Adsorption Air-Conditioning for Containerships and Vehicles

Final Report

Metrans Report 00-7

Craig Christy and Reza Toossi
California State University Long Beach

August 05, 2004

METRANS
Transportation Center
USC / CSULB

Disclaimer

Abstract

An investigation is undertaken into the feasibility of meeting the cooling needs for commercial tractor-trailer refrigeration and transit bus air conditioning (A/C) by utilizing their own exhaust heat to drive an adsorption refrigeration system.
An experimental vapor compression A/C system utilizing adsorption compression was refurbished and operated at CSULB to verify previously reported coefficient of performance (COP) and specific cooling power (SCP) values and to gain knowledge, experience, and insight into product design issues.

Industry reported cooling requirements for tractor-trailers and buses, the heat available from these vehicles' exhaust gases, and the experimental systems COP and SCP were reported and used as benchmarks to establish system requirements and design concepts for a novel system proposal. Design options were considered in relation to these benchmark values and evaluated based upon their packaging volume, weight, and their relative energy input requirements. The use of refrigerants R717 (ammonia) and R134a were explored and potential sorbent bed designs were extrapolated for each.
The expected reduction in emissions and fuel consumption from conventional refrigeration systems is discussed with respect to implementing the new adsorption compressor design.

The heat energy available from a large diesel engine's exhaust was found to be adequate to support an R717 adsorption system with reasonable size requirements and adequate cooling capacity to meet industry needs. In conclusion, design recommendations for an adsorption compressor cooling system applied to the transit industry are discussed. Utilizing R717 (ammonia) provides the best packaging design and satisfies the high end of typical cooling capacity requirements. An R134A systems performance is such that its overall size and heat rate required limit the systems practical cooling capacity to the realm of 30,000 to 60,000 BTU/hr. Potential for an R134A system to support the larger cooling requirements (60 to 120 kBTU/hr) are discussed utilizing a more complicated design layout.

Table of Contents

List of Figures

List of Tables

Disclosure

Project was funded in entirety under a contract to California Department of Transportation.

Acknowledgments

The authors would like to thank the United States Department of Transportation, California Department of Transportation, and METRANS for their interest and provision of grant support to make this project possible. We would also like to thank Roy Ganzer and team at Mechanic Refrigeration Company for their invaluable technical and operational support during the activation of the experimental system.

Phase II Deliverables

Task 1 Hardware Refurbishment. To prepare hardware to working conditions many components had to be either replaced or repaired. Specifically, the ammonia flow meter was repaired; all heat regenerative fluid flow control valves were rebuilt to remove contamination that had caused many valves to malfunction. A computer program using Labview was developed to provide this user-friendlier valve control method.

Task 2 - Experiments and Data Collection . Task 2 was completed with respect to operating the adsorption air-conditioning experimental unit to collect valuable performance data. Unfortunately, because of a refrigerant leak into the heat regenerative water system, only a limited number of experimental tests were performed. Development of practical system designs with details such as sizing, required heat input and cooling output was still possible using the best of data we had experimentally determined along with previously determined performance from other published studies.

Task 3 - Sorption System Characterization. This task involved a detailed system design, including size, weight, heat input, operational modes and layout configurations. An adsorption system with cooling capacity ranging from 30,000-120,000 Btu/hr and capable of maintaining temperatures in the range of 0^{o}F to 40^{o}F (for trucks with refrigerated containers) and 67^{o}F (for bus A/C systems) was designed.

Task 4 - Design of A/C systems applicable to transportation and containership units

In this task the application of the sorption refrigeration system in design of a practical A/C system for a refrigerated trailer and a bus air conditioner was completed. The system is also shown to be applicable to automotives, although detail designs were outside the scope of present work.

Nomenclature

Most of the nomenclature is defined as it is introduced or else is obvious from the context of its use. However, it is summarized here for convenience.

A/C	Air Conditioning
APU	Auxiliary Power Unit
ASW	Adsorption System Weight
Aux	Auxiliary
CO	Carbon monoxide
Comp	Compressor
COP	Coefficient of Performance
CSW	Conventional System Weight
C_p	Specific Heat at constant pressure
DCL	Dual Carbon Layer (bed cross sectional design option)
gal	Gallon
GPM	Gallon per minute
h	Convective heat transfer coefficient
h.p.	Horse power
$\dot{m}_w$	Mass flow rate of water
MEOP	Maximum Expected Operating Pressure
Mech	Mechanical
R	Specific Gas Constant
R134A	Tetrafluoroethane Refrigerant (chemical formula, CH_2 FCF_3)
R717	Ammonia refrigerant (chemical formula, NH_3)
Ref	Refrigerant
ROM	Rough Order of Magnitude
SCFM	Standard cubic feet per minute
SCL	Single Carbon Layer (bed cross sectional design option)
SPC	Specific Cooling Power
ΔT	Change in temperature
Q	BTU/hr heat rate
qty	Quantity

Introduction

Adsorption refrigeration systems have enjoyed limited use as research studies for advanced cryogenic coolers onboard surveillance satellites (NASA JPL Brilliant Eyes Ten K Sorption Cryo Cooler Experiment “BETSCE”), and recently, a commercial icemaker for diesel fishing vessels have been developed. However, this scarcely used technology could be extended to many other settings. Since the adsorption compressor is thermally driven, any number of acceptable energy sources such as electric heaters, solar energy, waste heat from exhaust, regenerative braking or engine coolant can provide power. This flexibility in input energy sources makes the system attractive when exploring new potentials.

Tractor-trailer refrigeration and bus A/C systems are especially attractive potential applications for this technology. The common automobile or bus air conditioner uses the vehicle’s engine to turn a mechanical compressor. Operating a mechanical compressor can consume as much as 10% of the engine’s output; and increase the engine load, operating temperatures, fuel consumption, and emissions production. An adsorption compressor could use thermal compression powered by the “free energy” of exhaust heat and not affect the vehicle’s engine performance. Since moving parts in an adsorption system are limited to valves; it is considerably simpler, requires no lubrication, and thus little maintenance. Other advantages include quiet operation, energy efficiency and potential modularity of the system to adjust the heating and cooling capacity by the addition or subtraction of sorbent beds.

The system takes advantage of the ability of activated carbon (sorbent material) or “charcoal” to adsorb a relatively large quantity of refrigerant vapor (sorbate) at a low temperature and pressure and desorb the refrigerant at a higher temperature and pressure. The compressor effect is generated by cyclically heating and cooling the sorbent material and refrigerant resulting in high pressure outward flow or refrigerant release during the hot desorption phase, and inward flow or low pressure suction during the cold adsorption phase.

The sorbent material is stored in multiple canisters (sorbent beds) that maintain a continuous flow of compressed refrigerant to the system by temperature cycling them in a staggered fashion. While the hot beds are releasing refrigerant vapor, the others are in different stages of cooling and adsorbing new refrigerant or being heated to generate refrigerant flow as their pressure increases. Each sorbent bed canister has two dedicated passive one-way check valves utilized to control refrigerant flow directions. The one valve allows low-pressure refrigerant flow into the bed from the evaporator and does not allow a hot bed’s higher pressure flow back into the evaporator. The other check valve allows hot beds’ high pressure flow out of the bed towards the condenser, but doesn’t allow the system’s high pressure to flow back into a colder, low pressure bed. The basic adsorption system hardware and operation was reported earlier.[1]

Principles of Adsorption

The ability of porous solids to reversibly adsorb large volumes of vapor was recognized in the eighteenth century by Scheele and Fontana[1.1]. The phenomenon of a solid surface (sorbent) absorbing a gas or liquid substrate (sorbate) is known as adsorption, which is distinguished from absorption. Adsorption refers to the binding of molecules (sorbate) to the surface of an inert material (sorbent) without any chemical change, whereas absorption refers to the uptake of molecules into the interior of another substance (suck up into as by capillary, osmotic, solvent or chemical action). Adsorption is better described as a physical process at the surface as opposed to the take up or in by molecular or chemical action such as absorption. Adsorption occurs because the atoms, molecules or ions at the surface of the sorbent are extremely reactive with unfulfilled valence requirements as compared to their counterparts in the interior, which have valence requirements satisfied. The unused bonding capability of surface atoms may be utilized to bond molecules of the sorbate to the surface of the sorbent. When a sorbent is a solid, such as carbon, it has the characteristics of, and is known as, a clathrate material. A clathrate is an organic compound that has 3-dimentional lattices that make up a network of micro pores or individual sites for the sorbate to reside. The sorbate, while attached to the sorbent surface, gets trapped in cavities of the sorbents' cage like crystals. Once a sorbate molecule is in a micro pore or site of the sorbent, it is somewhat concealed from other sorbate molecules; thus reducing or eliminating intermolecular forces (Van der Wals forces) between sorbate molecules and can result in a closer packaging of the sorbate molecules.

