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Simulation of a VRF system applied in electric buses in Taiwan

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ABSTRACT

Based on test data from an electric bus manufacturer, HVAC systems consume nearly 30% of the available energy, which heavily decreases the bus mileage. Therefore, improvements of the bus HVAC system design are needed.

In contrast to conventional HVAC systems for electric buses, the aim of this paper is to integrate and simulate a VRF (variable refrigerant flow) system as HVAC system.
The applied compressors provide adequate power, depending on the requirements of distributed heat exchanger units inside the bus cabin. This system provides either cooling or heating capacity depending on the need of the target space. Therefore, the energy consumption of the compressor will be reduced.

The simulation results show that the target temperature of 22°C and relative humidity of 50% can easily be reached with a VRF system for the complete bus. In conventional HVAC systems comfort in the middle of the bus will be reduced. Those systems usually use one big evaporator in the back of the bus which is used to cool down the passenger room, to set temperature.

The electrical energy consumption is very important for the mileage of an electric transportation bus. When a VRF system is used as AC system the total electrical power of the compressor can be reduced by 12% compared to the state of the art system as the simulation results show. The passenger comfort can be increased significantly due to the more continuous temperature distribution inside the bus.

Thanks to the dynamic simulation model developed in this study the optimal design specifications of the VRF HVAC system can be predicted and evaluated.

Keywords: VRF, electric buses, variable capacity, dynamic simulation, public transportation

1. INTRODUCTION

Nowadays, the HVAC systems of electric buses are the same as in traditional diesel engine buses. Such a system generates the necessary cooling or heating power by changing the speed of the compressor motor. Heat pumps are state of the art to use for air conditioning in electrical vehicles. The applied compressors are normally driven by a Lithium-Ion batteries or a diesel engines (ASHRAE Handbook, 2011). The performance of the used HVAC system has therefore a direct impact on the bus mileage. High efficient HVAC systems are important for all electric vehicles.

In the case of electric driven transportation buses that data from an electric bus manufacturer in Taiwan show that the cooled air has to be delivered to the very end of the bus, and all seats in between increase the load of the compressor and fans. Even if only half of the bus is occupied by passengers, all seats are cooled. On the other hand, a too high air velocity usually reduces passenger comfort (Fanger, 1970). Further problem can be an uneven temperature distribution.

The VRV (variable refrigerant volume) or VRF (variable refrigerant flow) technology is now widely adopted in home-used AC systems (Daikin AC, 2013). The compressors provide adequate power depending on the requirement from the terminal AC systems. This can be either cooling or heating, and the power is adjusted to meet the needs of the target space, as shown in Figure 1. The energy consumption of the compressor can be reduced effectively because energy is only consumed when really needed. Therefore, this technology seems to be a valid solution for saving energy in electric bus HVAC systems.

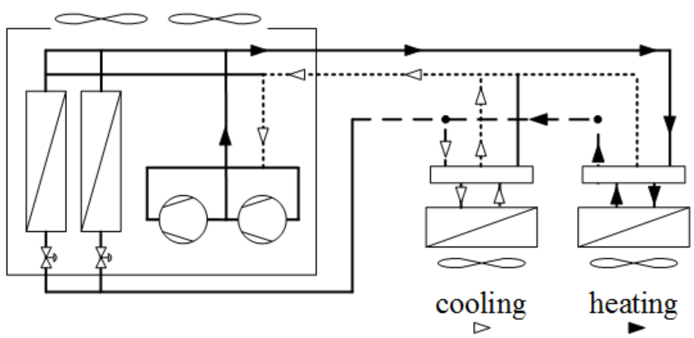

Figure 1: Example of a VRF system for household applications (cooling and heating),

VRF systems operate on the direct expansion (DX) principle, meaning that the heat is transferred to or from the space directly by circulating refrigerant to the evaporators located near or within the conditioned space. Refrigerant flow control is the key to many advantages, as well as the major technical challenge of VRF systems. The VRF technology typically uses inverter-driven scroll or rotary compressors (Xiaohong, 2006). With these variable speed compressors, a capacity range from 10% to 100% can be controlled. Separation tubes and / or headers are used for the refrigerant piping. Each indoor unit contains one indoor heat exchanger, and a separate electronic expansion valve or a pulse-modulating valve in the liquid line (Bhatia, 2016). With these extra expansion valves, it is possible to independently regulate the refrigerant mass flow in every unit of the bus and therefore, the capacity of each single unit inside the bus (see Figure 1). VRF systems are only applied for building applications, but not in electric buses.

