Investigating route gradient and thermal demand on hydrogen fuel cell electric bus energy consumption

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Abstract

In 2022 in the UK, the transport sector was the largest single contributing sector to greenhouse gas emissions, responsible 34% of all territorial carbon dioxide emissions [1]. In the UK there is growing uptake in zero emission powertrain technologies, with the most promising variants based on battery electric or hydrogen fuel cell electric configurations. Given the limited number of fuel cell electric buses currently in operation in Europe, vehicle models and simulations are one of the few methods available to estimate energy consumption and provide the necessary increased confidence in operating range. This paper investigates the impact of route characteristics, thermal demand and coefficient of performance of different heat source configurations on the operational energy consumption of fuel cell electric buses. Using a MATLAB/Simulink model, the total energy demand of a vehicle operating in different route/elevation profiles is considered. The findings from this study show that implementing a waste heat recovery system, which recovers excess heat produced by the fuel cell system, in a traditional heat pump based HVAC system can reduce the amount of excess fuel consumption required for providing HVAC services by up to 70% over the course of a duty cycle, so long as high levels of waste heat can be recovered. It also shows that through the use of waste heat recovery systems it is possible to replace the heat pump based HVAC system with a much simpler positive temperature coefficient heater system and still achieve or improve upon the performance of a traditional heat pump based HVAC system.

Introduction

Recent estimates report that 4,738 fuel cell electric buses (FCEB) are currently operating across 16 countries, with 88% of these vehicles operating in China. By 2030 it has been estimated that there will be 45,000 FCEBs in Europe, 1,200 FCEBs in Japan and 11,600 FCEBs and fuel cell coaches in China with increasing to 250,000 FCEB in global operations by 2050 [2]. FCEBs are zero emission at the point of operation, with no greenhouse gases emitted at the tailpipe (referred to as tank-to-wheel TTW emissions). Greenhouse gases are still emitted during the production of both the vehicle and the hydrogen which is used to drive the power train (referred to as the well-to-tank WTT emission) [3].

Given the small number of fuel cell electric buses currently in operation, rule of thumb methods and vehicle models simulations can be used to estimate energy consumption and examine performance of FCEBs. Within the literature, the energy consumption of FCEB has been approximated with rule of thumb methods, where average consumption data (kg-H2/km or kWh/km) is combined with distance to provide route level consumption. This method has been used as part of economic and environmental assessments [3], [4] with values of 1.8 kWh/km [4] and 10kg-H2/100 km [3] reported. Simulation based models which utilize parameters relating to vehicular and operational parameters alongside vehicle drive cycles that have examined FCEB performance have reported average consumption of 6.17 kg-H2/100km [5] for a double deck fuel cell bus and 1.849 kWh/km (5.55 kg-H2/100 km) and a standard deviation of 0.069 kWh/km (0.21 kg-H2/100 km) for a single deck FCEB [6]. Whilst there are a number of papers which examine the relative significance of parameters influence energy consumption of battery electric buses the same cannot be said for fuel cell buses. For battery electric buses route gradient, average journey speed and numbers of stops have all been shown to have a significant influence on energy consumption [7]–[9]. The effect of ambient temperature and resulting thermal demands on battery electric and fuel cell vehicles has likewise been recognized within the literature. Energy consumption on battery electric buses increased by up to 27% for a “worst-case” scenario (bus fully loaded with air conditioning on, operating in temperatures 30-35°C) when compared with the same vehicle operating in moderate ambient conditions (15-25°C) [10]. Another study by [11] undertook well-to-wheels (WTW) based analysis of fuel cell electric vehicle technologies for the US based school buses. They found that local climate and duty cycles impacted the fuel consumption of vehicles and that alongside regional electrical grids and fuel production pathways resulted in multilayered variations in WTW analysis. The study highlighted the need for adaptive load management approaches and accurate characterization of accessory loads. Overall previous studies have shown that there is large uncertainty associated with operational energy consumption values, with limited examination of the influence of route characteristics and thermal demands for FCEBs.

Within the transport sector proton exchange membrane (PEM) fuel cells are the dominant fuel cell technology and generate electricity to power the drive power train via a conversion of pressurized hydrogen gas. This electrochemical reaction is exothermic, with between 45-60% of the total hydrogen energy being converted into heat. Optimal efficiency of a PEM fuel cell requires an operational temperature of between 60-80 °C [12]. Temperatures in excess of this range will dehydrate the membrane and negatively impact the productivity, stability and long term durability of the fuel cell stack. Similarly the performance other components including batteries, motors and electronics are negatively impacted by temperatures requiring the need for thermal management systems. To ensure thermal comfort of passengers, extra heating is provided by PTC (positive temperature coefficient) heaters or air source heat pumps. The addition of these thermal systems however can increase the overall weight of the vehicle and decrease range and fuel economy [13]. Given the excess heat generated by fuel cells, there has been significant interest in the use of heat recovery systems and effective vehicle thermal management systems to offer improved performance efficiency. A paper by [14] examined different heat recovery options based on thermochemical heat storage units used within a for a high temperature fuel cell range extender vehicle for heating purposes, with range extended by up to 17%. Work undertaken by [13] simulated the performance of fuel cell trucks operating on four drive cycles. Three system variations were considered (including PTC, heat pump system supplied with external air and a heat pump system utilizing waste heat generated from the PEM fuel cell). Both heat pump configurations offered efficiency gains when compared to the PTC, with the largest savings (16.7%) for heat pump system utilizing recovered waste heat.

