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A multi-scale energy systems engineering approach to residential combined heat and power systems

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ABSTRACT

Distributed power production systems, such as residential cogeneration, are a promising alternative to the energy and environmentally intensive centralized power production. A key factor in realizing the potential of distributed systems is their operational capabilities under varying conditions and the interactions of the operation with the design configuration. In this work we present a systematic approach for the optimal design, control and operation of residential scale CHP units. Based on out recently presented PAROC framework, the approach features a detailed dynamic model describing the cogeneration of heat and power, combined with conventional and advanced control schemes. It is shown that a simultaneous approach delivers distinct energy, environmental and financial benefits.

1. Introduction

1.1. Decentralized energy systems

The sustainability of power generation, which encompasses aspects of (i) environmentally related issues, (ii) energy systems and networks configurations and design, (iii) optimal resources utilization and maintenance and (iv) financial aspects, has been a recurring area of interest for all participating shareholders; oil and gas companies (BP p.l.c., 2015), national and international policy makers (Organisation for Economic Co-operation and Development, 2008; European Commission, 2010), energy consultants and professionals (Ecuity Consulting llp, 2013; Clarke Energy, Residential CHP, 2015) and researchers (e.g. Burer et al., 2003; Pehnt, 2008; Bayod-Rújula, 2009) are all seeking for ways to reduce the impact that energy consumption has on the day-to-day as well as the industrial level. Among the alternative scenarios considered, the concept of decentralized energy production via distributed systems (presented in Fig. 1) is being considered as appealing for a variety of reasons summarized below (Bayod-Rújula, 2009; Akorede et al., 2010; Liu et al., 2013):

Efficiency: The production of several energy forms from a single intensified process results into an improved overall system efficiency. Distributed systems such as cogeneration units can produce, through a single process, the cooling, heating and electrical power requirements to a single or a series of residencies using the same amount of fuel.

Reduced power transfer related consequences: Traditionally, a central electrical power production system for covering electricity needs throughout large areas, suffers from losses associated with the transfer of power through a high voltage grid. These power losses cause (a) increased electricity production to cover for the losses (b) increased grid maintenance costs and (c) generalized outage in cases of severe grid malfunctions. Localized production of power removes most of those problems (Liu et al., 2013; Pepermans et al., 2005).

Renewable energy resources: Utilizing distributed systems offers the opportunity to include a large variety of renewable energy resources in the power production process. Solar energy, via the use of photovoltaic and solar panels, and wind turbines are the most common and widely spread. Other choices include the usage of biomass either for direct combustion (i.e. in Stirling engines, specialized boilers, etc.) or indirectly via the use of biofuels within the realm of cogeneration. The advantages posed by such an approach are mainly environmental as the greenhouse gas, sulfur and nitrogen oxide emissions are reduced (Akorede et al., 2010; Bayod-Rújula, 2009).

Other: A distributed network can act either as a supplement to the grid or as a standalone power provision solution. In the

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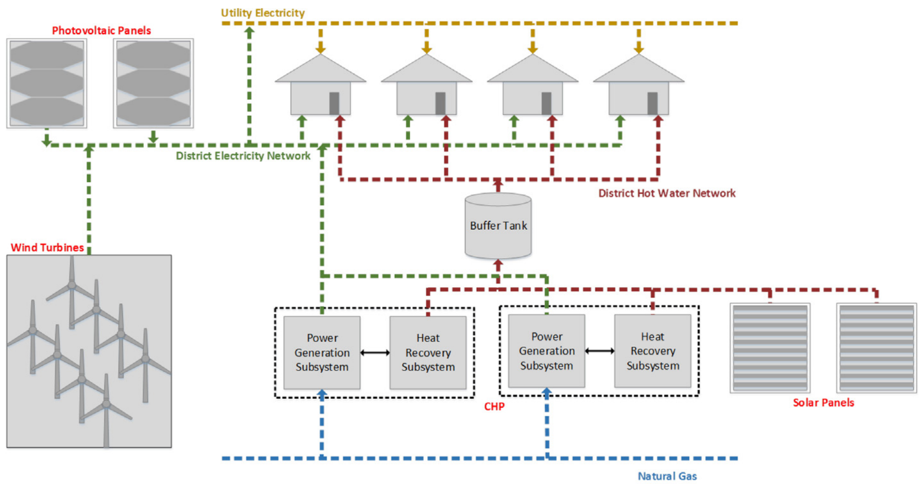
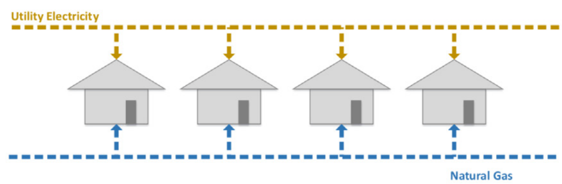


(a) Centralized system power provision



(b) Distributed system power provision

Fig. 1. Conceptual comparison of centralized via decentralized cogeneration in domestic complexes.

first case, the distributed system behaves as a backup in cases where the grid is unable to cover the demand. This may include, but is not limited to, peak demand time operation, cases of grid failure and time spans of expensive electricity provision through the central grid. In the second case, the provision of power to isolated areas is often tackled with the use of a distributed system. As an example, consider the provision of usable heat and power to a series of residencies in an isolated location, where a central power grid is not available. Based on the climatological and topological terrain, a series of alternatives can be considered, such as renewable energy sources (e.g. wind and solar power) as well as electrical generators powered by conventional means such as internal combustion diesel engines (Bayod-Rújula, 2009; Liu et al., 2013).

In this work we focus on quantifying the benefits from a combined heat and power system as part of a distributed energy system for a residential complex.

1.2. Combined heat and power systems

A sustainable, environmentally friendly and cost-effective approach to distributed systems is the concept of residential cogeneration of heat and power (CHP) (Organisation for Economic Co-operation and Development, 2008). Applicable in either a centralized or decentralized fashion (Fig. 2), it offers the advantages of a distributed system that utilizes natural gas to produce electrical power and usable heat for space and utility heating to cover the needs of an urban residential area. The in tandem operation of the unit(s) with the central electricity and natural grid networks offers the possibility of covering the electrical and heating demand from (i) the central grid, (ii) the cogeneration plant or (iii) a combination of both. Furthermore, trading energy from the cogeneration unit to the grid adds to the financial advantages of the approach, given the fact that the electrical power rates as well as the demand fluctuate throughout the day.