Some combinations of sorbate and sorbent material have high compatibilities with each other and reach high concentrations at saturation. Each combination has its optimum operating temperature and pressure regions. The adsorption capacity is a function of the certain characteristics of sorbate and sorbent such as, sorbent porosity, sorbate boiling point, and operational temperature and pressure. Maximizing the adsorption capacity is optimal as it allows for smaller packaging sizes for a given cooling capacity. Activated carbon is an excellent sorbent material because tarry carbonization products have been removed from the pores - resulting in a large surface area for interaction throughout the material. Conventional refrigerants such as R-134A and R-717 (ammonia) are readily available and perform in the desired temperature and pressure regions when used in an activated carbon adsorption system.

Application

Much of an internal combustion engine's heat from combustion is discarded out of the exhaust or carried away via the engine cooling water. All this wasted energy could be useful. The common automobile, truck or bus air conditioner uses shaft work of the engine to turn a mechanical compressor. Operating the mechanical compressor increases the load on the engine and therefore increases fuel consumption, emissions and engine operating temperature. With an adsorption compression system, we can utilize the exhaust heat and the heat absorbed by the engine's cooling water. This heat, which could be considered as free energy, would be enough to drive an adsorption refrigeration system. A diesel truck freightliner, which carries produce needing refrigeration, or a bus air conditioning system, could employ the adsorption refrigeration system, which would run from the heat of exhaust to cool the cargo.

Environmental Impact

One of the main objectives of implementing an adsorption compression refrigeration system on large refrigerated truck trailers and for air-conditioning (A/C) on large buses is to help reduce pollution emission into the atmosphere. The typical refrigeration system used on large refrigerated trailers and for A/C on large buses is a diesel powered vapor-compression refrigeration system. The standard and conventional refrigeration systems for these applications utilize a mechanical compressor that receives its input work, or shaft power, from a diesel engine's work output. In buses and light trucks, the compressor usually runs directly off the main engine, whereas in refrigerated trailers, an auxiliary diesel engine is dedicated to the refrigeration system. In either case, the A/C or refrigeration systems expend diesel fuel and add emissions to the atmosphere. We now attempt to quantify potential reductions of emissions if a large portion of the bus A/C and refrigerated trailer systems utilize the waste heat of the bus or truck main engine exhaust. The waste heat would be used as input power for an adsorption compression refrigeration system.

Diesel driven compressor data has been obtained from two leading transit refrigeration system companies, Carrier and Thermo King. Reviewing a large amount of manufacturer data showed that compressors typically require 25-35 h.p. to operate. The larger bus engines tend to be more efficient than the smaller auxiliary engine. Data on the pounds of diesel fuel consumed per hp-hr indicates smaller bore diesel engines consume 0.38 lbs/hp -hr and the larger bore engines consume 0.32 lbs/hp-hr.[7] For a 34 h.p. refrigeration compressor, this means that approximately 11-13 lbs of diesel fuel per hour of operation is being consumed. Wabash National Corporation, [8] a leading manufacturer of refrigerated trailers, reports that on average, refrigerator trailers operate between 1500 to 1700 hours a year and consume fuel at rate of approximately 12 lb/hr, in agreement with the previously determined values for 34 h.p. diesel engine operations. Therefore, the typical refrigerator trailer consumes from 9000 to 10200 pounds per year of diesel fuel.

Obert[7] gives a typical diesel-exhaust-gas analysis, which has determined the exhaust gas volume flow rate and carbon monoxide (CO) percent by volume specifically for a 26 h.p. diesel operating at 1400 RPM and consuming 12.5 lbs fuel per hour. The dry exhaust gas volume flow rate of 4050 scfh (standard cubic foot per hour) and a CO percent by volume of 0.027 is reported. The CO volume flow is determined as 1.1 scf/hr. Using standard conditions of 60^{o}F and 14.7 psia, and assuming ideal gas, the CO mass flow is 0.08 lbm/hr. From above, a typical refrigerated trailer is utilized 1600 hrs a year in which the refrigeration unit's 'ON' time is approximately 800 hrs per year, a 50 % duty cycle. The 800 hrs of use will produce 0.08 lbm/hr x 800 hrs = 64 lbs of CO. Therefore, one refrigerated trailer could possibly contribute 64 lbs of CO emissions to the atmosphere every year.

We will consider two different orders of refrigerated trailer activities; 10,000 and 100,000 units considered on the road. Assuming 10,000 trailer units, the potential reduction in CO is 640,000 lbs per year. For 100,000 units this amounts to 6,400,000 lbs of CO per year.

The data obtained from the California Air Resources Board gives the tons per day of various emissions from various sources. The data related to Heavy / Heavy Duty Diesel Trucks CO levels emitted were 95.6 tons/day for 746 trucks. Comparing this value to the above-determined refrigerated trailer's values is tabulated below considering reefer truck quantities at both 30 percent and 80 percent of the 746 trucks.

Table 1. Comparison of Emissions from Refrigerated Trailers and Heavy Duty Diesel Trucks

Assumed % of 746 Trucks Operating Reefer Trailers in Calif.	Reefer Trailer CO output per day in Tons	Calif. state wide CO emitted in tons for 746 trucks	Potential % reduction in CO emission
30% (224 units)	0.020	95.6	0.02 %
80% (597 units)	0.052	95.6	0.05 %

The above estimates show that implementing an adsorption refrigeration system on refrigerated trailers can reduce exhaust emissions, but the reduction is still a very small fraction of the total. A similar analysis performed on A/C systems used in large buses indicates significant savings. As stated in the performance parameters section of this report, buses typically have a larger cooling load than reefer trailers, including a higher duty cycle (75-100 percent).

Assuming that it takes an average of four days for a truckload of goods to be transported between New York to LA, approximately 100 gallons additional diesel fuel will be used to meet the refrigeration needs of the trailer. This equates to over 700 lbs of diesel fuel into exhaust and 0.67 pound of CO emission per truck per transcontinental trip. If we assume there are at least 10,000 refrigerated trucks on the US roads, this equates to 6300 lbs of CO emission, or 1575 lbs per day, in line with the 1753 lbs mentioned above.

Besides improving air quality, the new system saves a considerable amount of precious natural resources. If the adsorption system was totally implemented into buses and refrigerated trailers, hundreds of thousands of pounds of diesel fuel per year could be saved.

