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## ABSTRACT

In the United Sates, more than 80% of transit city buses are air conditioned. Vapor compression refrigeration systems are standard for air conditioning buses and account for up to 25% of fuel consumption in the cooling season. Vapor compression devices use chlorofluorocarbons (CFCs), chemicals that contributes to Earths's ozone depletion and to global warming. Currently, evaporative cooling is an economical alternative to CFC vapor compression refrigeration for air conditioning buses. It does not use CFCs but is restricted in use to arid climates. This limitation can be eliminated by dehumidifying the supply air using desiccants. We studied desiccant systems for cooling transit buses and found that the use of a desiccant-assisted evaporative cooling system is feasible and can deliver the required cooling. The weight and the size of the desiccant system, though larger than vapor compression systems, can be easily accommodated within a bus. Fuel consumption for running desiccant systems was about 70% less than CFC refrigeration systems, resulting in payback periods of less than 2.5 years under most circumstances. This preliminary study indicated that desiccant systems combined with evaporative cooling is a CFC-free option to vapor compression refrigeration for air conditioning of transit buses. The concept is ready to be tested in a full prototype scale in a commercial bus.

## INTRODUCTION AND BACKGROUND

The interior of a bus requires cooling because of heat loads gained by outside temperatures, solar gain, passengers, engine heat transmission, and air infiltration (ASHRAE, 1991; Kohler et al., 1990). Air conditioning of buses is now standard in all regions of the country to satisfy passenger comfort and, thus, to sustain a minimum number of patronage. Today, about 83% of city buses in the United States are equipped with air conditioning systems (APTA, 1990). Almost all buses use conventional *mechanical refrigeration* (vapor compression [V/C]) systems.

In a mechanical refrigeration system, a refrigerant fluid is maintained under pressure in a closed system consisting of a reservoir of liquid, an expander, an evaporator, a compressor, and a condenser (Figure 1). The process is as follows: Liquid at high pressure in the reservoir is allowed to "expand" into the lower pressure evaporator and transform into its vapor phase. In doing so, it absorbs heat from the surroundings, usually the space being cooled. The vapor then proceeds to the compressor where its pressure is increased. It is then fed to the condenser where it is allowed to return to its liquid state and accumulate in the reservoir. In this transformation, it releases heat to the surroundings of the condenser. Such a system is usually presented in a compact package of pressure tubing and vessels incorporating a compressor which consumes a significant amount of power to operate. Ancillary fans and blowers are required to move the air being cooled and the air used in cooling the condenser. The total power consumption for a typical V/C air conditioner for a bus is between 12 and 15 kW (16 and 20 hp) (see Table 1).

Usually, the refrigerant in these V/C air conditioning systems are chlorofluorocarbons (CFCs), chemicals that contribute to depletion of Earth's ozone layer and to global warming. CFCs are being subjected to increasingly restrictive

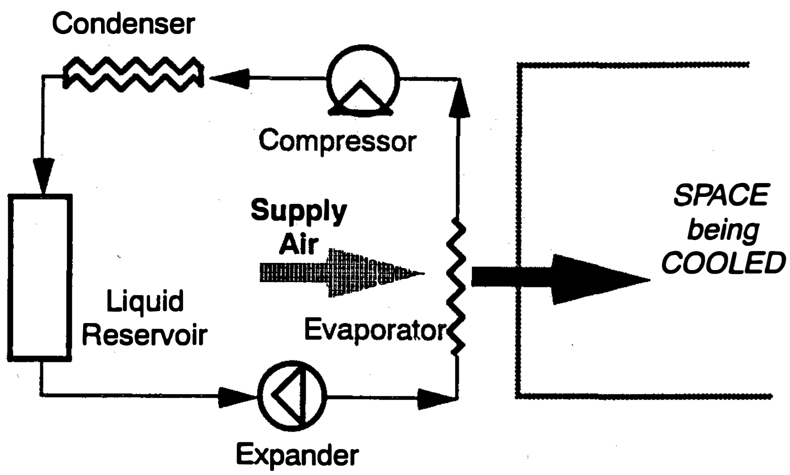


**Figure 1. Schematic of Vapor Compression Cooling System**

**Table 1**
**Summary of Bus Air Conditioners from Four Manufacturers**

| Unit | Manufacturer | Weight (Approx) (kg) | Machine Capacity (kW) | Flow Rate ($m^3$/min) | Compressor or Pump Power (kW) | Fan Power (kW) | Total Power (kW) |
|---|---|---|---|---|---|---|---|
| Capri 280 | Carrier | 340 | 23.5 | 69 | 13.1 | 2.0 | 15.1 |
| AC-31S | Suetrak | 340 | 25.5 | 61 | 10.5 | 1.5 | 12.0 |
| T Series | Thermo King | 410 | 24.0 | 68 | 11.8 | 1.9 | 13.7 |
| Transit-Aire | Climatran | 730 | 17.6 | 113 | 0.3 | 1.1 | 1.4 |

Notes:
1. All these units provide similar cooling performance. Tests with the Transit-Aire unit showed that it performs better in "pull-down" tests because of higher flow rate (Climatran, 1989).
2. The first three units work by mechanical refrigeration using R-12 refrigerant, but Transit-Aire works based on the principle of evaporative cooling.
3. The weight of the units includes supporting frames. The weight of the Transit-Aire unit contains 450 kg of water for evaporative cooling.

regulations and will be banned eventually. Venting CFCs is prohibited by law and any release during repair or maintenance can be subject to as much as a $25,000 fine. Although new safer refrigerants are on the horizon, these systems are relatively complex mechanically, and their failure is the number one cause of downtime for the vehicle and may also result in early retirement of an otherwise serviceable vehicle (Thermo King, 1985). An alternative to vapor compression air conditioning is *evaporative cooling*.