VRF systems promise a higher energy efficiency on somewhat higher costs (Goetzler, 2007). Compared to multi-split systems, VRF systems minimize the refrigerant path and require less copper tubing. VRF also distinguished from VAV (variable air volume systems), because VAV systems work by varying the airflow to the conditioned space based on the variation of room loads. For the knowledge of the authors, all currently used HAVC system for electric buses are based on VAV systems and not on VRF systems.

Advantages and disadvantages for adopting VRF technology to electric buses are summarized in the following list:

+ Independent control of the zones and lower air velocities increase comfort
+ Several heat exchangers are placed in the bus that could provide adequate cooling/heating power to the defined area depending on the number and location of the passengers.
+ Energy consumption will be lower with the VRF heat pump system.
o High efficient and high power electric compressor with variable refrigerant flow are needed
- More terminal systems are needed which increase the overall cost and weight, but might need less room than the air flow channels of conventional systems.
- Oil management and safety provisions must be examined.

The aim of this paper is to develop a dynamic simulation model of a public transportation bus, which uses a VRF system as AC system. In addition to the electrical consumption of the VRF system, the sizes of the compressor and indoor heat exchangers are evaluated for different ambient conditions in Taiwan.

2. SIMULATION MODEL

The goal of this physics based model is to define the size of the VRF system by simulating the cooling and heating demand of an electric bus. The overall heat gains and losses are calculated with respect to time. Unfortunately, there are no real measurement data available to validate the simulation results. The simulation model was designed for a summer scenario in Taiwan, when the VRF systems acts as AC system inside the bus. For the simulation, the bus was split up into 4 elements. Figure 2 shows the definition of all elements. Input parameters are the bus geometry, the ambient conditions, the maximum number of passengers as well as the bus material properties, which were given by the project partner ARTC. To simulate the dynamic behavior of the system the doors were opened every 4 minutes for a period of 2 minutes. During a simulated stop at a bus station, the passenger numbers were also changed. Only in element no. 1 (the driver section) the number of passengers (one) is constant during the whole simulation time. During the bus operation time, the cabin temperature inside the bus should not exceed 22°C, and the relative humidity should not be greater than 50%. These boundary conditions are valid for all bus elements.