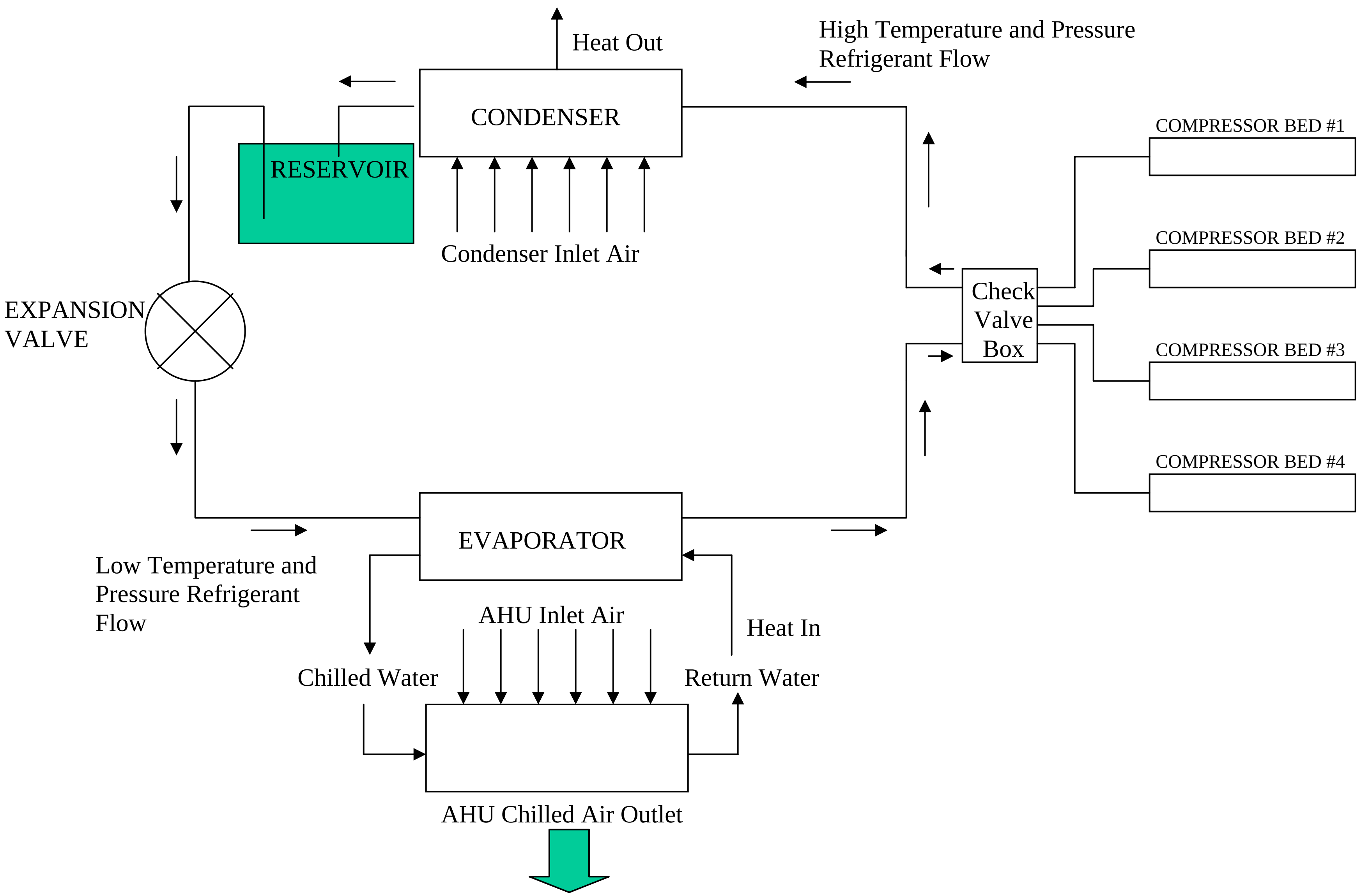
Experimental Data and Analysis

Figure 1 is a simple layout of the experimental adsorption A/C system. For simplicity, the sorbent compressor's complex multi-valve 'heating and cooling water' heat regenerative system is not shown. The Phase I report[1] of this research effort covers the heat regenerative system in detail. Heat regeneration utilizes a working fluid to heat and cool the sorbent beds and to utilize the heat of a hot bed to warm a colder bed. The A/C system consists of a standard ammonia R-717 vapor-compression refrigeration circuit and a chilled water circuit. The ammonia refrigerant circuit and chilled water circuit are coupled via the systems evaporator heat exchanger. The ‘chilled water circuit’ is chilled by the ammonia refrigerant in the evaporator. Heat is transferred from the surrounding air to the chilled water circuit via an air-handling unit. A 7 GPM pump is used to circulate the chilled water in this circuit. The four sorbent beds of the adsorption compressor are connected to the system through the ‘check valve box’.

Experimental Test Runs

Many experimental runs were performed over the course of this project. Unfortunately, because of an ammonia leak, only test data from two runs were considered to be entirely reliable. These two runs do have similar outcomes and the data does compare fairly well with previously documented predictions and tests.

Figure 1. System Schematic



Results

Table 2 below tabulates pertinent experimental data and associated performance of the two runs. The research team realizes that the two experimental runs are not sufficient for developing an optimal system. The optimum design requires collecting data over a wide range of operating temperatures, airflows, sorbent bed temperatures, and cycling times.

Table 2. Recorded Adsorption Compression System Test Data

System parameters	Run#1 7-20-01	Run#2 5-29-02	Comments
Evaporator Chilled Water Inlet Temperature	77.6 F	83.2 F	7 GPM water flow rate (60.14 lb/min)
Evaporator Chilled Water Outlet Temperature	73.6 F	80.1 F	7 GPM water flow rate (60.14 lb/min)
Chilled Water Temperature Change	4 F	3.1 F	
Cooling Capacity	13,568 BTU/hr	10,515 BTU/hr	1.13 Tons on 7-20-01 0.88 Tons on 5-29-02
5.3 Kw Heater Duty Cycle	40% on time	50% on time	5.3kW = 18,089 BTU/hr
Heater Power Input	7236 BTU/hr	9044 BTU/hr	
COP (cooling BTU/hr per BTU/hr Power Input)	1.87	1.16	Unexpectedly high
SCP (BTU/hr per lb sorbent)	950.8	736.9	14.27 lb carbon total

Average maximum temperatures were in the range of 180^{o}F, which is approximately 60^{o}F greater than the Phase I experimental activities. The water heater was cycled 'ON' and 'OFF' with approximately a 40 to 50 percent duty cycle to avoid overheating (maintaining less than 200°F) the recirculation water that is both supplying and regenerating heat between the sorbent beds. The measured wattage input with the electric heater 'ON' was 5.3 kilowatts, per a power meter's indication. The total heater power rate is determined by multiplying the heater output rate by the heater duty cycle. This heater input value is used for the System Coefficient of Performance evaluation.

The following is a sample analysis of the tabulated performance values of the systems above using the experimental data collected in Run#1.

Cooling Power Output of the Evaporator

A specific heat value of 0.94 BTU/lb/F is used for the 20% glycol – 80% water mix (chilled water circuit) applying the heat load to the evaporator. We use $Q=Mdot*C_p*(T_{in}-T_{out})$ to determine the cooling capacity of the system. T_{in} is the inlet temperature of the evaporators' water inflow and T_{out} is the outlet temperature of the evaporators' water outflow.

The evaporator 7 GPM glycol/water volumetric flow is converted to mass flow (20/80 mix S.G.=1.03 at 50F)

$\dot{m}_w$ = 60.14 lbs/minute.
$\Delta T = (T_{in} - T_{out}) = 77.6 - 73.6 = 4$ F (increased over 3.7 F during phase I)

$Q = \dot{m}_w * C_p * \Delta T = 60.14 lb/min * 0.94 BTU/lb/F * 4 F * 60 min/hr = 13{,}568$ BTU/hr

This value is used to determine the system's overall COP.

Adsorption Compression A/C Coefficient of Performance (COP)

The ratio of the evaporator refrigeration capacity and the sorbent beds heater power input determine the overall COP of the system. The refrigeration capacity has been determined in the previous to be 13,568 BTU/hr.

Total refrigeration in tons is

T= (13,568 BTU/hr)/ (12000BTU/hr/ton) = 1.13 tons of refrigeration

The approximate 1 ton value from this experimental adsorption compression system is a reasonable value since this system was originally designed for 1.25 tons, per initial design predictions. The originator of this system also observed actual runs in the order of 1 ton (or slightly less) with sorbent bed temperature extremes of 100^{o}F on the cold side and 400^{o}F on the hot. The experimental runs of this same system at CSULB created sorbent bed temperature extremes from a low of 80°F and a high of 150-200^{o}F. Safety issues with the relatively old system and the hazards of the scalding regenerative water prevented testing at higher temperatures.

The 5.3 kW electric power input to the sorbent bed's heating water (regenerative heating water) was applied with an average duty cycle of 50% resulting in an electric input power of 9044 BTU/hr (after conversion from measured kW units to BTU/hr).
The system COP is as follows, without consideration of power input to the regenerative water pump or condenser and air handling unit blowers.

System COP = (13,568 BTU/hr) / (9044 BTU/hr) = 1.5

The relatively high COP value can be attributed in part to the low environmental parasitic heat losses associated with the sorbent beds' relatively low upper temperature. It is also possible that it is in part due to human error during manual recording of heater "ON/OFF" times and/or data instrumentation calibration errors. As reported by Jones (5), the ideal COP for an ammonia/carbon adsorption system is 1.46, a COP of 1.0 has been predicted for a four bed regenerative system, and a COP as low as 0.7 has been accepted for adsorption systems without regenerative heating.

In the above analysis, heat loss in the regenerative water prior to getting to the beds is neglected. The regenerative heating water circuit has 20 bulky solenoid valves and a maze of over 40 feet of plumbing, which also have heat losses and in part, get temperature cycled. A more accurate evaluation of the adsorption compression cycle performance is to evaluate evaporator-cooling load with respect to the sum of actual heat input to each of the four sorbent bed compressors (versus heat into the heat regenerative water).

Experimental System Specific Cooling Power (SCP)

The SCP is determined by the systems overall cooling capacity per mass of sorbent material. It has been determined (See Appendix A-2) that the experimental system sorbent compressor has 14.3 lbs of sorbent material. The experimental run #1 data indicates a cooling capacity of 13,568 BTU/hr. The SCP is determined by simply dividing the cooling capacity by the sorbent material mass, which results in 951 BTU/hr / lb sorbent material. This value does agree well with previously published SCP predictions and test data, and therefore this value will be used in support of the following design efforts.