The use of cogeneration in the residential sector requires a series of decision making at different time scales. The design of the system is a key decision that can, in the long term, affect the environmental and financial impact of the plant. Consider a cogeneration plant that due to its limited size is unable to cover the electrical demand in times of "expensive" electrical power or one that is unable to produce heating power at the desired levels, eventually leading to discomfort within the residencies. On the other hand, a larger plant would be able to cover the demand but it would require a considerably larger investment cost thus rendering the overall acquisition and maintenance costs unsustainable. At the time scale of the system's operation, the rate fluctuations of the grid electrical power and the grid natural gas are key to the policy that is applied in order to cover the residential demand, which can be covered either from the cogeneration plant or external sources. Clearly, financial criteria prevail for these decisions, therefore the design of the system plays an important role here as well, since the financially optimal operation needs to be within the feasible operational space of the plant which depends on its design. The short term decisions should not be neglected either. The ability of a controller scheme to bring the system to the desired operating setpoints is subjected to feasibility and financially optimal operation, and also depend on the design of the plant. Process intensification is therefore necessary when considering such systems. These complex interactions between design, operations and control can be simultaneously analyzed through an integrated approach which is the subject of this work.

In particular, based on our recently presented PAROC framework (Multi-Parametric Optimisation and Control, 1999; Pistikopoulos et al., 2015) and our earlier works (Diangelakis et al., 2014, 2014, 2016) we develop an integrated framework for the simultaneous design and control optimization featuring:

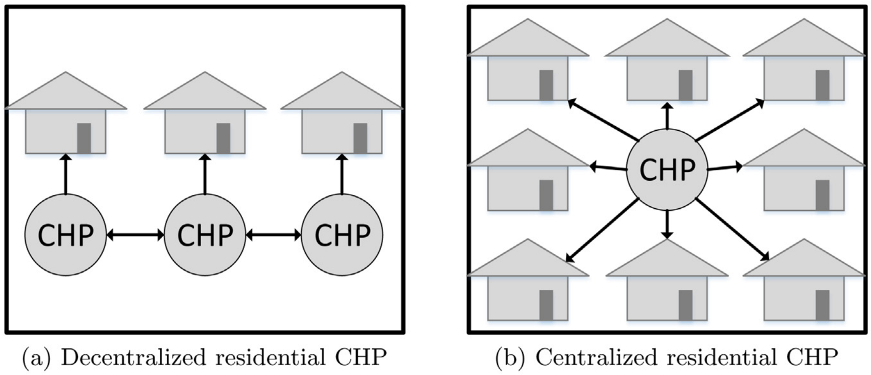


Fig. 2. Concept of centralized versus decentralized application of residential cogeneration.

1 a dynamic high-fidelity process model of the CHP system,
2 suitable approximate models that adequately capture its operation,
3 a dynamic optimization formulation that simultaneously optimizes a conventional PI control scheme and the design of the system,
4 a multi-parametric model predictive control formulation based on the optimal design of the approximate model,
5 a validation procedure of the control schemes against the original high fidelity model and
6 a comparison procedure to highlight the financial and energy benefits of the approach as well as the engineering relevance of the integrated design and control optimization approach.

The next section presents (i) the currently available residential scale CHP technology, (ii) the operational characteristics of a residential scale CHP system and (iii) the high-fidelity process model.

2. Domestic CHP model overview and operation characteristics

Cogeneration systems typically consist of a prime mover responsible for the generation of power. A large variety of prime movers can be utilized for cogeneration purposes, e.g. steam and combustion turbines, micro turbines, internal combustion engines and Stirling engines. An overview of those alternative as well as their advantages and disadvantages is presented in Table 1. The information in Table 1 is acquired through the works presented in Table 2.

Fuel cell technology is also a prime mover candidate for residential CHP applications. These quiet, compact sized generators show high electrical efficiency. In particular, proton exchange membrane fuel cells (PEMFC) are small, adjustable to power demands and resistant to corrosion. On the other hand the disadvantages of PEMFC are their high investment cost and their high sensitivity to impurities. PEMFC can be mostly exploited in small scale domestic CHP applications. The relatively new technology of the PEMFC lacks of liability due to limited operational testing for a significant amount of time but offers great advantages such as low acoustic noise and no vibration (Wu and Wang, 2006).

The of the works presented in Table 2 can be divided into two major categories based on the type of models they employ for their studies. On one hand, works that focus on the simulation and control of the CHP process focus on "high-fidelity" model development thus capturing the behavior of the system on a micro-operational level. On the other hand the works that focus on energy supply optimization treat the CHP system as part of a larger production system and therefore focus on either a simplified model or one that describes a steady-state operation (macro-operational level). In this work we incorporate the control decisions (micro-scale) into the design optimization of the system, a decision corresponding to the macro-scale as it remains fixed throughout its lifespan. In our previous works we have argued about the need to simultaneously considered the decisions in several levels (design of the system, scheduling of operation and optimal control) simultaneously in order to ensure minimum acquisition and operational cost without shortages (Pistikopoulos and Diangelakis, 2015).

In this paper, a natural gas fueled, internal combustion engine based cogeneration system is considered that produces electrical power and usable heat for space and utility water heating. The choice of the internal combustion engine as the prime mover is based on the broad use of such systems in real life applications. Furthermore, internal combustion engines are able to work with a variety of fuels compared to the premium fuels that other technologies require (Heywood, 1989). A well-known practice involves the modification of an existing internal combustion engine design to operate with various fuels, upon availability, an advantage for the residential sector, especially when remote areas are considered. A PEM fuel cell based CHP can be a viable alternative for the domestic sector application given its quiet operation. The operational range of turbines (steam or combustion) typically exceeds the needs of a residential area and is more suitable for an industrial setting, while micro-turbines technology is not yet mature for an operation of a 25-year lifespan (Diangelakis et al., 2014). A typical mathematical model developed in our previous works is presented in Table 3 (see Diangelakis et al., 2014 for details). The model considers every component of a domestic CHP individually. The behavior of each component is modeled based on first principles and correlations.[1] The interactions between the components result into a non-linear dynamic process model without spatial derivatives.