More recent work related directly to buses include [15], which examined four different thermal management scenarios modelled for a bus operating in Spain. Waste heat recovered from a PEM fuel was used for cabin heating, battery pre-heating and hybrid solution of cabin and battery preheating resulting in 7%, 4% and 10% energy savings respectively. Another study by [16] simulated the cooling and heating performance of fuel cell bus (80 kW fuel cell and 105 kWh battery) operating on the New European Driving Cycle (NEDC). They reported a range of 505.68 g to 840.16 g of equivalent hydrogen consumption depending on a range of ambient conditions, heat recovery rates and heat limits to temperature of battery and fuel cell. Both of these papers highlight the importance of effective thermal management and potential savings of heat recovery but neither paper examine directly the influence of different route characteristics on fuel cell operation, with route elevation profile omitted from both.

In order to accelerate the uptake of FCEB it is vital to understand the interdependency of operational parameters such as route characteristics, thermal demand and heat recovery in the context of the overall power train energy consumption. This would support public transport operators a detailed understanding of energy consumption of vehicles and ensure the fulfillment of timetable requirements and the appropriate planning for refueling infrastructure.

Methodology

This study aims to determine and compare the energy consumption of a generic fuel cell electric bus (FCEB) operating on drive cycles with varying road gradients, installed with either a heat pump (HP) HVAC system or waste heat recovery (WHR) HVAC system. Bus operational energy consumption is simulated for a range of synthetic drive cycles using a backwards facing powertrain model. This section discusses the methods for drive cycle selection, as well as vehicle modelling and simulation.

Fuel Cell Electric Vehicle Model

The fuel cell electric bus simulation model used in this study (Figure 1) was developed in order to predict the energy consumption of the FCEB powertrain, as well as the HVAC loads associated with the operational scenarios. The modelling approach and strategy, as well as the governing equations have been published previously by [5], [17]–[20]. Fully validated versions of the model for generic BEB and FCEB vehicles have been used in studies conducted by [21], [22].

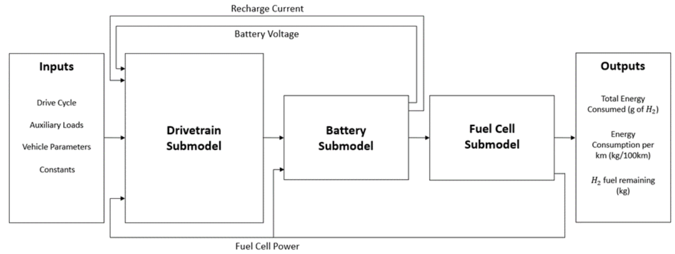

Figure 1. Schematic of Fuel Cell Electric Vehicle Simulation including model inputs and outputs.

Vehicle Operating Conditions

The generic double deck FCEB considered in this study is of the same chassis configuration of that described by [21], [22], with the specification that is realistic for today's public transport marketplace. The vehicle is assumed to have a 70 kW fuel cell based on those commercially available, with an average Balance of Plant efficiency of 90%. The vehicle was assumed to be operating at 50% seated capacity, the same test conditions as used in the ZEB certification process [23]. The key vehicle parameters, powertrain components and characteristics that are used to model the FCEB are summarized in Table 1.

Table 1. FCEB vehicle specification.

Vehicle Parameter	**Value**
Axle Configuration	4 x 2
Body Type	Double deck
Electric Motor	240 kW
Fuel Cell Power (Maximum)	70 kW
Balance of Plant Efficiency	90%
Hydrogen Tank	25 kg
Battery Capacity	27 kWh
Battery Chemistry	Lithium titanate (LTO)
Battery SOC Operating Window	65 – 80%
Number of Passengers	37
Unladen Weight	14,300 kg
Test Weight	16,900 kg
Frontal Area	10.269 m^2
Auxiliary Loads	4.7 kW
Rolling Resistance	0.007
Coefficient of Drag	0.6

This analysis will investigate the vehicle operation under a range of different thermal conditions utilizing three different thermal systems to provide heat to the vehicle cabin (Figure 2). The thermal load which these systems must provide to heat the cabin, known as the cabin thermal demand $\dot{Q}_{cabin}$, is scaled from 0kW to 20kW to represent a range of different ambient conditions.

Thermal Systems and Model

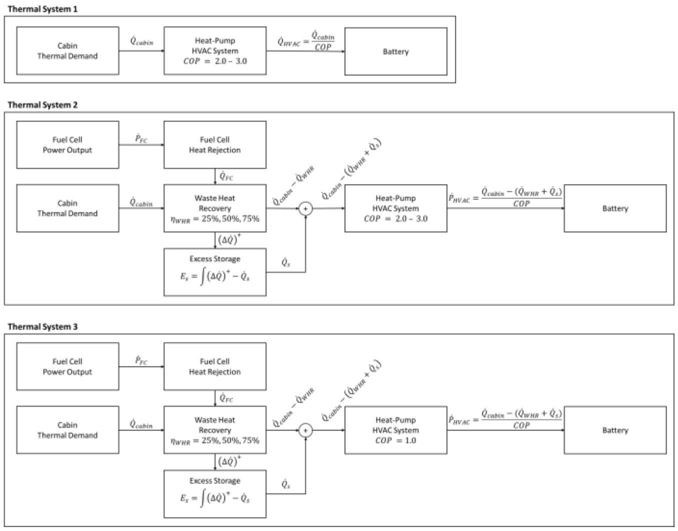

Figure 2. Overview of the three thermal systems; a standard HP-based HVAC system; a WHR system with a HP; a WHR system with a PTC heater.

