1}) \quad (11)$$

$$Q_{skin,loss1} = [UA]_{loss1} (T_{eng} - T_{room}) \quad (12)$$

$$Q_{overheat} = [UA]_{pro} (T_{eng} - T_{room}) \quad (13)$$

Control volume 3: Jacket water heat exchanger control volume

The governing equation of the jacket water heat exchanger control volume is as follow:

$$[MC]_{jacket,hx} \frac{dT_{cw,out1}}{d\tau} = m_{cw} \cdot Cp_{cw} (T_{cw,out1} - T_{cw,in}) + Q_{jacket,hx} \quad (14)$$

where $[MC]_{jacket,hx}$ is the thermal capacitance of the plate heat exchanger and the encapsulated water. Cp_{cw} is the average cooling water specific heat; m_{cw} is the mass flow rate of cooling water; $T_{cw,in}$ is the temperature of cooling water flowing into the unit; $T_{cw,out1}$ is the temperature of cooling water flowing out from the plate heat exchanger.

Control volume 4: Flue gas heat exchanger control volume

The governing equation of the flue gas heat exchanger control volume is as follows:

$$[MC]_{flue,hx} \frac{dT_{cw,out2}}{d\tau} = Q_{flue} - m_{cw} \cdot Cp_{cw} (T_{cw,out2} - T_{cw,out1}) - Q_{skin,loss2} \quad (15)$$

where $[MC]_{flue,hx}$ is the thermal capacitance of the flue gas heat exchanger and the encapsulated water.

The enthalpy change of the flue gas can be calculated as:

$$Q_{flue} = \varepsilon_{bypass} \cdot m_{flue} \cdot (Cp_{flue}(T_{flue,out1}) \cdot T_{flue,out1} - Cp_{flue}(T_{flue,out2}) \cdot T_{flue,out2}) \quad (16)$$

where ε_{bypass} is the fraction of flue gas bypassed to the flue gas heat exchanger. Then, the temperature of flue gas exhausted to atmosphere can be estimated as:

$$Cp_{flue}(T_{flue,mix})T_{flue,mix} = \dot{m}_{flue} \left[\varepsilon_{bypass} \cdot Cp_{flue}(T_{flue,out2}) \cdot T_{flue,out2} + (1 - \varepsilon_{bypass}) \cdot Cp_{flue}(T_{flue,out1}) \cdot T_{flue,out1} \right] \quad (17)$$

The transfer of heat from flue gas to cooling water is calculated as:

$$[UA]_{flue,hx} \cdot \Delta T = m_{cw} \cdot Cp_{cw} (T_{cw,out2} - T_{cw,out1}) \quad (18)$$

where $[UA]_{flue,hx}$ is the overall thermal conductance between flue gas and cooling water and ΔT is the logarithmic mean temperature.

$$\Delta T = \frac{(T_{flue,out1} - T_{cw,out1}) - (T_{flue,out2} - T_{cw,out2})}{\ln((T_{flue,out1} - T_{cw,out1}) / (T_{flue,out2} - T_{cw,out2}))} \quad (19)$$

The $UA_{flue,hx}$ can be estimated as [28]:

$$[UA]_{flue,hx} = \theta \cdot m_{flue}^{\alpha} \cdot \bar{T}_{flue}^{\beta} \quad (20)$$

where θ is a parameter determined by the heat exchanger structure, α and β are the power related to tube bank arrangement and thermo-physical properties of flue gas. These parameters can be

219 calibrated through the experimental data. $\bar{T}_{flue}$ is the average temperature of flue gas. The heat losses
 220 through the flue gas heat exchanger can be calculated as:

$$221 \quad \dot{Q}_{skin,loss2} = [UA]_{loss2} (T_{cw,out2} - T_{room}) \quad (21)$$

222

223 3.1.2 Standby phase

224 In standby phase, no electric or thermal energy is generated. Then no fuel is consumed by the unit.
 225 However, the control system needs some electricity $E_{standby}$. Thus:

$$226 \quad E_{net} = -E_{standby} \quad Q_{gross} = 0 \quad m_{fuel} = 0 \quad (22)$$

227 where E_{net} is the net electricity outputted from the unit. During this phase, the major thermodynamic
 228 process is the transfer of heat from the unit to surroundings.