Evaporative cooling is mechanically and operationally simpler, because it does not require a compressor or CFC fluid. Instead, only a water supply is required along with an ample supply of relatively dry air. Evaporative cooling works based on the principle of evaporating water by air. Liquid water is sprayed onto a porous pad with air blowing through it. Depending on the dryness of the blowing air, water is evaporated into the air. The heat of vaporization is supplied by the air, thus reducing the air temperature. Figure 2 shows a schematic of a direct evaporative cooling system. In direct evaporative cooling, the air is humidified as it is cooled. In indirect evaporative cooling, the supply air stream is cooled without increasing its moisture content. This is achieved by evaporative cooling an auxiliary source of air and passing it through the exhaust side of a heat exchanger while simultaneously passing the primary air stream through the opposite or supply side of the heat exchanger.

Evaporative cooling systems are simple in design, construction, and operation. Energy requirements are about 1.5 kW (2 hp) (see Table 1), much smaller than the power needed by V/C systems. Evaporative cooling systems consist

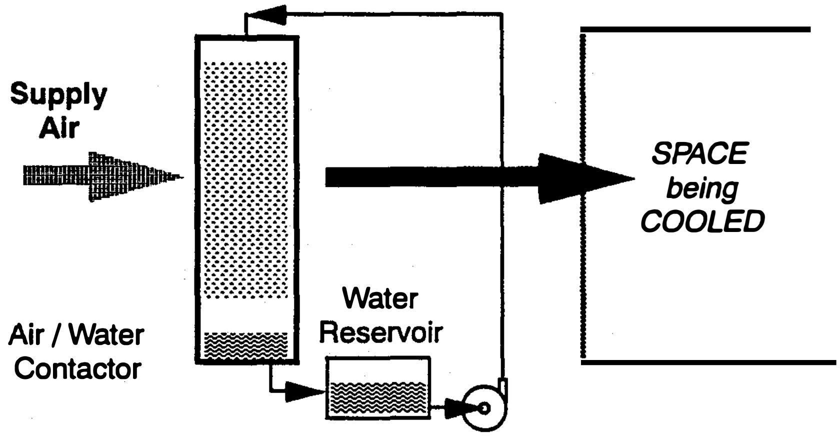


**Figure 2. Schematic of Evaporative Cooling System**

of fans or blowers to move the air through the contractor and around the space to be cooled and pump water over the pad. A water supply must be provided and replenished and may contribute to a slight increase in system weight and bulkiness. The water is usually treated to minimize fouling of contact surfaces. The extent of cooling that can be achieved depends on the extent of water vapor content (also characterized as wet bulb temperature) of the supply air and its actual temperature (called dry bulb temperature).

Evaporative cooling systems are widespread and popular throughout dry climate regions where they are used principally for the cooling of buildings. In the early 1980s, it was demonstrated that evaporative cooling could be applied to the cooling of buses in relatively dry climate areas with a significant (≥ 80%) reduction in power consumption (UMTA, 1983). In the past several years, a manufacturer (Climatran Corporation) has improved the design and economics and installed evaporative cooling systems on more than 400 buses.

As the wet bulb temperature increases, the effectiveness of evaporative cooling decreases. This limits the applicability of evaporative cooling systems to arid and semi-arid climates. Referring to Figure 3, direct evaporative cooling can be used in region I (e.g., Colorado, Arizona), and indirect/direct evaporative coolers can be used in region II (e.g., parts of California, Kansas, the Great Lakes). However, the stand-alone evaporative cooling systems are not effective in humid climates such as the southeast quadrant of the United States as represented by regions III and IV.

To expand the range of applicability of evaporative cooling to all climates, including regions II, III, and IV, the air supplied to the evaporative cooler has to be pre-dehumidified. This can be achieved by preconditioning supply air with desiccant-based dehumidifiers. The dehumidifier removes the water vapor from the supply air. In order for the dehumidifier to operate continuously, the desiccant needs to be regenerated, i.e., the removed water must be driven off to the exhaust side by hot air. The concept of desiccant-assisted evaporative cooling has been demonstrated for building applications (Harriman, 1990) in humid climates. In building air conditioning applications, natural gas or solar energy can provide the hot air. For a bus application the waste heat from the vehicle's engine can provide the necessary heat, reducing energy requirements.

The purpose of this study is to evaluate the feasibility of applying the concepts of *desiccant-assisted evaporative cooling* techniques to air conditioning of transit buses. Such a process would broaden the range of applicability of evaporative cooling to essentially all climatic regions. The incentives for the use of evaporative cooling in buses are the potential for:

- reduced fuel consumption,
- simplicity, reduced maintenance and down time,
- a viable alternative to CFC-based cooling systems at competitive cost.

## BACKGROUND INFORMATION

The initial step in the study was to gather relevant information concerning the vehicles, the air conditioning systems,and geographical climatic data across the country. Of particular interest were the nature and intensity of the energy loads and distributions on the vehicle, the specifications and performance characteristics of conventional V/C cooling systems, the characteristics of supply air, and the desired effect under the more demanding climatic conditions.

