LOAD CALCULATION AND SYSTEM EVALUATION FOR
ELECTRIC VEHICLE CLIMATE CONTROL

by

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ABSTRACT

Providing air conditioning for electric vehicles (EVs) represents an important challenge, because vapor compression air conditioners, which are common in gasoline powered vehicles, may consume a substantial part of the total energy stored in the EV battery.

This report consists of two major parts. The first part is a cooling and heating load calculation for electric vehicles. The second part is an evaluation of several systems that can be used to provide the desired cooling and heating in EVs. Four cases are studied. Short range and full range EVs are each analyzed twice, first with the regular vehicle equipment, and then with a fan and heat reflecting windows, to reduce hot soak.

Recent legislation has allowed the use of combustion heating whenever the ambient temperature drops below 5°C. This has simplified the problem of heating, and made cooling the most important problem. Therefore, systems described in this project are designed for cooling, and their applicability to heating at temperatures above 5°C is described. If the air conditioner systems cannot be used to cover the whole heating load at 5°C, then the vehicle requires a complementary heating system (most likely a heat recovery system or electric resistance heating).

Air conditioners are ranked according to their overall weight.

The overall weight is calculated by adding the system weight and the weight of the battery necessary to provide energy for system operation. The air conditioner with the minimum overall weight is considered to be the best.

The air conditioners under analysis are classified in two categories. Systems in the first category have a low overall weight, and therefore are competitive with respect to vapor compression air conditioners. The systems within this category are ice storage systems and adsorption systems. These systems, along with a vapor compression air conditioner (used as a basis for comparison), are evaluated in detail. For these three systems, the report presents a calculation of system weight, system volume, COP, and potential applicability for heating, for the four combinations of vehicle characteristics and driving ranges being considered. The report also includes a calculation on how the battery energy storage capacity affects the overall system weights and the selection of the optimum system.

Systems in the second category are capable of providing EV air conditioning. However, they have an excessive overall weight, due to high system weight and/or high energy consumption, and are therefore not considered to be competitive for EV cooling, at the current state of technology. Systems in this category are thermoelectrics, stirling cycles, brayton cycles, vacuum boiling of water, and heat-actuated air conditioners. These systems are briefly described, with a discussion of the reasons for being classified within this category, and possible applicability in the near future.

The results indicate that, for the batteries currently available for EV propulsion, an ice storage system has the minimum weight of all the systems considered. Vapor compression air conditioners become the system with the minimum weight for battery storage capacities above 270 kJ/kg.

NOMENCLATURE

A	area
a,b,c	heat exchanger dimensions (Table A.1)
B	extinction coefficient
C	constant for Equations (2.2) and (2.3)
C_f	clearness factor
c_v	specific heat at constant volume
c_p	specific heat at constant pressure
d	half of the distance between fins
F_{ij}	view factor
H_s	isosteric heat of adsorption
h	heat transfer coefficient
I	radiation intensity
I_d	diffuse solar radiation intensity
I_{dn}	direct normal radiation intensity
I_{ex}	extraterrestrial solar radiation intensity
I_r	reflected solar radiation intensity
J	radiosity
k	thermal conductivity
l	heat transfer path length
M	total mass
m	mass flow rate
Nfan	fan number (Table A.1)
NTU	number of heat transfer units
p	pressure
Q	heat transfer rate
q	heat transfer rate per unit area

q_s	water adsorbed per kg of zeolite
T	temperature
t	time
U	overall heat transfer coefficient
w	specific humidity
x	coordinate perpendicular to fins (Figure A.2)
α	absorptivity
β	solar angle with the horizon
δ	reflectivity
ϵ	emissivity
θ, ϕ	parameters in Equations (A.3), (A.4), and (A.5)
ρ	density
Σ	angle between a surface and the horizontal
σ	Stefan-Boltzmann constant
τ	transmissivity
τ_i	time constant for surface i

Subscripts

a	air
ad	adsorbate
amb	ambient
b	adsorption bed
c	coolant
db	dry bulb
g	ground
i	property of surface i
in	internal or inlet condition

m	metal
out	external or outlet condition
s	adsorbent
sky	sky
t	total
wb	wet bulb

1. INTRODUCTION

The state of California has mandated zero emission vehicles by 1998. This opens the market for introduction of electric vehicles (EVs), which currently represent the best developed technology for zero emission vehicles. However, before EVs can be successfully marketed, five major issues must be addressed. These are, cost, performance, convenience, passenger safety, and passenger comfort. This work focuses on the last issue, evaluating alternative systems to accomplish appropriate climate control for EVs.

Heating and air conditioning (HVAC) are by far the car accessories that consume the most energy. HVAC does not represent a challenge for amply-powered gasoline vehicles. On the other hand, EVs have a low energy storage capacity, and heating and air conditioning may consume a substantial amount of the total energy stored, considerably reducing the vehicle range, which is one of the most important parameters for EV acceptability.

Recent legislation has allowed the use of combustion heating for EVs at ambient temperatures below 5°C. This legislation simplifies the problem of EV heating, which can be accomplished by using waste heat or electric resistance heating (or using the air conditioner system as a heater, if appropriate), at temperatures above 5°C, and turning to combustion heating as ambient temperature falls below 5°C. Air conditioning of EVs has therefore become the most important problem associated with EV climate control.

This report analyzes four combinations of vehicle equipment and driving scenarios. Short range and full range EVs are each

analyzed twice, first with the regular vehicle equipment, and then with heat reflecting (hot mirror) windows (and a fan that circulates air when the vehicle is stopped, to reduce hot soak), to evaluate the effect that load reduction systems may have on the thermal load. Both vehicles are analyzed for a daily schedule of two trips, one starting from comfort conditions (assuming that the vehicle is cooled and protected from the sun during the charge process in the cooling season and assuming that the vehicle is heated during charge in the heating season), and the other starting from full hot or cold soak. The trips have a duration of 30 minutes each for the short range EV, and 60 minutes each for the full range EV. The 60 minute trip length for a full range EV is chosen according to the range specifications of currently existing EVs, as listed by Wyczalek, 1991. The applicability of the short range electric vehicle to fill a market niche is studied in detail by Theodore Barry and Associates, 1992, who indicate that over 80% of all vehicle trips in Southern California are shorter than 16 km. A full description of the EV characteristics for the four cases considered are listed in Table 1.1. The dimensions and styling of the vehicle are those of a subcompact car.

This report starts by showing the procedure for calculating cooling and heating loads for the different vehicle equipment and driving scenarios. The result for the heating load at 5°C ambient temperature yields the maximum heating power requirement for which no combustion heating can be used. Therefore, the vehicle must be equipped with an auxiliary heating system (most likely an electric heater, a waste heat recovery system, or an air conditioner that

Table 1.1. Electric vehicle characteristics.

1. All cases considered have the dimensions of subcompact vehicles, and include the following thermal load reduction equipment. Additional equipment is indicated for each case.
 - A) A roof insulated with 0.02 m of high performance thermal insulation.
 - B) Operation by minimizing the amount of external ventilation.
2. Vehicles are used for two daily trips. The first trip starts from comfort conditions (i.e. the vehicle is cooled and protected from the sun during the charge process in the cooling season, and is heated during charging in the heating season). The second trip starts from full soak conditions. Load calculation considers two occupants during summer operation, and a single occupant during winter operation. The four cases considered are as follows.
 - Case 1. Short range electric vehicle, used for two trips daily, each lasting 30 minutes. All the vehicle windows are made of regular glass, with heat absorbing coating on the top 0.10 m of the windshield.
 - Case 2. Short range electric vehicle, used for two trips daily, each lasting 30 minutes. All the vehicle windows are made of heat reflecting (hot mirror) glass. Also, a solar powered fan operates whenever the vehicle is left in the sun, to reduce hot soak temperature.
 - Case 3. Full range electric vehicle, used for two trips daily, each lasting 60 minutes. Same vehicle characteristics as for Case 1.
 - Case 4. Full range electric vehicle, used for two trips daily, each lasting 60 minutes. Same vehicle characteristics as for Case 2.

can be used for heating) to provide heating at ambient temperatures above 5°C.

The results for the cooling load are used for designing appropriate air conditioning systems. The aim is to find the air conditioner that represents the best choice under a given set of requirements, and how this best system changes as the requirements change. The figure of merit used is the overall weight of the system, which includes the weight of the battery required to provide the energy for operating the system. The best air conditioner has the minimum overall weight. This is equivalent to using the reduction in range as the figure of merit, considering that added system weight results in a decreased possibility of installing more batteries in the EV.

Many alternative systems for air conditioning have been analyzed, and have been classified in two categories. Systems in the first category are competitive with respect to vapor compression systems in overall system weight. These systems are described with detail in this report, including an evaluation of the capacity of these systems to provide the heating required at temperatures above 5°C. Systems within this category are an ice storage system and a zeolite-water adsorption system. A vapor compression system is also analyzed, to compare its performance and total weight with those of the alternative systems. Systems in the second category are capable of providing the required air conditioning. However, they either have an excessive system weight or a low efficiency, resulting in a high overall system weight, and are therefore not competitive with systems in the first category,

at the current state of technology. A brief description of these systems is given, along with the possibilities for future system applicability.

2. COOLING AND HEATING LOAD CALCULATION

This section shows the procedure and results of a cooling and heating load calculation analysis for the vehicles and driving conditions described in Table 1.1. Additional vehicle information, including heat transfer areas for the EV, as well as the heat transfer coefficients for a parked vehicle (necessary for the hot soak calculation) and an operating vehicle (necessary for the transient cooling and heating calculation) are given in Table 2.1. The load calculations assume the presence of two passengers inside the vehicle.

