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Evaluation of an ~~Alkaline Fuel Cell System~~alkaline fuel cell system as a ~~Micro-CHP~~micro-CHP

Ivan Verhaert^{a, c, *}

ivan.verhaert@uantwerpen.be

Grietus Mulder^d

Michel De De Paepe^b

^aEnergy and Materials in Buildings and Infrastructure (EMIB), Antwerp University - UAntwerpen, Campus Hoboken, Salesianenlaan 90, 2660 Hoboken, Belgium

^bDepartment of Flow, Heat and Combustion Mechanics, Ghent University - UGent, Sint-Pietersnieuwstraat 41, 9000 Gent, Belgium

^cThomas More, University College, Kleinhoefstraat 4, 2440 Geel, Belgium

^dVITO, Flemish Institute for Technological Research, Boeretang 200, 2400 Mol, Belgium

*Corresponding author at: Energy and Materials in Buildings and Infrastructure (EMIB), Antwerp University - UAntwerpen, Campus Hoboken, Salesianenlaan 90, 2660 Hoboken, Belgium.

Abstract

Micro-cogeneration is an emerging technology to reduce the non-renewable energy demand in buildings and reduce peak load in the grid. Fuel cell based cogeneration (CHP) has interesting prospects for building applications, even at relatively low heat demand. This is due to their partial load ~~behaviour~~behavior which is completely different, compared to other micro-CHP technologies. Within the fuel cell technologies suitable for small scale CHP or micro-CHP, the existing configuration of an alkaline fuel cell system is ~~analysed~~analyzed. This analysis is based on validated models and offers a control strategy to optimize both water management and energy performance of the alkaline fuel cell system. Finally, the model of the alkaline fuel cell system with optimized control strategy is used to compare its part load ~~behaviour~~behavior to other micro-CHP technologies.

Keywords: Alkaline ~~Fuel Cell~~fuel cell; Micro-CHP; Part load; Cogeneration; System model

Nomenclature

C

heat capacity (W/K)

P

power (~~W~~)(W)

Q

heat (~~W~~)(W)

T

temperature (°C)

c

specific heat capacity (J/kg · K)

h

specific enthalpy (J/kg)

ṁ

mass flow rate (kg/s)

α_{partial}

partial efficiency (%)

Δ

difference (-)

ε

effectiveness (-)

η

efficiency (-)

κ

model parameter (a.u.)

Subscripts and acronyms

AFC

alkaline fuel cell

CHP

combined heat and power

FC

fuel cell

C

cold

H

hot

HE

heat exchanger

KOH

electrolyte flow

air

air flow

aux

auxiliary equipment

e

electrical

fan

fan

inverter

inverter

mech

mechanical

misc

miscellaneous

pump

pump

ref

reference

s

saturated ~~vapour~~vapor

surr

surroundings

system

system

w

water

1 Introduction

Micro-cogeneration (micro-CHP) for building applications is getting more attention [\[1-4\]](#). Depending on size, several ~~technologies~~ technologies are available. The most mature technologies are combustion based technologies, like micro-turbines, internal combustion engines [\[5\]](#) and Stirling engines [\[6-8\]](#). For small sizes, they have a relatively low power-to-heat ratio. This makes them suitable for buildings with a relatively high

heat demand. For the existing building stock they can be an alternative solution to boiler driven heating systems. However, due to an improved insulation rate and an increasing number of electric appliances [9-11] in buildings, the energy profile in buildings is changing [12,13]. To meet these future demands in buildings, technologies are preferred, promising higher power-to-heat ratios even at small scale.

In this prospect, fuel cell based micro-CHPs are a promising alternative in buildings with a relatively low heat demand. Fuel cells have the potential for high electrical efficiencies, compared to other technologies [14]. Contrary to other types of CHP-technologies, their electrical efficiency is independent of size. Moreover the electrical efficiency even increases at part load. The application of fuel cell technology as micro-CHP refocused the research and development efforts on fuel cells [15]. Next to stack performance and lifetime, the ability of heat recovery in the system design became an important research topic for possible improvements [15-17].

Different types of fuel cell technology are suitable as micro-CHP for building applications [18]. Solide oxide fuel cell (SOFC) technology is the most mature technology as it can run on natural gas [19]. Within buildings SOFC show to be already competitive with fuel oil and electrical heating systems. The viability could be increased by improving utilization of waste heat [17]. Next to SOFC, also micro-CHP systems based on proton exchange membrane fuel cells (PEMFC) are (PEMFC) are developing rapidly [19-21]. The energetic-exergetic comparison in [22] showed that PEMFC based systems use their fuel energy input more efficiently than SOFC based systems for building application. This was expected, since building applications do not require the high temperature of the SOFC system. Like PEMFCs, alkaline fuel cells (AFCs) work at relatively low temperature, in the range of ~~50 to 90~~ 50-90 °C.

In Refs. [23,24,18] it is shown that within the domain of (micro-)CHPs for building applications, also an AFC-based system offers possibilities. Next to this, AFCs have the possibility to be manufactured at less cost, using non-noble materials [25]. Despite a diminished interest in the last decade in alkaline fuel cells, compared to PEMFC, the electrical efficiencies presented in Refs. [24,26] are still competitive to PEMFC and SOFC used for micro-CHP [27] and certainly to other CHP-technologies [1,2,4,5]. Recent studies show that efficiency improvements are still possible by using different preparation methods for the development of electrodes [25].

1.1 Scope

As AFC-technology has some promising evolutions, an analysis on a system level is necessary to gain more insight in its technical potential as a micro-CHP.

Hence, an existing system set-up is evaluated, considering the influence of the external heating circuit and load variation on system performance. In this way the results can be put in perspective in comparison with other technologies for micro-CHP.

The paper provides a simulation based analysis of the system set-up and possible control strategies in order to maintain water management and optimize energetic performance.

2 Methodology

In previous work a stack model is developed and validated [28,29] to analyse analyze the water management [29] and performance [30] of an alkaline fuel cell stack. The goal of our research is to evaluate the potential of a complete CHP-system based on AFC technology. Therefore, the stack-model developed in Refs. [28,29] is integrated in a complete CHP-system model. This will enable a simulation based comparison with other micro-CHP technologies.

Next to energetic performance, also safety and water management are important criteria. In Ref. [23] the safety is discussed of the present set-up. This work focuses on water management and energetic performance.