VEHICLE MECHANICAL INFORMATION - There are 60,000 mass transit buses in the United States (APTA, 1990). These numbers do not include intercity, school, military, and other buses. A transit bus is designed for frequent-stop service, has front and center side doors, is usually powered by a rear-mounted diesel engine, and offers low-back seating without storage compartments or restrooms. The majority of transit buses are of standard size, the length varying from 10.7 to 12.5 m (35 to 41 ft) (APTA, 1990). The standard bus characteristics selected for this study were length 12.20 m; width 2.6 m; height 3; capacity of 49 seats; and average weight 14500 kg. The standard engine for such a vehicle is 277 break horsepower (207 kW) diesel. Of particular interest to this

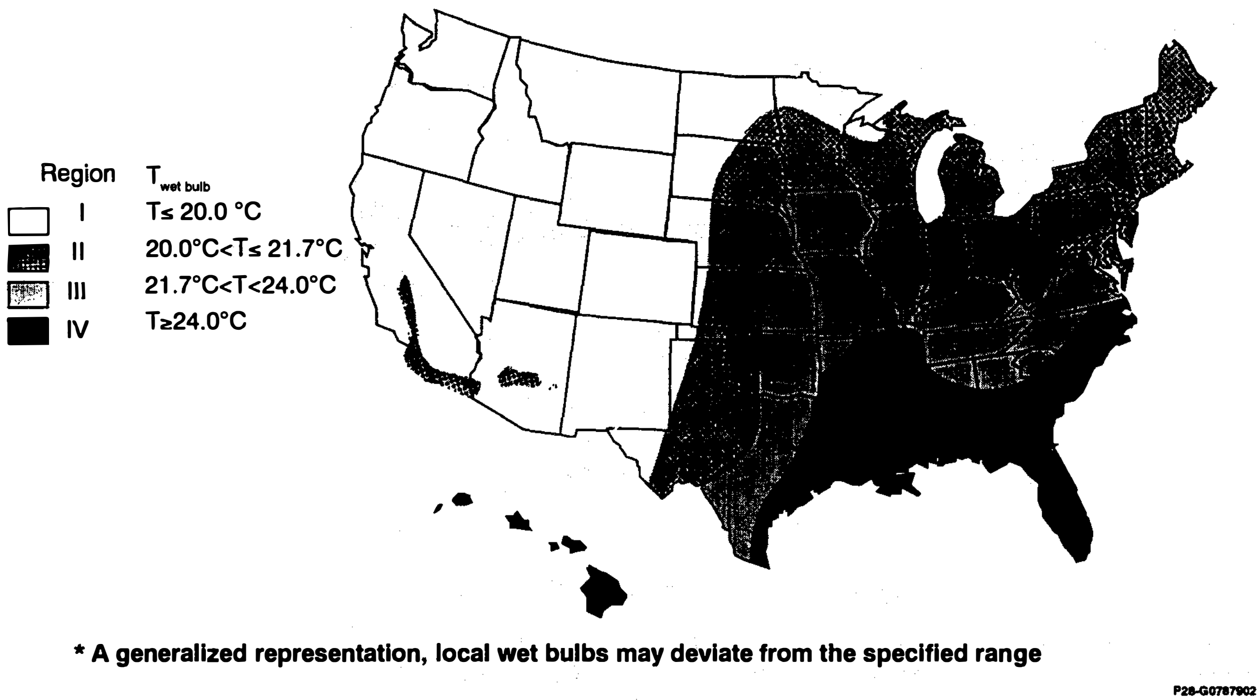


**Figure 3. Regions of Common Wet Bulb Temperature in the United States (excluding Alaska)**

design is the amount of 175 kW (600,000 Btu/h) engine heatthat has to be rejected and could be available for regeneration of a desiccant dehumidifier (Detroit Diesel Corp., 1984).

VEHICLE ENERGY CONSUMPTION INFORMATION - According to AiResearch (1980), the typical fuel consumption of a engine of a bus is on the order of 0.315 L/kWh (0.062 gal/hp-h). Another element in fuel consumption is weight. A 2% increase in vehicle weight results in an increase of 1% in fuel consumption (AiResearch, 1980). This translate into a 0.15 L/h (0.03 gal/h) increase in fuel usage for every 300 kg (500 lb) increase in weight.

VEHICLE COOLING LOAD AND AIR CONDITIONER INFORMATION - The cooling load in a bus consists of the sum of the heat loads associated with solar gain, heat conduction, passengers, engine heat transmission, and air infiltration. Table 2 summarizes the results of total cooling load estimates from four different sources. Estimates differ because of differences in assumptions and locations considered. For this study, we selected a value of 17.6 kW (60,000 Btu/h) for the total cooling load. Of this load, 75% corresponds to sensible heat, and 25% to latent heat. Table 1 presents some specifications of some air conditioners that are currently used in buses.

CLIMATIC CONDITIONS - A representative list of U.S. cities is given in Table 3 showing summer design climatic data (ASHRAE, 1989). In this study, we chose the most demanding conditions corresponding to the Houston, TX data of 35°C (95°F) dry bulb temperature and 25°C (77°F) wet bulb temperature. Also provided in Table 3 are the annual operating hours for bus air conditioners.