In this section, cooling and heating loads are calculated for steady-state, and also for operation starting from hot and cold soaks. The cooling load is calculated for two sets of ambient conditions. The first is $T_{db}=43^{\circ}\text{C}$, $T_{wb}=22^{\circ}\text{C}$, characteristic of the desert southwest, and the second is $T_{db}=36^{\circ}\text{C}$, $T_{wb}=27^{\circ}\text{C}$, characteristic of the southeast. Both of these sets of conditions are used in the calculation to decide which is the most critical. The design outdoor temperature for heating is 5°C , because, as previously established, current legislation allows the use of combustion heaters whenever the ambient temperature drops below this value. Comfort conditions are $T_{db}=25^{\circ}\text{C}$, $w=0.012$ Kg/kg, for cooling, and $T=23^{\circ}\text{C}$ for heating (ASHRAE, 1981).

2.1 COOLING LOAD, STEADY STATE

The cooling load has the following components.

2.1.1. Metabolic Load: It is taken as 150 W per occupant, or 300 W

Table 2.1. Heat transfer areas and heat transfer coefficients for the vehicle.

vehicle surface ¹	area m ²	heat transfer coefficients, W/m ² °C				
		inside ²		conduction	outside	
		no vent	vent	(k/l)	parked ³	moving ⁴
windshield	0.89	4.5	6.0	157	9.5	41.6
roof	1.47	4.5	6.0	1.0	9.5	48.4
rear window	0.52	4.5	6.0	157	9.5	51.1
side windows	1.46	4.5	6.0	157	9.5	79.2
rear gate	0.42	4.5	6.0	4.4	9.5	51.1
doors	1.33	4.5	6.0	4.4	9.5	79.2
side rear panels	0.93	4.5	6.0	4.4	9.5	79.2
floor	1.60	4.5	6.0	5.2	9.5	48.4

1. Surfaces are constructed as follows:

Roof: metallic plate, 0.02 m insulation and liner.

Gate, doors and panel: metallic plate, air space and padding.

Floor: metallic plate, liner and carpet.

2. Heat transfer coefficients for the vehicle interior are equal to 4.5 W/m²°C (natural convection) when there is no ventilation. Ventilation increases this value to 6 W/m²°C

3. Heat transfer coefficients for the vehicle exterior when parked are calculated from an equation given by Aschenbrenner and Anderson (1987), assuming a 1 m/s wind speed.

4. Heat transfer coefficients for the vehicle exterior when moving are calculated from correlations given by Shimizu et al. (1982), assuming a 20 m/s vehicle speed.

for two occupants. The sensible and latent components of this load are assumed to be approximately equal (ASHRAE, 1981).

2.1.2. Ventilation Load: The EV under analysis operates mainly on recirculated air to reduce energy consumption. Only the minimum amount of new air is used. The minimum ventilation needs for the occupants are $0.007 \text{ m}^3/\text{s}$ of ambient air per person, or $0.014 \text{ m}^3/\text{s}$ total (ASHRAE, 1977). This results in a sensible load of 277 W, and a latent load of -150 W, for the conditions of the desert southwest, and 171 W sensible and 265 W latent in the southeast. The negative latent load in the southwest indicates that the specific humidity at the comfort condition is higher than the specific humidity in the ambient.

2.1.3. Insolation Load: This solar load calculation follows a method given by ASHRAE, 1981, and assumes that the EV is moving on a flat space, with the sun perpendicular to the windshield (considered to be the worst case), on a clear day, taken as June 21. Under these conditions, the intensity of normal solar radiation is,

$$I_{dn} = \frac{I_{ex} C_f}{\exp(B/\sin\beta)} \quad (2.1)$$

where I_{ex} is the extraterrestrial solar radiation, B is the extinction coefficient, β is the solar angle with the horizon, and C_f is the clearness factor. Substituting values for a windshield angle of 40° ($\beta=50^\circ$), results in $I_{dn}=874 \text{ W/m}^2$ in the desert southwest ($C_f=1.05$), and $I_{dn}=750 \text{ W/m}^2$ in the southeast ($C_f=0.90$). For the side and rear windows, $I_{dn}=0$, since side and rear windows do not receive any direct solar radiation when the sun shines

perpendicular to the windshield.

The amount of diffuse radiation from the sky is,

$$I_d = C I_{dn} (1 + \cos \Sigma) / 2 \quad (2.2)$$

where Σ is the angle between the surface and the horizontal (40° for the windshield, 70° for the rear window, and 80° for the side windows), and C is a constant. The last component of the incident radiation is the radiation reflected by the ground. This is,

$$I_r = I_{dn} (C + \sin \beta) \delta_g (1 - \cos \Sigma) / 2 \quad (2.3)$$

where δ_g is the ground reflectivity, assumed equal to 0.22, corresponding to a concrete surface. The three components of the incident radiation on the vehicle surfaces are listed in Table 2.2, for the two locations being considered.

Once the amount of incident radiation is calculated, the total insolation load is estimated as follows,

$$q_t = I_t (\tau + U\alpha/h_{out}) \quad (2.4)$$

where I_t is the incident radiation, τ is the glass transmissivity, U is the overall heat transfer coefficient, h_{out} is the external heat transfer coefficient, and α is the glass absorptivity. The calculation has to be done separately for the diffuse and the direct radiation, because glass properties are a function of the angle of incidence. For direct radiation on the windshield, the angle of incidence is zero. For diffuse radiation, ASHRAE, 1981, recommends the use of a 60° angle of incidence. The results are also shown in Table 2.2, for an EV with clear windows, as well as

for an EV with heat reflecting (hot mirror) glass, for the two locations being considered. The glass properties for 0° and 60° incidence angle are given in Table 2.3 for clear glass, heat absorbing glass (used in the top 0.10 m of the windshield in the regular glass vehicle), and hot mirror glass.

The resulting insolation loads are, 888 W for the regular glass vehicle, and 565 W for the hot mirror vehicle, in the southwest. In the southeast, the loads are 761 W for the regular glass vehicle and 484 W for the hot mirror vehicle.

2.1.4. Conduction Load: Conduction load has many components. Energy is conducted through the roof, the glass surfaces, the bottom surface of the vehicle, the rear gate, and the doors. The overall heat transfer coefficient for each surface is calculated by using the heat transfer coefficients given in Table 2.1, considering a moving vehicle with ventilation inside. The conduction load is then found by adding $UA(T_{\text{amb}} - T_a)$ over all the surfaces, where T_a is the air temperature inside the conditioned space. However, the conduction load calculation for the roof is different, since the roof is subjected to direct solar radiation, and therefore its temperature is higher than the ambient temperature. The roof temperature is calculated by using the following equation,

$$I_t \alpha = h_{\text{out}} (T_{\text{roof}} - T_{\text{amb}}) + \frac{(T_{\text{roof}} - T_a)}{(1/h_{\text{in}} + 1/k)} + \epsilon \sigma (T_{\text{roof}}^4 - T_{\text{sky}}^4) \quad (2.5)$$

In this equation, I_t is the total incident solar radiation on the roof, calculated as previously done for the glass surfaces; α is the absorptivity of the roof, which is estimated here as 0.5

Table 2.2. Results for steady-state incident radiation and insolation loads for the two locations being considered.

Surface	Incident Radiation, W/m ²					
	southwest			southeast		
	direct	diffuse	reflected	direct	diffuse	reflected
	from sky	from sky	ground	from sky	from sky	ground
windshield	874	103.2	20.2	750	88.5	17.3
rear window	0	78.6	51.8	0	67.4	44.4
side windows	0	68.7	65.0	0	58.9	55.7

Surface	Insolation Load, W			
	southwest		southeast	
	regular	hot mirror	regular	hot mirror
windshield	703	458	603	392
rear window	48	28	41	24
side windows	137	79	117	68
Total insolation load	888	565	761	484

Table 2.3. Properties of the regular glass, the heat absorbing glass, and the heat reflecting (hot mirror) glass, for 0° and a 60° incidence angles.

	transmissivity		absorptivity		reflectivity	
	0°	60°	0°	60°	0°	60°
regular glass	0.84	0.70	0.08	0.08	0.08	0.22
absorbing glass	0.53	0.40	0.41	0.41	0.06	0.19
reflecting glass	0.52	0.40	0.11	0.11	0.37	0.49

1. Values at 0° incidence are taken from Dieckmann and Mallory (1991).

2. Values at 60° estimated from Chiou (1988), and ASHRAE (1981), which indicate that the major effect of increasing the incidence angle is an increase in reflectivity, while the absorptivity remains fairly constant.

(Shimizu et al. 1982); ϵ is the roof emissivity; σ is the Stefan-Boltzmann constant; and T_{sky} is the radiating temperature of the sky, calculated by the method given by Aschenbrenner and Anderson, 1987. The resulting roof temperature is 50°C in the southwest, and 42°C in the southeast.

The total conduction load for the vehicle is equal to 500 W in the southwest, and 305 W in the southeast.

2.1.5. Total Steady State Cooling Load: Adding all the contributions results in a total steady state cooling load of 1815 W for an EV with regular windows, and 1492 W for an EV with hot mirror glass, in the desert southwest. In the southeast, the steady-state loads are 1802 W and 1525 W respectively for the regular and hot mirror vehicles. In the southwest, the total thermal load is sensible, and there is no need to dehumidify the conditioned space. In the southeast the latent load is equal to 415 W.

The steady-state loads for the two locations being considered are very similar. However, the transient cooling load from hot soak is expected to be larger in the desert southwest, due to the higher ambient temperature and insolation load existing at this location. Therefore, it is considered that the desert southwest represents the most critical location within the country, and only the conditions corresponding to this location are considered in the next sections of this report.

2.2 COOLING LOAD FROM HOT SOAK

The cooling load from hot soak is calculated by using a

transient model for the vehicle interior. The model has two parts. The first part is a radiation heat transfer program that solves for the temperatures inside the EV when it is left in the sun for a long enough period of time to reach steady state conditions. The second part uses the results from the first part as an initial condition, and calculates temperatures and humidities inside the vehicle as a function of time, for given values of the air conditioner power. The model is used to calculate the power required to cool the EV to comfort conditions, in a given time period, starting from the initial hot soak condition. After the cool down period, the model calculates the required cooling power to keep the temperature and humidity at the comfort conditions. A period of 15 minutes is assumed to be the required cool down time.