229 3.1.3 Start-up phase

230 Two situations should be considered in start-up phase: cold start-up and hot start-up. More time and
 231 more fuel are taken during cold start-up than that of hot start-up. The rule during start-up phase can be
 232 expressed as follows:

$$233 \quad E_{net} = \begin{cases} E_{set} - E_{normal} & E_{set} < E_{start-up,max} \\ E_{start-up,max} - E_{normal} & E_{set} > E_{start-up,max} \end{cases} \quad (23)$$

$$234 \quad m_{fuel,start-up} = \begin{cases} m_{fuel} & hot \quad start-up \\ \lambda_{start-up} \cdot m_{fuel} & cold \quad start-up \end{cases} \quad (24)$$

235 where E_{set} is the setting value of generation for the unit. E_{normal} is the electricity consumption by unit in
 236 normal phase and start-up phase. $E_{start-up,max}$ is the maximum generation during start-up phase. $\lambda_{start-up}$ is
 237 the fuel consumption coefficient for cold start-up.

238 3.1.4 Cool-down phase

239 During cool-down phase, no electric or thermal energy is generated. Some electricity $E_{cool-down}$ is
 240 consumed to cool down the engine. Thus:

$$241 \quad E_{net} = -E_{cool-down} \quad Q_{gross} = 0 \quad m_{fuel} = 0 \quad \varepsilon_{pro} = 1 \quad (25)$$

242 3.1.5 Overheat protection phase

243 The overheat protection system consists of two part. The first part is to prevent water-glycol mixture
 244 in primary circulation from overheating. The second part is to prevent cooling water in secondary

245 circulation from overheating.

246 The first part can be triggered during any operation mode. The control principles are as follows:

$$247 \quad \varepsilon_{pro} = \begin{cases} 0 & T_{eng} < T_{eng,stop} \\ 0 & T_{eng,stop} \leq T_{eng} \leq T_{eng,max} \ \& \ \varepsilon_{pro,lasttime} = 0 \\ 1 & T_{eng} > T_{wg,max} \\ 1 & T_{eng,stop} \leq T_{eng} \leq T_{eng,max} \ \& \ \varepsilon_{pro,lasttime} = 1 \end{cases} \quad (26)$$

$$248 \quad E_{net} = \begin{cases} E_{gen} - E_{normal} & \varepsilon_{pro} = 0 \\ E_{gen} - E_{cool-down} & \varepsilon_{pro} = 1 \end{cases} \quad (27)$$

249 The second part would be triggered during the thermal priority mode, which controls the cooling
250 water temperature by bypassing flue gas and on/off the unit. The control principles are as follows:

$$251 \quad \varepsilon_{bypass} = \begin{cases} 1 & T_{cw,out2} < T_{thp,set} - \Delta T_{bypass} \\ 1 & T_{thp,set} - \Delta T_{bypass} \leq T_{cw,out2} \leq T_{thp,set} + \Delta T_{bypass} \ \& \ \varepsilon_{bypass,lasttime} = 1 \\ 0 & T_{cw,out2} > T_{thp,set} + \Delta T_{bypass} \\ 0 & T_{thp,set} - \Delta T_{bypass} \leq T_{cw,out2} \leq T_{thp,set} + \Delta T_{bypass} \ \& \ \varepsilon_{bypass,lasttime} = 0 \end{cases} \quad (28)$$

$$252 \quad \varepsilon_{unit} = \begin{cases} 1 & T_{cw,out2} < T_{thp,set} - \Delta T_{start} \\ 1 & T_{thp,set} - \Delta T_{start} \leq T_{cw,out2} \leq T_{thp,set} + \Delta T_{stop} \ \& \ \varepsilon_{unit,lasttime} = 1 \\ 0 & T_{cw,out2} > T_{thp,set} + \Delta T_{stop} \\ 0 & T_{thp,set} - \Delta T_{start} \leq T_{cw,out2} \leq T_{thp,set} + \Delta T_{stop} \ \& \ \varepsilon_{unit,lasttime} = 0 \end{cases} \quad (29)$$

253 where ε_{unit} represents the on(1)/off(0) control signal for the unit. The inputs, parameters and outputs
254 of the simulation model are shown in **Appendix A**. The transitional conditions between different
255 transient phases are shown in **Table 3**. The algorithm of the simulation model is presented in **Appendix**
256 **B**. According to the theory above, a new component is developed on the TRNSYS software platform.