## ANALYSIS, DESIGN, AND RESULTS

In order to design a desiccant evaporative cooling system, design constraints have to be identified. We discussed some of these design constraints in the previous section; they are summarized in Table 4.

Supply air flow rate is a critical parameter in designing and sizing any desiccant cooling system. The supply flow rate necessary to satisfy the minimum cooling requirements imposed on the vehicle depends on the dry bulb temperature supplied to the bus as shown on Figure 4. The supply air flow rate and supply air humidity were selected so that the air humidity in the interior of the bus did not exceed the maximum allowable air humidity. Based on our experience with desiccant evaporative cooling systems, a supply air temperature above 18°C (64°F) is obtainable. This resulted in a practical range of supply flow rates between 70 and 100 $m^3$/min (2500 and 3500 cfm). Higher flow rates would result in large component sizes and excessive and uncomfortable air velocities in the interior of the bus. To design a system, we considered three different configurations. Two used solid desiccant materials, and one used liquid desiccant material. Based on a selected flow rate, individual components for each system were designed and sized based on the desired performance. The performance of the complete system was

**Table 2**
**Estimates of Bus Cooling Load from Various Sources**

| Source | Conduction (kW) | Solar (kW) | Passenger (kW) | Engine (kW) | Fresh Air (kW) | Total (kW) |
|---|---|---|---|---|---|---|
| UMTA | 4.14 | 3.65 | 5.13 | -- | -- | 12.92 |
| Climatran | 2.29 | 3.68 | 2.92 | 2.92 | -- | 11.71 |
| AiResearch | 2.78 | 3.52 | 10.93 | -- | 10.95 | 28.18 |
| Lemke, 1975 | 6.15 | 1.82 | 5.98 | -- | 5.42 | 19.37 |

**Table 3**
**ASHRAE Summer Design Weather Data**
**for Several Major U.S. Cities (ASHRAE, 1989)**

| City | 2.5%[1] Dry Bulb (°C) | 5%[1] Wet Bulb (°C) | Estimated Seasonal A/C Operating Hours[2] |
|---|---|---|---|
| Boston, MA | 31.1 | 21.1 | 900 |
| Chicago, IL | 32.8 | 22.2 | 1200 |
| Denver, CO | 32.8 | 15.0 | 1000 |
| Houston, TX | 35.0 | 25.0 | 3000 |
| Las Vegas, NV | 41.1 | 18.3 | 3000 |
| Los Angeles, CA | 31.7 | 20.6 | 1500 |
| Miami, FL | 32.5 | 25.0 | 3000 |
| New York, NY | 31.7 | 22.2 | 1500 |
| St. Louis, MO | 34.5 | 23.3 | 1800 |
| San Antonio, TX | 36.1 | 22.8 | 2000 |
| Washington, DC | 32.8 | 23.3 | 1600 |

[1] 2.5% or 5% design condition means 2.5% or 5% of the summer cooling hours may exceed this (dry bulb or wet bulb) temperature. The concept of ASHRAE design conditions is intended to provide a basis for sizing heating, ventilating, and air conditioning equipment by assuring adequate capacity and avoid costly over design.
[2] Based on annual hours at various temperatures (U.S.A.F., 1987), hours that ambient temperature is higher than 21.1°C was considered since the bus is hotter as a result of solar gain and engine heat transmission.

**Table 4**
**Design Criteria for System Evaluation**

| | |
|---|---|
| Typical Heat Load to be Removed from a Bus | Sensible: 14.3 kW (45,000 Btu/h)<br>Latent: 4.8 kW (15,000 Btu/h) |
| Ambient Air Conditions (Extreme Case) | $T_{dry\ bulb}$: 35°C (95°F)<br>$T_{wet\ bulb}$: 25°C (77°F)<br>Humidity: 0.01591 kg water vapor/kg dry air |
| Bus Interior Air Conditions (Maximums)<br>(Exit from the Bus) | $T_{dry\ bulb}$: 26.7°C (80°F)<br>Humidity: 0.0150 kg water vapor/kg dry air |

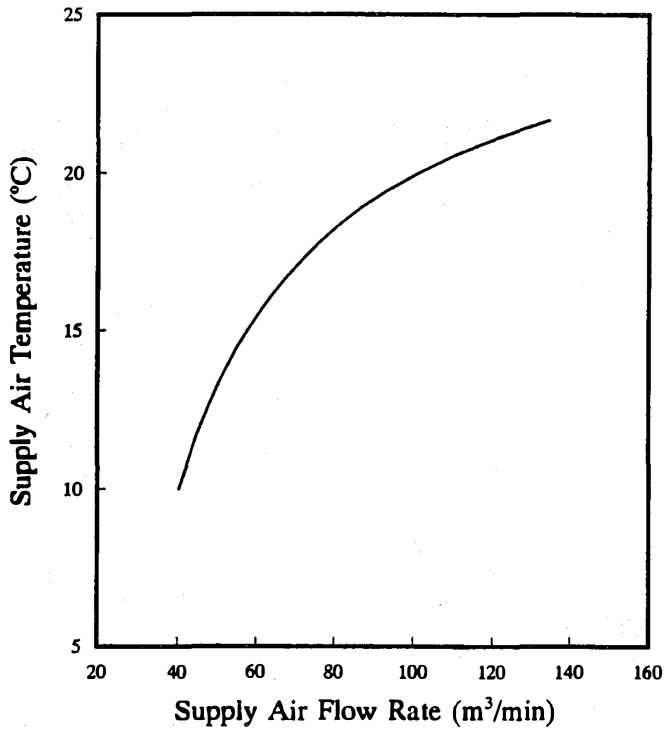


**Figure 4. Relation Between Supply Air Temperature and Supply Air Flow Rate**

estimated based on the individual performances to see whether the system can meet the required cooling load. The design/size of individual components was modified to satisfy the required load. We applied this approach for the three systemconfiguration and obtained three system designs. It should be noted that all three concepts have existing stationary versions operating in the processing industry or in building air conditioning applications. Here, we briefly discuss these three systems. More detailed descriptions and discussions of these individual systems can be found in Pesaran, et al. (1991).