2.1 Water management

An important evaluation criterion for a fuel cell system is the effectiveness of the water management. This implies energy friendly solutions to remove excess water in the electrolyte flow or to prevent net evaporation of the electrolyte.

Next to this, the robustness of the settings for nominal operation will be investigated. A robust or steady operation implies that small parameter changes result only in small net production or evaporation of water in the electrolyte flow. This will limit the range of steady operation of the fuel cell. The thermal response of the system is relatively slow, which can be deduced from the measurements on the water level in Ref. [29]: a response time of ~~10 to 20 minutes~~ 10-20 min of the water management on a new set-point is noticed. Therefore, a robust working point, regarding water management, is defined as a working point where the net water production or evaporation is kept lower than 0.2 ml/s, which corresponds to the contents of a glass of water every twenty minutes.

2.2 Performance

To evaluate the performance of a CHP-system and the effect of different operating and system parameters on the CHP-potential, the methodology used in [30] is repeated: the primary energy savings are calculated. These savings result from the comparison with a separate production of heat and electricity.

2.2.1 Efficiency of the CHP

To evaluate the CHP-potential of the fuel cell, the electrical and thermal efficiency are defined. These efficiencies are defined at higher heating value (HHV) as this facilitates comparing different fuel types and introducing necessary conversion factors.

- The electrical efficiency, α_e , is defined as the ratio of the net generated electric power, $P_{e,system}$, to the fuel input, $Q_{Fuel,system}$, Eq. (1). This efficiency, α_e , differs from the efficiency of the fuel cell stack itself, α_{FC} , which is discussed in [30] and defined by Eq. (2).

This difference is found both in the definition of the fuel input as in the definition of the generated power. The fuel input of the system, $Q_{Fuel,system}$, includes the hydrogen, which is purged into the environment to remove water drops at the anode side and in the piping at the stack inlet. This is defined by a purge efficiency, η_{purge} , Eq. (3). The generated electric power (DC) by the fuel cell stack, $P_{e,FC}$, will be converted into AC power, $P_{e,AC}$, by the inverter at efficiency of the inverter, $\eta_{inverter}$, Eq. (4). The difference between the generated AC power, $P_{e,AC}$, and the net generated electric power, $P_{e,system}$, is found in the electric load of auxiliary equipment, $P_{e,aux}$, Eq. (5). This load is the result of the electric energy needed for the electrolyte pump, $P_{e,KOHpump}$, fan power, $P_{e,airfan}$, monitoring and safety equipment, $P_{e,electronics}$, etc., $P_{e,misc}$, Eq. (6).

$$\alpha_e = \frac{P_{e,system}}{Q_{Fuel,system}} \quad (1)$$

$$\alpha_{FC} = \frac{P_{e,FC}}{Q_{Fuel,FC}} \quad (2)$$

$$\eta_{purge} = \frac{Q_{Fuel,FC}}{Q_{Fuel,system}} \quad (3)$$

$$P_{e,AC} = \eta_{inverter} \cdot P_{e,FC} \quad (4)$$

$$P_{e,system} = P_{e,AC} - P_{e,aux} \quad (5)$$

$$P_{e,aux} = P_{e,KOHpump} + P_{e,airfan} + P_{e,electronics} + P_{e,misc} \quad (6)$$

- The thermal efficiency, α_{th} , is defined as the ratio of the useful heat output, Q_{TH} , to the fuel input, $Q_{Fuel,system}$, Eq. (7). Depending on system set-up, this heat output will be the result of heat recovery in the electrolyte flow or in the air flow or in a combination. The useful heat output, Q_{TH} , is characterized by a temperature change of the water flow in the heating circuit, (Eq. (8))

$$\alpha_{th} = \frac{Q_{TH}}{Q_{Fuel,system}} \quad (7)$$

$$Q_{TH} = \dot{m} \cdot c_w \cdot (T_{depart} - T_{return}) \quad (8)$$

2.2.2 Primary ~~Energy~~energy

To evaluate the different parameters, the primary energy savings compared to separate generation of electricity and heat were calculated. For heating, a gas boiler with 90% efficiency ($\eta_{t,ref}$) is considered, based on the higher heating value (HHV). Electricity is bought from the grid. The amount of primary energy needed to produce this electricity is strongly dependent on the used technology and its efficiency, $\eta_{el,ref}$. In this way, in Belgium, two different values for $\eta_{el,ref}$ are interesting to compare with micro-CHP. However, these values are mostly reported at lower heating value (LHV). Therefore, a conversion factor has to be taken into account to allow comparison with the CHP efficiencies, defined in Section 2.2.1.

- To determine the quality of a cogeneration ~~CHP~~ unit, the CHP is compared with the best possible classic alternative: a combined cycle gas/steam turbine plant. A combined cycle plant has a typical efficiency between 53% and 56% (LHV). Considering transformation and transport losses, a realistic value of 50% (LHV) for the efficiency of separate electricity production ($\eta_{el,ref}$) is assumed [4]. Based on the average lower and higher heating values for natural gas, this can be converted in an efficiency at HHV of 45%.

- To calculate the actual primary energy savings, an efficiency of 40% (LHV) is used which is representative for the average Belgian power plant. For fossil fuel plants the calculated average is 42% (LHV) or 38% (HHV) and for nuclear plants 37% [4]. At higher heating value a reference efficiency, $(\eta_{el,ref})$, of 37% is taken into account.

The first values for $\eta_{el,ref}$ will be used to calculate possible financial support measures to promote CHP in Belgium. The second values will be used to calculate the primary energy use in buildings according to the European building directive [31]. The primary energy savings are an important criterion to evaluate the performance of CHP systems [4]. It is clear that for CHPs raising the electrical efficiency is worth more than raising the thermal efficiency. Based upon the quoted conversion efficiencies this can be quantified; improving the thermal efficiency with 10% corresponds to an electrical improvement of 5% or 4.1%, depending on the chosen reference situation.

2.2.3 Hydrogen carrier

In Section 2.2.1 the efficiencies of the CHP are defined, with respect to the energy in the used fuel at the inlet. As stated in Section 2.2.2 as base for comparison the primary energy consumption is used.