System 1 represents a basic heat pump style HVAC system with no consideration of waste heat recovery (WHR), this will provide a baseline to assess the impact of WHR. This type of system is common in BEB operation and will typically have a Coefficient of Performance (COP) of between 2.0 and 3.0; that is, for every 1 kW of electrical energy that is consumed they will produce 2 – 3 kW of heat energy to be supplied to the cabin. Under system 1 the electrical power draw from the battery required to provide HVAC services ($\dot{P}_{HVAC}$) will therefore be given by the equation:

$$\dot{P}_{HVAC} = \frac{\dot{Q}_{cabin}}{COP} \tag{1}$$

System 2 represents a similar setup, however this time including a WHR system. This system utilizes the heat generated by the fuel cell during its operation to provide heat to the cabin, reducing the amount of heat which must be supplied by the HVAC system. The amount of waste heat available ($\dot{Q}_{FC}$) is determined by the fuel cell power output $\dot{P}_{FC}$ and the fuel cell efficiency η_{FC}, which are both functions of the fuel cell performance. The fuel cell waste heat can be calculated using equation:

$$\dot{Q}_{FC} = \dot{P}_{FC}\left(\frac{1}{\eta_{FC}} - 1\right) \tag{2}$$

Not all of this heat will be recoverable however, therefore the amount of useable waste heat $\dot{Q}_{WHR}$ will be a function of the WHR system efficiency η_{WHR}:

$$\dot{Q}_{WHR} = \eta_{WHR}\dot{Q}_{FC} \tag{3}$$

There are limited examples in the literature detailing the waste heat recovery process for medium- to heavy-duty application fuel cells, therefore this study will consider three levels of WHR: low recovery ($\eta_{WHR} = 25\%$), medium recovery ($\eta_{WHR} = 50\%$), and high recovery ($\eta_{WHR} = 75\%$).

The HVAC system will primarily utilize the heat available from the WHR process to satisfy the cabin thermal demand. When the amount of heat recovered is greater than the cabin thermal demand any excess heat will be used to increase the temperature of the coolant in the WHR system, acting as a type of thermal storage device rather than simply rejecting the excess heat to the environment. The difference in the amount of heat recovered and the thermal demand is $\Delta\dot{Q} = \dot{Q}_{WHR} - \dot{Q}_{cabin}$, which can be broken down into $\left(\Delta\dot{Q}\right)^+$ when there is more waste heat generated than thermal demand (i.e. $\Delta\dot{Q} > 0$), and $\left(\Delta\dot{Q}\right)^-$ when there is less waste heat generated than thermal demand (i.e. $\Delta\dot{Q} < 0$).

The amount of thermal energy stored is given as E_s, where:

$$E_s = \int \left(\Delta\dot{Q}\right)^+ dt \tag{4}$$

When the waste heat is insufficient to meet the thermal demand then this gap will first be satisfied by the stored thermal energy through the load $\dot{Q}_s$; equation 4 can therefore be rewritten as:

$$E_s = \int \left(\Delta\dot{Q}\right)^+ - \dot{Q}_s \, dt \tag{5}$$

The amount of stored thermal energy which is supplied per second to fill the gap in demand is equal $-\left(\Delta\dot{Q}\right)^-$ when there is sufficient energy available, or the remaining stored energy if not. This can be written as:

$$\begin{aligned} \dot{Q}_s &= -\left(\Delta\dot{Q}\right)^-, & when\ E_s > -\left(\Delta\dot{Q}\right)^- \\ \dot{Q}_s &= \frac{E_{stored}}{\Delta t}, & when\ E_s \le -\left(\Delta\dot{Q}\right)^- \end{aligned} \tag{6}$$

The total amount of heat which must be supplied from the HVAC system can now be calculated by subtracting the recovered and stored heat from the cabin thermal demand, and the electrical power load can be determined by considering the COP of the system, giving the final equation:

$$\dot{P}_{HVAC} = \frac{\left(\dot{Q}_{cabin} - \left(\dot{Q}_{WHR} + \dot{Q}_{stored}\right)\right)}{COP} \tag{7}$$

For a heat pump system this COP will range from 2.0 – 3.0.

System 3 is based on system 2, however instead of using a heat pump to supply additional heat, a positive temperature coefficient (PTC) heater is installed. These systems are simpler than a heat pump system which has benefits of reducing the number of parts, cost, and maintenance for the vehicle. However, a PTC heater is only able to output as much heat as the electrical load provided, therefore these systems have a maximum COP equal to 1: every 1kW of heat required will consume 1 kW of power from the battery. The amount of power consumed by the HVAC system is therefore given by equation 7, using a $COP = 1.0$.