257 3.2 Steady simulation model

258 The steady model simulates the performance based on the energy performance of the unit in steady
259 state. It is mainly governed by the equations as follows:

$$260 \quad E_{gen} = E_{set} \quad (30)$$

$$261 \quad \eta_{el} = a_0 + a_1 \cdot E_{gen} + a_2 \cdot E_{gen}^2 + a_3 \cdot E_{gen}^3 \quad (31)$$

$$262 \quad \eta_{th} = c_0 + c_1 \cdot E_{gen} + c_2 \cdot E_{gen}^2 + c_3 \cdot E_{gen}^3 \quad (32)$$

263 where η_{th} is the total thermal efficiency. Q_{gross} can be calculated by eq.(3). The total recovery heat can
264 be calculated by:

$$Q_{total,th} = \eta_{th} \cdot Q_{gross} \quad (33)$$

The major distinction between dynamic and steady model can be concluded as follows:

- ◆ *Dynamic process.* The steady model does not consider the dynamic processes: start-up, cool-down and overheat protection controls. The electric generation cannot reach the setting point immediately when the unit is started up. The unit is cooled down once it is shut down. As a result, the unit should be warmed-up when started again. What's more, more fuel is consumed during start-up process than that at steady state in the same electric generation. When overheat protection control is on, the cooling water outlet temperature would fluctuate acutely.
- ◆ *Thermal capacity of the unit.* The steady model does not consider thermal capacity of the unit. So the heat output is only relevant with the electricity output. However, for dynamic model, some heat production should be used to warm-up the unit every time it starts up because of the thermal capacity of the unit.

3.3 Evaluation method

Primary energy saving ratio (PESR) is defined as the ratio of the saving energy of the CHP system to the energy consumption of the conventional system.

$$PESR = (Q_{CS} - Q_{gross}) / Q_{CS} \quad (34)$$

$$Q_{CS} = E_{gen} / \eta_{el,CS} + Q_{th} / \eta_b \quad (35)$$

CO₂ emission reduction ratio (CO₂ERR) is defined as the ratio of the reducing CO₂ emission of CHP system to the emission of conventional system.

$$CO_2ERR = (CO_2E_{CS} - CO_2E_{fuel}) / CO_2E_{CS} \quad (36)$$

$$CO_2E_{CS} = \mu_e E_{gen} / \eta_{el,CS} + \mu_{NG} Q_{th} / \eta_b \quad (37)$$

$$CO_2E_{fuel} = \mu_{fuel} Q_{fuel} \quad (38)$$

where μ is the CO₂ emission coefficient. The relative parameters used to calculate the evaluation criteria are shown in **Table 4**.

The cumulative prediction error of E_{gen} , Q_{th} , and η_{total} between predictions of simulation model and experimental results can be calculated as follows:

$$Error_E^\tau = \left(\sum_0^\tau E_{gen,sim} - \sum_0^\tau E_{gen,exp} \right) / \sum_0^\tau E_{gen,exp} \quad (39)$$

$$Error_Q^\tau = \left(\sum_0^\tau Q_{th,sim} - \sum_0^\tau Q_{th,exp} \right) / \sum_0^\tau Q_{th,exp} \quad (40)$$

$$Error_\eta^\tau = \frac{\left(\left(\sum_0^\tau E_{gen,sim} + \sum_0^\tau Q_{th,sim} \right) / \sum_0^\tau Q_{gross,sim} - \left(\sum_0^\tau E_{gen,exp} + \sum_0^\tau Q_{th,exp} \right) / \sum_0^\tau Q_{gross,exp} \right)}{\left(\sum_0^\tau E_{gen,exp} + \sum_0^\tau Q_{th,exp} \right) / \sum_0^\tau Q_{gross,exp}} \quad (41)$$