Since different fuels are used a conversion factor has to be taken into account. For fossil fuels, natural gas and diesel this factor can be 1 as they are primary energy sources. Hydrogen is an energy carrier, which can be produced in different ways like reforming and electrolysis. Depending on the production method and origin of the primary fuel a conversion factor can be proposed to calculate primary energy. It is chosen to work with the average reforming efficiency as conversion factor, as this is most representative for the production of hydrogen. In Ref. [32] a reformer efficiency is reported from 85%, which is used in this evaluation.

3 Analysis of an AFC-based CHP-system

A model-based approach is used to evaluate the present system set-up (See (see Ref. [23]).). In the original configuration the goal of the control strategy is to ensure safe operation and to proof the concept. A model-based approach enables exploration of other system parameters and working points.

3.1 Description of the AFC-system

In Ref. [23] the first AFC-based system useful for CHP-applications was presented. In Ref. [24] a first system evaluation was performed based upon measurements. In Refs. [28,29] an alkaline fuel cell stack model was developed, validated with new measurements on the same system design. Therefore, this system design, shown in Fig. 1, is discussed in the following section. The air flow is extracted and controlled by the fan out of the surroundings into the system, passing only a adsorptive ~~scrubber~~, CO₂-scrubber, and therefore has the same conditions as the surroundings. The fan uses electric power to deliver sufficient air flow rate.

Fig. 1 System set-up of original AFC-based CHP-system.

The air flow and hydrogen flow (Fuel energy) enter the fuel cell stack, in which the oxygen out of the air flow and the hydrogen react into water, electric power and heat. The hydrogen flow is directly proportional to the DC-current and therefore controlled by the electric load. An inverter converts the electric DC-power into AC-power.

Next to air and hydrogen, an electrolyte circuit is connected to the fuel cell stack, which ensures cooling and separation of the electrodes. The pressure and circulation of the electrolyte is delivered and controlled by an electric pump. The reaction heat results in a temperature rise of the electrolyte, in which part of the water is evaporated. The rest of the water is removed as water ~~vapour-vapor~~ in the overflow of the air stream. Both electrolyte and air flow are directed into the electrolyte tank, which has the purpose to evaporate the water out of the electrolyte into the air flow to ensure a constant electrolyte concentration. The air flow is ejected into the environment. The electrolyte is cooled to heat up an external heating circuit, after which it is redirected into the fuel cell stack.

Next to the electric energy ($P_{e,system}$) and thermal energy which heats up the heating circuit (Q_{useful}), there are still energy losses to the environment by conversion losses (in the inverter ($\eta_{inverter}$), fuel handling (η_{purge}), ...), by energy use of the auxiliary components ($P_{e,aux}$), by radiation losses to the environment (Q_{loss}) and by waste heat in the air flow (Q_{air}).

3.2 Model of the components

3.2.1 Fuel cell stack

The complete stack model developed in Ref. [29] is used in this system model. As input both air, hydrogen and electrolyte flow and surrounding temperature are used to predict the thermodynamics and water management of the output air, hydrogen and electrolyte flow and the generated electric DC-power ($P_{e,FC}$) and heat loss to the environment (Q_{surr}).

3.2.2 Electrolyte tank

As described in Ref. [29] the electrolyte tank is modelled as a function of the following parameters:

- electrolyte temperature
- electrolyte flow
- air flow
- air temperature
- relative humidity of air
- percentage of evaporation: 0 means no evaporation - 100 means that the air is completely saturated

This last parameter is a fitting parameter, $\kappa_{evaporation,tank}$, listed in Table 1. It is reasonable that the KOH tank has an influence. However, complete saturation will not be reached. Therefore, the parameter will be higher than 0% and lower than 100%. The actual value depends on the lay-out of the electrolyte tank. For the present set-up, a best fit could be found at about 40% [29]. For a lay-out with more contact surface between air flow and electrolyte, this evaporation rate can be increased. This will be simulated in the system evaluation. The model predicts the thermodynamics and water management of the output flows. The model equations are based upon a heat balance (Eq. (9)) of heat required for evaporation, Q_{vap} (Eq. (12)) and the heat delivered by cooling of both mass flows, Q_{KOH} and Q_{air} (Eqs. (11) and (10)). These mass and energy flows are illustrated in Fig. 2.

$Q_{vap} = Q_{air} + Q_{KOH}$ (9)

$Q_{air} = m_{air,in} \cdot c_{air,in} \cdot (T_{air,in} - T_{air,out})$ (10)

$Q_{KOH} = \dot{m}_{KOH,out} \cdot c_{KOH,out} \cdot (T_{KOH,in} - T_{KOH,out})$ (11)

$Q_{vap} = \dot{m}_{vap} \cdot (h_s (T_{air,out}) - h_w (T_{KOH,in}))$ (12)

If the electrolyte tank would be infinitely big the electrolyte and air flow will come out at the same temperature ($T_{balance}$) as a result of the zeroth law of thermodynamics. Next to this, the output air flow would be completely saturated, which means a maximum amount of water will be evaporated out of the electrolyte flow into the air flow. The heat required for this net evaporation will be delivered by cooling the air flow and/or electrolyte flow. This possibility corresponds to an evaporation rate of 100%. In the model, the evaporation rate is defined as

- the actual heat transfer for evaporation in relation to the heat exchange in an infinitely large electrolyte tank
- the temperature change of the electrolyte flow in relation to the temperature change in an infinitely large electrolyte tank.

Table 1 List of model parameters for the original system.

Model parameters	Value	Unit
$\kappa_{evaporation,tank}$	40	%

$\kappa_{pump,1}$	0.014	$W \cdot \min^3/l^3$
$\kappa_{pump,2}$	4.886	$W \cdot \min/l$
$\kappa_{fan,1}$	4.1	$W \cdot s^3/m^9$
$\kappa_{fan,2}$	678	$W \cdot s/m^3$

Fig. 2 Heat balance within the electrolyte tank.

These assumptions allow a unique solution for all ~~modelled~~-modeled output parameters (electrolyte temperature, electrolyte flow rate, air temperature, air flow rate and air humidity rate).