.

Mass of Refrigerant Adsorbed and Desorbed

Figure 2 is an isotherm plot for ammonia (sorbate) and carbon (sorbent) which is the sorbate and sorbent used in this experiment's compressor. The plots of Figure 3 are of an isotherm for R134A on carbon. An isotherm plot is a constant temperature curve that provides the ultimate mass of sorbate that can be adsorbed per mass of sorbent material as a function of sorbate pressure.

Figure 2. R717 Ammonia – Carbon Isotherm (curves of constant temperature)

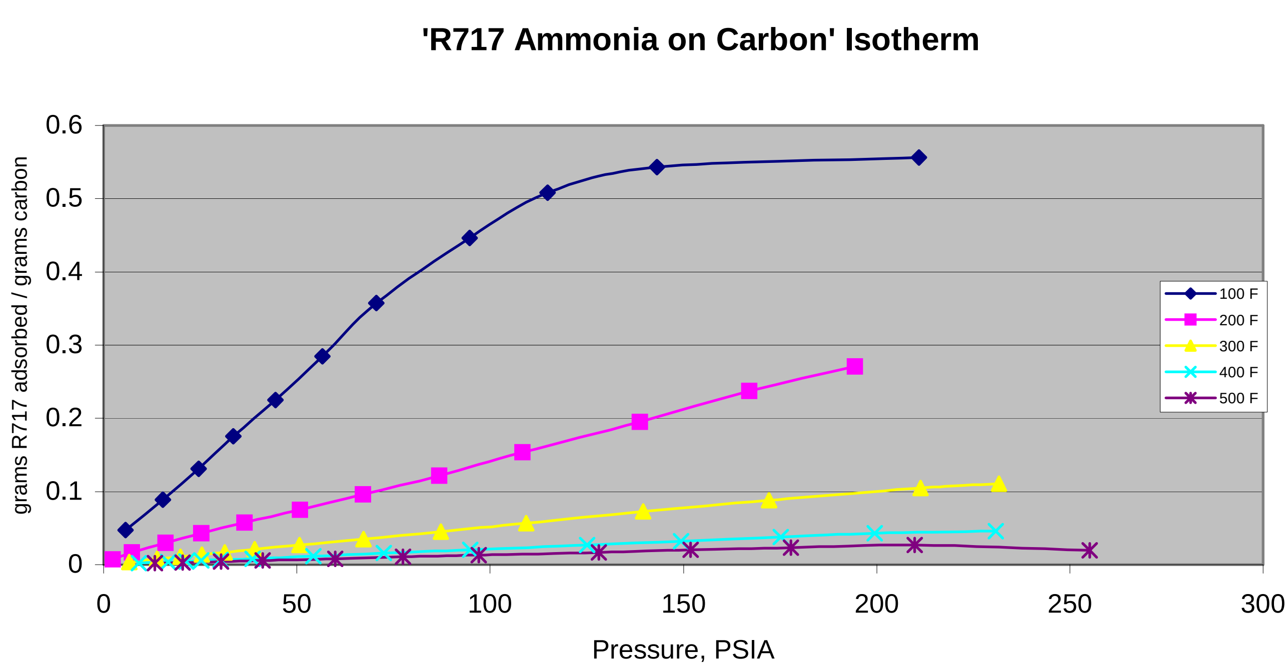
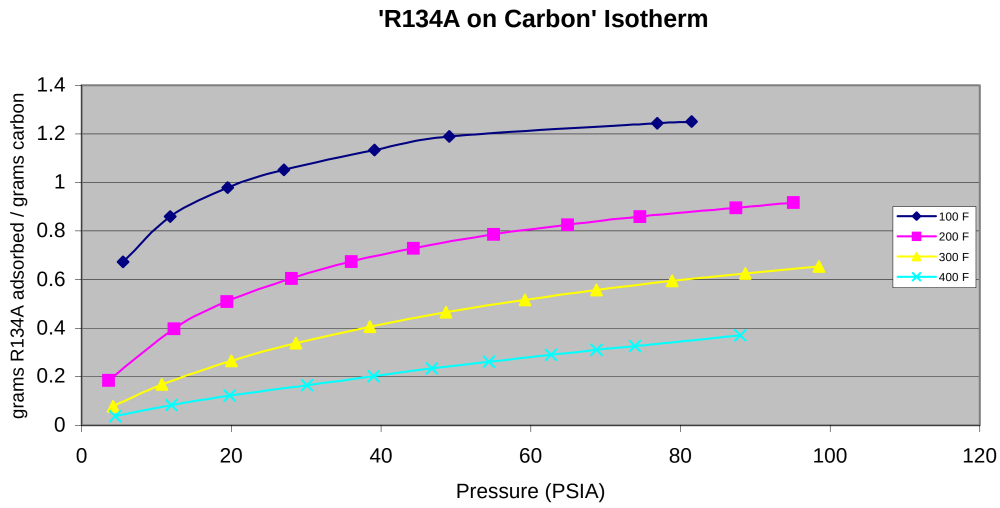


Figure 3. R134A – Carbon Isotherm (curves of constant temperature)



The isotherm data considers a totally saturated condition. Total saturation is time dependant. The time required to reach total saturation depends on the refrigerant's ability to diffuse into the sorbent material (mass transfer phenomena). The mass transfer rate depends on the sorbent's porosity and its relative exposed surface area per total mass. During a short temperature cycle, the sorbent material will most likely not reach a high percent of saturation as compared to a longer time at a single temperature and pressure condition. The adsorption phenomena is such that a dry, or lean, sorbent material will uptake the sorbate faster than if it is relatively more saturated, or richer. The same is true for chemisorption systems (such as ammonia – water absorption). This brings up two points of interest. If the sorbent's temperature cycle time is increased too much in an attempt to adsorb more refrigerant, the slowing of the adsorption mass transfer process, as relative saturation is increased, will also decrease the net refrigerant flow rate. This, in turn, lowers the systems Specific Cooling Power, SCP (evaporator cooling power per sorbent mass). The second point of interest is that the SCP will also decrease if the length of the temperature cycle is too short. Again, this is because the refrigerant flow rates will decrease as mass transfer (smaller percent of saturation) and heat transfer into the sorbent are limited in time. So, each system has an optimum cycle time to adsorb enough to produce adequate flow rates, but not stall out and waste time. It is desired for the SCP to be optimized as it relates directly to the unique configuration of the sorbent compressor. Increasing the temperature extremes of the bed can increase system performance and SCP to a point. A higher maximum and lower minimum temperature can increase the heat transfer rate and, to a point, reduce the mass transfer rate and cycle time. Once again, returns diminish if sorbent temperature is too high, resulting in greater heat losses to ambient The desire to shorten cycle time on the behalf of heat transfer and sorbent temperature doesn't always, but can, result in optimizing the adsorption mass transfer mechanism.

Jones[5] has performed experiments with both ammonia and R134A on a sorbent bed with 1.1 lb of carbon. The temperature cycling was from 100°F to 400°F with three-minute hot and three-minute cold cycle lengths. It was determined that the ammonia gave a cooling capacity of 1038 BTU/hr and the R134A gave 337 Btu/hr. This data indicates that three times the carbon quantity is required for a R134A adsorption system to perform as well as the ammonia adsorption system. The refrigerant performance difference is two fold. R134A does have a higher boiling point than ammonia (R717) resulting in better uptake (adsorbed sorbate mass) by the carbon. By evaluating the two isotherms, we know the R134A uptake exceeds R717 by a factor of 3. However, R134A requires 6 times the mass flow rate of R717 to yield comparable performance or cooling capacity (reference Refrigerant Flow Vs Cooling Capacity plot in the Performance Parameter section). The result is just as reported by Jones in that an R134A system requires approximately 3 times the carbon mass as an R717 adsorption system. R134A needs 6 times the mass flow to perform as well as R717. However, because the R134A uptake is 3 times that of R717, R134A can achieve similar performance using only twice the carbon of that required by R717.

Using the above isotherm plots and the previously determined required refrigerant flow to produce the reported cooling capacities; the actual uptake ratio is determined to be between 0.4 and 0.5 (uptake ratio is actual sorbate mass adsorbed/desorbed compared to the maximum value of sorbate adsorbed/desorbed per the isotherm). For the 1.1 pound carbon test, the R134A refrigerant flow rate required to maintain 337 BTU/hr cooling is 0.08 lb/min and per the isotherm; the amount of R134A adsorbed at 400°F and 140 psia is 0.48lb/lb carbon, and the amount adsorbed at 80°F and 48 psia is 1.19 lb/lb carbon. This difference over a three-minute period gives approximately 0.2 lb/min flow. The 0.08 lb/min refrigerant flow compared to the 0.20 lb/min maximum capability of the bed gives the 0.4 uptake ratio. A similar evaluation for the R717 also gives a 0.4 to 0.5 uptake ratio.