The thermal system is modelled as a sub-component within the inputs section of the FCEV model (Figure 1) and contributes to the auxiliary loads. Figure 2 shows how each of the three thermal systems are configured to calculate the required electrical load for the HVAC system; within the model only one component is implemented which resembles systems 2 and 3, and for system 1 analysis the waste heat recovery is set to 0%. The power output by the fuel cell stack along with its efficiency are determined by the Fuel Cell Submodel, and the amount of waste heat available is calculated based on equation 2. The amount of heat energy required by the HVAC system is found by subtracting the useable waste heat and the returned stored heat from the thermal demand. Limits are applied to stop the HVAC system being passed a negative required heat load, and to stop the thermal storage from having a negative supply of energy. In this study the thermal storage is treated as having an infinite capacity.

It has been assumed that the fuel cell specific thermal management system acts independently of the HVAC thermal systems considered within this research. The fuel cell thermal management system maintains a relatively constant thermal environment within the fuel cell stack, and it is the heat which would nominally be rejected by the stack which is recovered in the WHR process. Further, it is assumed that there is no weight benefit/penalty associated with changing between different thermal systems to allow for a more direct comparison of results.

Drive Cycle Generation / Selection

To compare the operation the FCEB with different fuel cell specifications and thermal systems, a range of synthetic drive cycles were created which cover bus operation on high, intermediate and low average road gradients. The synthetic drive cycles were created for the city of Belfast, using the same method as described by (Blades et al., 2021). For this study, drive cycle A was created to simulate driving on roads with a high average gradient (2.20%), drive cycle B to simulate driving on intermediate average road gradients (0.50%), and drive cycle C to simulate driving on low average road gradients (0.00%). Table 2 summarizes the synthetic drive cycles used in this study.

The drive cycles for the return journey of each of drive cycles A, B and C were also created and named A_{Rev}, B_{Rev} and C_{Rev}. Three duty cycles were created, each around 160 km in distance, in order to simulate the FCEB operating in a "route locked" manner, with the bus driving back and forth on the same route over an entire day. Duty cycle A (157.4 km) consists of 11 repeated runs of A followed by A_{Rev}, duty cycle B (155.1 km) consists of 9 repeated runs of B and B_{Rev}, and duty cycle C (167.5 km) consists of 12 repeated runs of C and C_{Rev}. By simulating on a duty cycle, it allows for the average energy consumption to be calculated across the operational range of the fuel cell power output and battery SOC.

Table 2. Drive cycles for the city of Belfast considered in this study.

Drive Cycle	A	A_{Rev}	B	B_{Rev}	C	C_{Rev}
Duration (mins)	26.3	26.4	35.2	35.1	19.5	19.2
Distance (km)	7.15	7.14	8.7	8.56	6.97	6.99
Average Road Gradient	2.20%	-2.18%	0.50%	-0.52%	0.00%	0.00%
Average Speed (km/h)	16.3	16.2	14.8	14.6	21.4	21.9
Maximum Speed (km/h)	46.6	46.4	48.3	42.8	45.9	46.6
Number of Stops per km	3.6	3.6	4.1	4.3	1.4	1.4
Percentage Time Driving	78.8	78.6	74.5	77.6	87.0	87.3
Percentage Time Stationary	21.2	21.4	25.5	22.4	13.0	12.7
Average Positive Acceleration (m/s2)	0.49	0.55	0.51	0.52	0.43	0.46

Results

In this section the results of the simulations for the generic FCEB are presented and discussed, for each of the thermal systems in Figure 2 over three duty cycles consisting of high, intermediate and low average road gradients. Analysis will initially compare simulation results for operation over the different duty cycles with zero heating load/thermal demand from the HVAC system. This allows the baseline energy consumption of the FCEB over the different road gradients to be determined for the driving of the vehicle alone, and so the energy consumption associated with the different thermal systems can be analyzed. The second set of simulations are for the FCEB operating over the different average road gradient duty cycles with the HP HVAC system (Thermal System 1). The third set of simulations are for the FCEB operating over the different average road gradient duty cycles with a WHR system supplementing the HP HVAC system (Thermal System 2), considering low recovery (25%), medium recovery (50%), and high recovery (75%). Finally, simulations are performed for an alternative HVAC system that consists of a WHR system along with PTC heater in the absence of a HP (Thermal System 3); again considering low, medium and high waste heat recovery. The analysis conducted is for the specified FCEB vehicle modelled and so the results are only valid for the exact specification considered.

Zero Heat Load Simulation Results

To gain an understanding of the energy consumption of the FCEB operating under different route gradients, with no thermal system, simulations were performed for the vehicle on each of duty cycles A, B and C. On duty cycle A, the FCEB is predicted to consume 8.563 kg-H2/100 km, on duty cycle B the energy consumption was predicted to be 6.938 kg-H2/100 km, and on duty cycle C an energy consumption of 5.630 kg-H2/100 km was predicted. These simulations show that as the average road gradient of the route increases, so too does the energy consumption of the FCEB. These points are marked as circles on Figure 3.