The operation of every cogeneration system is twofold. The CHP system considers two separate subsystems, the electrical power production subsystem and the heat recovery subsystem (Diangelakis et al., 2016). As a result, any operating policy for the CHP unit should be able to account for the interactions of the two subsystems and the requirements to produce both electrical power and heat (in the form of hot utility water).

Due to the restrictive power to heat ratio a certain electrical power operating point of the CHP dictates the maximum amount of heat recovered from the system operation. Equivalently, a certain heat production operating point dictates the operating level of the CHP system in terms of electrical power, therefore the power production level cannot be determined independently. This results into the two following modes of operation (Diangelakis et al., 2016):

[1] The correlations are mainly used for the derivation of the thermodynamic efficiency of the internal combustion engine and the characterization of the combustion.

Table 1
Prime mover technology for CHP systems.

Prime mover	Advantages		Disadvantages
Steam turbines	Reliability, long life span		Low electrical efficiency, slow start-up, poor partial load performance
Internal combustion engines	Reliable, low capital cost, fast dynamics, high efficiency in partial loads		High vibrations and emissions, acoustic noise, high maintenance cost, diverse temperature levels
Combustion turbines	vs. steam turbines	vs. ICE	Premium fuel, high capital investment, ambient conditions dependency
	Easy installation, less area intensive, lower capital cost		Lower maintenance, lower emissions
Micro turbines	General	vs. ICE	Unavailability, high investment cost, low electrical efficiency, ambient conditions dependency
	Fuel variety, low maintenance		Lower temperatures, lower NO_x emissions, less noise
Stirling engines	Fuel variety, low emissions, low noise, high efficiency, reduced vibration		Development phase, Limited operational data and testing, limited availability, high investment cost

Table 2
Small scale cogeneration modelling and power supply review – indicative list.

Authors (year)	Contributions
Míguez et al. (2004), Porteiro et al. (2004)	Design and development of domestic trigeneration systems
Savola and Keppo (2005), Savola and Fogelholm (2007)	Small scale CHP model development and MINLP formulation maximizing production objectives
Wu and Wang (2006)	Review on combined cooling, heat and power cogeneration systems
Onovwiona and Ugursal (2006)	Technology review on heat and power cogeneration systems
Videla and Lie (2006)	Modelling and dynamic simulation of small scale internal combustion based CHP systems
Cockroft and Kelly (2006)	Assessment of power sources for cogeneration
Dorer and Weber (2009)	Energy, environmental and performance studies on PEMFC, SOFC, stirling engine and internal combustion engine based residential cogeneration
Collazos et al. (2009)	Optimal control with mixed integer formulation and stochastic elements for micro generation in domestic applications
Konstantinidis et al. (2010)	Multi-parametric model predictive control of small scale CHP systems suitable for domestic/residential use
Diangelakis et al. (2014)	First principle modeling and design optimization of an internal combustion based residential scale CHP system
Menon et al. (2013)	Optimal design of polygeneration systems, including CHP technologies, under optimal control assumptions
Fazlollahi et al. (2012)	Multi-objective optimization techniques for the optimal design of complex energy systems, including CHP technologies
Mehleri et al. (2011, 2012)	Distributed energy systems optimization, based on superstructure approach
Kopanos et al. (2013), Kopanos and Pistikopoulos (2014)	Planning and reactive scheduling via multi-parametric programming of a network of micro-CHPs
Ondeck et al. (2015)	CHP as a viable means for providing district-level cooling, heating, and power to a residential district in a hot climate

Table 3
Residential CHP model overview. See Appendix A for Nomenclature.

Piece of equipment	Equations
Throttle valve – fuel and air manipulation	$\dot{m}_{th} = c_d A_{th} \frac{P_{ab}}{\sqrt{R_\beta T_{ab}}} \psi \left(\frac{P_{ab}}{P_{mn}}\right)$
Manifolds – pressure difference driven flow for the inlet air and exhaust gases	$\frac{d}{dt} m_{mn} = \dot{m}_{mn,in} - \dot{m}_{mn,out}$
	$\dot{V}_{mn,out} = cpf\,(P_{mn} - P_{mn,out})$
	$\dot{V}_{mn} \frac{d}{dt} E = \dot{m}_{mn,in} h_{mn,in} - \dot{m}_{mn,out} h_{mn,out}$
Internal combustion engine – energy and mass balances	$\dot{m}_{ex} = \frac{P_{mn}}{R_\beta T_{mn}} \frac{V_d}{4\pi} \eta_{vl} \omega_{en}$
	$\dot{m}_{mn,out}\left(h_{mn,out} + \sum_{i=aircomp.}\left(x_{air,i} h^o_{f,air,i}\right)\right) + \dot{m}_\phi\left(h_\phi + \sum_{j=fuelcomp.}\left(x_{\phi,j} h^o_{f,fuel,j}\right)\right)$
	$- \dot{m}_{ex}\left(h_{ex} + \sum_{k=exhaustcomp.}\left(x_{ex,k} h^o_{f,ex,k}\right)\right) = \dot{Q}_f + \dot{Q}_{cg\to cw} + \dot{W}_c + \dot{W}_{en}$
Crankshaft – torque generation	$To_{en} = \frac{P_{meb} V_d}{4\pi}$
Generator – power generation through torque	$\frac{d}{dt} To_{cl} = \frac{1}{Fl}(To_{en} - To_{cl})$
	$Pec = \eta_{en} To_{cl} \omega_{en}$
Engine cooling system – energy balances	$\frac{d}{dt} T_i = \frac{\dot{Q}_{in,i} - \dot{Q}_{out,i}}{m_i c_{p,i}}$
	$\forall i \in$ engine cooling system components
	$\dot{Q}_{a\to b} = TC_{ab} A_{ab} (T_a - T_b)$
	$\forall a, b \in$ engine cooling system components
Heat exchangers – energy balances	$Q = UA \Delta T_{mean}$
	$\Delta T_{mean} = \frac{\Delta T_{in} - \Delta T_{out}}{\log\left(\left|\frac{\Delta T_{in}}{\Delta T_{out}}\right|\right)}$

Operation mode 1: The power production driven operation corresponds to the operation during which a certain electrical output is produced but the flow rate of the water is such that the produced hot water is at a certain temperature, typically at 70 °C for residential applications. In this mode of operation the main product is the electrical power.