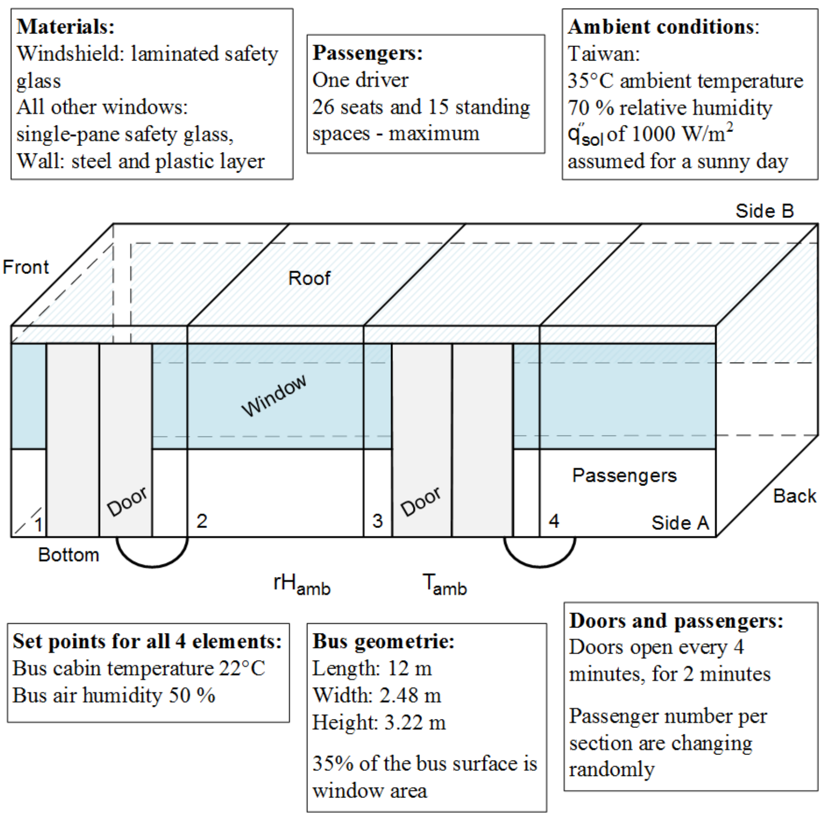

Figure 2: Definition of all bus elements

The geometry of each element is described separately. Each element of the bus is segmented in a Side A (door side), Side B, Front, Back, Roof and Bottom part. For example, element no. 1 has a front side but no back side, because it has element no. 2 as neighbor. The element no. 1 contains the driver section, the first door and the windshield. Element no. 2 has no door but contains passengers. The back door can be found in element no. 3. Most passengers are assumed to be seated in element no. 4. This element includes the back side of the bus, which has no window according to the specification of ARTC. All window or wall areas are defined separately for each element. The complete definition for element no. 1 is shown in Table 1.

Table 1: Description of element no. 1

Element 1	Side A	Side B	Front	Back	Roof	Bottom	Front Zone	Back Zone
A_{wall} [m^2]	5.75	5.75	4.70	0	6.75	6.75	0	0
A_{glass} [m^2]	2.33	2.33	3.00	0	0	0	0	0
q''_{sol} [W/m^2]	1000	1000	1000	0	1000	10	0	0
Neighbor	0	0	0	0	0	0	0	2

When a VRF system is used as HAVC system, each element contains a heat exchanger to fulfil the various required cooling or heating demands, as can be seen in Figure 3. Figure 3 also contains a simplified equivalent circuit of the heat pump system. The heat pump model was designed according to this scheme. The used heat exchanger surface for the cabin heat exchangers, is the same for all elements. For the simulation of the heat pump system the overall displacement of the compressor V_{cc}, the compressor speed, the evaporator and the condenser temperature were input values.

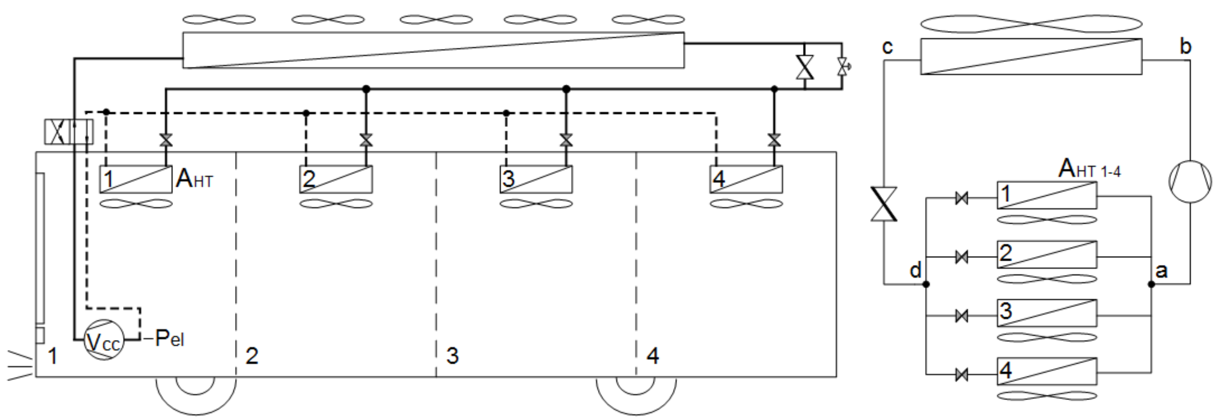

Figure 3: Left: Bus including the VRF system, Right: equivalent heat pump circuit diagram

To design the heat pump system, an energy and water mass balance was created for all bus elements individually. Figure 4 shows the involved heat gains and losses in detail.
As can be seen in Figure 1 the bus walls contain a window part and a solid wall part. Therefore, the heat losses over the bus hull were calculated for the glass and wall part separately. For the windows the transmission and reflection part depends on the glass type. For the simulation, window panes from the German company Polartherm were used. For the wall parts no direct radiation part was assumed.