Thermal System 1 – Heat Pump HVAC System Simulation Results

Figure 3 shows the results of the simulations conducted over each of the three duty cycles for the FCEB operating with a HP HVAC system to satisfy a range of cabin thermal demands. Assuming that the COP will be in the range of 2.0 – 3.0, the shaded bands on Figure 3 represent the range of expected results on each duty cycle. From the results it is possible to determine the additional fuel consumption associated with operating the HP HVAC system. It is clear to see that, as thermal demand increases, so does the hydrogen consumption of the FCEB. For all systems the maximum fuel consumption occurs at 20 kW of thermal demand, with a HP system COP of 2.0; for duty cycle A these conditions add an additional 4.410 kg-H2/100 km, or 51% of the baseline drive energy, to the FCEB energy consumption;

for duty cycle B this results in an additional 5.324 kg-H2/100 km, or 77% of the drive energy, to the energy consumption; and on duty cycle C this is predicted to add an additional 3.976 kg-H2/100 km, or 71% of the drive energy, to the energy consumption. These points are marked in diamonds on Figure 3.

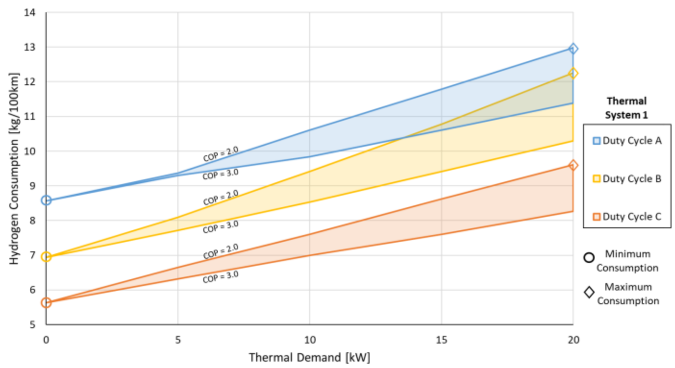

Figure 3. Fuel consumption predictions for the low, medium, and high average road gradient duty cycles over a range of thermal demands and system COP's.

Based on these results the largest increase in fuel consumption, both in terms of kg-H2/100km and percentage increase, occurs on the medium gradient duty cycle (B). This duty cycle is also the most sensitive to system COP, with a range in results at 20 kW of demand covering 10.300 – 12.262 kg-H2/100 km, which creates an overlap on Figure 3 with the results from the high gradient duty cycle (A). The fuel consumption on the medium gradient duty cycle is more affected by thermal demand when compared to the high gradient duty cycle due to the amount of regenerative energy gain available from high gradient routes. The regenerative loads affect the battery state of charge (SOC) and therefore fuel cell operation. An example of this can be seen in Figure 4, where a single drive cycle of duty cycles A and B are analyzed (i.e. route A & A_{rev}, and route B & B_{rev}). The profile times have been normalized to zero to allow for easier comparison, and the starting SOC will be dependent on the route gradient and fuel cell power output. The fuel cell control logic aims to maintain the SOC within a window of 65 – 70%, however the high negative gradients on route A cause the battery state of charge to increase above 70% which causes the fuel cell power output to reduce. On route B the road gradients are lesser, limiting regenerative braking and as such there are no large increases in battery SOC which would contribute to lowering the average fuel cell power output/fuel consumption.

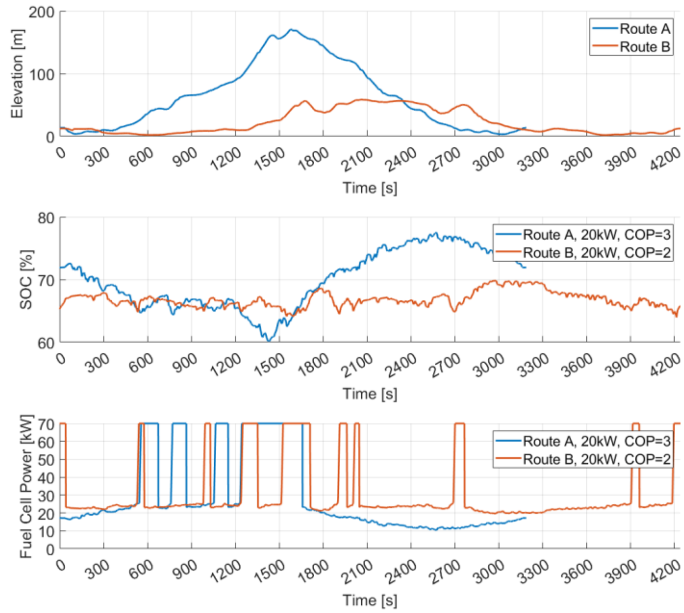

Figure 4. Vehicle elevation, battery SOC, and fuel cell power output profiles over 1 drive cycle on duty cycles A (high gradient) and B (medium gradient) at high thermal demand.