The radiation model uses a diffuse-gray approximation for all the surfaces in the enclosure, along with an approximate, 2-dimensional, configuration for the vehicle interior (Figure 2.1). The method consists of solving a system of equations for the surface radiosities. This system of equations consists of 11 equations (one for each surface) of the following type:

$$J_i - \sum F_{ij} J_j = q_i \quad (2.6)$$

where J_i is the radiosity of surface i , F_{ij} is the view factor from surface i to surface j , and q_i is the heat flux into surface i , as a result of conduction, convection or solar radiation (not including the radiative exchange within the enclosure). The values of q_i for each of the surfaces in the vehicle are given respectively in Tables 2.4 and 2.5, for a vehicle with regular

Figure 2.1. Two dimensional model of the vehicle interior used in the diffuse-gray simulation of the radiative heat exchange inside the EV.

glass windows and for a vehicle with hot mirror glass windows, at the ambient conditions of the desert southwest.

Once the radiosities are known, the temperatures can be calculated by using the following expression for each surface:

$$\epsilon_i \sigma T_i^4 = J_i - (1-\epsilon_i) \sum F_{ij} J_j \quad (2.7)$$

The emissivity of all the interior vehicle surfaces is taken as 0.9. The last step in the analysis consists of calculating the air temperature within the EV. This is done by using the following equation:

$$M_a c_{v,a} \frac{dT_a}{dt} = \sum h_i A_i (T_i - T_a) + m_a c_{p,a} (T_{amb} - T_a) \quad (2.8)$$

where M_a is the total mass of air inside the vehicle; and m_a is the air mass flow rate for the parked vehicle, which is equal to zero, unless outside air is introduced with a fan when the EV is stopped.

The air and surface temperatures resulting from solving Equations (2.6), (2.7) and (2.8) are shown in Table 2.6. The equations have to be solved iteratively, because the values of q_i are a function of temperature (Tables 2.4 and 2.5).

The surface temperatures are then used to calculate the required cooling power for the EV. This calculation is based on humidity and energy balances for the air conditioner as well as for the vehicle interior. The thermal load in the vehicle has the following components: conduction, ventilation, insolation, metabolic and transient. Calculation of the first four components follows the same procedure detailed for the steady-state load. Calculation of the transient load uses the surface temperatures

Table 2.4. Heat fluxes for the 11 surfaces in the vehicle. Vehicle with regular windows and no fan.

Surface	q_i for Equations (6), W/m^2
1	$4.5(T_a - T_1) + 9.5(T_{amb} - T_1) + 117 - \epsilon_1 \sigma (T_1^4 - T_{sky}^4)$
2	$4.5(T_a - T_2) + (T_{roof} - T_2) + 17.8$
3	$4.5(T_a - T_3) + 9.5(T_{amb} - T_3) + 10.4 - \epsilon_3 \sigma (T_3^4 - T_{sky}^4)$
4	$4.5(T_a - T_4) + 17.8$
5	$4.5(T_a - T_5) + 17.8$
6	$4.5(T_a - T_6) + 1.92(T_{amb} - T_6) + 17.8$
7	$4.5(T_a - T_7) + 17.8$
8	$4.5(T_a - T_8) + 315 + 17.8$
9	$4.5(T_a - T_9) + 1.92(T_{amb} - T_9) + 119 + 17.8$
10	$4.5(T_a - T_{10}) + 17.8$
11	$4.5(T_a - T_{11}) + 621 + 17.8$

1. The terms $4.5(T_a - T_i)$ indicates free convection of air inside the vehicle.

2. Terms that include the factor $(T_{amb} - T_i)$ are convection or conduction terms to the air outside the vehicle.

3. Direct solar radiation is incident on surfaces 8, 9, and 11. Each of these surfaces respectively receive 315, 119, and 621 W/m^2 . These values are calculated by considering that solar radiation is perpendicular to the windshield.

4. Diffuse solar radiation is assumed to distribute uniformly throughout the vehicle. This results in an incident radiation of $17.8 W/m^2$ in all the EV surfaces, except for the windows.

5. Surfaces 1 and 3 radiate energy into the sky, and also absorb part of the solar radiation incident on them. The amount of energy absorbed by surfaces 1 and 3 is respectively equal to 117 and 10.4 W/m^2 .

Table 2.5. Heat fluxes for the 11 surfaces in the vehicle. EV with hot mirror windows and fan operating when vehicle is stopped.

Surface	q_i for Equations (6)
1	$6.0(T_a - T_1) + 9.5(T_{amb} - T_1) + 109 - \epsilon_1 \sigma (T_1^4 - T_{sky}^4)$
2	$6.0(T_a - T_2) + (T_{roof} - T_2) + 10.9$
3	$6.0(T_a - T_3) + 9.5(T_{amb} - T_3) + 14.3 - \epsilon_3 \sigma (T_3^4 - T_{sky}^4)$
4	$6.0(T_a - T_4) + 10.9$
5	$6.0(T_a - T_5) + 10.9$
6	$6.0(T_a - T_6) + 1.92(T_{amb} - T_6) + 10.9$
7	$6.0(T_a - T_7) + 10.9$
8	$6.0(T_a - T_8) + 228 + 10.9$
9	$6.0(T_a - T_9) + 1.92(T_{amb} - T_9) + 70.7 + 10.9$
10	$6.0(T_a - T_{10}) + 10.9$
11	$6.0(T_a - T_{11}) + 370 + 10.9$

1. The terms $6.0(T_a - T_i)$ indicates free convection of air inside the vehicle.

2. Terms that include the factor $(T_{amb} - T_i)$ are convection or conduction terms to the air outside the vehicle.

3. Direct solar radiation is incident on surfaces 8, 9, and 11. Each of these surfaces respectively receive 228, 70.7, and 370 W/m^2 . These values are calculated by considering that solar radiation is perpendicular to the windshield.

4. Diffuse solar radiation is assumed to distribute uniformly throughout the vehicle. This results in an incident radiation of $10.9 W/m^2$ in all the EV surfaces, except for the windows.

5. Surfaces 1 and 3 radiate energy into the sky, and also absorb part of the solar radiation incident on them. The amount of energy absorbed by surfaces 1 and 3 is respectively equal to 109 and $14.3 W/m^2$.

obtained from the radiative model, along with an estimate for the thermal mass, the specific heat, and the surface area for each of the 11 surfaces within the car. The thermal mass in the vehicle interior was estimated as 50 kg by directly weighing the different components for a subcompact car. All surfaces are made of a cushion or padding covered by upholstery and supported by a steel structure. This analysis assumes that only the upholstery and the cushion or padding acquire the surface temperature as calculated by the radiation model. The steel structures and other components that are shielded from radiation are assumed to reach a temperature equal to the air temperature within the EV. The thermal properties for the different surfaces are shown in Table 2.6. The time constant for each surface is calculated as $\tau_i = M_i c_{p,i} / h_i A_i$. The heat transfer from each surface to the air inside the EV is calculated as,

$$Q_i = M_i c_{p,i} (T_i - T_a) / \tau_i \quad (2.9)$$

Figures 2.2 and 2.3 illustrate the results of the calculation. These figures show air conditioner cooling power and air temperature inside the vehicle, for the regular glass and for the hot mirror vehicle respectively. The regular glass vehicle requires 3080 W of initial power to reach comfort conditions in 15 minutes. After this point, the air conditioner cooling power is controlled to keep a fixed (comfort) temperature inside the vehicle, and therefore the required power drops with time, approaching the steady state value. The same behavior is observed in Figure 2.3 for the hot mirror vehicle, for which the required

Table 2.6. Hot soak temperatures and thermal transient properties for the surfaces inside the vehicle.

Surface	Temperatures in °C		Thermal Properties		
	regular	hot mirror	mass, kg	c_p , J/kg K	τ , s
1	72	54	0 ¹	-	-
2	81	52	1.0	1260	142
3	59	45	0 ¹	-	-
4	78	51	2.0	1670	1159
5	83	52	1.3	1260	455
6	78	51	1.0	1260	500
7	81	51	1.6	1260	444
8	109	82	1.6	1260	444
9	93	60	1.3	1260	568
10	95	59	4.4	1670	2560
11	122	90	4.4	1670	2560
air	95	50	2.5	1000	-
shielded surfaces ²	95	50	31.4	451	900

1. Windshield and rear window cool down rapidly when the vehicle is moving, and therefore their thermal mass may be considered equal to zero for the vehicle load calculation.

2. Shielded surfaces do not receive radiation, and therefore have a temperature equal to the air temperature. The value of τ for the shielded areas cannot be calculated easily. $\tau=900$ s is considered to be a good estimate.

Figure 2.2. Air conditioner cooling power and air temperature inside the vehicle, as a function of time, for the regular glass vehicle.

Figure 2.3. Air conditioner cooling power and air temperature inside the vehicle, as a function of time, for the hot mirror glass vehicle.

initial power is equal to 2070 W.

2.3 HEATING LOAD, STEADY STATE

The contributions to the heating load in steady state include metabolic (negative contribution), ventilation, conduction, and defrosting. These are estimated in this section. The design temperatures for the conditioned space and for the surroundings are respectively 23°C and 5°C.

2.3.1. Metabolic Load: The metabolic energy generation is equal to 150 W per person (ASHRAE, 1981), 75 W sensible energy and 75 W latent. For cooling, both of these contributions are included in the calculation. Latent loads are important for cooling because ambient humidity affects how efficiently the human body can dissipate energy by sweating. On the other hand, humidity does not play a major role in determining comfort conditions for heating. Therefore, only the sensible load is considered for this analysis. The total sensible load for two vehicle occupants is -150 W. Metabolic load is a negative contribution because it has the effect of heating the vehicle. Latent load does not directly affect the energy balance inside the vehicle, but it may cause condensation on the windows, introducing a requirement for defogging, as considered in the next paragraphs.