3.3 Calculation of experimental parameters

The total energy inputted into the engine can be calculated:

$$Q_{gross} = LHV_{fuel} \cdot \rho_{fuel} \cdot V_{fuel} \quad (42)$$

where V_{fuel} is the volume flow rate of the fuel gas. The recovered heat from jacket water and flue gas can be calculated as:

$$Q_{jacket,hx} = Cp_{cw} \cdot \rho_{cw} \cdot V_{cw} \cdot (T_{cw,out1} - T_{cw,in}) \quad (43)$$

$$Q_{flue} = Cp_{cw} \cdot \rho_{cw} \cdot V_{cw} \cdot (T_{cw,out2} - T_{cw,out1}) \quad (44)$$

where V_{cw} is the volume flow rate of cooling water. The electric efficiency and thermal efficiencies can be estimated as:

$$\eta_{jacket,hx} = Q_{jacket,hx} / Q_{gross} \quad (45)$$

$$\eta_{flue} = Q_{flue} / Q_{gross} \quad (46)$$

$$\eta_{th} = \eta_{jacket,hx} + \eta_{flue} \quad (47)$$

$$\eta_{total} = \eta_{jacket,hx} + \eta_{flue} + \eta_{el} \quad (48)$$

4. Calibration of parameters in dynamic simulation model

4.1 Calibration of η_{el} and η_{gen}

The values of steady performances are determined by the average of the data for one hour operating in steady state. **Fig. 4** plots the electric efficiency and thermal efficiencies as a function of the electric generation E_{gen} . It shows that the electric efficiency and heat recovery efficiency of the flue gas increase with the increase of E_{gen} . However, the heat recovery efficiency of the jacket water decreases

first and then increases. It can be found that the total efficiency is lower than 70 % when the E_{gen} is smaller than 15kW. The electric efficiency is just 12.24% when the E_{gen} is equal to 3kW. The fitting curves are also shown in **Fig. 4**. Considering the heat loss of engine control volume, η_{gen} is a little high than $\eta_{jacket,hx}$. Therefore, coefficients of fitting functions of η_{el} and η_{gen} are shown in **Table A.2**.

4.2 Calibration of $\Delta t_{start-up,cold}$, $\Delta t_{start-up,hot}$, $\Delta t_{cool-down}$, and $E_{start-up,max}$

Fig. 5 plots E_{gen} and V_{fuel} as a function of time in the start-up phases. **Fig. 5 (a)** shows the characteristics of the unit in cold start-up phase. The setting value of electric generation is 12kW. However, the generation stabilizes around 10kW during the warm-up phase. Then it increases to the setting value after the warm-up phase. Based on the results of other tests, the same situation appears when the setting value is larger than 10kW. For this unit, $E_{start-up,max}$ can be determined to be 10kW. **Fig. 5 (b)** shows the characteristics of the unit in hot start-up phase. The setting value is 9kW. It can be found that the electric generation would reach the setting value during the warm-up phase at this situation. The same result appears when the setting value is smaller than 10kW. Compared **Fig. 5 (a)** and **Fig. 5 (b)**, it can be derived that the duration of cold start-up is longer than that of hot start-up. Moreover, more fuel is needed for the cold start-up. An embossment exists on the curve of V_{fuel} during the cold start-up. The durations in different tests are summarized in **Table 5**. According to these results, $\Delta t_{start-up,cold}$ and $\Delta t_{start-up,hot}$ are determined to be 300 s and 125 s, respectively.

Fig. 6 shows the characteristics during cool-down phase. Cool-down phase starts when the fuel supply is stopped and finishes when the stand-by mode is entered (water pump P1 and P2 is stopped). When the cool-down phase starts, water pump P1 would be turned on and some of the water-glycol mixture is bypassed to the radiator. Then the primary circulation would be cooled down. Therefore, the temperature of the jacket water T_{eng} decreases with the time. It can be seen that the duration of cool-down $\Delta t_{cool-down}$ is 315 s, as shown in **Table 5**.

