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DESIGN OF COGENERATION: A CASE STUDY OF AN APARTMENT BLOCK

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ABSTRACT

Cogeneration has a great potential to save primary energy in buildings. However, these savings, and their associated economic benefits, are influenced by the design of the complete heating system, including a back-up boiler and storage tank. Also the hydronic configuration should be chosen, of which, in general, two typologies can be defined: serial and parallel. In this paper, both typologies are analysed and for the parallel configuration two different control strategies are compared. On top of that, a sensitivity analysis is performed on the storage tank size and CHP size. Simulations are used as a methodology for the comparison, which is based on a case study of an apartment block with 24 apartments. It is shown that the hydronic configuration is a crucial aspect in a design process and is, for the parallel configuration, substantially affected by the control strategy.

KEYWORDS: hydronic design, cogeneration, building system simulations

1.INTRODUCTION

1.1 Problem statement

The last decades cogeneration expanded its field of application; the technology is not only useful in typical domains like industrial processes and hospitals. Indeed, it also plays a growing role in energy-efficient and sustainable heat and electricity production in the residential building sector [13]. This makes sense, as the usage of these both forms of energy (i.e. heat and electricity) are often requested simultaneously in residential buildings. Also, depending on the behaviour of the inhabitants, this consumption might be somewhat synchronised with the typical peak demands of the electricity grid during morning and evening hours. Therefore, the share of electricity that is generated by the CHP, but not locally consumed, has a stabilising effect on the grid [8].

The efficiency of a complete heat production system with a CHP depends largely on its design. Strategies to develop a particular design were documented multiple times, e.g. [12], [10], [11], [9]. In general, these design processes include the following steps: quantifying the energy demand over time, sizing the different components (CHP, storage tank and auxiliary boiler) and choosing a control strategy.

However, in a recent project [3] it was shown that another design aspect is crucial for the total production efficiency: the *hydronic configuration*. This term refers to the way the different components are connected by pipes, pumps and valves. In literature and design guides this aspect is overlooked. Two general options for the hydronic configuration exist: serial and parallel (see Fig. 1). In previous work [16], these two options have been compared. It was shown that a serial configuration has a positive effect on the operating time of the CHP (1131 hours per year more then a parallel configuration). The authors showed that this difference was caused by an altered flow balance in the storage tank, which depends on the control strategy. Indeed, a simplistic open-closed control was applied to the flow through the boiler (see Fig. 1). However,

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the serial configuration showed a decreased boiler efficiency of 7.2 percentage points.

To conclude, it can be said that hydronic design has a major effect on energy savings at different levels. However, literature lacks information on decision rules regarding hydronic design and on the influences of the design context on that decision. Therefore, this paper compares two hydronic configurations: serial and parallel. To tackle the disadvantages of the parallel configuration, another control strategy is investigated and compared with the open-closed strategy discussed above. Also the influences of the storage tank size and CHP size are investigated. The comparison is based on the simulation of a case study.

1.2 Hydronics and control strategies

In this subsection the hydronic configurations, shown in Figure 1, will be discussed in more detail.