High road gradients can even have an impact on results at lower thermal demands, as the results for duty cycle A in Figure 3 show negligible difference between the fuel consumption at 5 kW thermal demand for a HP system with a COP of 2.0 compared to a HP system with a COP of 3.0. This is similarly due to the complex way in which regeneration of battery SOC affects the fuel cell control parameters. Figure 5 shows the SOC and fuel cell power output profiles for route A at 5 kW thermal demand; under these conditions with a HP system COP of 3.0 the SOC rises under the regenerative loads until it reaches 80% at which point it commands the FC system to power off. This offsets the higher power output at the start of the cycle and the net effect is that the fuel consumption is similar to a system with a COP of 2.0 which remains active for the duration of the cycle but does not require the same peak loads at the start of the cycle.

The impact of these results on vehicle operation is that FCEB range reduction due to thermal load conditions is not constant. The additional load due to providing HVAC services is a function of both the thermal demand, system COP, and the route profile itself.

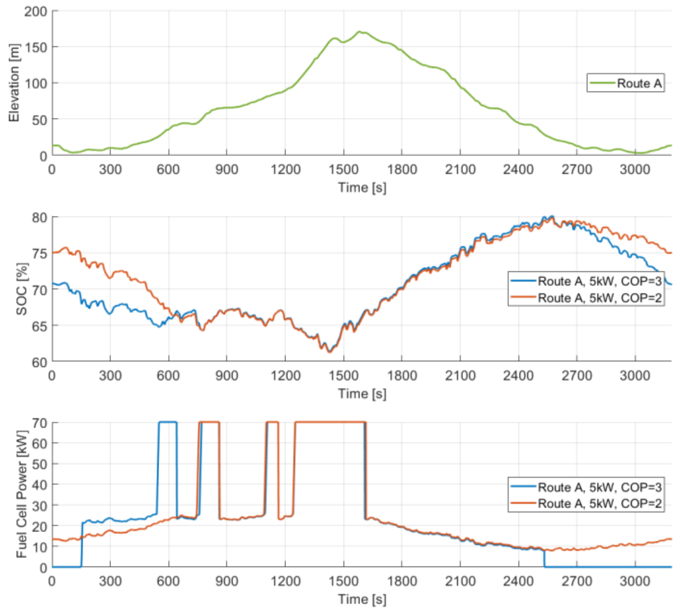

Figure 5. Vehicle elevation, battery SOC, and fuel cell power output profiles over 1 drive cycle on duty cycle A (high gradient) at low thermal demand.

Thermal System 2 – Waste Heat Recovery HP HVAC System Simulation Results

Figure 6 shows the results of simulations performed for the FCEB over duty cycles A, B and C, with the thermal demand satisfied using a HP HVAC system that also includes low (25%), medium (50%) and high (75%) WHR system. For comparison, the energy consumption for each duty cycle without WHR is also plotted from Figure 3. At low thermal demands, from 0 – 10 kW, WHR is active and can be seen to provide significant fuel savings in comparison to the HP HVAC system without WHR. When thermal demand is low, at 5kW, the amount of additional fuel saved can be as much as 100% for medium and high WHR systems, and over 72% for low WHR systems. This shows the significant benefits that can arise from implementing a WHR system.

Even under more challenging conditions when thermal demand is high (20 kW) fuel savings are significant. The energy consumption savings can range from 28 – 45% for low recovery systems, 47 – 60% for medium recovery systems, and up to 60 – 70% for high recovery systems. The amount of fuel savings is dependent on the road gradient profile. As the fuel cell system must work harder to provide the necessary drive power, it also produces more waste heat which can be recovered. This is why the largest savings are observed on the high gradient duty cycle (A), with a reduction of up to 0.828 kg-H2/100km in fuel consumption. The biggest percentage reductions, however, are observed on duty cycles B and C where their smaller initial fuel consumption results in larger percentage gains due to waste heat recovery. This has implications for route scheduling abilities, where these vehicles would see their total range decrease limited the most.

It is clear that the addition of the WHR to the conventional HP HVAC system results in the reduction of FCEB energy consumption. Even the low WHR system can be seen to result in high savings, and the savings at low thermal demands, where the bus will operate for the majority of its life span, are significant.