2.3.2. Ventilation Load: During the cooling season, the amount of new air can be reduced substantially. However, for heating, this is not possible, because the humidity generated by the occupants of the vehicle results in condensation on the vehicle windows for low ambient temperatures and low flow rates of new air (Kyle and

Sullivan, 1993). For example, using 0.016 kg/s of outdoor air saturated at 5°C results in a steady state dew point temperature inside the vehicle equal to 13°C, causing substantial condensation. A mass flow rate equal to 0.050 kg/s of outdoor air saturated at 5°C results in a 8°C dew point temperature. This value is considered low enough, and condensation on the windows can be eliminated completely by heating all the windows to temperatures above 8°C. The heating load for a 0.050 kg/s flow rate of new air is equal to 900 W for a 5°C ambient temperature.

2.3.3. Conduction load: As with the cooling load, this analysis uses the heat transfer coefficient data given in Table 2.2 to calculate the conduction load. There is no need for special treatment for the roof, as it was the case for the cooling load. However, windows require heating to at least the dew point temperature inside the vehicle (8°C), and therefore they are considered separately.

Adding the individual conduction heat transfer terms for all surfaces other than the windows results in a heating load of 216 W at 5°C ambient temperature. Keeping all the windows at 8°C during vehicle operation, either by blowing hot air over them or by using electric heating directly in the glass results in a 538 W heat loss due to convection to the vehicle exterior. Adding both contributions results in a total conduction load equal to 754 W.

2.3.4. Defogging load: Avoiding condensation on the EV windows requires enhanced ventilation and heating of the windows. These two contributions have been already calculated in the previous paragraphs, and therefore no additional energy is required for

defogging.

2.3.5. Total Steady-State Heating Load: Adding all the contributions results in a 1500 W total heating load at 5°C ambient temperature.

2.4 HEATING LOAD FROM COLD SOAK

The heating load from cold soak is calculated by assuming that all the vehicle surfaces have a temperature equal to the ambient temperature (5°C), at the moment in which vehicle operation starts. The required heating power and energy are calculated with the same computer code used for the transient cooling load analysis, assuming a heating period of 15 minutes, starting from hot soak.

Figure 2.4 illustrates the results. This figure shows the temperature inside the vehicle, as well as the heating power as a function of time. The initial power requirement to reach comfort conditions in 15 minutes is equal to 1830 W.

2.5 SUMMARY

Table 2.7 lists the energy required for cooling and heating the vehicle. The total energy requirement is obtained by adding the energy for the initial trip (starting from comfort conditions) to the energy of the second trip (starting from hot or cold soak conditions). The energy for the first trip is calculated by multiplying the steady-state load by the time length of the trip. The energy for the second trip is obtained by integrating the power requirement curves (Figures 2.2, 2.3, and 2.4). The results for the cooling load are obtained for the ambient conditions existing

Figure 2.4. Required heating power and air temperature inside the vehicle, as a function of time.

Table 2.7. Cooling and heating energy requirements for all the cases considered. First trip starts from comfort conditions. Second trip starts from a full hot or cold soak.

Cooling	Energy required, J		total
	1st. trip	2nd trip	
1. Regular glass, 30 min	3.27×10^6	5.11×10^6	8.38×10^6
2. Hot mirror, 30 min	2.69×10^6	3.55×10^6	6.24×10^6
3. Regular glass, 1 hr	6.54×10^6	8.90×10^6	1.54×10^7
4. Hot mirror, 1 hr	5.38×10^6	6.50×10^6	1.19×10^7
Heating			
1. Ambient 5°C, 30 min	2.70×10^6	3.19×10^6	5.89×10^6
2. Ambient 5°C, 1 hour	5.40×10^6	6.04×10^6	1.14×10^7

in the desert southwest, where the cooling load is close to the maximum in the country. The cooling loads are used in the next section to calculate the system weight and energy consumption for the air conditioning systems.

The results for the heating load indicate the maximum possible amount of heating that has to be provided to the vehicle before the combustion heater can be used. Although small compared to the heating requirements at low ambient temperatures if no combustion heating is allowed, the total energy requirement for heating is substantial, and could reduce the vehicle range considerably if only electric resistance heating is used to provide it. The following two alternatives are considered:

1. Use the air conditioner to provide heating. The applicability of this alternative depends on the type of air conditioner being used, and is discussed along with the system descriptions in the next chapter.
2. Use heat recovery from the power controller and motor to heat the vehicle. Assuming a 6 kW steady power for the vehicle, and power controller and motor efficiencies of 0.9 and 0.94 respectively, results in approximately 0.9 kW of waste heat, which is approximately equal to 50% of the required heating load, and could be used to complement the heating obtained from either resistance heating or air conditioner heating.

3. AIR CONDITIONER SYSTEM EVALUATION

This section gives a description and evaluation of three systems used to satisfy the cooling loads specified in the previous section. These systems, vapor compression, ice storage, and adsorption air conditioners, have been classified as having the best available characteristics for EV applications, and are described in detail, indicating the overall weight (system plus battery weight to provide energy for system operation), which is used as the figure of merit in this evaluation. System volume (including the volume of the batteries that provide the required energy) and COP are also listed with the results.

The storage capacity for lead-acid batteries, used in this analysis, is approximately 122 kJ/kg at no load; 94 kJ/kg at the vehicle driving load; and 64 kJ/kg, considering a discharge efficiency of 86% and a depth of discharge of 80%. These figures are listed by Hardin, 1992, as representative of high performance, current technology lead acid batteries. These values are used for the base designs. However, battery storage capacity is expected to increase in the future, as better storage technologies are developed, or existing technologies become less expensive. To take this possibility into account, Section 3.4 shows the effect of battery storage capacity on overall system weight for the three systems analyzed in this chapter.

System descriptions also include a calculation of the heating performance for the three systems. Applicability of the air conditioning systems for heating at temperatures above 5°C, for

which combustion heating is not allowed, is of great importance to avoid significant range reductions during the winter months. Additional systems not considered to be competitive in terms of overall system weight are described in Chapter 4.

3.1 VAPOR COMPRESSION AIR CONDITIONER

This system has been in use in most vehicles for a long time, and is therefore the system of choice, unless an alternative system proves to be substantially better in terms of range reduction and/or cost. Variable speed scroll compressors driven by electric motors have been developed recently, and represent an improvement that suits extremely well the applicability of vapor compression air conditioners to EVs.

The system and battery weight for the four combinations of vehicle and trip length are listed in Table 3.1. Two recent publications (Ikeda et al., 1990; Dieckmann and Mallory, 1992) have respectively indicated a weight of 14 kg and 10 kg for the compressor-motor unit. Dieckmann and Mallory, 1992, also indicate a total air conditioner weight of 31 kg, for a system with a 5.3 kW maximum capacity. The maximum AC power requirements for the vehicles considered in the present analysis are respectively 3.08 kW and 2.07 kW for the regular glass and for the hot mirror glass vehicles. While no manufacturer of variable speed compressors of the capacity required in the present work is known to the authors, a correlation has been derived from data published in manufacturer's catalogs for scroll compressor systems that predicts a 3 kg drop in system weight per kW of cooling power reduction.

Table 3.1. System weight by component, lead-acid battery weight to provide the required energy, overall weight, and total volume for a vapor compression air conditioner system, for the energy requirements given in Table 2.7 for the four cases being considered: Case 1. regular glass windows, two 30 minute trips; Case 2. hot mirror windows, two 30 minute trips; Case 3. regular glass windows, two 60-minute trips; Case 4, hot mirror windows, two 60 minute trips.

Case Number	1	2	3	4
System Component				
Air Conditioner weight (kg)	24	21	24	21
Battery Weight (kg)	60	44	109	85
Overall Weight (kg)	84	65	133	106
Total Volume (liters)	44	34	69	55

This correlation is applied to the calculation of the system weights given in Table 3.1.

The energy requirement calculation assumes a COP equal to 2.2, taken as an average of the values reported by Ikeda et al., 1990, once the parasitic losses are included. Considering charge and discharge efficiencies both equal to 0.86, the system COP (ratio of cooling energy to electricity from the grid required to charge the batteries) is equal to 1.65.

The vapor compression AC can easily be adapted to operate as a heat pump, which can provide all of the heating required at temperatures above 5°C. Considering a heating COP equal to 2.5, the system provides the total heating requirement while consuming the energy content of 37 kg of batteries for two 30 minute trips, and 71.5 kg of batteries for two 60 minute trips.

3.2 ICE STORAGE SYSTEM

In this system (Figure 3.1), ice is stored in the shell side of a shell and tube heat exchanger. Ice is made at the same time the batteries are recharged. Two system configurations are possible. The first one uses an external refrigerator, which is used to cool alcohol or any other coolant with a low freezing point and appropriate characteristics. The external refrigerator is connected to the vehicle during charge, and the coolant circulates through the ice storage until all of the water inside the storage freezes. Once the freezing process is completed, the external refrigerator is disconnected from the vehicle, remaining in the charging location. When air conditioning is required during

vehicle operation, the coolant is circulated through the ice storage and through a liquid to gas heat exchanger, cooling the air that circulates into the vehicle. This system has the double advantage of not requiring a refrigerating system on board, and requiring very little battery energy during operation (battery energy is only required for fan and pump operation). Both of these factors help in reducing overall system weight. Two disadvantages of this system are as follows. The first is the need to connect and disconnect the external refrigerator every day. This may be too complicated for widespread public acceptability. The second disadvantage is the requirement for an external refrigerator system at the charging location. The reasonable expectation would be that external refrigeration could be provided at the home charging station. If additional recharging is required away from home, it could be done by directly introducing ice into the ice storage.