Experimental Summary

The experimental run of the A/C unit with adsorption compression did provide valuable data. The initial tasks completed on the hardware and software at the start of phase II improved on the Phase I efforts such that experimental operations involving solenoid valve control and sorbent bed data collection were more efficient. The results of an overall COP of 1.5 is an exceptional value and is more than double the COP of 0.68 from the Phase I runs.

In comparing the experimental outcomes for this project (in which we had relatively low maximum sorbent bed temperatures) with other published experimental data, it may be concluded that practical COP values to be used for system design are in the range of 0.6 to 1 (closer to 1 with heat regeneration), specific cooling power values for ammonia systems are 950 BTU/hr per pound of carbon and that of an R134A system is 306 BTU/hr per pound of carbon. An R134A adsorption system requires three times the carbon mass for similar performance to that of an ammonia adsorption system. And using the isotherms as the maximum uptake performance at saturation, the actual uptake ratio is in the range of 0.4 to 0.5.

Cooling Requirements

Based on data provided by major A/C and refrigeration manufacturers (primarily Thermo King Corporation and Carrier Corporation), large refrigerated freight truck container cooling capabilities are in the range of 46,000 to 64,000 Btu/hr, maintaining 35^{o}F with 100^{o}F ambient temperature. Smaller to medium trucks' refrigerated containers require about 20,000 to 30,000 Btu/hr. Urban transit buses utilizing an air conditioning system have higher output requirements than that of large diesel truck refrigerated trailers. Large transit buses and coaches have a cooling output in the range of 75,000 to 120,000 Btu/hr. Large city buses require 82,000 to 113,000 Btu/hr of cooling. Small to mid-size buses need cooling in the order of 50,000 Btu/hr. Typical automotive A/C systems utilize approximately a 1 ton of refrigeration (12,000 Btu/hr). The above data is tabulated below for ease of reference.

Table 3. Refrigeration Requirements for Trailers and Buses in Btu/hr (95^{o}F ambient temperature)

Large Truck Refrigerated Trailer	Mid-Size to Small Truck Container	Large Transit Bus	Large City Bus	Small Bus
46,000 to 64,000	20,000 to 30,000	75,000 to 120,000	82,000 to 113,000	50,000

Specific Cooling Power

Specific Cooling Power (SCP) is a measure of the evaporator-cooling load per mass of sorbent material. An analytically predicted SCP can be used for the initial sizing exercises of an adsorption compressor in order to meet specific refrigeration requirements. As discussed earlier in the experimental section of this report, the values from both this effort and from tests by Jack Jones[5] are compared; resulting in the acceptance of the R717 systems' 944 BTU/hr per pound of carbon, and R134A systems' 306 BTU/hr per pound of carbon, to be used during the compressor design and sizing effort.

Increasing cycle time is important in order to raise the SCP, in turn, decreasing the compressor size. Increasing the maximum and decreasing the minimum bed temperatures will increase heat transfer rate into the bed and can reduce cycle time. As the temperature gets too great and cycling times decrease, the available time for the adsorption and desorption process of refrigerant diffusing into and out of the sorbent material matrix is reduced. The proper bed heating and cooling operations depends on many variables, such as rate of heat and mass transfers and sorbent porosity.

Adsorption Compressor

The following lists sorbent mass and heat input required based on SCP, COP, and packaged carbon sorbent densities (the ratio of carbon mass to volume occupied after carbon is packaged in the bed integral with an 8% aluminum foam structure). The plot of Figure 4 is based on the SCP values for R717 and R134A on carbon. The plot is a quick reference to determine the sorbent mass in a compressor for a desired cooling capacity.

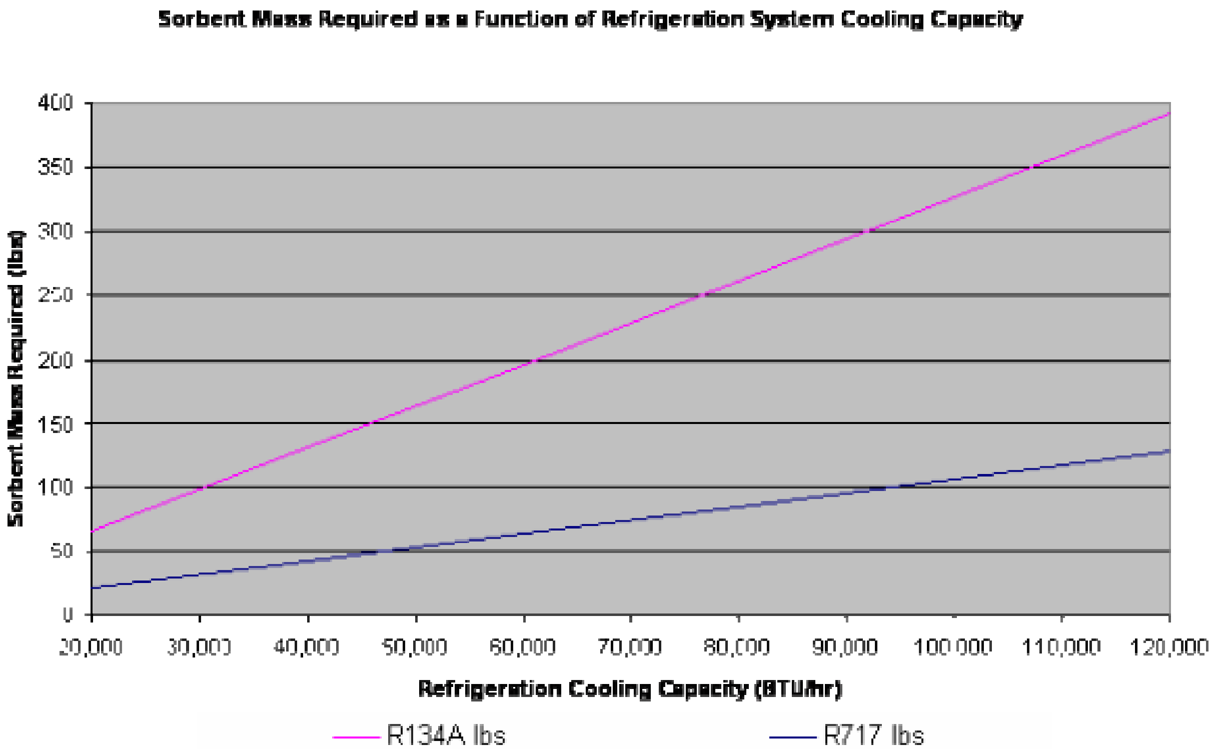
R717 Specific Cooling Power, SCP = 944 BTU/hr/lb sorbent
R134A Specific Cooling Power, SCP = 306 BTU/hr/lb sorbent
(ref. Performance Requirements, SCP)

COP considered for calculated heat input to R717 sorbent bed = 1

Density of Sorbent fully packaged/integrated into the 8% aluminum foam structure = 0.016 lb/cuin (ref. appendix)

Requirements for:	R717 Ammonia System Compressor			R134A System Compressor	
Refrigeration Cooling Requirement BTU/hr	Required Sorbent Mass R717 lbs	Sorbent Bed Required Heating Power BTU/hr	Volume Occupied by Packaged Sorbent cu ft	Required Sorbent Mass R134A lbs	Volume Occupied by Packaged Sorbent cu ft
20,000	21	20000	0.8	65	2.4
40,000	42	40000	1.5	131	4.7
60,000	64	60000	2.3	196	7.1
80,000	85	80000	3.1	261	9.5
100,000	106	100000	3.8	327	11.8
120,000	127	120000	4.6	392	14.2

Figure 4. Sorbent Mass Required as a Function of Refrigeration System Cooling Capacity for R717 and R134A



Note that the mass and volume information is for the sorbent material alone and does not include the mass and volume of the canister structure (packaging design). The remaining volume and mass of the canister structure design is considered in the following "System Design" section. The required heat listed above, considering a COP of 1, is in the realm of available exhaust heat from a large diesel engine, tabulated in Table 5 of the following section.