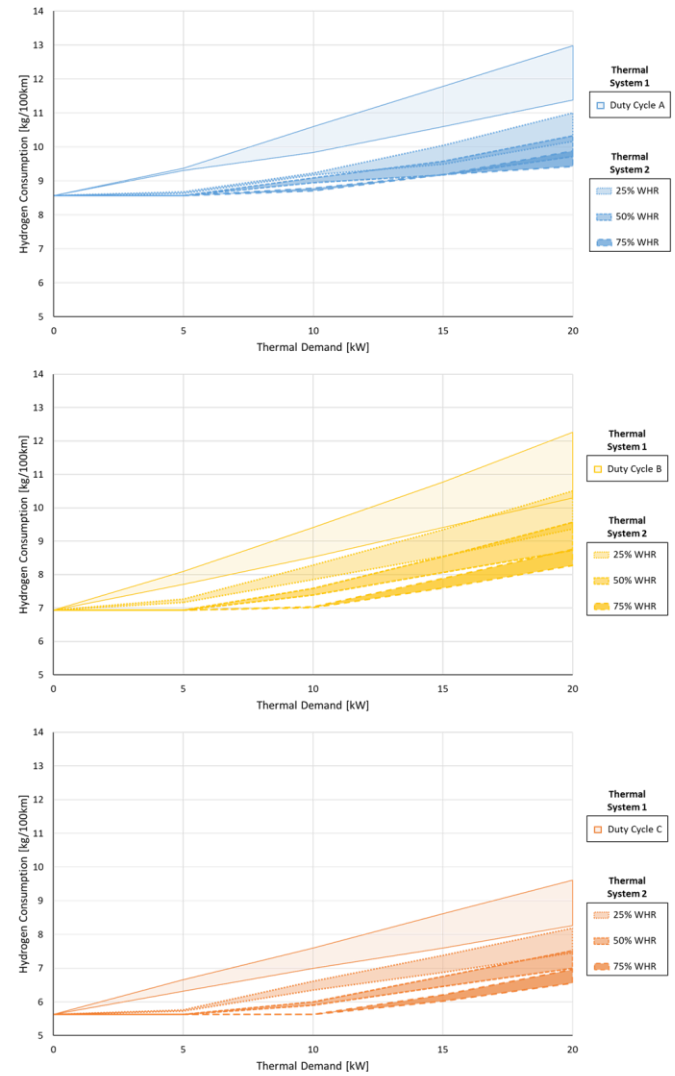

Figure 6. Fuel consumption predictions over the three duty cycles comparing a standard HP-based thermal system performance to a WHR/HP system with low, medium, and high levels of waste heat recovery.

Thermal System 3 – Waste Heat Recovery PTC HVAC System Simulation Results

The results of model simulations conducted for the FCEB over duty cycles A, B and C, with the thermal demand satisfied using low, medium, and high WHR, and a PTC heater system, are shown in Figure 7. In this case the HP has been replaced entirely, with thermal demand fulfilled mostly using WHR. The energy consumption for each duty cycle with the conventional HP HVAC system is also plotted in Figure 7 for comparison.

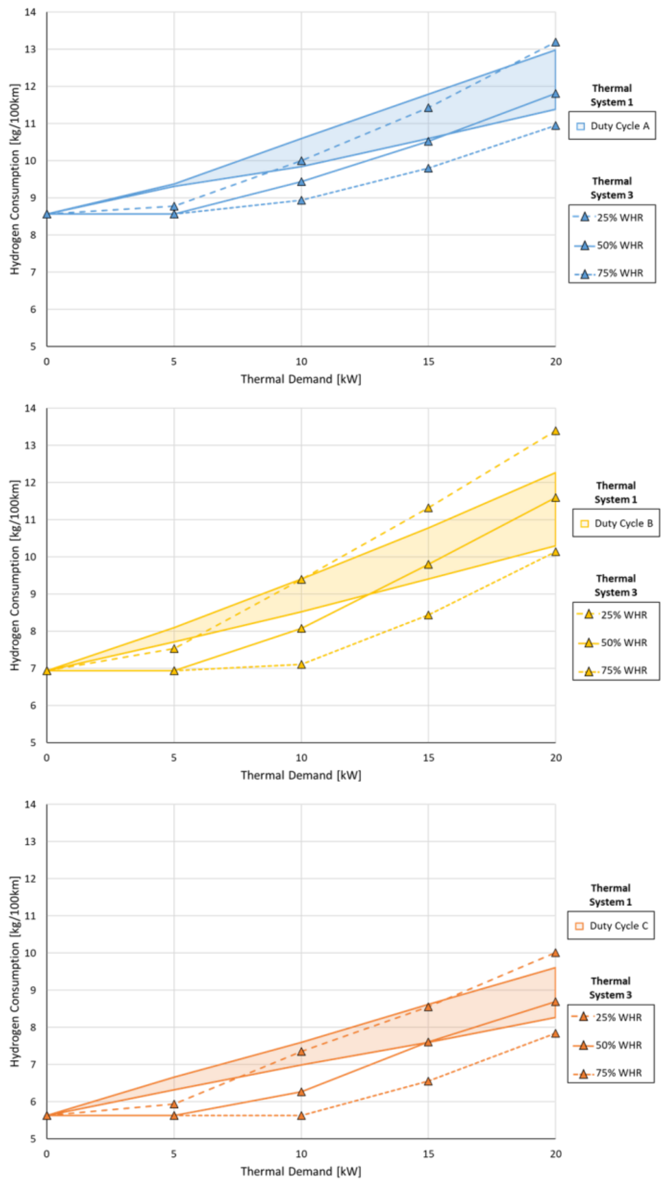

Figure 7. Fuel consumption predictions over the three duty cycles comparing a standard HP-based thermal system performance to a WHR/PTC system with low, medium, and high levels of waste heat recovery.

The results in Figure 7 show the effect that this alternative PTC HVAC system with WHR would have on fuel consumption across the three duty cycles. At low levels of thermal demand (<5 kW) this system will always outperform a HP system with no WHR in terms of energy consumption, however, above 10 kW of thermal demand the low recovery system starts to lose performance across all of the duty cycles. This is because at low levels of thermal demand the FC waste heat recovered is always sufficient to limit the need for use of the PTC heater. At higher levels of thermal demand however the efficiencies of the WHR/PTC system begin to significantly impact the results, and at 20 kW of thermal demand the use of a WHR/PTC system with low levels of heat recovery (25% WHR) performs worse than a conventional HP system. It can be seen that on duty cycle B, at 20 kW thermal demand, the PTC WHR system will consume up to twice as much additional fuel as the conventional HP system operating at a COP of 3.0 because the waste heat recovery is so low that the PTC heater is supplying most of the heat energy to meet thermal demand.