On the in- and outside of the bus forced convection is assumed due to the driving speed and the ventilation of the heat pump system. The bus hull is never completely sealed; an energy leakage factor $\dot{Q}_{leak}$ is introduced to consider this fact in the simulation.

All elements are thermally linked to each other by an internal energy exchange called $\dot{Q}_{int}$ and an air mass flow between the elements $\dot{m}_{int}$, which simulates the air temperature and air humidity exchange between the elements.

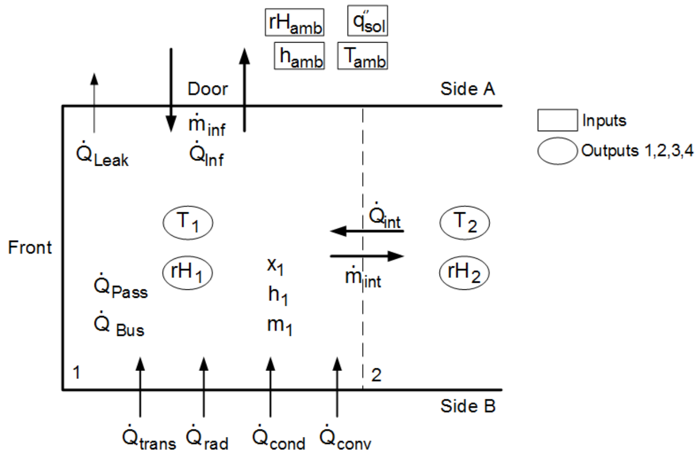

Figure 4: Detailed element calculation for one element – cooling scenario

Depending on the number of passengers in each element $\dot{Q}_{pass\ el}$ is calculated.

$$\dot{Q}_{pass\ el} = Q_{pass\ heat} * A_{pass} * n_{el\ pass} \tag{1}$$

For the simulation Taipei is chosen as location of interest. Taipei is located at 25°2' N/121°32' E. The elevation of the sun on the 21th of July is considered with the factor s_{factor} depending on the azimuth (Markstaler, 2015). Therefore, the direct radiation of the sun $\dot{Q}_{sol\ el}$ is:

$$\dot{Q}_{sol\ el} = Q''_{sol} * A_{el\ gl} * s_f \tag{2}$$

$\dot{Q}_{bus}$ describes the heating up of the bus and its complete interior.

$$\dot{Q}_{bus\ el} = U_{bus} * A_{bus} * (T_{bus} - T_{el}) \tag{3}$$

$\dot{Q}_{int}$ calculates the energy transfer between one element and its neighbours.

$$\dot{Q}_{int\ el} = \dot{m}_{int} * (h_{el+1} - h_{el}) \tag{4}$$

$\dot{Q}_{inf}$ is the energy exchange between the bus air and the surroundings when the doors are open. The infiltration mass flow $\dot{m}_{inf}$ is calculated by half the door surface A_{door} the ambient air velocity $\mathrm{v}_{air\ out}$ and the ambient air density $\rho_{el\ air}$.

$$\dot{Q}_{\mathrm{inf}\ el} = \dot{m}_{inf} * (h_{amb} - h_{el}) \tag{5}$$

$$\dot{m}_{inf} = \mathrm{v}_{air\ out} * A_{door} * \rho_{el\ air} \tag{6}$$

Depending on the bus surface (glass window or plastic/steel wall - inside or outside) the radiation is calculated at the outer surface $A_{el\ wall}$ of the bus against the surroundings and from the inside bus wall against the bus cabin. Equation (7) shows the radiation part $\dot{Q}_{rad\ el}$ of the inside wall of the bus.

$$\dot{Q}_{rad\ el} = \varepsilon_{wall} * A_{el\ wall} * (T_{el}^4 - T_{wall}^4) \tag{7}$$

Also for the convection calculation $\dot{Q}_{conv\ el}$ it is necessary to distinguish between the convection happening on the outside or inside of the bus. Formula (8) shows the calculation towards the inside part.

$$\dot{Q}_{conv\ el} = \alpha_{air} * A_{el\ wall} * (T_{el} - T_{wall}) \tag{8}$$