The sorbent beds' two main design requirements and constrains are as follows. Provide proper sorbent material quantity per information from the above graph and provide adequate maximum and minimum temperature extremes. Temperature swings of the bed should be at least 200 to 300°F to produce comparable refrigerant flow rate to that from previous experimental efforts upon which the performance parameters are based. The second major design requirement is to minimize exhaust backpressure to levels below the engine manufactures' maximum allowable. For example, Cummins large diesel engines (M11 and N14 series) exhaust maximum allowable backpressure created by the piping and silencer is 40 inches of water [9] (less than 1.5 psi). The specified normally acceptable inside diameter of the exhaust pipe is 5 to 6 inches (20 to 28 sq.in).

Diesel Engine Exhaust Available Heat Rate

The available exhaust heat and the maximum temperatures are taken from Cummins M11, N14 and ISC engines. The Cummins M11, N14 and ISC are used for transit buses and medium and large truck applications. Both the engine's coolant system and exhaust gases could be used to provide necessary heating requirements. Both are evaluated in the following.

We first consider the available heat from the engine coolant fluid. A minimal coolant temperature of 180°F and maximum temperature of 212°F are used for best available conditions, as suggested by the manufacturer. The coolant is a 50/50 mixture of ethylene glycol antifreeze and water with specific gravity of 1.025. The N14's coolant flow rate at the higher RPM is 98 GPM on a 525 H.P. engine and 84 GPM on the 330 H.P. engines. Information on the M11 is not available, but it is a predecessor of the Cummins ISC; available data on the 225 H.P. ISC is 90 GPM coolant flow rate. Nominal operation will provide a coolant temperature difference of 7°F from radiator inlet to outlet. Using a nominal flow rate of 90 GPM (768 lb/min 50/50 E.G./water mix with a Cp of .86 Btu/lb/F at 160°F and 0.88 Btu/lb/F at 212°F [2]) and a temperature difference of 5°F, analysis shows that approximately 198 kBTU/hr is available. Considering an 80% coolant flow rate, 158 kBTU/hr is available. The available heat from 100% and 80% of maximum coolant flow rates are tabulated below.

Table 4. Available Heat from the Engine Coolant Circuit of a Large Diesel

Diesel Engine Type	Percent Maximum Diesel Engine Coolant Flow	Engine Coolant Mass Flow (lbm/min)	Coolant Temperature Difference Considered (^oF)	Coolant Heat Available (BTU/hr)
225 – 525 H.P.	100	768	5	198,000
225 – 525 H.P.	80	614	5	158,000

Now we consider the available heat from a large diesels exhaust. Diesel engines commonly have an exhaust driven turbo-fuel injector (turbo-charger). The maximum exhaust temperature after the turbo-charger on the 525 H.P. N14 is 912°F at a peak power RPM of 1700. Otherwise, regular operating exhaust temperatures at the turbo-charger are from 600°F to 800°F. The maximum exhaust flow is about 3000 SCFM. The 225 H.P. ISC provides a maximum exhaust flow of 1400 SCFM at the maximum temperature of 860°F. In order to provide margin in the analysis and account for moderate running conditions, a reduced amount (60 percent) of these maximum values will be used as a baseline. A typical specific heat of exhaust gases will be considered at 0.355 Btu/lbm/F at 500° F and 0.367 Btu/lbm/F at 900°F considering dry exhaust constituents of 81% N_2, 10% CO_2, 6% CO, and 3% H_2. The following table tabulates heat quantities available considering both 60 and 100 percent of the maximum exhaust flow rates and considering an exhaust gas temperature drop of 100°F.

Table 5. Available Heat from the Engine Exhaust of a Large Diesel

Large Diesel Engine Type	Diesel Engine Exhaust Mass Flow (lbm/min)	Considered Exhaust Temperature Loss to Bed (degrees F)	Available Exhaust Heat (BTU/hr)
525 H.P.	223.5 (max. exhaust flow)	100	483,000
525 H.P.	134.1 (at 60% of max. flow)	100	290,000
225 H.P.	104.3 (max. exhaust flow)	100	225,000
225 H.P.	62.61 (at 60% of max. flow)	100	135,000

The available exhaust heat values listed above compare with the 70 kW minimum available recovered exhaust heat from a standard 44-foot, 50 seat, 277 h.p. bus as reported by Li Zhi Zhang (ref. 6). Note that the 70 kW reported by Zhang converts to 239,000 BTU/hr.

Refrigeration Thermodynamic Parameters and Flow Rate

The vapor-compression refrigeration cycle temperatures and pressures at each point of the cycle are required to perform an evaluation and prediction of the refrigeration system performance and, ultimately, the required refrigerant flow rate. The points of concern will basically be at the inlet and/or outlet of each component. This evaluation will support the proper and detailed design and sizing of the system. The refrigerants considered for this evaluation will be the high performance, non ozone-depletion-potential (ODP) ammonia R717, and the common medium-temperature refrigerant R134A. Evaporator temperature and cooling loads are different for use in each bus air-conditioning and refrigerated trailers. A separate evaluation will be performed for each. The bus and refrigerated trailer return air temperatures and cooling loads are determined from existing operational systems and their respective manufactures specifications.

Refrigerated Trailer and Bus A/C System Operating Temperatures and Pressures

Reference the adsorption system shown in Figure 1 of the “Experimental Data and Analysis” section of this paper. The refrigerated trailer has a relatively low evaporator temperature and low cooling load in comparison to the bus A/C requirements. Refrigerated trailer evaporator temperatures will vary depending on the trailer contents such as a frozen versus a produce load. Consider a R717 system with evaporator temperatures varying from 0 to 40°F and corresponding cooling loads from 46,000 to 64,000 BTU/hr. The corresponding evaporator pressures will be approximately 15 and 55 psig. These evaporator pressures are the sorbent beds' adsorption pressures at the bed minimum temperature, which is considered to be approximately 10 to 15°F above ambient temperature, giving a 110°F adsorption temperature (95°F ambient + 15°F = 110°F adsorption temperature). The beds' compressor outlet temperature is considered to be the beds' maximum heated temperature of 400°F at the condensing pressure of 230 psig. Compressor outlet conditions equal condenser inlet conditions. An optimum condenser outlet temperature could be 110°F (if we consider 95°F ambient temperature and add 15°F), and outlet pressure is considered the same as (actually slightly less than) inlet pressure. The expansion valve actually has an inlet liquid pressure and temperature slightly less than condenser outlet conditions given, but for this evaluation the losses will be considered negligible. These operational parameters can be used to determine the required flow rate of refrigerant for a given cooling capacity and to determine the maximum expected refrigerant flow rate using the isotherms.

Temperature and pressure values for refrigerated trailers using R134A and for bus A/C systems using both R717 and R134A are now listed using a similar evaluation to that above, such that the condenser and adsorption compressors' operating temperatures are the same as above.

Refrigerated trailer using R134A:
Evaporator temperature of 40°F.
Evaporator pressure of 35 psig, which is the compressors' low temperature adsorption pressure.
Condenser pressure of 145 to 150 psig that is the compressors high temperature desorption pressure.
Bus Air-Conditioning System using R717: The typical evaporator temperature coincides with or is slightly less then the desired return air temperature of approximately 67°F.
Evaporator temperature of 67°F.
Evaporator pressure of 105 psig, which is the compressors' low temperature adsorption pressure.
Condenser pressure of 230 psig that is the compressors high temperature desorption pressure.
.
Bus Air-Conditioning System using R134A:
Evaporator temperature of 67F.
Evaporator pressure of 65 psig, which is the compressors low temperature adsorption pressure.
Condenser pressure of 145 psig that is the compressors high temperature desorption pressure.