ROTATING SOLID DESICCANT SYSTEM - Satisfactory cooling performance was achieved with no recirculation, a supply air flow rate of 82 $m^3$/min (2880), a delivery temperature of 18.4°C (65°F), a water consumption rate of 176 kg/h (80 lb/h) and a power consumption of about 3 kW (4 hp). In this configuration, a solid adsorbent material (lithium chloride or silica gel) is supported on a matrix structure that offers low pressure drop to the processed stream. The adsorbent structure and containing system are designed in such a way that the adsorbent continuously moves from the process air stream to the regenerating air stream and back. This mode of operation allows for compact packaging and a reduced inventory of desiccant. Disadvantages are tied to the mechanical complexity of rotating equipment and the seal requirements for isolation of the two flowing air streams. Figure 5 presents a schematic of the rotating solid desiccant system. Systems of this type are commercially available for building air conditioning systems. In this particular evaluation, lithium chloride was selected as the desiccant material, and system performance was determined for a range of air flow rates and recirculation ratios. The heat required for regeneration was about 58 kW (200,000 Btu/h) which could be easily supplied by the engine cooling system.

FIXED SOLID DESICCANT SYSTEM - Satisfactory cooling performance was achieved with no recirculation, a supply air flow rate of 100 $m^3$/min (3500), a delivery temperature of 20°C (68°F), a water consumption rate of 140 kg/h (63 lb/h), and a power consumption of about 3.7 kW (5 hp). The adsorbent matrix used in this preliminary design study is comparable to the one used in the rotating device described above except that it is stationary, and silica gel was used as the adsorbent material. The system described in Figure 6 presents two fixed adsorbent structures exposed to the air flow streams. Here, the process and regeneration streams get switched between fixed adsorbent matrices. Advantages include fewer moving parts, ease of separating flow streams, and ease of maintenance. Disadvantages are related to the need for a potentially larger inventory (two desiccant beds) and the size of an individual bed as dictated by allowable air velocity. Here also, the engine cooling system could provide more than adequate heat for adsorbent regeneration.

LIQUID DESICCANT SYSTEM - With this system design we obtained adequate cooling performance with a 50/50 blend of outside air and recirculated air, a supply air flow rate of 86 $m^3$/min (3025), a delivery temperature of 19°C (66.5°F), a water consumption rate of 374 kg/h (170 lb/hr), and a power consumption of about 2.4 kW (3 hp). In this approach, air to be dehumidified flows through an absorber in which a rich (concentrated) lithium bromide solution comes in contact with the moist air. Moisture passes from the air to the bromide solution. Before being recycled to the absorber, the diluted bromide solution is heated in a regeneration step. This process is shown schematically in Figure 7. An advantage of this system is the possibility for modular construction. Modules can be located in different parts of the vehicle and linked through liquid transfer lines. Other advantages include reduced volume of components and the potential for adjustable performance (cooling capacity) by changing the liquid desiccant flow rate. Disadvantages are corrosion, weight, and the need to control liquid sloshing and entrainment. The heat from engine cooling is sufficient to provide the heat needed for desiccant regeneration.

**DISCUSSION**

All three approaches studied can supply conditioned air to meet the temperature, humidity, and flow rate constraints imposed. Furthermore, system components and hardware can be designed that could satisfy the space limitations associated with the vehicle (see Pesaran et al., 1991 for details). The overall volume of the liquid desiccant system is about 2 to 2.5 times larger than existing V/C systems. The overall volume for the solid desiccant systems is about 3 to 4 times the V/C system. Figure 8 gives an idea of space filling and possible location of components of the various systems studied.

The most significant difference between the desiccant cooling systems and the V/C system lies in the amount of energy consumed to handle the cooling load. Even when corrected for the increase in weight (conservative values), the