4.3 Calibration of ΔT_{bypass} , ΔT_{stop} , and ΔT_{start} in thermal priority mode

The characteristics of the unit operating in thermal priority mode are shown in **Fig. 7**. The setting value of the outlet temperature of cooling water is 45 °C. The thermal load is turned on, turned off, turned down and turned up to adjust the outlet temperature, as shown in **Fig. 7**. When the $T_{cw,out2}$ is higher than 46.42 °C, the bypass mode is on. The bypass valve opens slowly. More and more flue gas is let out to atmosphere directly. Therefore, the Q_{flue} decreases sharply. When the $T_{cw,out2}$ is higher than 53.17 °C, the unit is shut down and cooled down. The unit is started up when the $T_{cw,out2}$ is lower than

39.35 °C. When $T_{flue,out2}$ is lower than 43.47°C, bypass mode is turned off and Q_{flue} increases. As we can see, much heat of the flue gas is lost when the bypass mode is turned on. Therefore, these dynamic processes should be considered when evaluating the performance of a Micro-CCHP system. The bypassing of flue gas and the shutdown of the unit should be avoided whenever possible. According to the results above, the setting value of ΔT_{bypass} , ΔT_{stop} , and ΔT_{start} can be achieved, as shown in **Table A.2**. The setting value of $T_{eng,max}$ and $T_{eng,stop}$ are calibrated in section 5. The other parameters are calibrated by using the method presented in ref. [24].

5. Validation of dynamic model and analysis

The validation of the model includes two parts: steady and dynamic performances. For steady performance, the comparison between experimental data and prediction results of the model during normal operation phase is shown in **Table 6**. The simulation results of electric output are in a good agreement with the experimental data (the maximum error is 1.86%). The prediction error of the recovered heat is also small (the range is from 1.97% to 6.04%). Then the predicted total efficiency agrees well with experimental data (the maximum error is 4.86%). From the comparison above, it can be deduced that the dynamic model can simulate the steady performance of the unit accurately.

Fig.8 shows the validation of temperature of flue gas out from the engine $T_{flue,in}$ and the power to thermal ratio during steady state. The simulation results are close to the experimental results. It can be seen that both $T_{flue,out1}$ and $PtoT$ increase with the increase of E_{gen} . The $T_{flue,out1}$ increase from 315 °C (3kW) to 542 °C (22kW). The flow rate of flue gas is also changed a lot. According to **Eq. (20)**, the heat transfer coefficient would be impacted significantly. Therefore, if the E_{gen} of the unit change frequently, it will significantly influence the performance of the heat recovery device. The maximum $PtoT$ ratio is 0.53 (18kW) and the minimum value is 0.23 (3kW).

For dynamic performance, the experimental results of the unit operating in thermal priority mode are used to validate the model. This process contains the start-up phase, the transition phase, the cool-down phase, the normal operation phase and the bypass mode operation. It can be used to validate the dynamic performance of the simulation model comprehensively. **Fig. 9 (a)** shows the comparison of trends of flue gas temperatures between experimental data and simulation results in thermal priority mode. It shows that the model can predict the dynamic trends of flue gas temperatures well. The major differences appear in the time of unit shut down and bypass mode turned on. When the unit is shut

down, no flue gas is generated. At these times, the temperature of the flue gas is assumed to be equal to that of the engine by the model. When the bypass mode is on, $T_{flue,out,sim}$ is much higher than $T_{flue,out,exp}$. The reason is that no insulation measure is used on the bypass pipe in the actual structure. And the transfer of heat through this pipe is neglected by the model.

Fig. 9 (b) shows the comparison of the outlet cooling water temperature and thermal energy output between experimental data and prediction results. The trends of simulation results are basically in accordance with those of test results. During transition phase, even though the model cannot simulate the high-frequency fluctuation, it can predict the average value. Since the temperature of water-glycol mixture in primary circulation is lower than that of cooling water in secondary circulation, some heat in the cooling water is lost through the plate heat exchanger during the cool-down phase, as show in **Fig. 9 (b)**. To avoid heat loss, the cooling water should avoid flowing through the jacket water heat exchanger during cool-down phase.