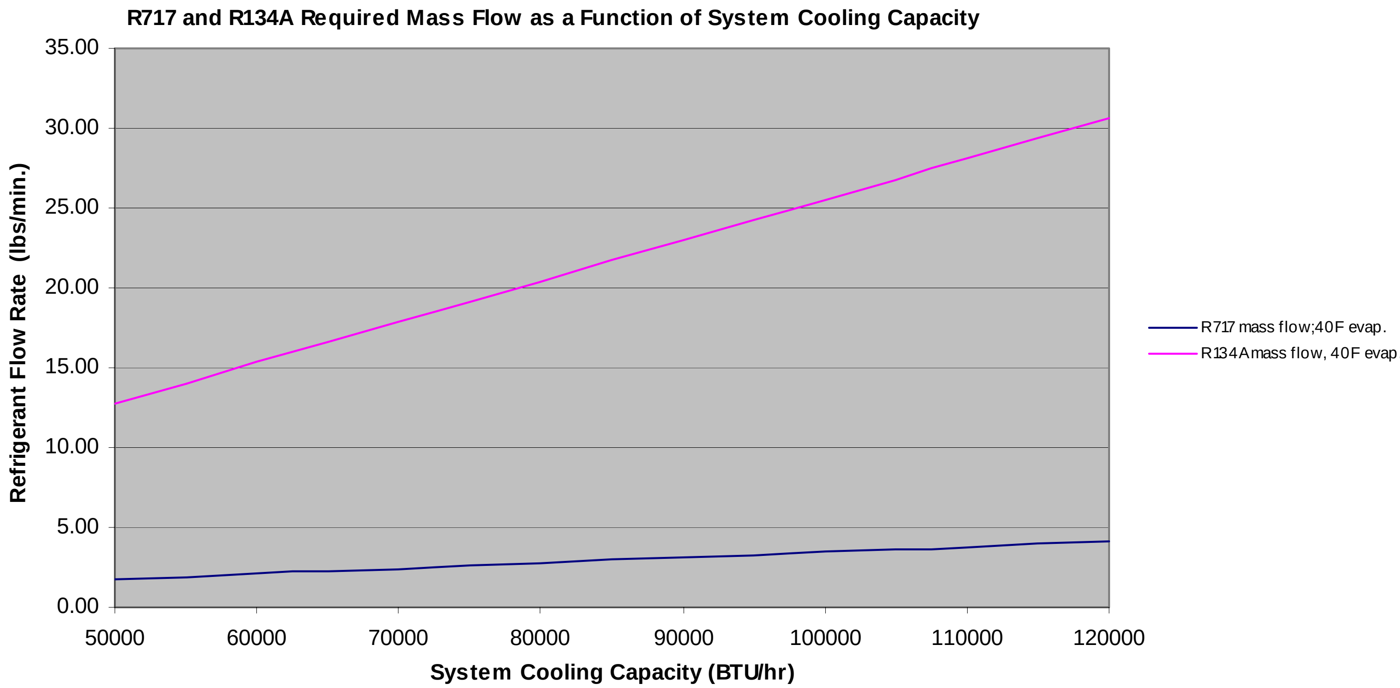
Refrigerant Flow Rate and Plumbing Sizes

Using the enthalpy values of the refrigerants, the required flow rate per cooling capacity is determined and plotted below.

Refrigerant
Mass flow (lb/hr) = required cooling capacity (BTU/hr) / refrigerant enthalpy change across the evaporator (BTU/lb)

The refrigerant pressure drop across the expansion valve is an isenthalpic process, or called a constant enthalpy throttling process. Since the enthalpy across the expansion valve is constant, the enthalpy at the expansion valve outlet is that of the condenser outlet. The enthalpy at the expansion valve outlet is the enthalpy of the evaporator inlet. So, the refrigerant enthalpy values at condenser temperatures of 100-110°F and corresponding pressures, and the 40°F evaporator temperature and corresponding pressure are used; the following graph of refrigerant flow rate as a function of cooling capacity is created.

Figure 5. Refrigerant R717 and R134A Required Flow Rates as a Function of Cooling Capacity



In fact, the refrigerant flow rates indicated above are also accurate for 20°F and 0°F evaporator temperatures.

Required plumbing sizes are determined based on minimizing refrigerant flow losses in the system. The rule of thumb for liquid flow (incompressible flow) is to maintain velocities well below 10 ft/sec (2ft/sec used here). The rule of thumb for gas flow (compressible flow) is to maintain velocities well below 20% of the speed of sound (mach1) in the gas at the flow temperature (0.1 mach used here). The R717 liquid flow for a 64,000 BTU/hr system requires a 3/8" diameter minimum line size and for a 120,000 BTU/hr system it requires a 1/2" diameter minimum line size. The R717 gas flow for a 64,000 BTU/hr system requires a 1/2" minimum line size, for a 120,000 BTU/hr system it requires a 5/8" minimum line size. The R134A liquid flow for a 64,000 BTU/hr system requires a 5/8" diameter minimum line size and for a 120,000 BTU/hr system it requires a 7/8" minimum line size. The R134A gas flow for a 64,000 BTU/hr system requires a 5/8” diameter minimum line size and for a 120,000 BTU/hr system it requires a 1” diameter minimum line size.

System Design

The System Design section covers basic system operation modes, weight, required heat input, and sorbent bed and compressor assembly designs; including packaging, size and layout configurations. Both, the high performing R717 (ammonia) and the more user friendly R134A are investigated.

First some basic design ground rules are identified. The basic design concept discussed in detail does not have a heat regenerative system, thus eliminating the associated volume and heat sink effects that come with the bulky hardware. The heat regenerative system could be practical on a refrigerated trailer when the compressor bed is located on the trailer such that it utilizes a working fluid to transfer the heat into and out of the sorbent beds. But the added complications of the extra working fluid subsystem weight, volume and required valve control distract from the feasibility of an adsorption system for transit applications.

It is desired to have the simplicity of the direct exhaust heating with forced air cooling of the sorbent beds

Operating Plan and Modes

Hybrid System Complications and Operational Options

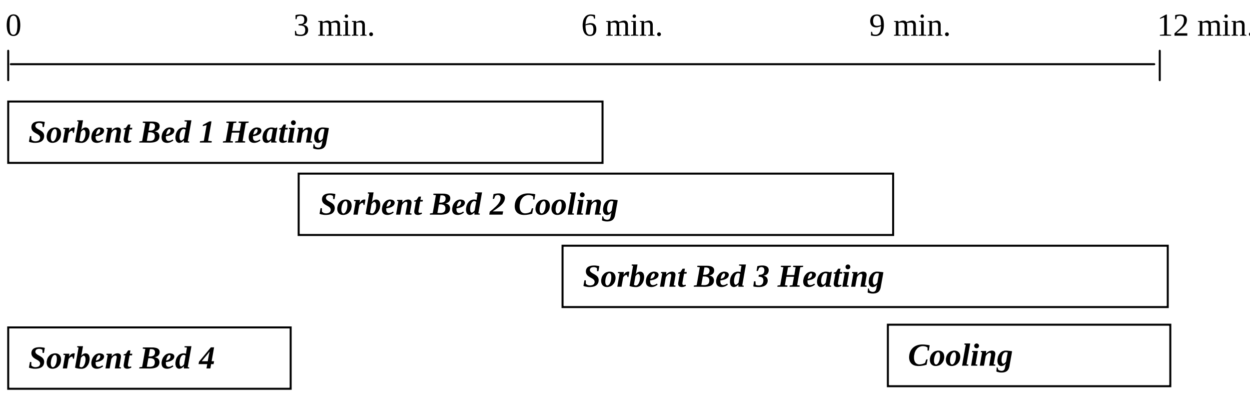
In various refrigerated trailer transit scenarios, the trailer and cargo are desired to be self-sufficient. Without the tractor truck attached to the trailer with its engine running, the adsorption system will not create refrigeration. One solution is to consider a hybrid system with both an adsorption compressor and a diesel powered mechanical compressor as back up. Isolating one compressor type (adsorption or mechanical) out of the system and including another by way of valves is not a problem and can easily be done if packaging volume allows. The problem arises with the oil used in typical mechanical compressors, as it will contaminate the refrigerant. This oil contaminant can be adsorbed into the sorbent bed and never fully desorb, resulting in a lower sorbent bed performance and can ultimately kill off the bed’s adsorbent properties. One design option is the use of filters at the outlet of the mechanical compressor or at the inlet to the sorbent beds. If the mechanical compressor is used with filters, system maintenance frequency, such as refrigerant and filter change-out, will be dependant on duration and frequency of the mechanical compressor’s use.

One option to increase the refrigerated containers’ short-term ability to be self-sufficient is to sub-cool the refrigerated space prior to shutting down the truck’s main engine. Of course this might not be as effective in harsh environments. Another alternative is for adsorption compression refrigerated trailers to be operated such that a dedicated driving team maintains continuous transit from refrigerated payload pick-up to delivery with only short stops.

Valve Control System

In the case of the direct exhaust heating and forced air-cooling of the sorbent beds, there are valves requiring control to change from one mode, heating or cooling, to the other. Also, there is an exhaust bypass valve that needs control to avoid runaway sorbent bed temperature and overheating. A simple representation of these valves is a flapper type shown in Figure 14.

In order to temperature cycle the sorbent beds, each bed will have a flapper type valve that either directs hot exhaust during a heating mode, or forced ambient air during a cooling mode, through the bed's heating/cooling flow path. A complete cycle for all four beds would be 12 minutes such that every quarter cycle (3 minutes), one of the beds will change modes from heating to cooling or visa versa. This results in each bed remaining in a heating or cooling mode for 6 minutes prior to a change in mode.