Operation mode 2: The heat recovery driven operation ensures the production of a certain flow rate of hot water at a certain temperature. During this mode of operation the production level of the total power requirements may not be guaranteed. The main product in this case is the hot water.

It becomes clear that the design characteristics of the CHP unit are of vital importance to the operation of the plant. The size of the prime mover determines the maximum amount of electrical power that can be produced from its operation as well as the maximum amount of hot water. Furthermore, the size of the prime mover affects the thermal and electrical efficiency of the plant. Therefore, different designs may result into different operational characteristics with different economies. Subsequently, it is clear that the design of the system affects it in a manner that makes the simultaneous consideration of design operations and control optimization schemes necessary.

3. Design and control of the domestic heat and power system

The interactions of design and control have been investigated in detail in the past (e.g. Tong et al., 2014; Gutierrez et al., 2014; Mehta and Ricardez-Sandoval, 2016; Washington and Swartz, 2014; Baldea, 2015; Wang and Liu, 2013; Pistikopoulos and Diangelakis, 2015 and references therein). As a method of process intensification it provides an in depth understanding of the interactions between the design characteristics of a process and its control and operational principles. In this work two approaches are investigated in terms of the integration of design and control. The first approach involves the simultaneous design and PI control of the CHP system based on an approximate model. The second approach enhances the first by designing an advanced model based controller for the optimally designed CHP. The steps of the procedure are explained below:

State-space model approximation: The procedure is initiated by approximating the original high-fidelity model of Table 3. The System Identification toolbox of MATLAB© is utilized in order for adequately accurate linear state space models to be identified. The approximation step is based on the operational principles of the CHP system that employs the two-subsystem approach, i.e. two linear state-space models are approximated corresponding to the power generation subsystem and heat recovery subsystem (Eqs. (1) and (2) respectively). For the next steps of the approach, it is key to preserve the correlation of the system output and the design of the system. Eq. (1) shows that V_d, the design variable is part of the approximate model formulation. Note that the correlation of the design variable to Eq. (2) is indirect, via the power generation level *Pec*.

Design and PI control optimization: Based on the approximate models of the previous step a mixed integer dynamic optimization problem is formulated. The MIDO optimizes the design of the CHP system as well as the gains of the PI controllers. Furthermore, based on electrical power and heat demands (fixed throughout the system's operation[2]) and the rate of electrical power and natural gas acquisition from a central grid (lower electrical power rated during nigh-time) a discrete decision is made regarding the operating mode of the CHP, i.e. Operating Mode 1 or Operating Mode 2. The latter affects the controller scheme of the system and which operating setpoint will be tracked (power generation or heat recovery). The outcome of this step is the optimal design of the CHP system and the optimal control scheme based on whether the CHP system operates during nigh-time or daytime.

Multi-parametric model predictive control: The optimization results of the previous step are utilized to apply a mp-MPC scheme on the system. The optimal design and the optimal switching time is used to design and apply the advanced controllers on the CHP. For the design of the controllers the approximate models are utilized (for details on the approach see Diangelakis et al., 2016).

Validation and scheme evaluation: In this step we apply the optimized PI and mp-MPC controllers on the high-fidelity model using the optimal design. We compare the performance of the system in terms of its energy and financial impact. The comparison takes place using a scenario where the entirety of the demand would be covered by conventional methods (i.e. electrical provision from the grid and heat provision from a natural gas thermal boiler). We also consider two cases where the operational information derived from the CHP model was unavailable, i.e. we compare the operational scheme that the operating mode alternation provides to the cases where either (i) only Operating Mode 1 is available or (ii) only Operating Mode 2 is available.

3.1. State-space model approximation

Adequately precise approximate models are key to this approach. The choice of linear state-space models enables the direct application of multi-parametric programming techniques that are applied further on in this work. Eqs. (1) and (2) present the approximate models for the power production subsystem and the heat recovery subsystem, respectively, as derived from the System Identification Toolbox of MATLAB©:

$$\begin{aligned}\frac{d}{dt}x &= -0.2029\,x(t) + 0.06393\,u_{th}(t) + 6.582\times 10^{-7}\,V_d \\ Pec(t) &= 7.839\,x(t)\end{aligned} \tag{1}$$

u_{th} is the discrete throttle valve control signal, *Pec* is the electrical power generated from the power generation subsystem and V_d is the displacement volume of the internal combustion engine, in this case treated as a measured uncertainty. By manipulating the discrete throttle valve position, the amount of air and fuel that enters the CHP system is affected, therefore a different production level of electrical power is achieved.

$$\begin{aligned}\frac{d}{dt}x &= \begin{bmatrix}-1.401 & -0.01659\\ -1.316 & -2.282\end{bmatrix}x(t) + \begin{bmatrix}0.129\\ -0.3904\end{bmatrix}Pec(t) + \begin{bmatrix}1.121\\ 5.467\end{bmatrix}u_{p_2}(t)\\ T_{water}(t) &= \begin{bmatrix}4.584 & 36.27\end{bmatrix}x(t)\end{aligned} \tag{2}$$

u_{p_2} is the water flow rate of the external circuit (heat recovery subsystem) and T_{water} is the temperature of water at the outlet of the heat recovery subsystem. Based on the mode of operation of the CHP unit, *Pec* is treated as a control variable when the unit is operated in a hot water production driven mode and u_{p_2} is treated as uncertain. The value for the latter is determined by the hot water

[2] Note that we assume a steady state demand in this work. It is clear though, from our previous works, that an operating setpoint for the CHP is correlated to fluctuating demands and electricity rates and is subjected to mid-term optimization techniques and their interactions with the current design and control approach (Kopanos and Pistikopoulos, 2014).

demand. The exact opposite holds true for a power production driven mode of operation.

3.2. Simultaneous design and PI control optimization

The approximate models derived in the previous section for the two subsystems are used to design and solve the simultaneous equipment design and control optimization. The outcome of the optimization is the derivation of:

Outcome 1 the size of the internal combustion engine displacement volume V_d.
Outcome 2 the gains of the PI controllers for the CHP system.
Outcome 3 the mode of operation of the CHP system based on the electrical power rates and assuming fixed electrical and heat demands.