The second possible configuration includes a small, low power vapor compression refrigerator as a part of the vehicle equipment, to freeze the water during charging. This increases the system weight. However, this may be the system of choice if the two disadvantages listed above rule out the applicability of the first configuration. The vapor compression freezer is designed to operate only when the batteries are being charged, limiting the amount of battery energy required for system operation.

The system weights for the four combinations of vehicle characteristics and trip lengths are shown in Table 3.2. Weights for the two system configurations described above are identical, with the only exception that the second configuration includes the

Figure 3.1. Schematic diagram of an ice storage air conditioning system.

Table 3.2. System weight by component, lead-acid battery weight to provide the required energy, overall weight, and total volume for the two configurations of an ice storage air conditioner system, for the energy requirements given in Table 2.7 for the four cases being considered: Case 1. regular glass windows, two 30 minute trips; Case 2. hot mirror windows, two 30 minute trips; Case 3. regular glass windows, two 60-minute trips; Case 4, hot mirror windows, two 60 minute trips.

Configuration 1. No freezer in the vehicle.

Case Number	1	2	3	4
System Component				
Ice Stored (kg)	25	19	46	35
Ice Storage/Heat Exchanger (kg)	7	6	11	9
Water to Air HX, pump and fan (kg)	7	9	8	10
Heat Recovery System (kg)	10	10	10	10
Battery Weight (kg)	3	3	6	6
Overall Weight (kg)	52	47	81	70
Total Volume (liters)	75	58	109	88

Configuration 2. Freezer in the vehicle

Case Number	1	2	3	4
System Component				
Ice Stored (kg)	25	19	46	35
Ice Storage/Heat Exchanger (kg)	7	6	11	9
Water to Air HX, pump and fan (kg)	7	9	8	10
Freezer (kg)	20	20	20	20
Heat Recovery System (kg)	10	10	10	10
Battery Weight (kg)	3	3	6	6
Overall Weight (kg)	72	67	101	90
Total Volume (liters)	85	68	119	98

additional weight of the refrigerating system, estimated at 20 kg. The system is designed by using a computer model which calculates the required coolant tube spacing inside the ice storage to provide appropriate cooling power in all cases. The ice storage (shell and tube heat exchanger) and the water to air heat exchanger are made of aluminum for reduced weight. The design process includes a calculation of the energy transfer from the ambient to the ice storage. This loss in cooling power is estimated as less than 5% of the total energy stored, even in the worst case (smallest storage), assuming that 0.02 m of high performance insulation is used in the storage, and a time period of 6 hours from the time the freezing is completed to the time the air conditioner starts operating. Losses in the water and coolant ducts during charging and operation are also found to be under 5% of the total energy stored. Considering 10% total losses, and an average freezer COP equal to 1.1 (ASHRAE, 1983), the system COP is estimated as 1.0. Both configurations of the system have a lower overall weight than the vapor compression air conditioner, making this system an attractive choice for EV air conditioning. An additional advantage is the possibility of being able to easily remove the ice storage (or at least the water) from the vehicle whenever cooling is not required, therefore reducing vehicle weight. System volume is, however, higher than the volume for the vapor compression system, due to the relatively low density of ice. This may be a limitation for high energy requirements.

Table 3.2 shows an estimated weight of 10 kg for a heat recovery system, which includes a heat exchanger, ducts, and a fan.

This extra weight is necessary because ice storage system performance for heating is limited. The only way in which this system can be used for heating is to heat the water in the storage during the charge process and use the hot water for heating during vehicle operation. If water can be heated to 90°C, and heat losses are limited to 10%, the system designs for the regular glass vehicles (Cases 1 and 3) can provide approximately 80% of the heating load at 5°C (Table 2.7). The system designs for the vehicle equipped with hot mirror windows (Cases 2 and 4) can provide about 60% of the heating load at 5°C. A heat recovery system can provide approximately 50% of the heating load (Section 2.5), and provides an adequate complement to the ice storage system for heating.

3.3 ADSORPTION AIR CONDITIONER

Adsorption systems operate by taking advantage of the affinity that some substances, known as adsorbents (silica gel, zeolites, activated carbon, etc.), have for certain liquids or gases (adsorbates). This affinity tends to reduce the vapor pressure of the adsorbates, which, for liquid adsorbates, results in partial evaporation, and cooling of the remaining liquid. The cold liquid can then be used for air conditioning.

Figure 3.2 illustrates an adsorption air conditioner. The operation cycle is divided in two parts, adsorption and regeneration. The adsorption cycle produces the cooling effect in the evaporator. For adsorption, the valves between the evaporator and the adsorbent are opened, which results in evaporation and a

cold adsorbate that can then be used for cooling. However, adsorbents can only adsorb a limited amount of adsorbate. Once this amount is adsorbed, it is necessary to desorb (regenerate the adsorbent) to continue operating the system. The regeneration cycle is usually accomplished by heating the adsorbent to temperatures above 100°C. Once the regeneration cycle is completed, the cooling cycle can be started again. This cyclic operation is very well suited for EVs, since the regeneration cycle can be done during the charge period, and the cooling effect is obtained during vehicle operation, without consuming battery power, except for the power required for fan and pump operation. For a more detailed description of adsorption systems, consult Suzuki, 1990.

Attraction between the adsorbent and the adsorbate is due to either surface forces (physical sorption), or chemical bonds (chemisorption). Most systems studied to date, including the system analyzed in this section, are physical sorption systems, for which property data is more readily available. Rockenfeller et al. 1992, have described several chemisorption systems, and have claimed a very high adsorbing capacity in the adsorbent. However, chemisorption systems are known to have slower kinetics and larger heats of adsorption than physical sorption systems (Wilkinson, 1980), which results in an increased need for heat exchange area, and higher system weight. Mei et al., 1992b, also point out that physical sorption systems have a higher thermal efficiency than chemical sorption systems.

This section analyzes a zeolite-water adsorption system.

Figure 3.2. Schematic diagram of an adsorption air conditioning system.

Zeolite is used because it can adsorb a great amount of vapor, and water is used because it has the highest latent heat of vaporization among the common liquids, providing a high cooling effect for a given adsorbed mass. This high cooling effect for water makes it the best choice, even though water adsorption systems necessarily operate under vacuum conditions, requiring the use of pressure vessels within the system, which add to the system weight.

The resulting weights and volumes for the adsorption systems designed for the four cases being considered, are given in Table 3.3. System designs are obtained from a detailed simulation and optimization code, described in the Appendix, which calculates the system dimensions required to provide the desired cooling load at all times during the operation cycle, and then optimizes the system to obtain a minimum overall system weight. The overall system weights for the adsorption systems are very similar to those obtained for the vapor compression system, with a slight advantage for the adsorption systems for high heating loads. System COPs are on the order of 0.4, inferior to those obtained with the other systems described in this section.

Adsorption cycles can provide heating without difficulty, by using the energy released by the adsorbent when adsorption takes place. The heat of adsorption for the zeolite-water system is approximately 30% higher than the latent heat of vaporization of water. Therefore, the zeolite-water system has the (theoretical) capability to provide 30% more heating than its cooling effect. Consulting the heating requirements given in Table 2.7, it can be

Table 3.3. System weight by component, lead-acid battery weight to provide the required energy, overall weight, and total volume for a zeolite-water adsorption air conditioner, for the energy requirements given in Table 2.7 for the four cases being considered: Case 1. regular glass windows, two 30 minute trips; Case 2. hot mirror windows, two 30 minute trips; Case 3. regular glass windows, two 60-minute trips; Case 4, hot mirror windows, two 60 minute trips.

Case Number	1	2	3	4
System Component				
Adsorbent (kg)	28	19	44	32
Adsorbent Storage (kg)	27	19	38	29
Water (kg)	8	8	11	10
Evaporator (kg)	4	4	5	4
Condenser (kg)	4	4	5	4
Hot HX, pump and fan (kg)	6	5	6	6
Cold HX, pump and fan (kg)	6	5	6	6
Battery Weight (kg)	6	6	12	12
Overall Weight (kg)	89	70	127	103
Total Volume (liters)	107	77	152	119

seen that the four adsorption systems are capable of providing enough heating at a 5°C ambient temperature.

3.4 BATTERY CAPACITY

The previous sections present weight calculations for the four systems (vapor compression, adsorption, and two configurations of ice storage) considered as appropriate air conditioners for EVs. These results are calculated for lead-acid batteries, which are currently the best developed technology for EV propulsion. However, alternative batteries with a higher energy storage capacity are currently being developed, or are available at a high cost. As new technologies become available, or the cost of existing technologies drops, the results of the calculations shown in the previous sections are expected to change. Figures 3.3 to 3.6 show the effect of the battery capacity on system weight for the four systems described in the previous sections, for the four combinations of vehicle characteristics and trip length being considered, as described in Table 1.1. Figure 3.3 is for Case 1, Figure 3.4 is for Case 2, Figure 3.5 is for Case 3, and Figure 3.6 is for Case 4. Battery storage capacities shown in the figures indicate energy storage at zero load (equal to 122 kJ/kg for lead-acid batteries), without taking into account the discharge efficiency and the maximum depth of discharge, which reduce the amount of battery energy that can be used in the vehicle. As expected, vapor compression air conditioners benefit the most from improved batteries, and surpass the ice storage system as the lightest system for large battery storage capacities. The battery

storage capacity at which the vapor compression air conditioner becomes the lightest system is a function of the energy requirement for the system, and is approximately equal to 270 kJ/kg for Case 1 (regular windows, two 30 minute trips), dropping to values in the range between 200 kJ/kg and 250 kJ/kg for the other cases.

The figures also indicate that the ice storage system that includes the freezer is also lighter than the vapor compression air conditioner, for energy capacities under 150 kJ/kg, for all cases except Case 2 (the lowest energy requirement), for which the vapor compression system is lighter even for lead-acid batteries. Adsorption systems have a lower system weight than vapor compression systems only for low battery capacities and high thermal loads (Cases 3 and 4).