3.2.3 Heat exchanger

There are several ways to model a heat exchanger: based on temperature difference, on heat transfer coefficient or on effectiveness, ϵ_{HE} [33]. This last method, based on following equations, is used to integrate the heat exchanger in our system.

$$Q_{HE} = C_H \cdot (T_{H,in} - T_{H,out}) \quad (13)$$

$$Q_{HE} = C_C \cdot (T_{C,out} - T_{C,in}) \quad (14)$$

$$\Delta T_{max} = T_{H,in} - T_{C,in} \quad (15)$$

if $C_H < C_C$,

$$\epsilon_{HE} = \frac{T_{C,out} - T_{C,in}}{\Delta T_{max}} \quad (16)$$

else,

$$\epsilon_{HE} = \frac{T_{H,in} - T_{H,out}}{\Delta T_{max}} \quad (17)$$

In these equations,

- Q_{HE} represents the transferred ~~heat~~heat,
- C_i represents the total heat capacity of the flow ~~h~~i,
- subscripts i,j indicate flow ~~(+)(-)~~(+)(-), which can be ~~cold(C)~~cold (C) or ~~hot(H)~~hot (H), and flow direction ~~(+)(-)~~(+)(-).

For a water/water heat exchanger an effectiveness of 0.8 is a realistic average value. In the analysis other effectiveness will be simulated. In this case, it has to be taken into account that a higher effectiveness mostly results in

higher pressure losses, resulting in more pump energy [33].

3.2.4 (Electrolyte) pump

The pump model is only used to calculate the electric power, $P_{e,pump}$, needed to supply a sufficiently high electrolyte flow rate, $\dot{m}_{KOH}$. For the electrolyte pump an empiric equation is formulated to calculate mechanical pumping energy, P_{mech} , based on electrolyte flow rate, $\dot{m}_{KOH}$. This equation (Eq. (18)) is based on flow and pressure measurements on the electrolyte pump in the system. An overall pump efficiency, η_{pump} , is introduced to complete the model. The average value for this efficiency is set at 50%, but can be altered in the model input to evaluate its effect on system performance.

$$P_{mech,pump} = \kappa_{pump,1} \cdot \dot{m}_{KOH}^3 + \kappa_{pump,2} \cdot \dot{m}_{KOH} \tag{18}$$

$$P_{mech,pump} = \eta_{pump} \cdot P_{e,pump} \tag{19}$$

The parameters used in Eq. (18) are listed in Table 1.

3.2.5 CO2-scrubber

In the scrubber the air is passed through a matrix of adsorption material resulting in a large pressure drop. This pressure drop will be balanced by the fan. In the scrubber CO₂ is removed from the inlet air. No temperature change is detected. Therefore, the scrubber is excluded from the system model. The pressure drop is taken into account in the fan model. The air is assumed to be free of CO₂.

3.2.6 (Air) fan

Similar to the pump model, the fan model calculates the consumed electric energy, $P_{e,fan}$, based on the air flow rate, $\dot{m}_{air}$, and the overall efficiency, η_{fan} , according to an empirically determined equation (Eq. (20)).

$$P_{mech,fan} = \kappa_{fan,1} \cdot \dot{m}_{air}^3 + \kappa_{fan,2} \cdot \dot{m}_{air} \tag{20}$$

$$P_{mech,fan} = \eta_{fan} \cdot P_{e,fan} \tag{21}$$

For this fan the model parameters are based upon flow and pressure measurements on the fan integrated in the present system (See (see Table 1)).

3.3 Evaluation of the original system set-up

To evaluate the different system components a sensitivity study is performed on following parameters.

- η_{purge} , the purge efficiency, defined in Eq. (3).
- $\eta_{inverter}$, the efficiency of the inverter.
- η_{pump} and η_{fan} , the efficiencies of the electrolyte pump and air fan(s).
- $c_{evaporation,tank}$, percentage of evaporation in the electrolyte tank.
- ϵ_{HE} , the effectiveness of the heat exchanger.

3.3.1 Influence of purge efficiency

The purge efficiency, η_{purge} is altered from 90% to 100%. Because of its definition (Eq. (3)) it effects all other efficiencies equally and proportionally. For the same output also the water management will remain constant. The hydrogen concentration can be an issue if the system is placed indoors, however since the original set-up is meant to be outdoors, no problems are expected.

3.3.2 Influence of the inverter efficiency

The inverter efficiency, $\eta_{inverter}$ is altered from 82% to 100%. It only affects electrical output since there is no heat recovery of the conversion losses. Every % improvement in efficiency will increase electrical output by 1% and primary energy savings by 2-2.5%, depending on reference efficiencies (See (see Section 2.2.2)).

3.3.3 Influence of pump and fan efficiency

This last remark also counts for the pump and fan efficiency, η_{pump} and η_{fan} , which are investigated together. The effect of their efficiency on the total electrical power output is negligible. For auxiliary equipment twice as efficient, the gain in electrical output is limited to 0.3% even at small loads. At large load this effect is even smaller. Therefore, a cheap and standardized pump and fan selection is acceptable and will not affect efficiency.

3.3.4 Influence of the rate of evaporation in the electrolyte tank

As described in Section 3.1 the goal of the air flow through the electrolyte tank is to evaporate and evacuate the reaction water. A different ‘tank’ configuration, allowing more contact surface or atomization of the electrolyte, could increase the percentage of evaporation in the electrolyte tank. This percentage, $c_{evaporation,tank}$, will have a large influence on the thermal power output as shown in Table 2. This influence is most pronounced at high loads and high return temperatures, whereas e.g. at 100 A and 45 °C, an increased evaporation rate results in a drop in thermal power output from about 7 kW to about 5 kW. At lower load factors and/or lower return temperature the influence of changes in evaporation rates drops. Higher evaporation rates result in higher heat losses in the air flow. The effect on electrical power output is also negative, since those heat losses cause lower electrolyte temperatures, which affect electrical efficiency [30]. Table 2 shows however that the effect is limited to about 0.05%, which is negligible. The heat loss on the thermal power output due to a higher evaporation rate, $\kappa_{evaporation,tank}$, will be compensated by an improved water management. Table 2 also illustrates how a higher evaporation rate results in a reduced net water production.

Table 2 Summary of sensitivity analysis of the evaporation rate of the electrolyte tank.