Note that unlike **Outcome 3**, **Outcomes 1** and **2** are decisions that remain fixed throughout the life-cycle of the plant. The mode of operation is subject to change based on several criteria. Most residential scale CHP units are operated in a heat production driven manner thus treating the electrical power as a by-product. We attempt to show that such an approach is rather limited and underestimates the potential of the CHP unit, from an engineering point of view. In order to ensure that outage in electrical power or hot water is prevented, we assume that the CHP unit is operated in a residential environment where the houses are connected to the electrical power grid as well as a central natural gas network. Potentially insufficient power or hot water generation is therefore covered by utilizing the central grid utilities. Similarly, excess electrical power is traded to the grid but excess hot water is discarded at a cost.[3] The following subsections describe the PI control scheme used in this work and the handling of the discrete component of the CHP system operation.

3.2.1. Conventional PI control

The CHP system is controlled by classical PI controllers. Three controllers are utilized.

1 The controller for the power generation subsystem manipulates the throttle valve opening to produce electrical power of a certain setpoint.
2 The controller for the heat recovery subsystem:
- manipulates the water flow rate of the external circuit of the subsystem to produce water of a certain temperature setpoint while the power generation subsystem follows the an electrical demand or
- it manipulates the required electrical power in such a manner so that a temperature setpoint is met for a water stream of a certain flow. In that case the electrical power demand is dictated by the PI controller of the heat recovery subsystem.

3 A third PI controller is required for the heat recovery driven operation that controls the water flow in the external circuit.

The general form of the PI controller used in this work is presented in Eq. (3).

$$u(t) = K_p e(t) + K_i \int_0^t e(\tau) d t \tag{3}$$

where $u(t)$ is the control variable, $e(t)$ is defined as the deviation of the output compared to a setpoint. K_i and K_p are the design parameters of control and are subjected to optimization. In the case of the power generation subsystem the control variable is the throttle valve opening and the output is the power generation level. In the case of the heat recovery subsystem the output is the temperature of the water while the control variable is (i) the water flow rate in the power generation driven operation or (ii) the power generation level setpoint in the heat recovery driven operation.

3.2.2. Cost function

The cost function of the dynamic optimization problem takes into consideration several aspects.

- The annualized cost of acquisition of the CHP plant is a function of the size of the internal combustion engine. For this, a 25 year life time has been assumed.
- The cost of electricity from the grid is a function of time. In particular, electricity from the grid is cheaper to consume during night time than day time. The difference can be up to 30% according to the regulations of the electricity provider.
- The cost of the natural gas does not fluctuate with time. Note that typically an internal combustion engine with a larger displacement volume consumes more fuel for a certain valve opening setting than an engine with a smaller displacement volume. Furthermore, the size of the engine affects the thermal and electrical efficiency of the plant. Therefore a larger engine could be more useful for a standalone system whereas a smaller one could be more suitable for every-day mild use.
- The trading of electricity with the grid is taken into consideration in the objective function. Fluctuation to the price is also taken into consideration.
- Scaled terms that ensure the minimization of the error associated with the control formulation are also present in the objective function formulation.

Following these points the objective function is of the form of Eq. (4).

$$\begin{aligned} f(V_d, K_p, K_i, y_{relax}, t) = {} & \text{Cost of equipment acquisition} \\ & + \text{Cost of production} + \text{Revenue of trading} \\ & + \text{Scaled deviation penalty} \end{aligned} \tag{4}$$

The overall form of the design and control optimization problem is presented in Eq. (5).

$$\begin{aligned} & \min_{d(t), K_p, K_i, y_{relax}(t)} \quad J = \int_0^t f(d(\tau), K_p, K_i, y_{relax}(\tau), \tau) dt \\ & \text{s.t.} \quad \frac{d}{dt} x = Ax(t) + Bu_{(t)} + Cd_{(t)} \\ & y(t) = Dx(t) \\ & u(t) = y_{bin}(t)(u(t) = K_p e(t) + K_i \int_0^t e(\tau) dt) \\ & e(t) = y_{sp}(t) - y(t) \\ & y_{bin}(t) = \tanh(M y_{relax}(t)) \\ & 0 \le K_p \le K_{p,\max} \\ & 0 \le K_i \le K_{i,\max} \\ & 0 \le y_{relax}(t) \le 1 \\ & d_{\min}(t) \le d(t) \le d_{\max}(t) \end{aligned} \tag{5}$$

where $x(t)$ are the states, $u(t)$ are the control variables, $y(t)$ and $y_{sp}(t)$ are the outputs and output setpoints respectively, $d(t)$ are the disturbances, $e(t)$ are the setpoint deviations, $y_{bin}(t)$ are the binary

[3] In future works, the utilization of a hot water storage tank is worth investigating in.

variables, y_{relax} are the relaxed variables, K_i and K_p are the integral and proportional gains for the controllers and M a sufficiently large number. In the case of the CHP system the optimization problem is set based on the systems defined in Eqs. (1) and (2). Note that the displacement volume of the internal combustion engine is a design variable that remains fixed throughout the optimization horizon. The same holds true for the gains of the PI controllers.

The objective of this optimization problem is to show the significance of the dual control approach and the effect of the design. Furthermore, the size of the internal combustion engine displacement volume is expected to affect not only the control setting of the degrees of freedom of the system but the ratio of electricity and heat that is produced from the CHP.