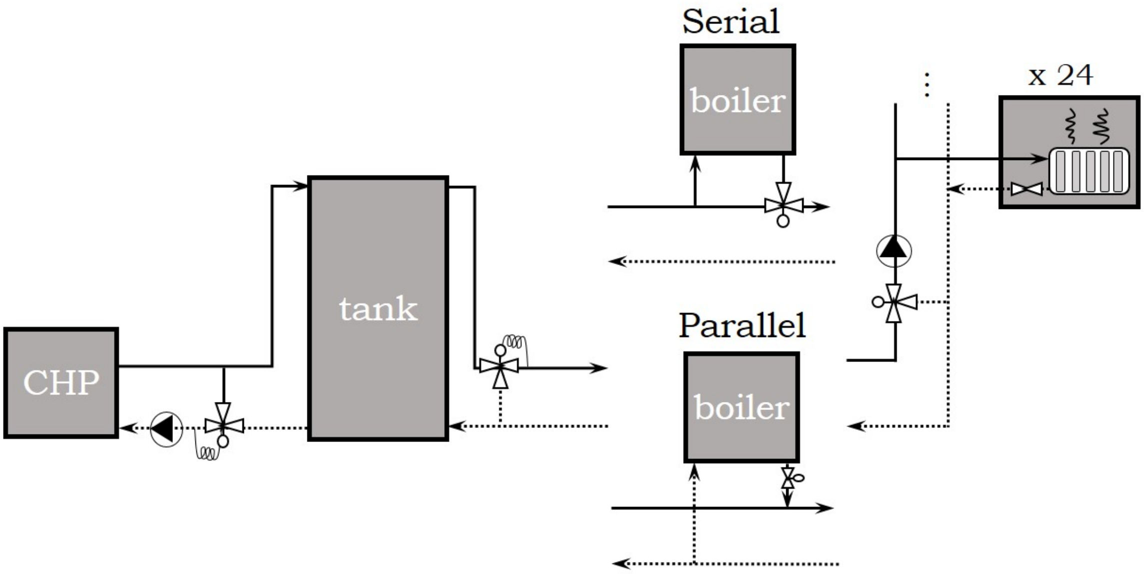

Figure 1: Schematic representation of two hydronic designs: a serial and a parallel connection of the boiler.

The CHP (gas-engine) is supplied with a three-way valve to ensure a proper operation of the engine (minimal ingoing temperature of 60°C). To increase operating hours, the engine is allowed to operate in part load when the ingoing temperature exceeds 70°C (up to 50% of its nominal electrical power). If the ingoing water temperature rises above 75°C or the outgoing one exceeds 95°C, the CHP is shut down. The mass flow rate through the CHP is designed to reach an outgoing temperature of 80°C (for the given ingoing temperature of 60°C).

The CHP is coupled to a storage tank, which is equipped with two temperature sensors (one at the top and one at the bottom, not shown on the figure). If the temperature measured by the lower sensor exceeds 75°C, the storage tank is assumed to be fully loaded and hence the CHP is shut down. It only starts up again if the temperature measured by the upper sensor drops below 75°C. This hysteresis control increases the operation time of the CHP.

The boiler can be integrated in different ways, as already discussed in the problem statement. The first option is to connect it in series with the supply pipe from the storage tank. In that case, the three-way valve shown on the figure opens fully when the boiler is on (or off but still warm) and closes fully if the boiler is off. The main disadvantage of this configuration is that the water going in the boiler is already heated by the CHP. In turn, this increased temperature will result in a lower efficiency of the boiler.

The second option is to connect the boiler in parallel, which will result in colder ingoing water. In that case, two control strategies for the two-way valve are considered: open-closed (as a reference, see problem statement) and a modulating one (as a proposed improvement). Both strategies allow flow through the boiler when this latter is on (or off but still warm) by opening the two-way valve. However, the opening position of that valve is controlled differently:

- The 'open-closed control': if the two-way valve opens, it always opens completely. Therefore, the rate at which water flows through the boiler is always a fixed share of the total water flow rate of the water in the supply and return pipes at the right hand side of the boiler in Fig. 1. As a design choice, this share is set proportional to the ratio ($1 - x^{chp}$, see next subsection) of the nominal boiler load relative to the total production system load (boiler plus CHP). This strategy was already shown to decrease the deloading of the tank in [16] (more often a higher flow rate at the source-side of the tank then the sink-side of the tank), as discussed in the problem statement.
- Instead of opening completely, the 'modulating control' adjusts the two-way valve if a flow through the boiler is required (on or off but still warm). The developed strategy is intended to maximise the deloading of the storage tank, in order to ensure maximal operating time of the CHP. Therefore, the flow through the boiler is minimised. However, a minimal flow is set to ensure the boiler is not overheated.

For all configurations and control strategies, the boiler controls the supply water temperature to reach its set point. An on/off signal is generated by a hysteresis and modulation is determined by a PI-controller.

The risers are equipped with a three-way valve to limit the supply temperature, thereby limiting pipe losses. An apartment block with 24 families, each living in a separate apartment unit, was considered. All apartment units are identical, with exception of their solar orientation, and characterised by $5kW$ heat loss at design conditions (22°C indoor and −8°C outdoor).

1.3 Other design parameters

Besides the hydronic aspects discussed above, the influence of two other design parameters will be investigated: the size of the storage tank and the size of the CHP. The CHP size is expressed as a percentage (x^{chp}) of the total production heat load (CHP plus boiler), which is here equal to ($24 * 5kW$=) $120000W$. The CHP size (i.e. its nominal heat load in kW) is then equal to $\dot{Q}^{chp}_{max} = x^{chp} * 120000$. The storage tank size is expressed as a minimal operating time of the CHP in hours, t^{sto}. To translate this time into a volume, it is assumed that the tank is filled with water at a temperature equal to the design return temperature T^{des}_{return}, and is heated completely to the design supply temperature T^{des}_{supply}. The formula: $\dot{Q}^{chp}_{max} * t_{sto} * 3600 = 4187 * 1000 * V^{sto} * (T^{des}_{supply} - T^{des}_{return})$ gives this relation, assuming a constant density and heat capacity of $1000kg/m^3$ and $4187J/kgK$.