Working point				Thermal power output			Electrical power output			Net water production		
DC-current (A)	Return temperature (°C)	Air condition		Evap.rate		Relative deviation (%)	Evap.rate		Relative deviation (%)	Evap.rate		Relative deviation (%)
		T (°C)	RH (%)	10% (kW)	85% (kW)		10% (kW)	85% (kW)		10% (ml/s)	85% (ml/s)	
40	25	1	1	2.61	2.45	6.26	2.50	2.50	0.01	0.30	0.27	12.77
		20	20	3.02	2.87	5.24	2.51	2.51	0.01	0.31	0.26	17.27
		20	80	3.13	2.96	5.43	2.51	2.51	0.01	0.34	0.29	17.08
	45	1	1	1.88	1.54	19.63	2.55	2.55	0.02	0.20	0.12	53.76
		20	20	2.28	1.90	17.40	2.55	2.55	0.02	0.21	0.07	89.58
		20	80	2.36	03	14.97	2.55	2.55	0.02	0.24	0.13	62.04
100	25	1	1	7.99	7.08	12.00	5.40	5.40	0.06	0.72	0.43	50.38
		20	20	8.49	7.53	11.85	5.41	5.41	0.06	0.73	0.36	64.16
		20	80	8.63	7.74	10.81	5.41	5.41	0.06	0.78	0.49	45.35
	45	1	1	6.52	4.86	28.91	5.51	5.50	0.07	0.52	−0.16	306.03
		20	20	6.97	5.19	28.45	5.51	5.51	0.09	0.51	−0.26	387.69
		20	80	7.11	5.35	27.47	5.51	5.51	0.10	0.56	−0.11	274.00

At higher return temperatures, at higher currents and at dryer air flows this effect is more pronounced and the net water production becomes even negative. At higher return temperatures and higher currents more heat is available to heat up the air, which results in a higher saturation temperature. This temperature and the (lack of) water vapour-vapor content determine the capacity of the air flow to remove water out of the system.

At higher evaporation rates, at about 70%, the drying effect slows down in relatively wet or cold conditions. This is due to the fact that at this rate of evaporation almost complete saturation is achieved in the output air flow. To determine an optimal evaporation rate both primary energy savings as water management has to be taken into account.

Based on an hypothesis that an electric resistance has to provide the heat needed to achieve zero water production, an optimum can be found between 40% and 70%. Therefore, the present set-up is an acceptable solution, because higher evaporation rates will result in larger set-ups and/or higher costs. As can be seen in Section 3.4, it will be possible to control water management by adjusting operating parameters.

These conclusions are based on a steady-state analysis of the CHP-system. Dynamic analysis could give an indication of the improvement potential in a reduced size of the evaporation tank. A smaller evaporation tank with a similar contact surface could reduce the thermal response time of the AFC-based CHP-system.

3.3.5 Influence of the effectiveness of the heat exchanger

Due to its definition, the effectiveness of the heat exchanger, ϵ_{HE} , will influence thermal performance. Starting from a fixed return temperature of the external heating circuit, the effectiveness will also determine the temperature of the electrolyte circuit. This will have a similar influence on thermal performance, because at a lower effectiveness a higher electrolyte temperature is necessary to exchange heat at a fixed return temperature. In [30] is found that higher electrolyte temperatures result in a lower thermal performance.

Since the electrical efficiency increases at higher electrolyte temperature [28,30], the effectiveness will have a negative impact on electrical performance. Since both thermal and electrical output experience opposite effects, primary energy savings are taken into account to evaluate the effect of the effectiveness of the heat exchanger. In Table 3 it is shown that the positive influence on thermal efficiency is larger than 2-to-2.5-2-2.5 times (See (see Section 2.2.2) the negative influence on electrical efficiency, so a reduction in primary energy consumption will be achieved by improving the heat exchanger.

Table 3 Summary of the sensitivity study, concerning the effect of the effectiveness of the heat exchanger on the system performance.

Working point				Thermal power output			Electrical power output			Net water production		
DC-current (A)	Return temperature (°C)	Air condition		Effectiveness HE		Relative deviation (%)	Effectiveness HE		Relative deviation (%)	Effectiveness HE		Relative deviation (%)
		T (°C)	RH (%)	30% (kW)	90% (kW)		30% (kW)	90% (kW)		30% (ml/s)	90% (ml/s)	
40	25	1	1	2.23	2.61	14.93	2.53	2.50	−1.11	0.22	0.29	23.21
		20	20	2.54	3.03	16.98	2.54	2.50	−1.25	0.22	0.29	26.70
		20	80	2.62	3.12	16.98	2.54	2.50	−1.29	0.24	0.32	24.95
	45	1	1	1.19	1.74	39.88	2.56	2.55	−0.50	0.09	0.15	45.26
		20	20	1.44	2.14	40.27	2.57	2.55	−0.61	0.07	0.16	58.54
		20	80	1.51	2.25	40.53	2.57	2.55	−0.64	0.10	0.18	50.71
100	25	1	1	5.18	7.69	37.43	5.53	5.39	−2.54	0.08	0.61	92.20
		20	20	5.44	8.18	38.68	5.54	5.39	−2.65	0.02	0.62	103.34
		20	80	5.54	8.35	38.97	5.54	5.39	−2.70	0.06	0.67	97.30
	45	1	1	3.58	6.05	51.26	5.59	5.49	−1.63	−0.35	0.26	275.12
		20	20	3.82	6.53	52.35	5.59	5.50	−1.73	−0.44	0.25	340.13
		20	80	3.89	6.68	52.60	5.60	5.50	−1.76	−0.41	0.30	289.49

Table 3 shows that at high return temperature the positive influence of a larger heat exchanger is most pronounced. Not only the increase on thermal power output is most pronounced here, but also the decrease in electric power is the lowest. For high return temperature the improvement in thermal power output is about 40% for low loads (or currents) and over 50-%-50% at nominal load, (100 A), compared to 15% and 38% at lower return temperatures. Whereas for the electric power the decrease at high return temperatures is only about 0.6-%-0.6% at 40 A and 1.7-%-1.7% at 100 A, compared to 1.2-%-1.2% and 2.7-%-2.7% at low return temperatures.

Especially at higher return temperature a higher effectiveness can result in primary energy savings.

High effectivenesses imply enlarging or improving (by integration of fins) the heat exchanging surface. However, this will increase cost. Therefore, based on the findings out of the simulation and the fact that the costs for the heat exchanger rise exponentially with the effectiveness, a heat exchanger with an average effectiveness of about 60% can be justified.

In [Table 3](#) is shown that at nominal load (at about ~~100A~~ 100 A DC) the net water production is strongly influenced by the effectiveness of the heat exchanger, since lower effectivenesses correspond with higher temperatures of the electrolyte circuit. This is implicated by the definition of ~~(See~~ ϵ_{HE} (see Eqs. (16), (17)) and the reduction in useful heat at higher electrolyte temperature. At higher electrolyte temperature less useful heat is available for the same fuel input, because more electrical power is generated and more heat is lost to the surroundings [\[30\]](#).