The heat conduction $\dot{Q}_{cond}$ through the bus hull is calculated in the following way

$$\dot{Q}_{cond} = \frac{1}{\Sigma \frac{th_{gl/wall}}{\lambda_{gl/wall}}} * A_{el\ gl/wall} * (T_{amb} - T_{el}) \tag{9}$$

These calculations (Dewitt and Incropera, 2002) are made for each element. The outside conditions, the passengers, the infiltration due to the bus doors and the internal leakage factor will contribute to heating up the bus. Only the AC system $\dot{Q}_{HP}$ reduces the temperature and the relative humidity. The, equations above determine the energy balance for each element. $\dot{Q}_{AC}$ and $\dot{m}_{AC}$ are calculated separately in a subprogram.

$$(h_{el\ new} - h_{el\ old}) * \frac{m_{air\ el}}{\Delta t} = \dot{Q}_{inf} + \dot{Q}_{int} + \dot{Q}_{pass} + \dot{Q}_{leak} + \dot{Q}_{conv} + \dot{Q}_{rad} + \dot{Q}_{bus} - \dot{Q}_{AC} \tag{10}$$

Due to the humid conditions in Taiwan, the dehumidification of the bus air (which takes place during the summer time) based of the AC system as well as the vapor created by the passengers through respiration, a water mass balance is needed for the simulation.

$$\frac{m_{el\ water\ new} - m_{el\ water\ old}}{\Delta t} = \dot{m}_{inf} * (x_{amb} - x_{el}) - \dot{m}_{int} \left(x_{el\ neighbour} - x_{el}\right) + \dot{m}_{pass} - \dot{m}_{AC} \tag{11}$$

The simulation model was developed to easily change of the time step Δt, duration of the dynamic simulation and all element design values.

To define the needed cooling capacity for reaching the set point of the bus cabin, first the cabin air requirements $\dot{Q}_{air\ el}$ are calculated for every bus element. $\dot{Q}_{air\ el}$ (12) consist of the sensitive (13) and latent (14) energy of the air. It is assumed that the exit air humidity of the indoor heat exchanger is equivalent to the evaporator temperature and 100% relative humidity.

$$\dot{Q}_{air\ el} = \dot{Q}_{sen} + \dot{Q}_{lat} \tag{12}$$

$$\dot{Q}_{sen} = \frac{\mathrm{m}_{air\ el}}{\Delta t} * c_{p\ air\ el} * (T_{el} - T_{set}) \tag{13}$$

$$\dot{Q}_{lat} = \dot{m}_{HEX\ air\ el} * (x_{el} - x_{HEX\ exit}) * \Delta h_{phase\ change} \tag{14}$$

From the assumption for the heat pump size, the evaporating and condensing temperature, displacement volume of the compressors, the heat transfer surface of the cabin heat exchangers $\dot{Q}_{evap\ tot}$ and $\dot{Q}_{HEX\ el}$ can be calculated, like described in Figure 3. The heat exchanger efficiency $\varepsilon_{HP\ hex}$ is assumed to be 80% and the U_{HEX} value 0.4 [kW/m^2 K].

$$\dot{Q}_{evap\ tot} = \dot{m}_{ref\ tot} * (h_d - h_a) \tag{15}$$

$$\dot{Q}_{HEX} = \varepsilon_{HEX} * U_{HEX} * A_{HEX} * \left(T_{el} - T_{evap}\right) \tag{16}$$

R134a is used as refrigerant for the simulation. The compressor model assumes an isentropic efficiency of 0.6 and the volumetric efficiency of 0.85. These values need to be changed or replaced with actual compressor data to figure out the optimal compressor design for specific operation points.

Based on the equations, water mass flow $\dot{\mathrm{m}}_{water\ AC}$ that condenses on the heat exchanger surface can be calculated

$$\dot{\mathrm{m}}_{water\ AC} = \dot{m}_{HEX\ air\ el} \ (x_{el} - x_{HEX\ exit}) \tag{17}$$

Since the cooling requirements of each element are different, some case distinctions must be made. First, the required cooling capacity from the bus air cannot be greater than the maximum capacity of the indoor heat exchangers due to its heat transfer surface A_{HEX}. Furthermore, the sum of all cooling capacitates from the indoor heat exchangers cannot get larger than the maximum overall cooling capacity that can be provided by the heat pump system.