3.5 SUMMARY

The overall weight for all the systems described in Sections 3.1 to 3.3 above are listed in Table 3.4 for comparison. The ice storage system with no freezer has between 60% and 70% of the overall system weight of the vapor compression air conditioner, and therefore should be considered as the best choice for cooling the vehicle. Ice storage systems that include the freezer in the vehicle are also lighter than vapor compression air conditioners, for all cases except Case 2 (lowest thermal load considered), and represent a simple compromise to charging infrastructure. Volumes for the ice storage systems are up to twice as large as the volumes for vapor compression air conditioners, and this may restrict some applications, especially at high energy storage requirements.

Figure 3.3. System weight as a function of battery capacity for the four systems described in this chapter, for a vehicle with regular glass windows, used daily for two 30 minute trips (Case 1). Battery capacities shown in the figures indicate energy storage at zero load (equal to 122 kJ/kg for lead-acid batteries), without taking into account the discharge efficiency and the maximum depth of discharge.

Figure 3.4. System weight as a function of battery capacity for the four systems described in this chapter, for a vehicle with hot mirror glass windows, used daily for two 30 minute trips (Case 2). Battery capacities shown in the figures indicate energy storage at zero load (equal to 122 kJ/kg for lead-acid batteries), without taking into account the discharge efficiency and the maximum depth of discharge.

Figure 3.5. System weight as a function of battery capacity for the four systems described in this chapter, for a vehicle with regular glass windows, used daily for two 60 minute trips (Case 3). Battery capacities shown in the figures indicate energy storage at zero load (equal to 122 kJ/kg for lead-acid batteries), without taking into account the discharge efficiency and the maximum depth of discharge.

Figure 3.6. System weight as a function of battery capacity for the four systems described in this chapter, for a vehicle with hot mirror glass windows, used daily for two 60 minute trips (Case 4). Battery capacities shown in the figures indicate energy storage at zero load (equal to 122 kJ/kg for lead-acid batteries), without taking into account the discharge efficiency and the maximum depth of discharge.

The adsorption systems studied in this report have approximately the same overall weight as obtained for the vapor compression air conditioner, and they can provide all the heating required down to 5°C. However, their weight is high when compared to the ice storage systems. Also, their COP is lower than for all the other systems analyzed in this chapter. Therefore, it is considered that further improvement in these systems is required before they can be extensively applied to EV air conditioning.

Table 3.4. Overall weight for vapor compression, ice storage with and without freezer, and adsorption air conditioners, for the energy requirements given in Table 2.7 for the four cases being considered: Case 1. regular glass windows, two 30 minute trips; Case 2. hot mirror windows, two 30 minute trips; Case 3. regular glass windows, two 60-minute trips; Case 4, hot mirror windows, two 60 minute trips.

Case Number	1	2	3	4
System	Weight in kg			
Vapor Compression	84	65	133	106
Ice Storage	52	47	81	70
Ice Storage and Freezer	72	67	101	90
Adsorption	89	70	127	103

4. OTHER AIR CONDITIONING SYSTEMS

This section lists additional air conditioning systems which have been analyzed and considered inappropriate for EV cooling, due to a high system weight or high energy consumption, or both, resulting in a high overall system weight, which is not competitive with respect to those listed in the previous section of this report. These systems are as follows.

4.1. THERMOELECTRICS

Current thermoelectric coolers have a low COP at the conditions existing in EV cooling (estimated as 0.42 by Mei et al., 1992a), and therefore consume a significant part of the total battery energy. Research projects currently under way are aimed at substantially improving thermoelectric cooler performance. However, even obtaining thermoelectric coolers with a figure of merit, Z^* , equal to 0.006, which is twice as large as available for current thermoelectrics, yields a COP of only 1.35, calculated for a 40°C temperature difference between hot and cold sides. Consult Angrist, 1976, for a description of the calculation method and a definition of Z^* . A COP of 1.35 for a thermoelectric system will undoubtedly enhance their applicability in a great variety of conditions. However, it is still too low to compete against the systems listed in the previous chapter. An even higher figure of merit ($Z^*=0.01$), is required for a COP=2, which is high enough to make these systems competitive. Improved EV batteries can also help in making thermoelectrics a better choice for EV cooling.

4.2. STIRLING CYCLES

Stirling cycles have a theoretical efficiency equal to the Carnot efficiency, and therefore have the potential to reach higher efficiencies than those possible with a vapor compression air conditioner. Stirling cycles are commonly used for cryogenic applications, and have the additional advantage of not requiring a CFC refrigerant. However, at this time, Stirling cycles are not competitive in cost, performance (and therefore overall system weight) or reliability when compared to vapor compression air conditioners, for vehicle cooling applications (Mei et al., 1992a). Development continues, and it is possible that Stirling cycles become a better choice in the near future.

4.3. BRAYTON CYCLES

Brayton air conditioners have as their major advantage a low system weight, which has resulted in a broad applicability to aircraft. However, their COP is usually less than 1, which results in an excessive energy consumption for EVs. Dieckmann and Mallory, 1991; and Mei et al., 1992a, have indicated that it is possible to obtain COPs of the order of 2, if very efficient compressors and turbines are used. However, it is doubtful if turbomachinery with the high efficiencies required will be developed in the near future.

4.4. VACUUM BOILING OF WATER

In this system, a vacuum pump is used to reduce the pressure of a water container, to the point in which the water boils. This

causes a drop in temperature for the liquid water that remains in the container, which can then be used for air conditioning. The system has a high maximum theoretical efficiency, but the vacuum pump required to boil enough water to cool the vehicle is excessively heavy, resulting in a large overall weight.

4.5. HEAT ACTUATED AIR CONDITIONERS

These systems, which include ejector, absorption, adsorption with regeneration, and metal hydride systems (Mei et al., 1992a), cannot be applied with advantage to EVs due to the small amount and low temperature of the waste heat available, and also due to the low COP that these systems normally reach.

5. CONCLUSIONS

This report has presented an evaluation of several systems that can be used for providing air conditioning in an EV. Four systems are selected as the best suited to accomplish the cooling. These are, a vapor compression air conditioner, an ice storage system (with and without a freezer in the vehicle), and an adsorption system. For these systems, the report includes a calculation of system weight and COP. All systems are evaluated at four different combinations of vehicle equipment (regular and hot mirror windows), and operating time (two daily trips, each lasting 30 or 60 minutes). The load calculation for these four cases are also presented in this report. The best system is chosen as the one with the least overall weight, where the overall weight is calculated by adding the system weight and the weight of the battery necessary to provide the energy for system operation. The report also shows a calculation of how the battery storage capacity affects the overall system weight. The most important conclusions obtained from this report are as follows:

1. Heat transfer into the EV can be decreased considerably by using special window glasses, extra insulation, reduced external ventilation, and operating a fan when the vehicle is left in the sun to reduce hot soak. Reducing the thermal loads is very important for providing appropriate air conditioning without an excessive system weight.
2. Ice storage systems may be installed with or without a vapor compression refrigerator (freezer) in the vehicle. While using an

external freezer is more desirable in terms of system weight, it may be necessary to install a freezer in the vehicle if using the external freezer is considered impractical. In either case, ice storage systems consume very little energy during EV operation, and have the lowest overall weight of all the systems analyzed. Ice storage systems represent the best alternative for EV air conditioning.

3. Ice storage systems present two disadvantages. The first is the large volume that they require, especially for large cooling requirements. The second disadvantage is being incapable of providing enough heating for an ambient temperature of 5°C . Therefore, this system needs to be complemented with a heat recovery system to provide all the heating required.

4. The zeolite-water adsorption systems analyzed and optimized in this report have approximately the same overall weight as vapor compression air conditioners. While they can provide all the required energy for heating down to 5°C , their COP is low, and they probably require further improvement before they can be extensively applied to EV air conditioning.

5. Increasing the battery storage capacity from its value for lead-acid batteries (122 kJ/kg) results in a great weight reduction for vapor compression air conditioners. These systems eventually reach and surpass ice storage systems as the system with the least overall weight. The battery storage capacity at which vapor compression air conditioners become the lightest system varies as a function of the cooling energy requirement, from approximately 220 to 270 kJ/kg. Vapor compression air conditioners will become

the best method for EV cooling once batteries with this capacity become available at a reasonable price.

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APPENDIX

SIMULATION AND OPTIMIZATION OF AN ADSORPTION AIR CONDITIONER FOR ELECTRIC VEHICLE APPLICATIONS

Adsorption systems operate by taking advantage of the affinity that some substances, known as adsorbents (silica gel, zeolites, activated carbon, etc.), have for certain liquids or gases (adsorbates). This affinity tends to reduce the vapor pressure of the adsorbates, which, for liquid adsorbates results in partial evaporation, and cooling of the remaining liquid. The cold liquid can then be used for air conditioning.

Figure A.1 illustrates an adsorption air conditioner. The operation cycle is divided in two parts, adsorption and regeneration. The adsorption cycle produces the cooling effect in the evaporator. For adsorption, the valves between the evaporator and the adsorbent are opened, which results in evaporation and a cold adsorbate that can then be used for cooling. However, adsorbents can only adsorb a limited amount of adsorbate. Once this amount is adsorbed, it is necessary to desorb (regenerate the adsorbent) to continue operating the system. The regeneration cycle is usually accomplished by heating the adsorbent to temperatures above 100°C. Once the regeneration cycle is completed, the cooling cycle can be started again. This cyclic operation is very well suited for EVs, since the regeneration cycle can be done during the charge period, and the cooling effect is obtained during vehicle operation, without consuming battery power, except for the power required for fan and pump operation. For a

more detailed description of adsorption systems, consult Suzuki, 1990.