The performance of overheat protection operation is an important aspect of the dynamic performance of the unit. **Fig. 10** presents the validation results of the model during overheat protection operation. According to the experimental result, $T_{eng,max}$ and $T_{eng,stop}$ are 88 °C and 75 °C, respectively. The overheat control operation makes outlet cooling temperature fluctuate acutely. The amplitude of fluctuation of cooling temperature is about 7 °C, as shown in **Fig.10**. It would make negative effect on the performance of subsequent devices. It can be seen that the predicted amplitude and period of temperature fluctuation are close to the results of test data. This model can be used to study the impact of the overheat protection control on the performance of heat recovery devices, such as absorption chiller and waste heat boiler.

6. The improvement of prediction accuracy by considering dynamic processes

Fig. 11 presents the distinctions among experimental data, prediction results by dynamic model and prediction results by steady model in electric generation, thermal energy output and total efficiency. Both dynamic and steady model can predict electric generations well. Compared to the steady model, dynamic model can predict thermal energy output and total efficiency of the unit better in dynamic process. The performance of the unit before becoming steady is much poorer than that in steady state. Therefore, the major distinctions between dynamic model and steady model lie in the time before the unit become steady. Once the unit is stopped, the engine would be cooled down. Thus it should be

warmed up again when it is started. According to **Fig. 11**, it can be deduced that start-stop interval would significantly influence the prediction differs of dynamic and steady model. The following analysis is based on the operation process shown in **Fig. 11**.

In order to evaluate the improvement of prediction accuracy of models considering dynamic processes, the cumulative prediction errors between simulation model and experimental results for different start-stop interval are shown in **Fig. 12**. The cumulative prediction errors decrease with the increase of the start-stop interval. When the start-stop interval increases from 0.25 h to 2 h, the Q_{th} prediction error of dynamic model decreases from 8.48 % to -0.42 %. The η_{total} prediction error of dynamic model decreases from 11.57 % to 1.35 %. It can be deduced that the lower start-stop frequency of the unit is, the more accurate simulation results could be achieved by the simulation model. By considering dynamic processes, 5.18% of Q_{th} cumulative prediction errors are decreased when the start-stop interval is 0.5h. At the same time, 3.20% of η_{total} cumulative prediction errors are decreased.

Fig. 13 presents the trends of PESR and CO₂ERR of the unit as function of start-stop interval. When the start-stop interval is equal to 2.0 h, the differences between the predictions of simulation model and experimental results are small. However, when start-stop interval is equal to 0.25 h, the simulation models vastly overestimate the performance of the unit. At this time, the PESR is much smaller than 0%, which means that the energy consumption of micro-cogeneration unit is larger than that of conventional system. The reason is that much heat produced by the unit is used to heat the unit during start-up phase. It is suggested that the start-stop interval for the unit in this paper should be longer than 0.5 h. The PESR and CO₂ERR performances of the unit are improved with the increase of the start-stop interval. Both dynamic and steady model are over estimate the PESR and CO₂ERR performance of the unit. And the difference is increase with decrease of start-stop interval. It is said that the shorter of the start-stop interval during the operation process, the more of the performance is over estimated by the simulation model.

7. Conclusions

In this paper, a dynamic simulation model, which considers the overheat protection control and status parameters of exhaust gas, is built as a new component on TRNSYS software platform. This model is calibrated and validated by experimental data during steady and dynamic processes. What's

more, the improvement of prediction accuracy of the model considering dynamic processes is analyzed.

Then some main conclusions can be drawn as follows:

◆ Through the validation by experimental data, it proves the dynamic can well predict the changing trends of physical parameters during the operation of overheat protection controls, start-up, and cool-down.

◆ Through the analysis of improvement of prediction accuracy of the model considering dynamic processes, it can be concluded that the cumulative prediction errors of the dynamic model decrease with the increase of start-stop interval. By considering dynamic processes, 5.18% of Q_{th} cumulative prediction errors are decreased when the start-stop interval is 0.5h. At the same time, 3.20% of η_{total} cumulative prediction errors are decreased.

◆ When the start-stop interval is shorter than a certain value, the energy consumption of micro-cogeneration unit would be larger than that of conventional system. It is suggested that the start-stop interval for the unit in this paper should be longer than 0.5 h.