Therefore, a priority of the provided cooling capacity for all elements is introduced (defined by ARTC). The driver section (element no. 1) has the first priority. After that, the second priority is given to element no 3., because of the opening of the back door. Element no. 4 has 3rd priority and element no. 2 last priority. On very hot and humid days, it can happen that there is not enough cooling capacity left to reach the set point in element no. 2. The temperature in element no. 2 will increase and therefore, the specifications cannot be fulfilled.
The complete structure of the dynamic simulation model, Figure 5.

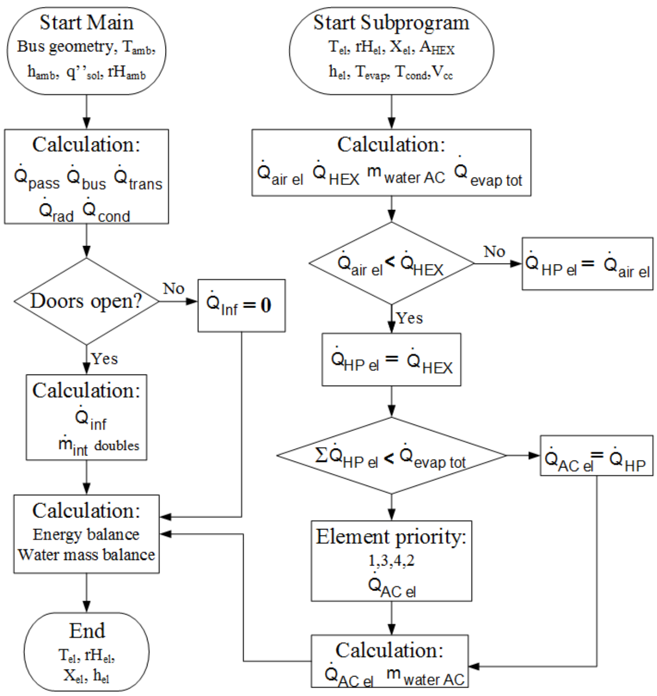

Figure 5: Program structure for all elements

Based on the energy balance, the cabin air enthalpy can be calculated for each element. The cabin enthalpy together with the water quantity of the cabin air (out of the water mass balance) implies the actual bus element temperature T_{el} and relative humidity rH_{el} for the next iteration step.

3. SIMULATION RESULTS

For the dynamic simulation results two sets of simulation were carried out. One with the VRF system (left column in Figure 6) and a second with a state of the art AC system (right column in Figure 6). The main difference between both systems is that the state of the art AC system has one big heat exchanger in element no. 4, to cool the passenger area, whereas the VRF system has four heat exchangers, one for each element, to fulfill the cooling demand of the bus cabin.

Table 2 contains all important parameters for both simulations. For the state of the art simulation the heat exchanger size A_{HEX} and the air velocities between the elements v_{int} and $\dot{m}_{AC\ air}$ were changed. $\dot{m}_{AC\ air}$ is the mass flow rate of air through the AC heat exchanger. Doubling the value of $\dot{m}_{AC\ air}$ the passengers comfort will decrease.

Table 2: Input parameters of the simulation

	VRF system	State of the art AC system
T_{evap}	12°C	12°C
T_{cond}	45°C	45°C
A_{HEX}	10 m^2 per heat exchanger	40 m^2 for one heat exchanger in the back
V_{cc}	170 cm^3	170 cm^3
v_{int}	0.1 m/s	0.5 m/s
$\dot{m}_{AC\ air}$	1.5 kg/s	3 kg/s

The starting values were set according to the target conditions (shown in Figure 2). Figure 6 shows the simulation results for 2 "doors open" cycles after the bus reached steady state. Only when the doors are open, infiltration takes place. For the state of the arte system a two-point controller is introduced. The AC starts when the average bus temperature is above the set point plus 0.5°C and turned off when the average temperature is below the set temperature minus 0.5°C.