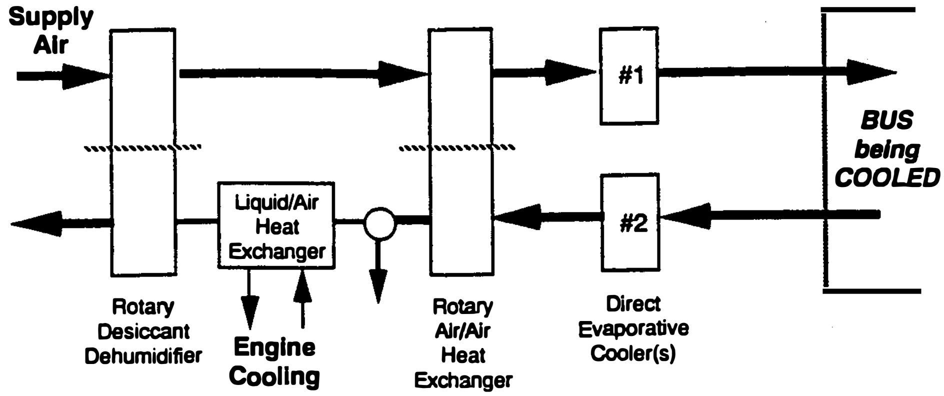


**Figure 5. Schematic of the Proposed Rotating Solid Desiccant System**

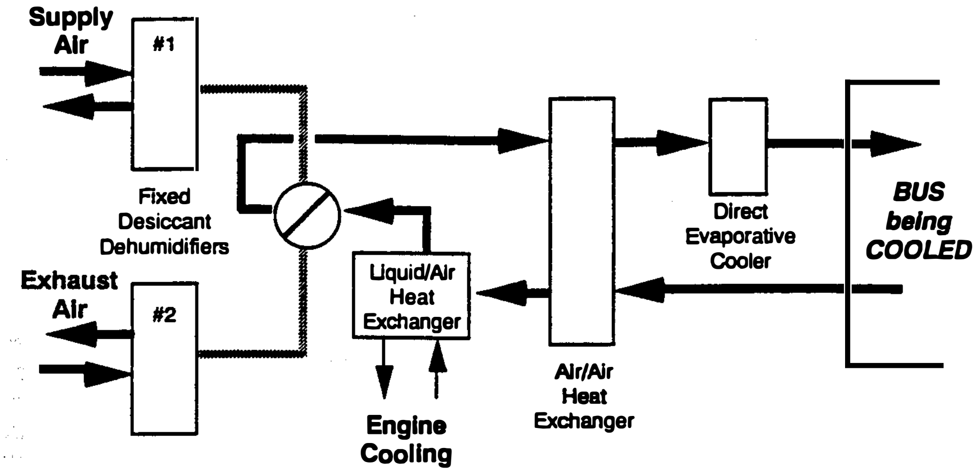


**Figure 6. Schematic of the Proposed Fixed Solid Desiccant System**

power consumption estimates are still far below the typical requirements for V/C systems. This is shown in Table 5, where the impact of the lower power requirement is translated into fuel economy between 2.3 and 3.5 L/h (0.6 to 0.93 gal/h), a fuel saving on the order of 70%.

In cities with 3000 h of cooling season such as Houston and Miami, this corresponds to savings on the order of 6800 to 10,230 L (1800 to 2700 gal) per bus per year. In cities with less demanding loads and schedules (Table 3), such as Washington, DC, and New York, the savings are still significant, on the order of 3790 to 5300 L (1000 to 1400 gal) per bus per year. Assuming that 50% of existing V/C air conditioning systems in buses were replaced with a desiccant-assisted evaporative cooling device, the total annual diesel fuel savings would be between 114 million and 303 million liters (30 million and 80 million gal).

The economic competitiveness of the desiccant cooling system over the V/C system is an important consideration for the end user considering the implementation of the new technology. The economics depend on capital (first) cost, annual operating hours, and annual fuel and maintenance costs. We estimated the simple payback periods for desiccant cooling systems for various scenarios. The simple payback period is defined as *capital cost differential* divided by *annual fuel and maintenance cost savings.* Assumptions considered include first cost of the desiccant system equal to 150% and 125% of a comparable V/C system ($10,000), maintenance cost equal or 50% less than the V/C system ($1,000/year), and operating times of 1500 and 3000 h/yr. Fuel cost is taken to be $0.264/L ($1/gal). The results are presented in Figure 9. As it can be seen from the analysis of the cases studied, the payback period ranges from less than 1 year to 2.5 years in Miami. This value can be reduced further by anticipated capital and maintenance cost reductions.

## CONCLUSIONS AND RECOMMENDATIONS

This study looked at the possibilities for using desiccant dehumidification as a pre-conditioner for an evaporative cooling system to be used in a vehicle such as a bus making use of waste heat from the engine for regeneration. The concept is found to be technically and economically feasible within the limitations imposed by engineering