3.3. Multi-parametric model predictive control

We use the state-space models that are presented in Eqs. (1) and (2) to design multi-parametric Model Based Controllers for the CHP system. At this stage, the value for the design variables is fixed to the value provided by the simultaneous design and control optimization of the previous section. Furthermore, the operating strategy follows the optimal strategy that was derived in the previous section. The formulation of Eq. (6) describes the mp-MPC problem at hand.

$$\begin{aligned} \min_{x,y,u} \quad & J = x_N^T P x_N + \sum_{k=1}^{N-1} \left(x_k^T Q_k x_k + \left(y_k - y_k^R\right)^T QR_k \left(y_k - y_k^R\right) \right) \\ & + \sum_{k=0}^{M-1} \left(\left(u_k - u_k^R\right)^T R_k \left(u_k - u_k^R\right) + \Delta u_k^T R1_k \Delta u_k \right) \\ \text{s.t.} \quad & x_{k+1} = A x_k + B u_k + C d_k \\ & y_k = D x_k + E u_k + e \\ & u_{\min} \le u_k \le u_{\max} \\ & \Delta u_{\min} \le \Delta u_k \le \Delta u_{\max} \\ & x_{\min} \le x_k \le x_{\max} \\ & y_{\min} \le y_k \le y_{\max} \end{aligned} \tag{6}$$

The mp-MPC problem formulation is set for the discrete time counterparts of the subsystems described in Eqs. (1) and (2). For instance, when the power generation subsystem controller is formulated the expression $x_{k+1} = A x_k + B u_k + C d_k$ corresponds to the discretized expression $\frac{d}{dt}x = -0.2029\, x(t) + 0.06393\, u_{th}(t) + 6.582 \times 10^{-7}\, V_d$ of Eq. (1) for a time-step $\tau_s = 0.1$ s. The algebraic counterpart of the state space representation $y_k = D x_k + E u_k + e$ corresponds to $Pec(t) = 7.839\, x(t)$ of the same equation. Similarly to the PI approach, three mp-MPC controllers are formulated for the power generation subsystem, the heat recovery subsystem and the water flow.

The design parameters for all the controllers are reported in Appendix B.

4. Scheme evaluation

The system is optimized for operation corresponding to the following production characteristics:

- The CHP system can either operate on an electricity driven operation producing 2.8 kW or on a heat recovery driven operation producing 0.1 kg/s.
- In cases where the production cannot match the aforementioned demands in either mode of operation (production below demand), the mismatch is met by utilizing grid resources.
- It is possible that an electricity driven operation mode results in heat recovery higher than the demand and vice versa. The extra electrical power is sold to the electricity grid, while the extra hot water production is penalized.
- The optimal design, as a result of the simultaneous design and PI control optimization, is defined in the high fidelity model. The latter is simulated using both the PI control scheme and the mp-MPC scheme.
- The operation follows the production modes resulting from the dynamic optimization.
- The two control schemes are compared with each other in terms of fuel and cost savings and with the case where the demand is met through the grid.

Table 4
Relative energy and cost savings comparison among the three alternatives.

CHP	PI control	mp-MPC control
Operating cost	12.2%	8.8%
Fuel consumption	11.95%	12.2%
Investment cost	4988.82 m.u.	4988.82 m.u.
Total savings (10 yr)	16495 m.u.	16897 m.u.

Table 4 shows the comparison between the different alternatives. The results of this approach yield an internal combustion engine volume of approx. 1520 cc and a change in the operating mode based on the price fluctuation of electric power. More specifically, during daytime, when the acquisition of electrical power from the grid is more expensive, a power generation driven approach is chosen, while a heat recovery approach is preferred during night time. The total savings shows the amount of monetary units saved assuming a ten year operation of the CHP plant. The relative analysis is performed on the basis that 100% corresponds to the case where the electrical power and heating demands are covered without the use of a CHP system. In that case the investment cost is assumed equal to zero.

The total savings have been calculated based on a ten year operation of the system at hand and after considering the attenuation of the initial investment.

Throughout this work we have considered alternating operating modes of the CHP system. It is worth investigating the impact that this approach has on the operation. For this purpose we also consider operation of the system under the same demand (and for the optimal design) but in this case we operate the system in different ways, considering the following three scenarios:

Scenario 1: An alternating operating mode approach is considered where the system operates under the electrical power driven operating mode in times of high electricity rates and under heat recovery driven mode otherwise.
Scenario 2: An electrical power driven mode is applied. The system treats the heat as a byproduct.
Scenario 3: A heat recovery driven mode is applied. The system treats the the electrical power as a byproduct.

Table 5 summarizes the results. **Scenario 1** corresponds to the base case for which the relative comparison that takes place. It is

Table 5
Relative energy and cost savings comparison when alternating operating modes are considered.

	Scenario 2 Power generation driven	Scenario 3 Heat recovery driven
Operating cost	+112.61%	+123.295%
Fuel consumption	+7.60%	+7.45%
Total savings (10yr)	−22.8%	−25.1%

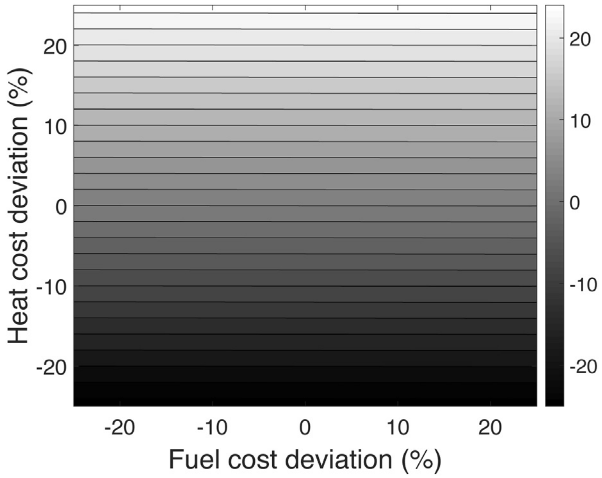


Fig. 3. Simulation driven operational cost sensitivity analysis for fuel and heat cost fluctuations.

clear that when only one mode is considered both the fuel consumption and the operating cost of the system are higher than when an alternating approach is applied. As a direct result of this, the total savings in a ten-year period of operation are also affected.

The fluctuation of the cost of (a) discarding heat or utilizing a complementary boiler, (b) fuel and (c) electricity are factors that can affect the economical benefits presented in this paper. In order to identify which of those factors has a greater impact on the economics of the optimally designed system the following simulation based sensitivity analysis was performed:

- The optimally designed CHP system was operated for 24 h.
- The full scale model was utilized for that purpose, under the mpMPC control described in Appendix B.
- The electricity driven operation aimed at a 2.8 kW production while the heat recovery driven operation for a 0.1 kg/s of water at 70 °C.
- Heat recovery driven operation was assumed during night time.
- The electricity pricing scheme involved reduced night cost.
- The cost of heat, fuel and electricity were fluctuated by ±25% from the nominal cost.