Acknowledgements

This work was supported by National Science Foundation of China under the contract No. 5151101162 and The Walt Disney Company (Shanghai) Limited.

Appendix A. Model inputs, parameters and outputs

Table A.1 Inputs of the model

No.	Symbol	Description	Unit
1	Mode	Control method: 0=deactivated 1=Manual control 2=thermal priority control 3=electricity priority control	-
2	E_{demand}	For Mode=0,1,3(elctricity demand)	kW
3	$E_{thP,set}$	Power output setting for thermal priority mode(Mode=2)	kW
4	T_{room}	Room air temperature	°C
5	$T_{cw,in}$	Water temperature at inlet	°C
6	m_{cw}	Flow rate at inlet	kg/s

451 Table A.2 Parameters of the model

No.	Symbol	Description	Value	Unit
1	$E_{gen,max}$	System maximum power	25000	W
2	$E_{gen,min}$	System minimum power	2000	W
4	$T_{eng,max}$	Maximum engine temperature	85	°C
5	$T_{eng,stop}$	Minimum temperature for the protection system stop	76	°C
6	$T_{thP,set}$	Outlet temperature for the thermal priority mode	45	°C
7	ΔT_{bypass}	Temperature difference for bypass mode	1.5	°C
8	ΔT_{stop}	Temperature difference for stopping the unit	8.0	°C
9	ΔT_{start}	Temperature difference for starting the unit	5.0	°C
10	LHV_{fuel}	Liquid fuel lower heating value LHV	119.8	MJ/m ³
11	m_{CO2}/m_{fuel}	Liquid fuel CO2 factor	0	kgCO2/kg
12	χ_{H2}	% Hydrogen in fuel gas mixture	0	mol/mol
13	χ_{CH4}	% Methane in fuel gas mixture	0.0021	mol/mol
14	χ_{C2H4}	% Hydrocarbon in fuel gas mixture	0	mol/mol
15	χ_{C2H6}	% Ethane in fuel gas mixture	0	mol/mol
16	χ_{C3H6}	% Cyclopropane in fuel gas mixture	0	mol/mol
17	χ_{C3H8}	% Propane in fuel gas mixture	0	mol/mol
18	χ_{C4H8}	% Butylene in fuel gas mixture	0	mol/mol
19	χ_{C4H10}	% Butane in fuel gas mixture	96.8	mol/mol
20	χ_{C5H10}	% Cyclopentane in fuel gas mixture	0	mol/mol
21	χ_{C5H12}	% Pentane in fuel gas mixture	0	mol/mol
22	χ_{C4H14}	% Hexane in fuel gas mixture	0	mol/mol
23	χ_{CH3OH}	% Methanol in fuel gas mixture	0	mol/mol
24	χ_{C2H5OH}	% Ehtanol in fuel gas mixture	0	mol/mol
25	χ_{CO}	% Carbon monoxide in fuel gas mixture	0	mol/mol
26	χ_{CO2}	% Carbon dioxide in fuel gas mixture	0	mol/mol
27	χ_{N2}	% Nitrogen in fuel gas mixture	2.98	mol/mol
28	χ_{O2}	% Oxygen in fuel gas mixture	0	mol/mol
29	$(dP_{net}/dt)_{max}$	Maximum rate of change in power output	1000	W/s
30	$[MC]_{eng}$	Power system thermal mass	85000	J/K
31	$[MC]_{jacket,hx}$	Jacketexchanger thermal mass	4350	J/K
32	$[MC]_{flue,hx}$	Flue gas exchanger thermal mass	2050	J/K
33	$[UA]_{jacket,hx}$	Effective heat recovery UA coefficient	5530	W/K
34	$[UA]_{loss1}$	Effective heat loss UA coefficient (engine)	55	W/K
35	$[UA]_{loss2}$	Effective heat loss UA coefficient (flue hx)	20	W/K
36	$[UA]_{pro}$	Effective heat transfer UA coefficient (protection system)	980	W/K
37	$[UA]_{eng}$	Effective engine	55	W/K
38	$\Delta t_{start-up,cold}$	Cold start-up period duration	300	s
39	$\Delta t_{start-up,hot}$	Hot Start-up period duration	125	s
40	$\Delta t_{cool-down}$	Cool-down period duration	315	s
41	$E_{cool-down}$	Electricity consumption by unit in cool-down phase	585	W