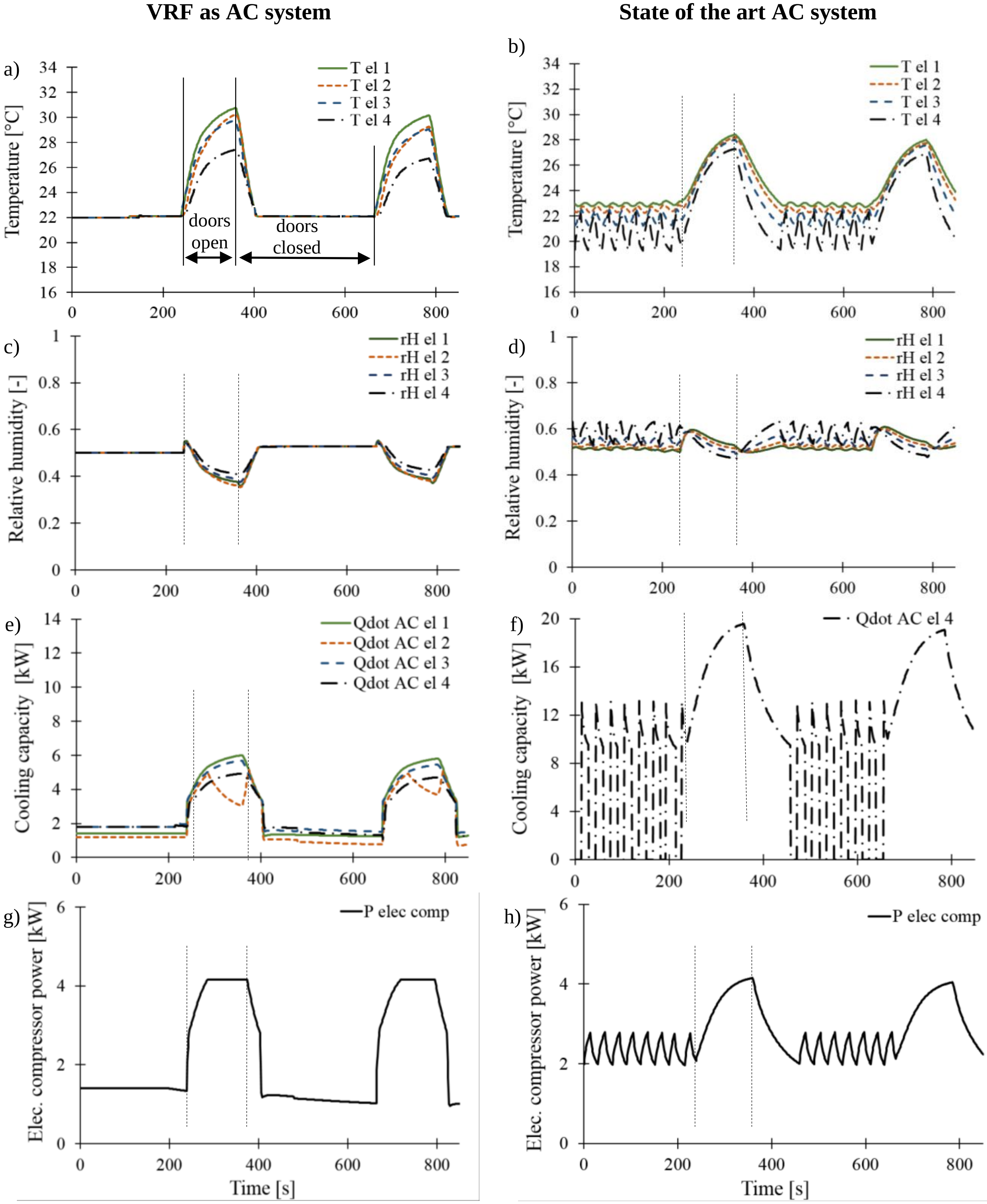

Figure 6: Comparison of a VRF system with a state of the art AC system. The bus cabin temperatures, the relative humidity, the AC capacity and the electrical power of the compressor for each element can be seen