It was shown that the electricity price fluctuation had a negligible effect on the overall production cost that did not exceed ±0.5%, regardless of the fluctuation of the two other factors. This is attributed to the ability of the power subsystem controller that is able to follow the demand accurately. The fluctuation of the fuel price was also small but managed to exceed ±1% in extreme cases. On the contrary to the two previous cases, the fluctuation of heat cost was the most deterministic as it affects the economic operation of the system the most. Fig. 3 shows the contour of the fluctuation of heat cost and fuel cost. The dependence on the heat cost on the *y* axis is obvious compared to the less obvious change along the *x* axis, showing the fuel cost fluctuation.

5. Conclusion and future work

In this work we demonstrated the application of the PAROC framework for the intensification of a domestic CHP process via techniques for design and control optimization. We showed how the proposed approach manages to significantly reduce the operating costs of such a unit as well as its positive energy impact. The simultaneous approach to design and PI control optimization was followed by the design of a mp-MPC. Simulation results of the optimized system, operated by PI controllers and mp-MPC controllers, were compared against each other and against the conventional case were only conventional electrical power and heat was used. Furthermore, we considered different operational strategies for the CHP system and showed the engineering relevance of the dual-mode operation of the CHP versus a monolithic approach of either only power generation driven operation or only heat recovery driven operation.

In future work, the PAROC framework will be used to simultaneously optimize the design of the CHP system while the latter is controlled by a design dependent mp-MPC. The problem described in this work assumes that the CHP system operates under fixed operating conditions throughout the day and in either one out of two modes. Note that, in the cases where the dynamic optimization of the CHP system started under a cold start, i.e. from rest, the results yield a heat recovery driven approach *regardless of the price of the electricity*. This is attributed to the overall behavior of the system to reach steady-state in that mode in less cost. A different steady-state could yield different results. As a result, an overall operational optimization of the CHP system would require to take into account the effect that the electrical power and heat real-life demands would have on the system. Furthermore, uncertainty on the cost of fuel, heat and electricity form external sources needs to be taken into consideration. Therefore, the incorporation of a mid-term operational optimization approach, such as a reactive scheduling formulation, would reveal the true energy and financial savings as well as the engineering relevance of such integrated approaches.

Acknowledgments

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Appendix A. Nomenclature for the model overview of Table 3

See Table A.6.

Table A.6
Nomenclature

Symbol	Physical meaning
m_{th}	mass of air entering the throttle valve
c_d	valve discharge coefficient
A_{th}	open area of the throttle valve
P_{ab}	ambient environment pressure
R_β	ideal gas constant
T_{ab}	ambient environment temperature
$\psi\left(\frac{P_{ab}}{P_{mn}}\right)$	laminar and turbulent flow indicator
m_{mn}	mass of fluid in the manifolds
$m_{mn,in}$	mass of fluid entering the manifolds
$m_{mn,out}$	mass of fluid exiting the manifolds
$V_{mn,out}$	volume of fluid exiting the manifolds
cpf	pressure – flow coefficient
P_{mn}	pressure in the manifolds
$P_{mn,out}$	pressure at the outlet of the manifold
V_{mn}	volume of fluid in the manifold
E	internal energy

Table A.6 (*Continued*)

Symbol	Physical meaning
$h_{mn,in}$	mass specific enthalpy of fluid entering the manifold
$h_{mn,out}$	mass specific enthalpy of fluid exiting the manifold
m_{ex}	mass of exhaust gasses
T_{mn}	temperature in the manifold
V_d	ICE displacement volume
η_{vl}	volumetric efficiency
ω_{en}	ICE angular velocity
$x_{air,i}$	mass fraction of air components
$h^o_{f,air,i}$	mass specific enthalpy of formation of air components
m_ϕ	mass of fuel
h_ϕ	enthalpy of fuel
$x_{\phi,j}$	mass fraction of fuel components
$h^o_{f,fuel,j}$	mass specific enthalpy of formation of fuel components
h_{ex}	enthalpy of exhaust gases
$x_{ex,k}$	mass fraction of exhaust gasses components
$h^o_{f,ex,k}$	mass specific enthalpy of formation of exhaust components
Q_f	heat of friction
$Q_{cg\rightarrow cw}$	heat from cylinder gaskets to cylinder walls
W_c	work of compression
W_{en}	usable ICE work
To_{en}	usable ICE torque
P_{meb}	mean effective break pressure
To_{cl}	flywheel torque
Fl	flywheel inertia
Pec	electric power
η_{en}	ICE efficiency
T_i	temperature of system equipment *i i*
$Q_{in,i}$	heat transferred into system equipment *i*
$Q_{out,i}$	heat transferred out of system equipment
m_i	mass of system equipment *i*
$c_{p,i}$	mass specific heat capacity of system equipment *i*
$Q_{a\rightarrow b}$	heat transfer from system component *a* to *b*
TC_{ab}	heat transfer rate coefficient from ICE to ambient environment
A_{ab}	area of heat transfer from ICE to ambient environment
T_a	temperature of fluid *a*
T_b	temperature of fluid *b*
UA	overall heat transfer coefficient multiplied by the area of the heat exchangers
ΔT_{mean}	logarithmic mean temperature
ΔT_{in}	temperature difference between the inlet streams of the heat exchangers
ΔT_{out}	temperature difference between the outlet streams of the heat exchangers

Appendix B. Controller designs and dynamic system behavior

Here we report the optimal controller design and demonstrate the behavior of the system under both PI and mp-MPC control through gPROMS©. In particular:

- Table B.7 reports the controller gains for the PI control scheme per subsystem and operating mode, which is presented in Eq. (3).
- Tables B.8 and B.9 reports the parameters of the optimization problem associated with each mp-MPC, which is presented in Eq. (6).
- Fig. B.4 presents the response of the optimally designed CHP system under (a) PI control and (b) mp-MPC control.

Table B.7
PI controller configuration for all subsystems for 1520 cc engine displacement volume.

	Power generation subsystem	Heat recovery subsystem	
		Power driven	Heat driven
Kp	9987.1	36688.63	988.1965
Ki	9840.5	100000	984.1965

Table B.8
mp-MPC controller configuration for the power generation subsystem.

mp-MPC design parameters	Value
N	4
M	3
$QR_k, \forall k \in \{1, \ldots, N\}$	31.25
$R_k, \forall k \in \{1, \ldots, M\}$	1
$R1_k, \forall k \in \{1, \ldots, M\}$	0.1
x_{min}	0
x_{max}	5
u_{min}	0
u_{max}	1
y_{min}	0
y_{max}	11
d_{min}	1500
d_{max}	5000
Δu_{min}	0.1
Δu_{max}	−0.1

Table B.9
mp-MPC formulation data for the heat recovery subsystem.