T^{des}_{supply} and T^{des}_{return} were set at 75°C and 60°C, respectively. The considered values for t^{sto} were 0.1, 0.25, 0.5, 1 and 1.5 hours; those of x^{chp} 5, 15, 25, 35, 45 and 55 %. All combinations of these two parameters were tested, for both the serial and the parallel configuration, and for both types of control strategy of the parallel configuration ($5 * 6 * 3 = 90$ possibilities).

2.METHODOLOGY: MODELS, SIMULATION AND METRICS

2.1 General simulation method

To make the comparison, a model for all different components in Figure 1 was implemented in Matlab. Simulations were performed with a step size, Δt, of 10 seconds. The hydraulic behaviour was simplified, by characterising pumps and valves by a maximal mass flow rate ($\dot{m}_{max}$). Indeed, control signals were assumed to adjust the mass flow rate directly, rather then adjusting rotational speed (in case of a pump) or opening position (in case of a control valve). The models to simulate thermal behaviour of the components will be discussed in more detail in the following two subsections. Multiple of these models are represented by a non-homogeneous linear ODE (ordinary differential equation) with constant coefficients a and b.

$$\frac{\mathrm{d}y(t)}{\mathrm{d}t} = -a * y(t) + b \tag{1}$$

These coefficients will be discussed further for each model separately. The solution after $t = \Delta t$ seconds, with initial condition $y(0)$ and a not equal to zero, is:

$$y(\Delta t) = y(0) * \exp(-a * \Delta t) + b/a * (1 - \exp(-a * \Delta t)) \tag{2}$$

Heat production models: CHP and boiler

In the first control volume, the chemical energy of the fuel is converted to thermal energy and electrical energy (respectively $\dot{Q}_{fuel}$ (W), $\dot{Q}_{th}$ (W) and P_{el} (W) if expressed in energy flow rates). The physical burning process is, however, not modelled as such. Instead, a *performance map* is used which defines the relation between $\dot{Q}_{fuel}$, $\dot{Q}_{th}$ and, in the case of the CHP model, P_{el} by efficiencies at stationary operation, as in [14]:

$$\eta_{el} = \frac{P_{el}}{\dot{Q}_{fuel}} \tag{3}$$

$$\eta_{th} = \frac{\dot{Q}_{th}}{\dot{Q}_{fuel}} \tag{4}$$

The thermal efficiency, η_{th}^{boi}, of the boiler is calculated with 3D interpolation of a dataset: the ingoing temperature, the part load ratio ($PLR = \dot{Q}_{th}^{boi}/\dot{Q}_{th,max}^{boi}$ with $\dot{Q}_{th,max}^{boi}$ the nominal thermal heat production) and the mass flow rate, as described in [17]. For the CHP, η_{el}^{chp} and η_{th}^{chp} were fitted on manufacturer's data of a $36kW_e$ gas-engine CHP, resulting in the following equations:

$$\eta_{el}^{chp} = -0.13 * (1 - ELF) + 0.26 \tag{5}$$

$$\eta_{th}^{chp} = 0.0025 * (T_{in,ref} - T_{in}) + 0.13 * (1 - ELF) + 0.7 \tag{6}$$

With $T_{in,ref}$ some reference of the ingoing temperature for scaling the equation, in this case equal to 40 °C. ELF is the *Electric Load Factor*, defined as $ELF = P_{el}/P_{el,nom}$.

$\dot{Q}_{th}$ is taken up by a lumped capacity *, which loses energy to the surrounding air (assumed to have a constant temperature of $T_{sur} = 20°C$). The water that flows in the boiler/CHP with mass flow rate $\dot{m}$ and temperature T_{in}, takes up $\dot{Q}_{hyd}$ from the lumped capacity and is heated to its temperature T_{out}. This temperature change is described by Eq. 2, with T_{out} as y, and a and b equal to:

$$a = \frac{UA_{loss,skin} + c * \dot{m}}{C} \tag{7}$$

$$b = \frac{\dot{Q}_{th} + UA_{loss,skin} * T_{sur} + c * \dot{m} * T_{in}}{C} \tag{8}$$