This appendix analyzes a zeolite-water adsorption system. This system is chosen because zeolite can adsorb large amounts of water, and water has the highest density and latent heat of vaporization of all the common liquids (Tchernev and Emerson, 1988). This results in a large cooling effect per unit mass of adsorbent, and a small adsorbent mass to satisfy the cooling load. These characteristics make zeolite-water a very good combination for EV air conditioning, even though water adsorption systems operate at extremely low pressures and cannot be used for cooling below 0°C (Critoph, 1988).

The next section presents a description of the adsorption system applied to EV air conditioning. Later sections show the analysis and optimization of the system.

A.1 DESCRIPTION OF THE ADSORPTION SYSTEM

The adsorption system analyzed in this section is a zeolite-water air conditioner. The major components are illustrated in Figure A.1. All heat exchanger surfaces (including the adsorption bed) are made of aluminum for reduced weight and enhanced heat transfer.

Adsorbents release energy (the heat of adsorption) during the adsorption process. This energy has to be withdrawn from the system to obtain a good cooling performance, because high temperatures reduce the capacity of the adsorbent to adsorb vapor. Cooling of the zeolite is obtained by constructing the adsorbent

Figure A.1. Schematic diagram of an adsorption air conditioning system.

bed as a plate fin heat exchanger, with a coolant (water) running inside the tubes, and with the adsorbent pellets stored between the fins (Figure A.2). The water used for cooling the bed is then circulated through a heat exchanger, where heat is transferred to the environment. The external walls of the adsorption bed, evaporator and condenser have to be designed to withstand the external atmospheric pressure, since water adsorption occurs at very low pressures.

The evaporator stores the cold water obtained from the adsorption process. Water from the evaporator is circulated into a heat exchanger to cool air. This air is then used for cooling the vehicle interior. The valves between the evaporator and the adsorbent bed control the amount of water evaporation, and therefore the evaporator temperature. This analysis considers that these valves allow the evaporation required to keep a constant water temperature in the evaporator, compensating exactly for the required cooling load, except on the first few seconds, when it is necessary to cool down the evaporator, from the ambient temperature to the design condition for cooling.

The condenser is of importance only in the desorption process, during which it cools down and condenses the water vapor flowing out of the adsorbent bed. Desorption is performed at the time when the EV batteries are being charged, by connecting electric resistance heaters located inside the adsorption bed. This allows system operation with no battery energy consumption, except as required for fan and pump operation.

Figure A.2. Schematic diagram of the adsorbent bed. The coolant (water) flows inside the tubes, and the adsorbent pellets are stored between the fins (Figure A.2).

A.2 ANALYSIS

This analysis uses a distributed model of the adsorbent bed, and a lumped model (effectiveness-NTU) of the bed cooler heat exchanger. The main simplifying assumptions are as follows.

1. The adsorption bed is isobaric throughout. This assumption is valid due to the large voids existing between the adsorbent particles, the high porosity of the particles, and the relatively low flow rate of adsorbate.
2. Temperature changes inside the adsorption bed occur slowly, so that there is a local equilibrium between the vapor phase and the adsorbate concentration at all locations in the adsorption bed.
3. The evaporator temperature is assumed constant throughout the process. Controlling this temperature involves operating the valves between the evaporator and the adsorption bed so that only the precise amount of vapor flows into the adsorbent, generating the required cooling power to compensate for the heating load into the vehicle.

Assumptions 1 and 2 have been tested against experiments with good results (Guilleminot et al., 1987; Aittomaki and Harkonen, 1988). The third assumption is in contrast with most analyses that assume that adsorption occurs at constant pressure in the adsorbent bed (equal to the vapor pressure of water in the evaporator), as obtained by keeping the valves open during the whole adsorption process. Keeping the valves open results in substantial adsorption during the initial seconds of operation, which is enough to freeze part of the water in the evaporator (Schwarz, 1992). The control strategy used in this analysis, although more complicated, is more

appropriate for EV air conditioning, because it reduces system losses. Generating a block of ice in the evaporator whenever air conditioning is required results in increased heat transfer from the environment. Also, any amount of ice left in the evaporator at the end of a trip is likely to melt before the vehicle is used again, reducing the cooling capacity of the system. Due to the energy limitations existing in battery powered EVs, it is important to reduce energy losses as much as possible.

The mathematical description of the adsorption bed is given by the following set of equations. The first equation is the energy balance for the adsorbent.

$$\rho_s c_{p,b} \frac{\partial T_b}{\partial t} = \frac{\partial}{\partial x} \left[k_b \frac{\partial T_b}{\partial x} \right] + \rho_s H_s \frac{\partial q_s}{\partial t} \quad (\text{A.1})$$

This equation assumes that the bed temperature T_b and the adsorbate concentration q_s (kg of adsorbate per kg of adsorbent) can be described accurately in terms of a single coordinate x and time t . The coordinate x is measured perpendicular to the fins, as shown in Figure A.2. In Equation (A.1), ρ_s is the density of the dry adsorbent, and $c_{p,b}$ is the effective bed specific heat, given as,

$$c_{p,b} = c_{p,s} + q_s c_{p,ad} \quad (\text{A.2})$$

According to the assumptions given above, the adsorbate concentration in every point of the bed is equal to the equilibrium concentration, given implicitly by the following set of equations for the zeolite-water system (Karagiorgas and Meunier, 1987),

$$p = 101 \exp(\Theta + \Phi/T) \quad (\text{A.3})$$

with

$$\theta = 20.49 - 60.4 q + 787 q^2 - 2.14 \times 10^3 q^3 \quad (\text{A.4})$$

and

$$\phi = -8013 + 33.83 \times 10^3 q - 3 \times 10^5 q^2 + 7.9 \times 10^5 q^3 \quad (\text{A.5})$$

The isosteric heat of adsorption, H_s , can be calculated from the equilibrium concentration equations (see, for example, Valenzuela and Myers, 1989). The effective thermal conductivity of the bed, k_b , is a function of many factors, the most important being the thermal conductivities of the solid and vapor phases, the porosity of the bed, the size and shape of the adsorbent particles, the vapor pressure inside the bed, the mechanical and optical properties of the particles, and the thermodynamical and optical properties of the vapor. Pressure is usually not an important factor in determining bed conductivity. However, in this analysis, the combination of low vapor pressure and small dimensions of the bed flow paths result in rarefied gas behavior of the vapor; thus the bed thermal conductivity is a function of pressure. The bed conductivity is calculated by using the Zehner-Bauer model, as described by Tsotsas and Martin, 1987. This model takes into account all the effects listed above, and has proven to be accurate in a wide range of experimental conditions. This model is also used to find the thermal conductivity at the wall. Suzuki, 1990, has indicated that the thermal conductivity near the wall is not the same as the thermal conductivity far from the wall, because bed porosity increases near the wall. To take this effect into account, this model uses a bed porosity function that increases as the distance to the wall becomes of the order of the particle

diameter.

The temperature of the metallic heat exchanger surfaces (fins and tubes) is assumed to be constant throughout the adsorption bed. This is considered to be a good assumption for surfaces made of aluminum, especially because the results indicate that the temperature of the cooling water increases only by approximately 1°C as it flows through the adsorption bed. The equation for the metal temperature is,

$$M_m C_{p,m} \frac{dT_m}{dt} = k_b A_b \frac{\partial T_b}{\partial x} - h_{out} A_{out} (T_m - T_{amb}) - m_c C_{p,c} (T_{c,out} - T_{c,in}) \quad (A.6)$$

In this equation, the first term in the right hand side is the heat transfer from the adsorbent into the metallic surfaces. The second term represents the heat losses from the external surface of the container to the ambient, and the third term is the energy transfer from the metallic surfaces to the cooling water.

Assuming a constant temperature in the metal results in the following expression for the water temperature flowing out of the adsorption bed,

$$T_{c,out} = T_m - (T_m - T_{c,i}) \exp(-NTU) \quad (A.7)$$

where $NTU = h_c A_c / m_c C_{p,c}$.

The coolant temperatures, $T_{c,in}$ and $T_{c,out}$ are also related by the equations that define the performance of the bed cooler heat exchanger. This is selected as a finned flat tube heat exchanger, with a fan to produce an air flow through the surfaces. This heat exchanger is analyzed with the usual efficiency-NTU method, as detailed by Kays and London, 1984.

The system of Equations (A.1) through (A.7) is solved iteratively, by assuming a vapor pressure inside the adsorption bed and calculating the bed temperatures and concentrations, as well as the metal and coolant temperatures. Bed temperatures and concentrations are obtained from Equation (A.1) by using an implicit, finite difference method and the Thomas algorithm. New guesses for the pressure are generated until the amount of vapor adsorbed is equal to the evaporation required to keep a constant evaporator temperature. Once this iteration converges, it is necessary to iterate further to match the temperatures $T_{c,in}$ and $T_{c,out}$ obtained from Equation (A.7) with those obtained from the effectiveness-NTU analysis of the bed cooler heat exchanger ($T_{c,out}$ must be equal to the temperature at the bed cooler heat exchanger inlet, and $T_{c,in}$ must be equal to the temperature at the bed cooler heat exchanger outlet, see Figure A.1). When convergence is obtained, the process is complete, and the program moves into the next time step. For each time step, the program reads the required cooling power (given in Table 2.7 and Figures 2.2 and 2.3) from a data file.

The air conditioner is designed for two daily trips, separated by a period in which the vehicle is not operated. While some cooling of the bed may occur in the period between trips, the thermal mass of the bed is large and cooling during this period can only be due to natural convection. Therefore, it is assumed that the bed temperature at the start of the second trip is equal to the bed temperature at the end of the first trip. This assumption represents the worst possible case, since any cooling of the bed

between the trips increases the capacity of the system.