3.4 Performance study on operating conditions

Next to a sensitivity study on system parameters, different operating conditions are ~~modelled~~ modeled to evaluate possible control strategies and to define boundary conditions for the working range of the system. This evaluation is based on water management and performance, which includes electrical and thermal output as stated in [Section 2.2](#). The selected operating parameters can be divided in three groups:

- operation modus, which is defined by the electric load and the return temperature of the heating ~~circuit~~ circuit,
- control parameters, which can be adjusted to optimize performance or water management: air ratio, electrolyte flow rate and flow rate within the heating ~~circuit~~ circuit,
- environmental conditions or weather conditions, since the system is placed outdoors: air temperature and humidity.

The influence of all these parameters is examined by changing its value in different set points. In [Table 4](#) an overview is given of all these parameters and its nominal value and/or its range in which it is varied to examine its influence.

Table 4 List of operation parameters with nominal value and simulated range of variation.

Operating parameter		Value	
Description	Unit	Nominal	Min-max
DC-current	(A)	100	20–140
Return temperature (CV)	(°C)	40	20–65
Air ratio	(-)	2.5	1–5.5
Electrolyte flow rate	(kmol/h)	20.5	-
Flow rate heating circuit	(l/s)	0.278	0–0.556
Temperature surroundings	(°C)	20	1–35
Relative humidity	(%)	40	10–90

3.4.1 Operation modus

The operation mode is defined by return temperature and electric load (DC-current).

In [Fig. 3](#) is shown that electrical efficiency rises with increasing return temperature and drops with increasing current, while thermal efficiency reacts in an opposite way. As discussed in [Section 2.2](#), the (relative) primary energy savings are considered to define the overall performance. Similar conclusions as posed in Ref. [\[30\]](#) for the stack evaluation are valid. For every return temperature an optimal current can be found, which maximize system performance based on primary energy savings. For a return temperature of ~~65 °C~~ 65 °C, this is about ~~60A~~ 60 A (DC). For lower temperatures, the optimal current is lower.

Fig. 3 The thermal and electrical efficiency of the simulated system for different return temperatures as a function of DC-current.

Above the optimal current, the relative primary energy savings (RPES) are driven by the slightly decreasing electric performance. The RPES drop but remain acceptable, similar to the electric performance. At currents lower than optimal the RPES drop rapidly, similar to the thermal performance. Therefore, the optimal current can also be seen as a minimal electric load.

If we set the water management as boundary conditions for the possible operation modes, Fig. 4 illustrates in which range of the return temperature the water level can be kept under control (between -0.2 – 0.2 and 0.2 ml/s). Fig. 4 shows that under the conditions listed in Table 4, a return temperature between 46 °C and 53 °C offers the widest range for different electric loads. It is also shown that for increasing electric load the upper limit shifts to lower temperatures, while the under limit shifts to higher temperatures. Therefore, at high electric loads it is critical to work with the most appropriate return temperature. At high electric load not only a lot of water (vapour)-(vapor) is formed in the reaction, which has to be removed from the system, but also a lot of heat is generated which has to be cooled away to prevent dry heating.

Fig. 4 The net water production of the simulated system for different DC-currents as a function of return temperature.

3.4.2 Control parameters

The control parameters can make the system more flexible for these different operation modes. Three different parameters can be distinguished:

- electrolyte flow rate
- air ratio
- flow rate of the external heating/cooling circuit

Although electrolyte flow rate can be adjusted, this parameter is not altered, since in [29] the effect on water management was not proven and the high flow rate is necessary to enable a stable working of the fuel cell. At lower flow rates the chance of a too low pressure in the electrolyte rises, which could result in a direct hydrogen flow through the electrolyte into the oxygen gas chamber or vice versa. This has to be prevented at all time. Therefore, alternation of the set point for electrolyte flow rate is not considered.

Air ratio is an adjustable parameter which could be used to optimize performance. For different air conditions, temperature and humidity, the air ratio is altered for several working points, with DC currents from 40 A to

100A-100 A and return temperatures from 20 °C to 65 °C. Regarding thermal power output a negative influence of increasing air ratio is found in almost all cases. Only at very low return temperature and low currents this effect can be positive, if the electrolyte temperature is lower than the input air temperature and if almost no water will be evaporated. For higher currents (100A) (100 A) and at higher return temperatures (65 °C) the negative effect is more pronounced (up to 50% for an air ratio 5 times higher). Increasing the energy content of the air (higher temperatures and higher humidity) has almost no effect on the absolute loss in thermal power output by increasing air ratio. The relative effect will be smaller, since at higher temperatures of the surroundings, less heat will be lost. Since electrical performance is higher at higher electrolyte temperature the influence of air ratio is also negative and more pronounced at higher return temperatures, but compared to thermal performance the effect is limited (up to 5.5%) for an air ratio 5 times higher. The impact on water production of air ratio is more significant, but its influence depends on air humidity.

In Fig. 5 is shown that in dry conditions for return temperatures higher than 35 °C an air ratio can be found for which the water management is kept under control (See Section 2.1). At high currents and high return temperature this air ratio is set to a minimum, 1, meaning that there is no air excess and a high chance on the formation of water droplets and pressure drops in the air channel. In moist conditions this is not a problem, but the lower limit for return temperatures rises up to 50 °C.

Fig. 5 The net water production of the simulated system as a function of air ratio for different working points in dry and moist conditions.

A last control parameter is the flow rate of the external heating circuit. For the same heat exchanger with a certain heat transfer coefficient, h_{HX} , the effectiveness, ϵ , changes with different flow rates. At higher flow rates lower temperature changes and therefore lower effectiveness will be obtained Eqs. (16) and (17). To evaluate this control parameter the model of the heat exchanger is adapted, using follow expressions, which are based on the expression for a counter-flow heat exchanger.

$$T_{C,out} = \frac{T_{H,in} \left[C_H \cdot \left(1 - \exp \left(\frac{hA_{HX}}{f} \right) \right) \right] + T_{C,in} \cdot [C_C - C_H]}{C_C - C_H \cdot \exp \left(\frac{hA_{HX}}{f} \right)} \quad (22)$$

$$T_{H,out} = T_{H,in} - \frac{C_C \cdot [T_{C,out} - T_{C,in}]}{C_H} \quad (23)$$

$$f = \frac{C_C \cdot C_H}{C_H - C_C} \quad (24)$$

Effectiveness is influenced by flow rate. Therefore, using effectiveness as a model parameter for the heat exchanger does not allow an evaluation of the flow rate as a control parameter. The value for the new model parameter, the heat transfer coefficient of the heat exchanger, is based on an effectiveness of 80% and the nominal flow rate of the external circuit, 1000 l/h.