In these equations, c is the specific heat capacity of water (J/kgK) and C, $UA_{loss,skin}$, T_{in} and $\dot{m}$ are the overall heat capacity of the thermal mass (J/K), the overall heat transfer coefficient (W/K), the ingoing temperature (°C) and the mass flow rate (kg/s) of the heat producer (i.e. boiler or CHP), respectively. C and $UA_{loss,skin}$ depend on the device considered and were therefore fitted on a dataset of 27 boilers and 11 gas-fired ICE-CHPs on the equations $C = c_C * \dot{Q}_{nom}$ and $UA_{loss,skin} = c_{UA} * \dot{Q}_{nom}^{2/3}$. c_C and c_{UA} were found to be $5.5s/K$ and $0.0005W^{1/3}/K$ for the boilers and $3.98s/K$ and $0.0142W^{1/3}/K$ for the CHP devices.

<u>Other models</u>

The storage vessel model is based on the stratified thermal storage tank model of TRNSYS (type 4 [5]). The insulation was chosen to match an *A*-energy efficiency class ecolabel [6] and an aspect ratio of 2.4 was considered.

The pipe models are based on a plug-flow model (type 31 [5]) with an insulation in accordance with the regulation described in [1].

Each of the 24 apartment units in the building is simplified into a single zone which is heated by an emitter. The emitter is assumed to be perfectly mixed. All the zones are modelled by assuming a lumped capacity for the air and one for the wall. All three sub-models can be described by Eq. 2, with y, a and b according to Table 1. $\dot{Q}_{em,air}$ and $\dot{Q}_{em,wall}$ are the shares of the total exchanged energy rate of the emitter $\dot{Q}_{em} = c * \dot{m} * (\bar{T}_{em} - T_{in}(0))$ with the air and wall, respectively. The horizontal line above the character indicates that a mean value within the time step of the variable is used. The loss from air to wall $\dot{Q}_{air,wall}$ is calculated as: $\dot{Q}_{air,wall} = UA_{air,wall} * (\bar{T}_{air} - T_{wall})$. To emulate the behaviour of multiple zones for each apartment, the mass flow rate is controlled by both an on-off thermostat with hysteresis and proportional control

*For the boiler, the lumped capacity represents the heat exchanger and its water content and for the CHP it represents the engine block and its cooling water.

Table 1: y, a and b from Eq. 2 for the three sub-models of the apartments. (0) means that the value at the beginning of the time step is used. The water flows in the emitter with temperature T_{in} and with a mass flow rate of $\dot{m}$. All C's and UA's are the overall heat capacity and heat transfer coefficients, respectively.

	y	a	b
emitter	T_{em}	$\frac{UA_{em}+c*\dot{m}}{C_{em}}$	$\frac{UA_{em}*T_{air}(0)+c*\dot{m}_{em}*T_{em,in}(0)}{C_{em}}$
air	T_{air}	$\frac{UA_{air,wall}}{C_{air}}$	$\frac{\dot{Q}_{em,air}+\dot{Q}_{sol,air}+UA_{air,wall}*T_{wall}(0)}{C_{air}}$
wall	T_{wall}	$\frac{UA_{wall,ext}}{C_{wall}}$	$\frac{\dot{Q}_{em,wall}+\dot{Q}_{air,wall}+\dot{Q}_{sol,wall}+UA_{wall,ext}*T_{ext}(0)}{C_{wall}}$

(which emulates the behaviour of a thermal radiator valve). $\dot{Q}_{sol}$ is the solar gain, of which half is taken up by the wall ($\dot{Q}_{em,wall}$) and half by the air ($\dot{Q}_{sol,air}$).

Temperature set points of all the apartment units were generated with a tool developed in the context of the projects [4] and [2]. This generator is based on statistical occupancy data of multiple types of families. These types were chosen to match a typical distribution of Flemish families, as is documented at [7]. The set point values were set at 22°C for comfort mode and 15°C for absent or night mode. It is assumed that domestic hot water is not supplied with the collective heating system and therefore it is not taken into account in this paper.

2.2 Metrics

The metrics quantifying the buildings performance are: the Room Temperature Lack (RTL) [15], Room Temperature Excess (RTE)[15] and total thermal energy consumption of the building (Q_{hyd}).

The following metrics were used to quantify the performance of the production system: r^{chp} (percentage of Q_{hyd} that is covered by the CHP; the remaining share is, obviously, covered by the boiler), t_{cyc}^{chp} (the mean continuous operating time in hours of the CHP), η^{boi} (efficiency of the boiler in %), t_{cyc}^{boi} (analogue to t_{cyc}^{chp} but for boiler) and $RPES$ (relative primary energy savings, with a reference boiler efficiency of 90% and a reference efficiency of the electric production 37% as in [18]).