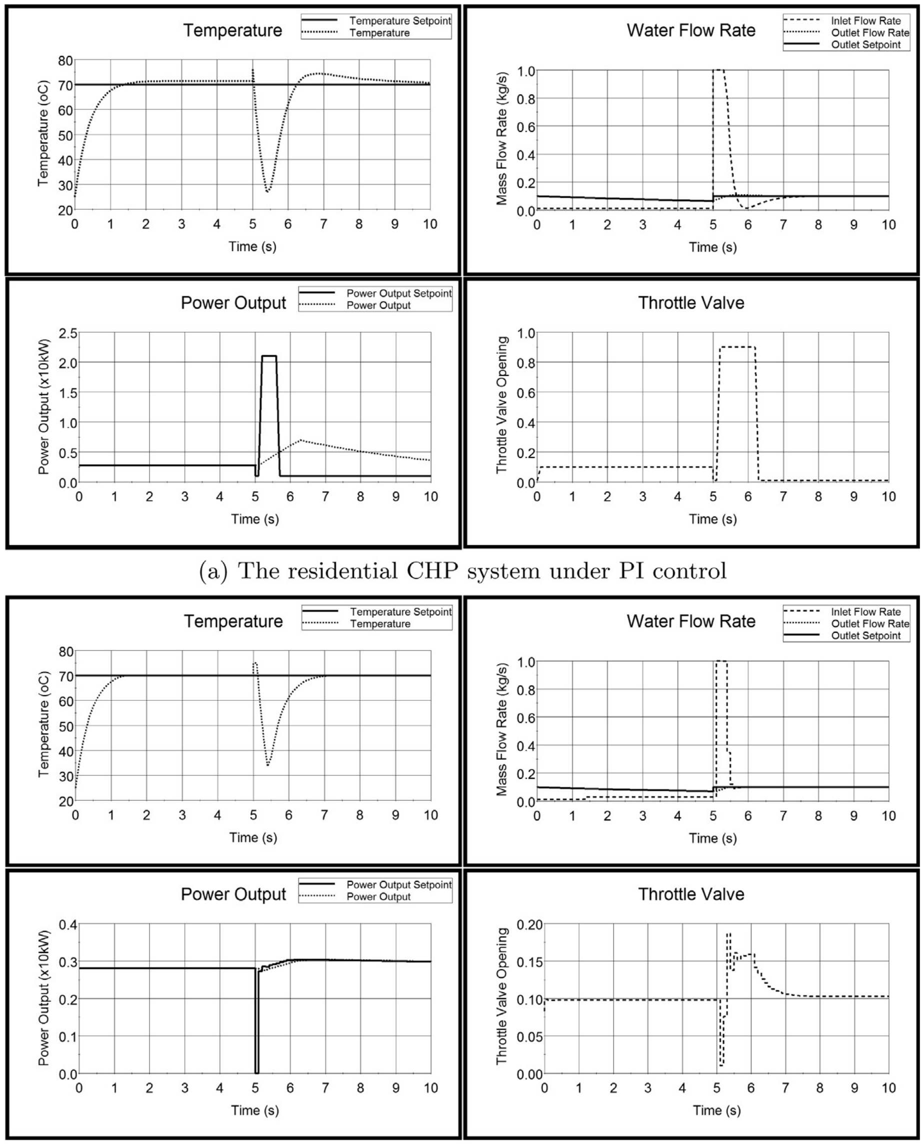
Power generation driven		Heat recovery driven	
mp-MPC design parameters	Value	mp-MPC design parameters	Value
N	3	N	2
M	3	M	2
		$Q_k, \forall k \in \{1, \ldots, N\}$	1
$QR_k, \forall k \in \{1, \ldots, N\}$	10000	$QR_k, \forall k \in \{1, \ldots, N\}$	10
$R_k, \forall k \in \{1, \ldots, M\}$	1	$R_k, \forall k \in \{1, \ldots, M\}$	10
$R1_k, \forall k \in \{1, \ldots, M\}$	10	$R1_k, \forall k \in \{1, \ldots, M\}$	10000
x_{min}	$[-1, -3]^T$	x_{min}	$[-1, -3]^T$
x_{max}	$[1, 3]^T$	x_{max}	$[1, 3]^T$
u_{min}	0	u_{min}	0
u_{max}	5	u_{max}	11
d_{min}	0	d_{min}	0
d_{max}	11	d_{max}	5
y_{min}	0	y_{min}	0
y_{max}	100	y_{max}	100
Δu_{min}	0.1	Δu_{min}	0.01
Δu_{max}	−0.1	Δu_{max}	−0.01

Appendix C. MIDO formulation – handling the discrete alternatives

In order to handle the discrete alternatives in the MIDO problem a smooth approximation is used similarly to Bansal (2000). According to this a function is used to relax the binary variables to continuous. It is well known that such an approach can settle for relaxed values that are different than 0 or 1. The formulation we use in this work though is free of such a constraint. The underlying idea of such an approximation is presented in Eq. (C.1).

$$y = \tanh(My_{relax}) \tag{C.1}$$

where $y \in \{0, 1\}$ and $y \in [0, 1]$. M is a large number. The function in Eq. (C.1) settles to 0 when $y_{relax} = 0$ and to 1, otherwise. The key for the successful application of the method is the choice of M. As mentioned before, the entire optimization problem is handled by gPROMS©. By setting the effective zero of the optimization algorithm of the software to a value of 10^{-x}, where x is a positive number and $M = 10^x$ we achieve the following: the only values of y_{relax} that would result into a value of y other than 0 or 1 are in the scale of 10^{-x}. Such values of x though are treated as zero by the optimization algorithm because of the effective zero setting. In this work $x = 8$, therefore $M = 10^8$ and the effective zero is set to 10^{-8}.



(a) The residential CHP system under PI control

(b) The residential CHP system under mp-MPC control

Fig. B.4. 10 s operation of the CHP system under PI and mp-MPC control. Mode swift occurs at 5 s.

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