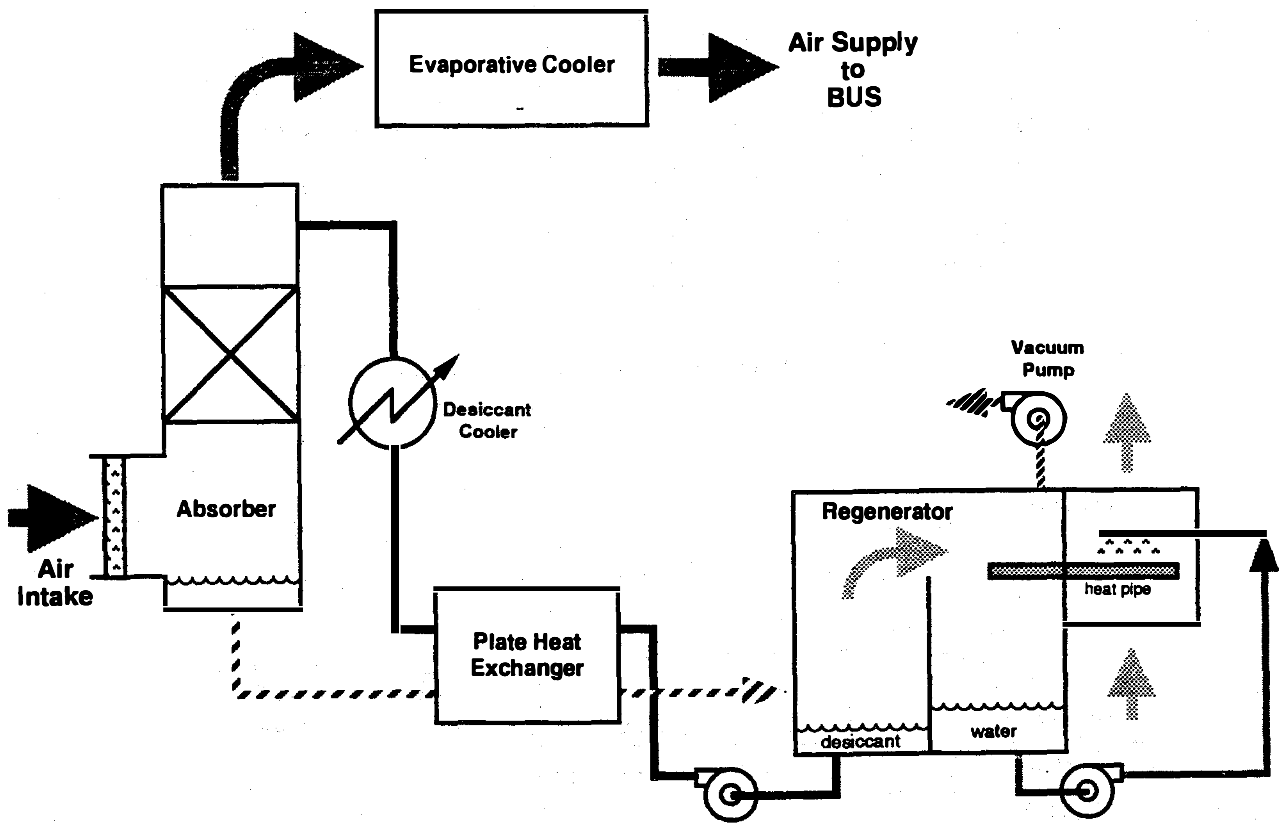


**Figure 7. Schematic of the proposed Liquid Desiccant System**

estimates. From the technical perspective, the three systems studied (rotating solid, fixed solid, and liquid desiccant) are all viable alternatives to the vapor compression system. In a first sweep estimate, all three systems show comparable cost factors and can be conceived to fit, within a reasonable amount of space, onto the vehicle. None of the desiccant systems use CFCs, and all offer significant (about 70%) fuel savings in operating the air conditioner.

To the level of engineering pursued in this report, all these desiccant configurations have advantages and disadvantages, which makes a rational selection of one system over another difficult. It is essential that a bench-scale investigation be carried out to fully assess the potential and value of these systems as the next step toward commercial viability. Full-scale prototypes must be fabricated and installed in buses and tested in a few different climates for a sufficient amount of time. Only then can we answer questions regarding costs, operation, performance, maintenance, electric and fuel consumption requirements, and the reliability of desiccant cooling systems for buses.

## ACKNOWLEDGMENTS

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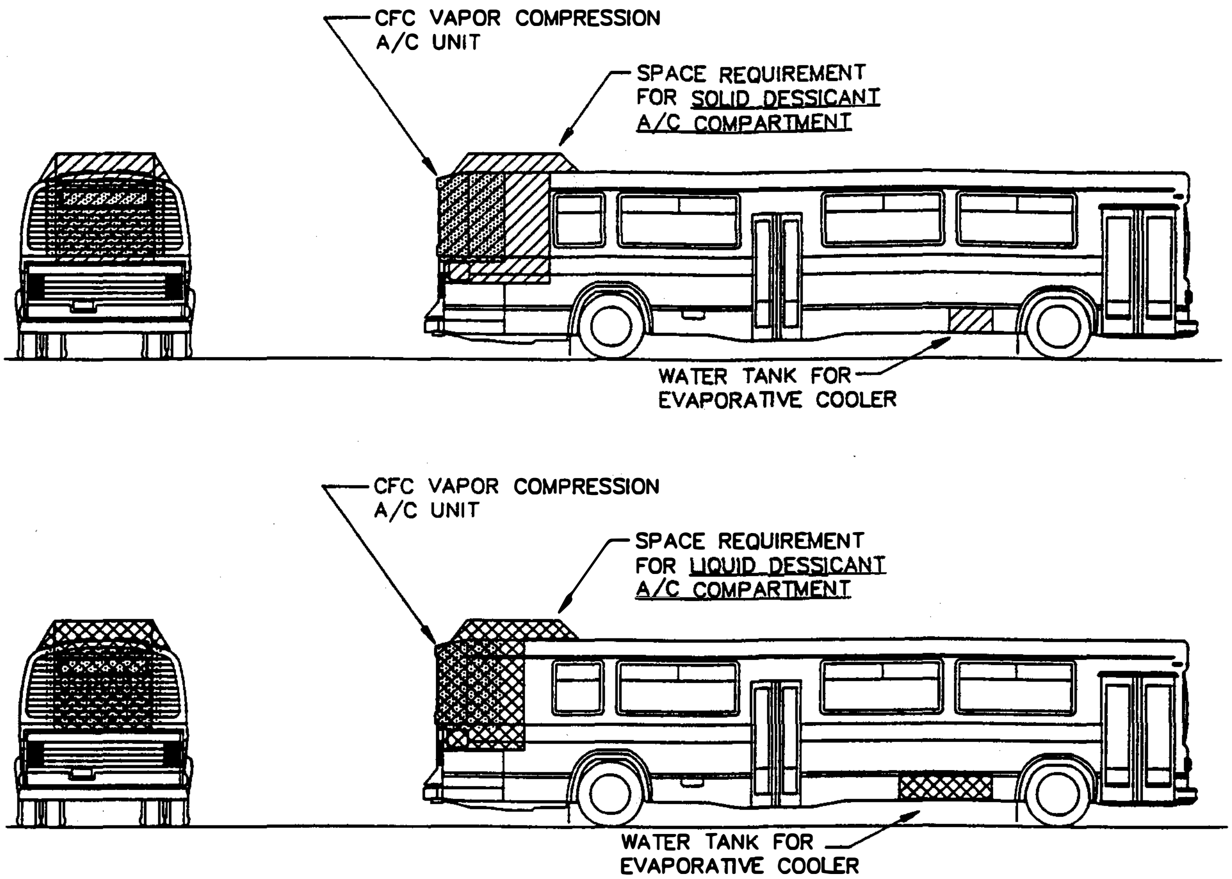