3.RESULTS AND DISCUSSION

3.1 General building performance

First of all it should be said that the performance of the building was similar for all simulations. Indeed, RTL, RTE and Q_{hyd} varied less then respectively 6.0%, 1.1% and 0.6%. Therefore, the rest of the discussion will focus on the performance of the production system, which is shown in Figure 2.

3.2 CHP performance

As already discussed in [16], it is clear that a parallel configuration results in a lower r^{chp} then a serial one if an open-closed control strategy is applied to the two-way valve of the boiler (see blue versus red/'par0' versus 'ser', respectively). Indeed, for the parallel configuration this variable is, depending on x^{chp}, only 64% to 77% of the serial configuration. However, when a modulating control is applied (green color/'par1'), the parallel configuration is able the reach the same r^{chp} (or even 1% higher). It should be noted that the storage size does not affect r^{chp}. However, this statement *should be interpreted with great care*. Indeed, in the performed simulations, no restrictions were set to the daily number of start-ups of the CHP. Off course, this is not true in practice as maintenance contracts of ICE-CHP-devices limit the number of start-ups to a few times a day to prevent to frequent maintenance.

And yet, the sensitivity analysis on tank size can be interpreted. Indeed, the mean number of hours of uninterrupted operation, t_{cyc}^{chp}, is substantially affected by that size for all configurations and control strategies, and for all x^{chp}. However, the parallel configuration with open-closed control shows for all tank sizes and all x^{chp} a lower t_{cyc}^{chp}. Again, the modulating