42	$E_{standby}$	Electricity consumption by unit in standby phase	100	W
43	E_{normal}	Electricity consumption by unit in normal phase	495	W
44	$E_{start-up,max}$	Maximum electric generation during start-up phase	10000	W
45	$T_{eng,initial}$	Initial temperature of the engine	25	°C
46	$\lambda_{start-up}$	Fuel consumption coefficient for cold start-up	0.05	-
47	e_{air}	Excessive air coefficient	1.30	-
48	a_0	Coefficient of electric efficiency	9.32	-
49	a_1	Coefficient of electric efficiency	1.45	-
50	a_2	Coefficient of electric efficiency	-0.037	-
51	a_3	Coefficient of electric efficiency	0.00048	-
52	b_0	Coefficient of heat generation efficiency of jacket water	68.901	-
53	b_1	Coefficient of heat generation efficiency of jacket water	-0.3243	-
54	b_2	Coefficient of heat generation efficiency of jacket water	-0.0305	-
55	b_3	Coefficient of heat generation efficiency of jacket water	0.0	-

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453 Table A.3 Outputs of the model

No.	Symbol	Description	Unit
1	T_{eng}	Temperature of the engines thermal mass control volume	°C
2	$T_{cw,out1}$	Outlet cooling water temperature of jacket water HX	°C
3	$T_{cw,out2}$	Outlet cooling water temperature of flue gas HX	°C
4	$T_{flue,out2}$	Outlet temperature of the gas heat recycle control volume	°C
5	E_{gen}	Steady-state electrical net power output	W
6	Q_{gen}	Steady-state heat generation from jacket water	W
7	m_{fuel}	Fuel flow	kg/s
8	m_{cw}	Flow rate of cooling water	kg/s
9	Q_{gross}	LHV flow rate	W
10	$Q_{skin,loss1} + Q_{skin,loss2}$	Skin losses to the containment room	W
11	$Q_{jacket,hx}$	Heat transferred from jacket water to cooling water	W
12	Q_{flue}	Heat transferred from flue gas to cooling water	W
13	η_{el}	Electrical efficiency	-
14	η_{th}	Thermal efficiency	-
15	$\eta_e + \eta_{th}$	Total efficiency	-
16	LHV_{fuel}	Lower heating value of the fuel	J/kg
17	m_{flue}	Flue gas flow rate	kg/s
18	m_{CO2}	Average CO2 emissions	kg/s
19	Cp_{flue}	Specific heat at constant pressure	kJ/kg*K
20	$Q_{overheat}$	Overheat or cool down system flow rate	W

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Appendix B. Algorithm of the simulation model

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Figure captions:

Fig.1 Scheme of experimental apparatus.

Fig.2 Photographs of the experimental facility under study: LPG, Fan Coil, Storage Tank and cogeneration unit.

Fig.3 Control volumes for the dynamic simulation model.

Fig.4 Steady state values of $\eta_{th,flue}$, $\eta_{th,jw}$, η_{th} , η_{total} and η_{el} as a function of E_{gen} .

Fig.5 The unit characteristics during cold start-up (a) and hot start-up (b).

Fig.6 The unit characteristics during cool-down operation.

Fig.7 The characteristics of the unit operating in thermal priority mode. (The setting value of the $T_{cw,out2}$ is 45 °C).

Fig.8 Comparison of steady state value of $T_{flue,in}$ and the power to thermal ratio between experimental data and simulation result.

Fig.9 Comparison of the trends of $T_{flue,out1}$, $T_{flue,out2}$, $T_{cw,out2}$ and Q_{th} between experimental data and simulation result in thermal priority mode.

Fig.10 Validation results of the model during overheat protection operation.

Fig.11 Distinctions among experimental data, prediction by dynamic model and prediction by steady model in (a) electric generation, (b) thermal energy output and (c) total efficiency.

Fig.12 Cumulative prediction errors between simulation model and experimental results for different start-stop interval.

Fig.13 The trends of PESR and CO₂ERR of the unit as function of start-stop interval.