The simulation results show that the medium and high WHR systems are able to achieve parity with a HP system at 20 kW of thermal demand. The FCEB operating with the medium WHR system is predicted to have an energy consumption which falls inside the range of the HP system for all of the duty cycles. The high WHR system is predicted to show a slight improvement when compared to the HP system, with energy consumption found to be 5 – 16% lower across all of the duty cycles.

The results of these simulations show that if the WHR system can achieve at least 50% WHR then a WHR/PTC thermal system should be able to meet or exceed the performance of a HP HVAC system, negating the need for a HP. At lower levels of thermal demand as the excess heat recovered is able to satisfy the cabin heat requirements without the need to run additional heaters. However, as thermal demand increases the benefits of a WHR/PTC system becomes proportionally smaller.

Implementing a WHR system within a HP system shows the most potential for satisfying the thermal demand whilst keeping the energy consumption low. This type of system has all the benefits of the WHR system at low levels of thermal demand, where the waste heat offsets the thermal demand and effectively nullifies the additional fuel consumption. At higher levels of thermal demand the WHR reduces the amount of work the HP system has to input and so still shows significant improvements, even when only small amounts of waste heat are recoverable. For WHR systems which are able to recover at least 50% of the heat rejected by the fuel cell it has been shown that the HP system can be replaced with a simpler PTC heater system and this system will still meet or exceed the performance of a traditional HP HVAC system.

Summary/Conclusions

This study investigates the impact of route characteristics, thermal demand and coefficient of performance of different thermal system configurations on the operational energy consumption of fuel cell electric buses. The generic double deck FCEB was modelled using a MATLAB/Simulink vehicle model, using synthetic drive cycles with high, intermediate, and low average road gradients. The performance of different thermal systems were analyzed across a range of heating requirements, spanning 0-20 kW of cabin thermal demand. Three different thermal systems were considered:

- Conventional HP HVAC system.
- Conventional HP HVAC system with low (25%), medium (50%) and high (75%) WHR from the fuel cell.
- Alternative HVAC system with PTC heater and low (25%), medium (50%) and high (75%) WHR from the fuel cell.

Simulations for the FCEB operating with the conventional HP HVAC system (Thermal System 1) showed that:

- At zero thermal load the energy consumption will increase with increasing average road gradient.
- An increase in thermal demand results in an increase in operational energy consumption.
- Regenerative braking on high average road gradients offsets some of the increase in fuel consumption caused by thermal loads, signifying that additional fuel consumption

is a factor of both the thermal load required and the route characteristics.

For the HP HVAC system with high (75%), medium (50%), and low (25%) WHR (Thermal System 2), simulations showed that:

- The addition of WHR (low, medium, or high) results in significant savings in energy consumption compared to the conventional HP HVAC system.
- At low thermal demands (0 – 10 kW), where the bus will be operating for the majority of its life, the WHR provides the most significant savings, as the thermal demand is mostly satisfied by the waste heat recovered from the fuel cell heat rejection.
- At high thermal demands (10 – 20 kW) the WHR system is able to offset up to 70% of the additional fuel consumed to provide HVAC services by reducing the amount of energy the HP system must supply.

For the alternative HVAC system with the PTC heater and low, medium and high WHR (Thermal System 3), the simulations showed that:

- At low thermal demands (<5 kW) this system is predicted to outperform the conventional HP HVAC system without WHR, in terms of energy consumption. This is again due to the ability of these systems to recover sufficient heat from the FC to meet the thermal demand.
- Above 10 kW of thermal demand, this system starts to lose performance across all duty cycles, however above 50% WHR it will still perform as well as/better than a conventional HP system. When operating with high and medium WHR, this system can achieve parity with the HP HVAC system at 20 kW thermal demand. With medium WHR, the system falls inside the range of the HP HVAC system for all duty cycles.
- Below 50% WHR these systems begin to reach parity with a conventional HP system around 10 kW of thermal demand, and above 15 kW thermal demand they perform worse than a HP system.
- Modelling shows that if the system can achieve at least 50% WHR, this system can meet or exceed the energy consumption performance of the conventional HP HVAC system, which would have cost benefits for manufacturers.

Overall, the HP HVAC system with WHR shows the greatest potential for satisfying the thermal demand of the FCEB whilst keeping energy consumption low, as it provides all the benefits of WHR at low thermal demands, whilst reducing the work of the HP at higher thermal demands. However, it has been shown that a thermal system with WHR and a PTC heater is a viable low cost solution that the achieve similar levels of performance to current conventional HP HVAC systems if WHR from the fuel cell can be achieved at medium or high levels.

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Definitions/Abbreviations

BEB	Battery electric bus
COP	Coefficient of performance
FC	Fuel cell
FCEB	Fuel cell electric bus
HVAC	Heating ventilation air conditioning
HP	Heat pump
OCV	Open circuit voltage
PTC	Positive temperature coefficient
SOC	State of charge
WHR	Waste heat recovery
WTW	Well-to-wheels
ZEB	Zero emission bus