It can be seen in Figure 6a) that the set temperatures is reached quickly after the doors are closed. Element no. 4 has the lowest increase of temperature when the doors are open because, it contains no door and interacts only with element no. 3. However, element no. 2 has no door but interacts with element no. 1 and 3. The doors are placed in element no. 1 and 3. The simulation results for the state of the art system Figure 6b), where only one evaporator is placed in element no. 4, show that the set temperatures cannot be reached even after the doors have been closed. Due to the one large heat exchanger in element no. 4. the air temperature is even lower than the set temperature. Therefore, the needed energy amount to reach the set temperature in the whole bus is higher compare to the VRF system. Hence, the energy losses are greater in the state of the art system. The highest temperature is reached in element no. 1 which contains the windshield. Element no. 1 contains the highest amount of window area. The temperature distribution which appears in the state of the art system (Figure 6b)) lowers the comfort of the passengers. Another effect which lowers the passengers comfort is the start/stop behavior of the of the AC system.

Figure 6c) shows that the relative humidity rises for a short time in element no. 1 and 3 when the doors are open. This is due to the very humid ambient conditions of Taiwan. This effect happens in both systems (Figure 6c) and d)). The bus cabin air gets dehumidified in every element due to the evaporators in the VRF system. Therefore, the set point of 50% relative humidity can be reach when a VRF system is used as AC system. The temperature is cooled down so intensely in element no. 4 that the relative humidity is the highest in this element. Furthermore, this element contains the highest number of passengers.

Figure 6e) shows the cooling capacities $\dot{Q}_{AC\ el}$ according to the priorities of the elements. When the doors are open, element no. 2 has the lowest cooling capacity, element no. 2 has the lowest priority according to the definition from ARTC. Due to the mass flow rate regulation, cooling capacity is only provided when it is needed in a VRF system. On the other hand, the cooling capacity of the state of the art system needs to be larger (due to its single heat exchanger) to cool down the bus cabin to a comfortable temperature in all segments.

The compressor of the VRF system needs, for this simulation an average 2.22 kW of electrical power, whereas the compressor of the state of the art system needs 2.76 kW. Therefore, with a VRF system the electrical power consumption can be slightly reduced by about 12%.

4. CONCLUSION

The simulation results show that when a VRF system is used in public transportation busses, the passengers comfort can be increased significantly. The temperature and relative humidity distribution is much smaller compared to the state of the art system. Furthermore, the needed electrical power is by the compressor approximately 12% smaller, which has a positive impact on the bus mileage.

Thanks to the dynamic simulation model developed in this study, the optimal design specifications of the VRF HVAC system can be predicted and evaluated.

Furthermore, specific evaluation studies can be made with this simulation model. For example, to show bus manufactures the impact of a modification of the insulation material for the bus walls. The simulation structure of the bus hull can also be used for any geometry e.g. a room with all its heat gains and losses.

This project has been developed in collaboration with the Automotive Research and Testing Center of Taiwan ARTC and the NTB Interstate University of Applied Sciences of Technology, Buchs.

5. NOMENCLATURE

A	Surface area [m^2]
c	Specific heat capacity[kJ/kg K]
h	Enthalpy [kJ/kg K]
m	Mass [kg]
$\dot{m}$	Mass flow [kg/s]
n	Number [-]
Q	Energy [kWs]
$\dot{Q}$	Capacity [kW]
q''	Solar intensity [W/m^2]
Q''	Horizontal solar intensity [W/m^2]
s	Altitude of the sun [-]
Δt	Time interval [s]
T	Temperature [°C]
th	Thickness [m]
U	U-Value [kW/m^2 K]
v	Velocity [m/s]
x	Quantity [kg/kg]

Greek symbols

α	Convection heat transfer coefficient [kW/m^2 K]
ε	Efficiency [-]
ε	Emission [-]
λ	Thermal conductivity coefficient [kW/m K]
ρ	Density [kg/m^3]

Subscripts

a	Suction point - compressor
amb	Ambient
AC	Air conditioning
cond	Conduction
conv	Convection
d	Point in p-h diagram, after expansion of the refrigerant
el	Element
evap	Evaporator
f	Factor
gl	Glass
HEX	Heat exchanger
HP	Heat pump
inf	Infiltration
int	Intersection
lat	Latent
leak	Leakage
pass	Passenger
phase change	Change of water between vapor to liquid
rad	Radiation
ref tot	Total refrigerant mass flow
sen	Sensitive
trans	Transmission

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