In Fig. 6 the net water production is evaluated at different flow rates, electric loads and return temperature. The condition of the surroundings are set at 20 °C with a relative humidity of 40%. It is shown that for every return temperature of the external circuit, lower than 55 °C, a flow rate can be found for which the net water production is zero. At higher temperatures, there will always be evaporation at the nominal conditions of the surroundings. At flow rates lower than in the electrolyte circuit, the flow rate has a large impact on the heat transfer (See Fig. 7). At flow rates higher than the electrolyte flow rate, which is about 0.1025 l/s, the thermal power output and thereby also the net water production remains almost constant.

Fig. 6 The net water production of the simulated system as a function of the flow rate of the external cooling/heating circuit for different electric loads and return temperatures.

Fig. 7 The thermal power output of the simulated system as a function of the flow rate of the external cooling/heating circuit for different electric loads and return temperatures.

The optimal flow rate regarding water management, will be higher for higher electric loads, because more cooling of the electrolyte will be necessary. Since at high return temperatures, the cooling effect is decreased, flow rate has to be increased at higher return temperature to optimize water management. As higher flow rates (still lower than the electrolyte flow rate) result in more cooling of the electrolyte, electric power output will decrease with flow rate as electrolyte temperature will drop, while thermal power output will increase.

To optimize system performance both water management and energy performance are to be taken into account. In Ref. [29] it is found that for every electric load an optimal electrolyte temperature can be chosen, based on relative primary energy savings (See (see Section 2.2.2)). Both return temperature and flow rate will influence electrolyte temperature (See (see Eqs. (23) and (24)). Therefore, several energetically optimal combinations of return temperature and flow rate can be found for a given electric load. As stated earlier, for most combinations of electric load and return temperature a flow rate can be found which enables a stable water management.

Combining both criteria for every electric load an optimal return temperature and flow rate can be found, enabling a stable water management and maximizing primary energy savings. A summary of these results are presented in Table 5.

Table 5 Model results for to optimize the external cooling circuit(return temperature and flow rate of the cooling circuit), based on primary energy savings and a balanced water management.

Set point	Optimal heat integration		Thermal output		Electric output	
DC-current (A)	Return temperature (°C)	Flow rate (l/s)	Power (kW)	Efficiency (%)	Power (kW)	Efficiency (%)
40	34	0.039	1.43	24.65	2.57	44.2
80	39	0.082	5.6	38.62	5.53	38.1

Based on electric load, the temperature range of the external circuit shifts from 43–34 °C for 40A 40 A to for 80A, 56–39 °C for 80 A. These temperature ranges are common for heating systems with radiators. At higher working points the temperature range can even be used to heat up sanitary water. In this way they can be compared to other heating technologies like condensate boilers, since they require a similar heating system. As heat pumps require a (more expensive) heating system at lower temperature (typically 35–25 °C) a comparison based only on production efficiencies would be incomplete.

The same evaluation could be made based on a fixed return temperature. In this case higher electric loads will result in better performance. The only limitation here is water management. A high return temperature will limit electric load in order to enable a stable water management.

4 Comparison with other technologies

With the results of the analysis in Section 3.4.1, the part load ~~behaviour~~-behavior of the AFC-based micro-CHP-system is discussed in comparison with other technologies. Next to a comparison of part load ~~behaviour~~-behavior, also the influence of supply temperature on performance will be discussed.

It has to be stated clearly that the authors have no intention to compare the actual efficiencies themselves of all the different technologies, as these were tested on different fuels (~~e.g.~~-(e.g. hydrogen versus natural gas) and based on different data. However, they still give a representative indication of the ~~behaviour~~-behavior in part load operation.

4.1 Description of the systems used in this comparison

4.1.1 Load factor and external heating circuit of the AFC

To compare the alkaline fuel cell system with other technologies a load factor and return temperature, should be defined. The nominal electric current of the present system is set at 100 A. This value is used to determine the nominal load. The electric load factor is defined as the ratio of the generated electric power to the electric power at nominal load. Based on an optimized heat integration (discussed at the end of Section 3.4) the nominal return temperature is set at 40 °C. At nominal conditions this corresponds to a heating system working on a ~~regime~~-60/40 °C-regime, which is acceptable for building heating systems. To complete the thermal integration the flow rate of the external circuit will be used as a control parameter for the water management. In this way a realistic comparison can be made with other technologies.

4.1.2 Data for other technologies

The AFC-system is compared with other technologies suitable for micro-CHP. The used CHP technologies are listed in Table 6.

- Two gas engines running on natural gas: the Senertec Dachs [5,4] and the Ecopower by PowerPlus. Their partial load and temperature ~~behaviour~~-behavior is discussed in [34] and combined with the total efficiencies in Refs. [5,4]. In Ref. [5] also more data about the partial load performance of the Ecopower was presented.
- Two Stirling engines on natural gas: Solo and the Whispergen/Microtech. The partial load ~~behaviour~~-behavior is described in Ref. [5]. the partial load ~~behaviour~~-behavior of the Whispergen/Microtech is ~~modelled~~-modeled in [35] to be constant, since it is on/off regulated.
- A micro turbine on natural gas. The partial load ~~behaviour~~-behavior is discussed in ~~ref~~-Ref. [36].
- A diesel engine. In ~~ref~~-Ref. [37] the partial load ~~behaviour~~-behavior is simulated in Aspen for small CHP units. The partial load ~~behaviour~~-behavior is also presented in ~~ref~~-Ref. [34], where similar results were presented.
- A hydrogen engine. In ~~ref~~-Ref. [37] the hydrogen engine is compared with the diesel engine. These data were used to make a comparison.
- A PEM and a SOFC based fuel cell system. Next to a list of heat based micro-CHP-systems, the AFC-system is also compared to other fuel cell technology. Therefore, also a hydrogen ~~fuelled~~-fueled PEM fuel cell and a gas filled SOFC are considered as alternative CHP-systems. The data used in this comparison are based on the PEMFC system evaluation made in Ref. [38] and on an techno-economic analysis of a SOFC-based micro-CHP in Ref. [39].