**Figure 8. Schematic Requirement in a Bus for Air Conditioning Systems**

**Table 5**
**Fuel Savings Potential of Desiccant Systems Relative to Vapor Compression**

| Desiccant System | Power Consumed (kW) | Fuel[1] for Power (L/h) | Weight[2] Differential (kg) | Fuel[3] for Weight diff. (L/h) | Total Fuel Consumption (L/h) | Fuel Savings (L/h) |
|---|---|---|---|---|---|---|
| Rotating Solid Desiccant | 3 | 0.95 | 590 | 0.30 | 1.25 | 2.50-3.45 |
| Fixed Solid Desiccant | 3.75 | 1.17 | 500 | 0.25 | 1.42 | 2.33-3.22 |
| Liquid Desiccant | 2.25 | 0.72 | 910 | 0.46 | 1.18 | 2.57-3.52 |
| **Vapor Compression** | 12 - 15 | 3.75 - 4.70 | 0 | 0 | 3.75 - 4.70 | 0 |

[1] Fuel consumption for a bus is estimated to be 0.315 L/kW/h (AiResearch, 1980).
[2] Weight of Vapor Compression system taken to be 320 kg.
[3] Each 300 kg of additional weight increases the fuel consumption by 0.15 L/h (UMTA, 1983).

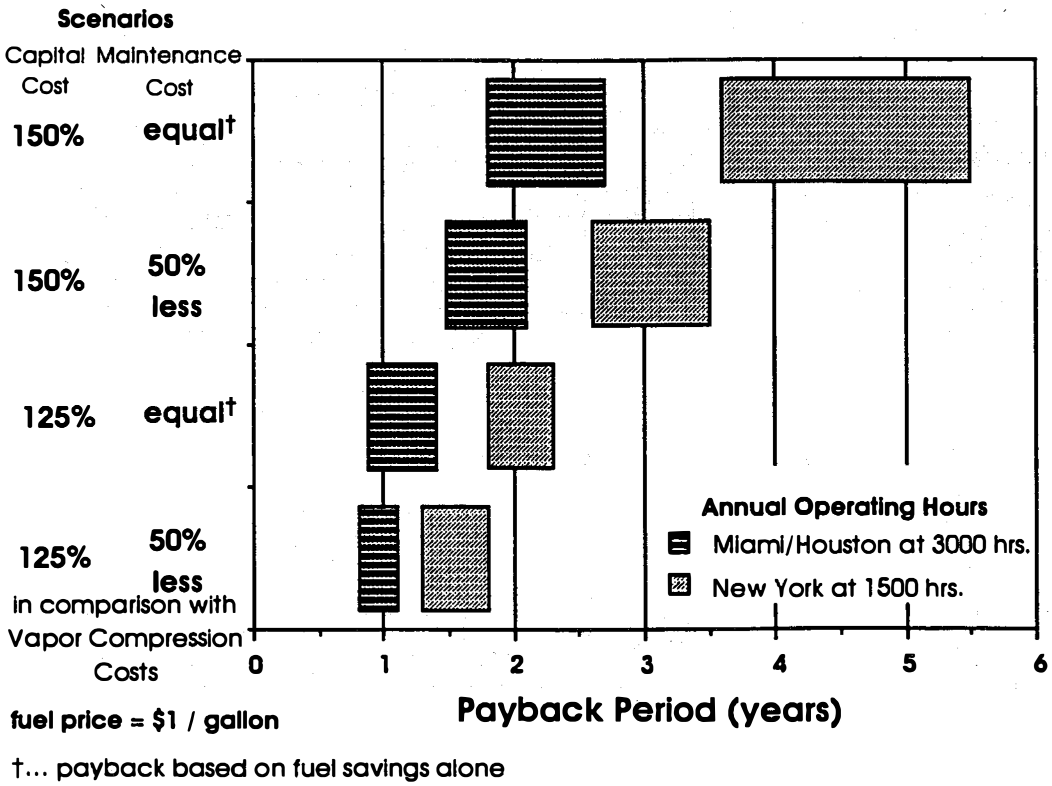


**Figure 9. Simple Payback Periods for Desiccant Cooling Systems**


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