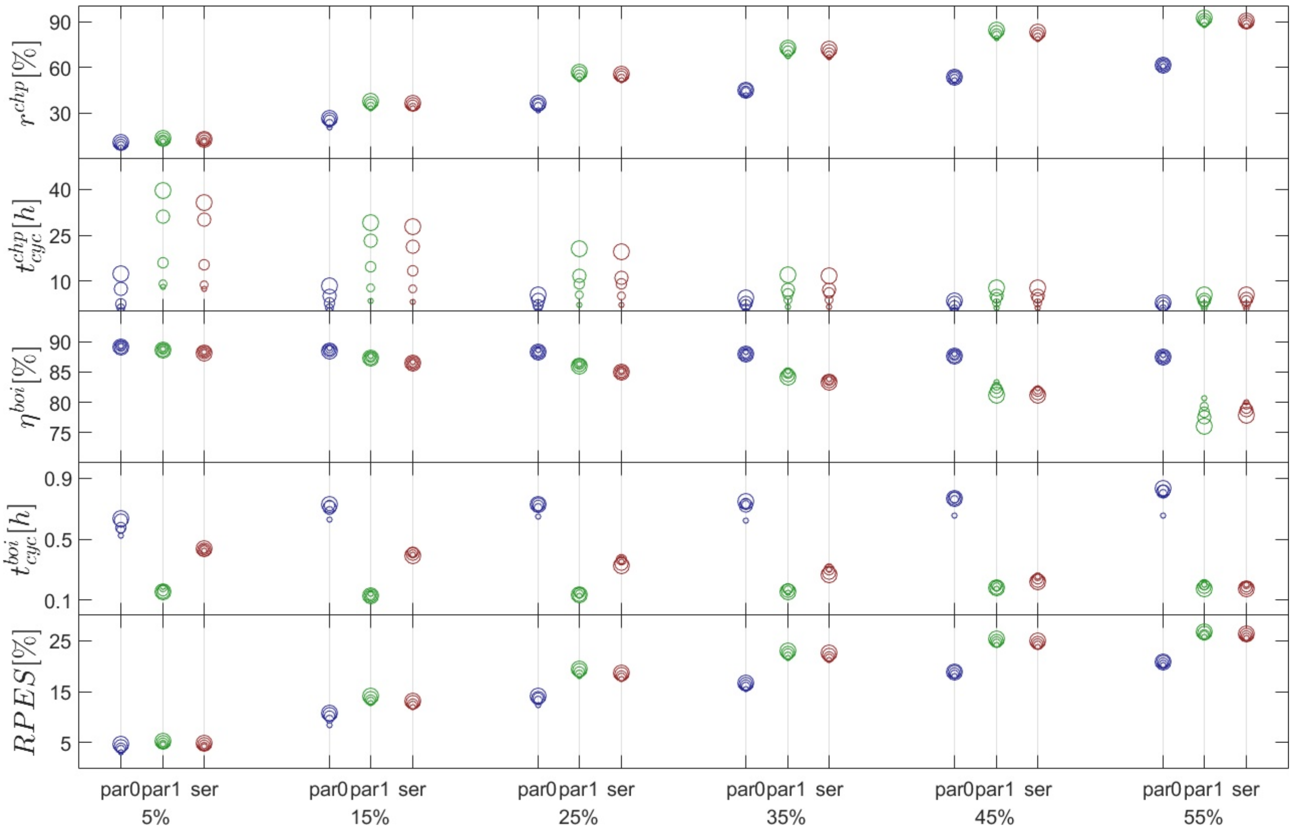

Figure 2: Results of the simulations. 'par0' refers to a parallel configuration without modulating two-way valve control (i.e. open-closed) and corresponds to a blue color; 'par1' refers to a parallel configuration with modulating two-way valve control and corresponds to a green color; 'ser' refers to a serial configuration and corresponds to a red color. The percentages on the x-axis indicate the nominal thermal load of the CHP, relative to the total nominal load of the production system (x^{chp}). The sizes of circles refer to different design load times of the storage tank, t^{sto}; from smallest to largest: 6, 15, 30, 60, 90 minutes. The reader should notice that apparently thick circles represent actually multiple data points with different storage sizes but a similar value.

control can increase this towards the same values as (or even slightly higher then) the serial configuration.

3.3 Boiler performance

Also the performance of the boiler is affected by the hydronic design. η^{boi} is, with exception of $x^{chp} = 55\%$, higher for the parallel configuration for both control strategies. This makes sense, as the the water going in the boiler has a low temperature (i.e. it is not pre-heated by the CHP as is the case for a serial configuration). However, η^{boi} is only slightly higher for the parallel configuration with modulating control. This can be explained by the minimal flow rate through the boiler. Indeed, for the same heat load, a smaller flow rate corresponds to a higher supply temperature, which negatively affects the boilers efficiency. The relative effect of CHP size on the difference in η^{boi} is limited. Indeed, while it increases with increasing x^{chp}, also the heat produced by the boiler decreases with increasing x^{chp}.

t^{boi}_{cyc} indicates that a parallel configuration with open-closed control results in the most stable boiler operation (i.e. the least frequent short-cycling). However, if the modulating control is applied the operation is less stable, also compared to the serial configuration.

3.4 Primary energy savings

The resulting primary energy savings depends on both the CHP and boiler performance. However, extra fuel consumption required for start-ups are not taken into account. Also, as discussed before, no maximal number of start-ups were considered. This explains the limited effect of storage size on $RPES$.

In the lower sub-plot of Figure 2, it can be seen that the serial configuration and the parallel configuration with modulation result for all CHP sizes in the highest savings. This means that the higher fuel consumption of the boiler of these designs do not eliminate the fuel savings corresponding to the increased CHP operation. The difference in savings between the parallel configuration with modulation and the serial configuration are, however, limited.

4.CONCLUDING REMARKS

In this paper, two types of hydronic configuration of heat production systems including an ICE-CHP, storage tank and auxiliary boiler were investigated: a parallel and serial configuration. For the parallel configuration, two different control strategies of a two-way valve were compared: one with open-closed control and one with modulation. On top of that, a sensitivity analysis was performed on the size of the storage tank and the size of the CHP. The comparison was elaborated for a case study and was based on dynamic building simulations.

Results indicate that, compared to a serial configuration, a parallel one decreases the operating time of the CHP (and therefore also energy savings) substantially if an open-closed control of the two-way valve is used. However, it was shown that if this control is 'upgraded' to the modulating one, the CHP performs the same as in the case of a serial configuration. Both the parallel with modulating control and the serial configuration have a lower boiler performance compared to the parallel configuration with open-closed control. In terms of primary energy savings, the parallel configuration with modulating control and the serial configuration performed the best. The sensitivity analysis showed that these conclusion are independent of the of the storage tank and CHP size.

In general, it can be concluded that a serial configuration is the most robust, and therefore a saver choice in a design process: with a simple control strategy, the CHP is able to operate both often as well as continuous. While the parallel configuration can increase the boiler efficiency, the CHP performance depends largely on the control strategy. Therefore, this latter configuration is less robust. In future work the modulating control of the two-way valve of the parallel configuration will be further optimised to combine maximal CHP operation with a high boiler efficiency.

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