The vapor pressure inside the bed increases as the adsorption process advances. When the pressure inside the bed is equal to the vapor pressure of the water in the evaporator, no more evaporation can occur, and the adsorption process stops. This limits the cooling capacity of the system. It is desirable that the adsorption system has enough power to satisfy the cooling load, without having excess capacity, which would increase the system weight. The simulation code calculates this system design, by adjusting iteratively the mass of adsorbent in the system until the adsorption process stops exactly at the moment in which the overall heating load is satisfied.

Once the design that satisfies exactly the cooling load is obtained, the simulation code calculates the conditions for desorption. During desorption, the system is heated by connecting electric resistances located inside the adsorption bed. Thermal gradients inside the adsorption bed during desorption are considered small, and therefore the calculations assume a constant temperature throughout the adsorption bed, equal for the adsorbent and the metal. The simulation code calculates the temperature and the water concentration in the adsorbent as a function of time, the time period required for desorption, and the COP for the overall process.

All pressure vessels in the system are designed according to the ASME Pressure Vessel Code (ASME, 1983). The use of aluminum as material for the pressure vessels limits the maximum operating temperature (and therefore the desorption temperature) to 200°C.

A.3 OPTIMIZATION

The four zeolite-water air conditioners (one for each of the four driving scenarios described in Table 1.1) are optimized with respect to the space between the fins inside the adsorption bed, the three dimensions of the bed cooler heat exchanger, and a selection of three fans used in the bed cooler heat exchanger. The figure of merit is the overall system weight, which includes the system weight and the battery weight required for operating the system. The optimum design is obtained in all cases as a trade off between enhanced bed heat transfer performance (which allows a reduction of the adsorbent mass required to meet the cooling demand), and the added system weight that results from the heat transfer enhancements. For instance, using a larger fan results in a more efficient bed cooling, improving the performance. However, the larger fan weighs more, and consumes more energy, increasing the battery weight required to provide the air conditioning. See Table A.1 for a description of the decision variables. Other system parameters are listed in Table A.2.

A.4 RESULTS

Table A.3 shows the optimum designs for each of the four cases being considered. The optimum values of the decision variables are listed, along with the weight of the different system components. All optimum designs use $N_{fan}=1$, since the increase in battery weight necessary to power a bigger fan is larger than the reduction in heat exchanger weight that can be obtained by using the bigger fan. Heat exchanger dimensions are only slightly larger for the

Table A.1. Decision variables used in the adsorption air conditioner optimization.

Variables	Description
a,b,c	<p>water-to-air bed cooler heat exchanger dimensions</p> <p>a and b are perpendicular to the air flow direction a and c are perpendicular to the water flow direction</p>
d	distance between fins inside the adsorption bed
Nfan	<p>fan number. Select a fan out of the following three:</p> <p>Nfan=1, fan power 50 W Nfan=2, fan power 83 W Nfan=3, fan power 108 W</p> <p>Fan dimensions and performance curves are obtained from a manufacturer's catalog, and incorporated into the simulation code.</p>

Table A.2. Adsorption system parameters for analysis and optimization.

Parameter	Value
ambient temperature during adsorption	43°C
ambient temperature during desorption	33°C
evaporator temperature	10°C
desorption temperature	200°C
bed porosity	40%
bed porosity at the wall	70%
particle diameter	0.001 m
bed density (dry)	560 kg/m ³
heating power for desorption	2000 W
number of points for temperature calculation in half of the distance between fins	20
time step	10 s

Table A.3. Optimum designs for the adsorption air conditioners, including the values for the 5 decision variables, system weight by component, battery weight to provide the required energy, and overall weight for a zeolite-water adsorption air conditioner, for the energy requirements given in Table 2.7 for the four cases being considered: Case 1. regular glass windows, two 30 minute trips; Case 2. hot mirror windows, two 30 minute trips; Case 3. regular glass windows, two 60-minute trips; Case 4, hot mirror windows, two 60 minute trips.

Case Number	1	2	3	4
Decision Variable				
a (m)	0.35	0.30	0.35	0.30
b (m)	0.35	0.30	0.35	0.30
c (m)	0.12	0.10	0.12	0.12
d (m)	0.004	0.004	0.005	0.005
Nfan	1	1	1	1

Case Number	1	2	3	4
System Component	Weight in kg			
Adsorbent	28	19	44	32
Adsorbent Storage	27	19	38	29
Water	8	8	11	10
Evaporator	4	4	5	4
Condenser	4	4	5	4
Hot HX, pump and fan	6	5	6	6
Cold HX, pump and fan	6	5	6	6
Battery Weight	6	6	12	12
Overall Weight	89	70	127	103

regular vehicles (Cases 1 and 3), as compared with the hot mirror vehicles (Cases 2 and 4). The required adsorbent mass, however, increases almost linearly as the cooling load increases. Heat exchanger dimensions for Case 1 are very similar to those obtained for Case 3, and the same similarity exists between Cases 2 and 4. The reason for this behavior is the fact that the required cooling rate, and not the total cooling, is what determines the optimum heat exchanger size. The value of d (half the distance between fins inside the adsorption bed) for the optimum designs, changes from 0.004 m for Cases 1 and 2 to 0.005 m for Cases 3 and 4. This reflects the fact that beds for Cases 1 and 2 (for 30 minute trips) operate at a higher power rate per unit mass of adsorbent, and therefore require additional fin surface to avoid excessive heating, which reduces the adsorption capability.

Figures A.3 through A.6 are used to illustrate the results. These figures show results for Cases 2 and 3, respectively the smallest system and the largest system, which are considered as representative of the conditions existing in the other cases. Figures A.3 and A.4 show the average adsorbent temperature, metal temperature, coolant temperature flowing into the adsorption bed, and coolant temperature flowing out of the adsorption bed, as a function of time. Figure A.3 is for Case 2, and Figure A.4 is for Case 3. Both figures show temperatures that initially increase rapidly as the heat of adsorption is released. Temperature levels then approach steady-state values, where the heat of adsorption being released is transferred into the ambient by the flowing coolant. After the first trip has been completed (30 minutes for

Case 2 and 60 minutes for Case 3), the vehicle remains unused for a period of time, during which the bed temperature is assumed to remain constant. The period between the two trips is not shown in the figures. When the second trip starts, the temperatures increase rapidly, because the second trip begins with the vehicle at hot soak conditions, and the required power is higher than for the first trip (Table 2.7). After 15 minutes, comfort conditions are reached, and the required power drops. As the cooling power drops, the temperatures stop increasing, reach a peak and start dropping.

Figures A.3 and A.4 also illustrate the small difference between inlet and outlet coolant temperatures, which supports the constant metal temperature assumption used in this model.

Figure A.5 shows the concentration of water adsorbed as a function of time during the operation cycles, for Cases 2 and 3. The amounts adsorbed during a time interval are proportional to the cooling load, calculated in Chapter 2. Figure A.5 shows that both systems adsorb approximately 0.15 kg of water per kg of dry zeolite, with the larger system adsorbing more than the smaller system. The larger system can adsorb more water per kg of zeolite because the cooling power per unit mass of adsorbent is smaller for this system. This results in lower adsorption bed temperatures for the larger system, as can be seen from Figures A.3 and A.4. A lower temperature results in a higher capacity to adsorb water.

Figure A.6 shows the bed temperature and the water concentration in the adsorbent as a function of time during the desorption process, for Case 2. The ambient temperature for

Figure A.3. Average adsorbent temperature, metal temperature, coolant temperature flowing into the adsorption bed, and coolant temperature flowing out of the adsorption bed, as a function of time, for a vehicle with hot mirror windows used daily for two 30 minute trips (Case 2).

Figure A.4. Average adsorbent temperature, metal temperature, coolant temperature flowing into the adsorption bed, and coolant temperature flowing out of the adsorption bed, as a function of time, for a vehicle with regular windows used daily for two 60 minute trips (Case 3).

Figure A.5. Concentration of water adsorbed as a function of time during the operation cycles, for Case 2, hot mirror vehicle and two daily 30 minute trips; and Case 3, regular glass vehicle and two daily 60 minute trips.

desorption is 33°C , lower than the ambient temperature during adsorption (43°C), because desorption normally takes place at night, at the same time the EV batteries are charged. Desorption is obtained by heating the adsorption bed with electric resistance heaters. The total heating power during desorption is 2000 W. The figure shows that desorption does not start until 2400 s after the heaters are connected. Before this time, the bed temperature is too low for any desorption to occur. Once water starts desorbing, the slope of the temperature line decreases, as part of the energy is applied to providing the heat of desorption. The process is concluded when the maximum operating temperature (200°C) is reached. This temperature determines the amount of water adsorbed at the end of the desorption process, an important factor in the calculation of the cooling power.

The system weights shown in Table A.3 are very similar to the system weights obtained for the vapor compression air conditioner (Table 3.4), with a slight advantage for the adsorption system for large systems (Cases 3 and 4). However, ice storage systems are substantially lighter, and have a higher COP. Adsorption systems require additional improvements before they can be applied with advantage to EV air conditioning. A recent survey (Mei et al., 1992b), has indicated current adsorption projects and possibilities for future improvement.

A.5. CONCLUSIONS

This appendix has shown the procedure and the results for the calculation of adsorption systems to provide the required air

Figure A.6. Bed temperature and water concentration in the adsorbent as a function of time during the desorption process, for Case 2, hot mirror glass vehicle and two daily 30 minute trips.

conditioning for the four cases being considered in this report. The model assumes a uniform pressure inside the adsorption bed, an adsorbent temperature that varies as a function of time and distance from the fins, and a uniform evaporator temperature.

The four systems being considered are optimized with respect to five decision variables. The results show the optimum values of all the variables, as well as overall system weights for all the configurations. The results also include temperatures and amounts adsorbed, for adsorption as well as for desorption.

System weights for the adsorption systems are very similar to the system weights for the vapor compression air conditioners analyzed in Chapter 3 of this report. However, ice storage systems can provide the required cooling load with a lower system weight, and therefore it is concluded that adsorption systems require further improvement before they can be extensively applied to EV air conditioning.

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