Table 6 Specifications (defined at LHV) and references of the micro-CHP technologies, used for comparison with the AFC.

CHP	P_E	η_E (%)	P_T	η_T (%)	References
Gas engine (Senertec)	5.5	27.7	12.5–15	63.6	[5,4,34]
Gas engine (Ecopower)	4.7	24.7	12.5	64.2	[5]
Stirling (Solo)	2–9.5	26.8	8–26	71.7	[5,4]
Stirling (Whispergen/Microtech)	1.1	12–15	6.6	80–90	[35,4]
Micro turbine	30	27	54	49	[36]
Diesel engine	6.53	31	10.7	50	[37,34]

Hydrogen engine	6.50	30	11.8	54	[37]
PEMFC-system	4.8	48	4.2	42	[38]
SOFC-system	5.0	38	6.8	52	[39]

4.2 Part load ~~behaviour~~behavior

Figs. 8-11 show a comparison of the part load ~~behaviour~~behavior of several micro-CHP-technologies. The data is a collision of experimental, simulated and extrapolated references, which are listed in Table 6. In the references most efficiencies were based on lower heating values (See(see Table 6). In the figures these efficiencies were translated into efficiencies on higher heating value, allowing a more appropriate comparison, since the prime energy source differs for the different options. Although a complete and honest comparison isn't possible for this reason, it is shown that the ~~behaviour~~behavior of fuel cell-based CHP-systems is completely different from the other technologies.

Fig. 8 Comparison of electric efficiencies (defined at HHV) at partial load of several micro-CHP-technologies listed in Table 6.

Fig. 9 Comparison of thermal efficiencies (defined at HHV) at partial load of several micro-CHP-technologies listed in Table 6.

Fig. 10 Comparison of total efficiencies (defined at HHV) at partial load of several micro-CHP-technologies listed in Table 6.

Fig. 11 Comparison of heat to power ratio at partial load of several micro-CHP-technologies listed in Table 6.

In Fig. 8 is shown that the electric efficiency of the fuel cell-system rises at part load. This **behaviour** is completely different for all other technologies. Next to that the electric efficiency lies a lot higher than the others, which indicates the potential. It has to be taken into account however that hydrogen is used as primary energy source and that a reforming step could further reduce this efficiency, next to other system losses.

In Fig. 9 is shown that the thermal efficiency is rather low, certainly at low load, due to a poor system design regarding heat losses.

The total efficiency reaches the same level of other technologies from a load factor of about 80%. This is shown in Fig. 10. At smaller loads the total power **utilisation** is a lot smaller due to the poor thermal efficiency. This is compensated by a low heat to power ratio (See Fig. 11), which is interesting for applications with a low heating demand, like low energy buildings.

Comparing the fuel cell-based systems, it is shown that, in stationary conditions, the presented AFC-system is competitive with the PEMFC-based systems, regarding electrical performance. Besides this, AFC-systems have a higher thermal efficiency. This thermal performance is possible, because the circulating electrolyte facilitates the heat transfer from the reaction to the external circuit. SOFCs have to deal with higher operating temperatures which results in a poor thermal efficiency at part load.

4.3 Influence of supply temperature

Within building applications, supply temperature is highly dependent on configuration of the heating system. Therefore, its influence is investigated. The influence of supply temperature on system performance varies for different technologies. In Section 3.4.1 it is shown that supply temperature has a positive influence on electrical performance (about +3% for a temperature rise of 30 °C). A similar effect is to be expected for PEMFC, since the operating temperature of low-temperature fuel cells is strongly influenced by the supply temperature. For SOFC this effect is to be neglected, since operating temperature is hardly affected, due to the high operating temperatures.

In [5] some heat based micro-CHPs were tested. It was shown that for a Stirling based micro-CHP, the electrical efficiency drops about 10% for a temperature rise of 30 °C. For gas engines no influence is noticed in [5].

Regarding thermal efficiency, supply temperature has a negative influence for all systems, since less heat sources are available for useful applications and/or due to higher radiation losses.

5 Conclusion

An alkaline fuel cell model is integrated in a model of a CHP-system, according to an existing configuration. First, a sensitivity study on component performance is executed. These results can be used in a cost-effective system design.

- Improvements in inverter and purge efficiencies will directly result in a higher performance. It is recommended to select a high quality inverter, depending on cost and the number of operating hours.
- The selection of pumps or fans isn't critical for system performance, therefore it is recommended to select standardized pumps and fans to reduce costs.
- The effectiveness of the heat exchanger can be chosen about 60%, which implies a relatively small heat exchanger.

With a fixed flow rate of the cooling circuit (See (0.278 l/s (see Table 4))), a return temperature between and 46 °C and 53 °C has to be maintained to enable a stable working range regarding water management. At higher electric loads this working range is narrowed to a single return temperature. For this return temperature an optimal electric load can be found, although higher loads will only result in small performance reductions.

To control water management and optimize energetic performance, several options were examined. Since an adjustable air ratio has only a significant influence at the right environmental conditions, the flow rate of cooling circuit is a much better control parameter to control water management.

For every electric load a range of return temperatures can be found for which the water management can be kept under control by adjusting flow rate of the cooling circuit. Within these boundaries, an optimal return temperature (combined with a certain flow rate) can be found for every electric load, based on primary energy savings. This optimal return temperature remains more or less the same and is about 40 °C. This means the present set-up can be used in a building with a conventional heating system.

A comparison is made with other micro-CHP-technologies. It is shown that for buildings with a decreasing heat demand and increasing electricity demand a fuel-cell based micro-CHP has better results. These results are also influenced by the temperature of the heat distribution system. Compared to a PEMFC based system the AFC system has a better thermal performance and similar electrical performance in stationary conditions. Further research should be focused on the more dynamic aspects of thermal behaviour behavior of the micro-CHP.

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Highlights

- Sensitivity analysis on system configuration of the AFC as a micro-CHP.
- Flow rate in the secondary heating circuit can be used to control water management.
- Part load behavior of fuel cells is compared to other micro-CHP technologies.
- For future energy demand in buildings fuel cells have the best performance.

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