Although time phasing could change for certain designs, this is the scheme used for the evaluation of the systems studied in this report (ref. Appendix B).

The other important valve requiring control is the previously mentioned exhaust bypass valve. A closed loop feedback control system would be required to change the position of the exhaust bypass valve and direct the exhaust through the standard exhaust pipe outlet, thus removing heat input from the sorbent beds. This feature would allow the tractor operation without the refrigerant system operation and is also a safety feature to maintain the beds below a predetermined maximum temperature, approximately 400 F.

Sorbent Bed Cross-Sectional Layout Designs

Two different basic bed designs are considered (ref. Figure 6). One design has a single path axial flow of the heating/cooling fluid through the center area of the 6" diameter tube with the sorbent material (single carbon layer) region bonded on the tube's outer periphery. In this design there is only a single carbon layer (SCL). The other design is more complex, with increased performance, and has two heating/cooling fluid flow paths (dual flow). One heating/cooling flow path is through the central 6" diameter flow duct, with the sorbent material in an annulus region between the inner/central heating/cooling flow path and the other outer heating/cooling flow path that is down an outer annulus region that is on the outer periphery of the sorbent material annulus region (ref. Figure 6). This design has a dual carbon layer (DCL) and can be more easily visualized in an isometric cross-sectional view in Figure 7. The dual flow canister has two main benefits. One is the increased amount of usable waste heat extracted from exhaust gases; the other is a more compact compressor (the added outer heating/cooling fluid flow path allows the addition of an outer sorbent layer which provides for a shorter bed with increased sorbent material per unit length). The added complexity and cross-sectional material of the dual flow design is overcome by the shorter bed length required (for a constant carbon quantity) such that it has less mass and requires less heat input when compared to the single flow SCL design. There is a point where the system size (and therefore a sorbent bed canister mass) and its thermal capacity can overwhelm the output of available exhaust waste heat. The added length of the single flow compressor bed, with the single sorbent layer, falls into this high-heat-input category which, in most cases of high-cooling-capacity requirement, exceeds the available heat. The bed length can also cause excessive exhaust back pressure.

Each of the two cross-sectional designs have been mathematically modeled to determine proper diameters, sorbent holding capability per unit length, total mass including canister hardware, volume, and required heat input rate (ref. Appendix B). The DCL design is the main focus of this study due to its higher performance. The DCL design displays a practical application with respect to volume (mostly overall length) and heat input.
As previously mentioned, the sorbent bed design considered focuses on the direct exhaust heating option <u>versus</u> use of a working fluid to extract heat from the exhaust and deliver it to the sorbent beds. The direct exhaust heating allows better utilization of the potentially available heat versus the losses associated with the heating and cooling required of the working fluid. Also of importance is the difficulty associated with the high temperatures the heat transfer working fluid must endure. But, with a heat transfer working fluid (versus direct exhaust heating and air cooling), the sorbent bed canister structure could potentially be lighter with aluminum, as opposed to stainless steel (with the proper choice of working fluid) and could therefore require less heat input. This heat transfer working fluid option is considered for the larger bed requirements on an R134A system.

For the direct exhaust heating design, all compressor tubing is stainless steel (AISI 316 type for high temperature) and the sorbent assembly typically constructed by packing carbon onto the 8% aluminum foam material. The carbon is mixed with a binder into a slurry mud and packed into the aluminum foam (ref. example of Figure 8). More details of the carbon to foam weight ratios are in the Carbon Density section A-2 of the appendix. Figures 9 and 10 show the DCL bed design end view (no cross-section) and a total bed isometric view respectively.

A major parameter in the bed design is the tubing's allowable wall thickness. Wall thickness plays an important roll in the robustness, safety, and long-life operation of the material and overall component. Considering an R717 ammonia system, if nominal operating high pressures are in the realm of 200 psi and system relief valves are set at 275 to 300 psi, the systems' maximum expected operating pressures (MEOP) are considered to be the relief valve setting pressure. Considering a 300 psi MEOP and a factor of safety of 2.5, a design pressure of 750 psi is determined. Using tabulated data[10] and the classic Barlow formula (or Modified Lame' Formula for high temperature conditions and creep) a stainless steel wall thickness for a 5 to 6 inch diameter 750 psi tube is 0.10 inch (using a 0.030" corrosion factor). In the case of the dual flow design, the outer shell tube does not see high pressure refrigerant and the exhaust pressures are not too high allowing the outer shell tube to have a thinner wall. For this evaluation, commonality is kept between all SS tubing for robustness and the fact that the extra wall thickness of the outer shell will also help account for concentric tubing secondary supporting structure not otherwise considered.

Figure 6. Sorbent Bed Cross Sectional Design Options

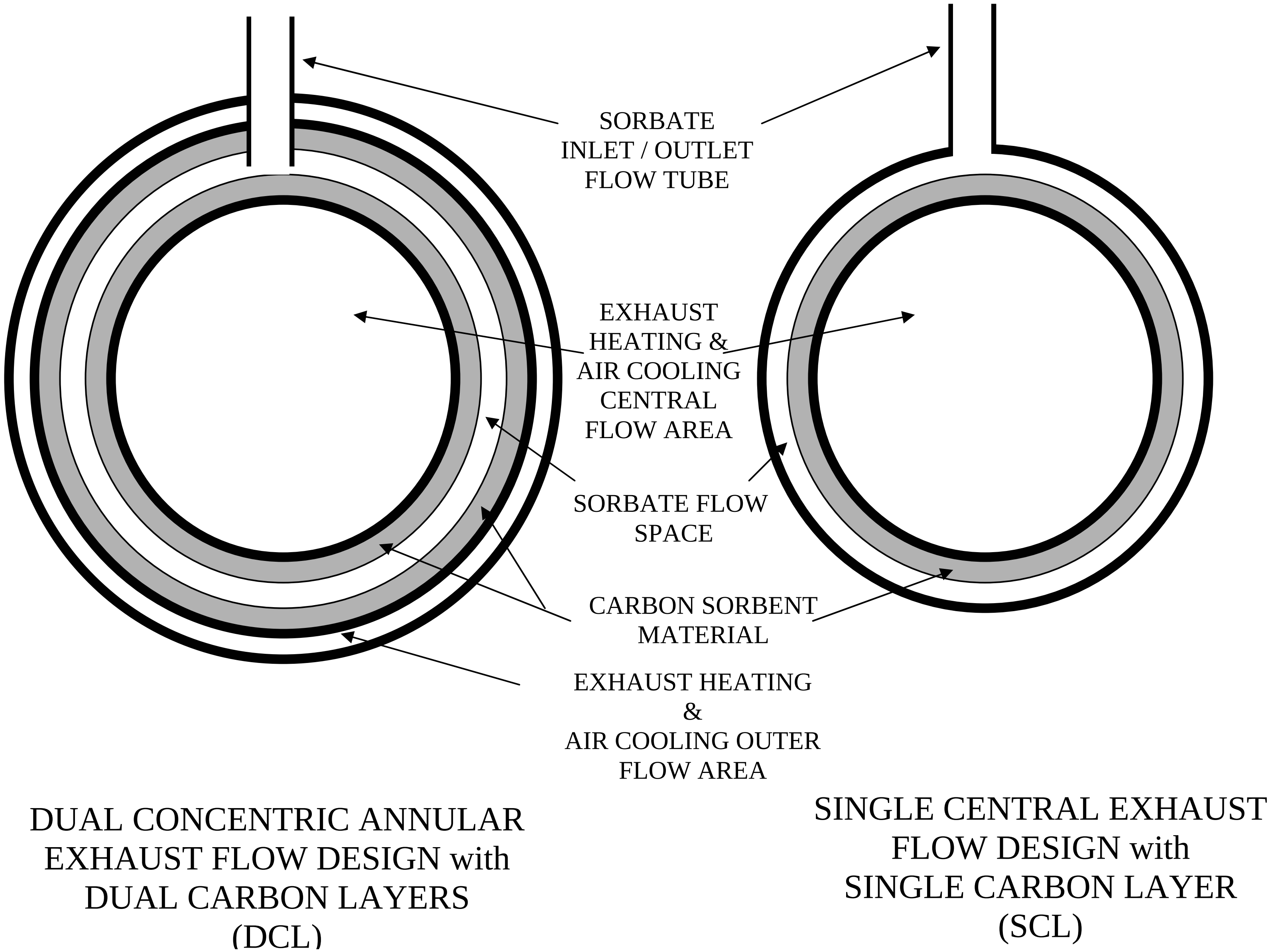


Figure 7. DCL Sorbent Bed Cross-Sectional Isometric View

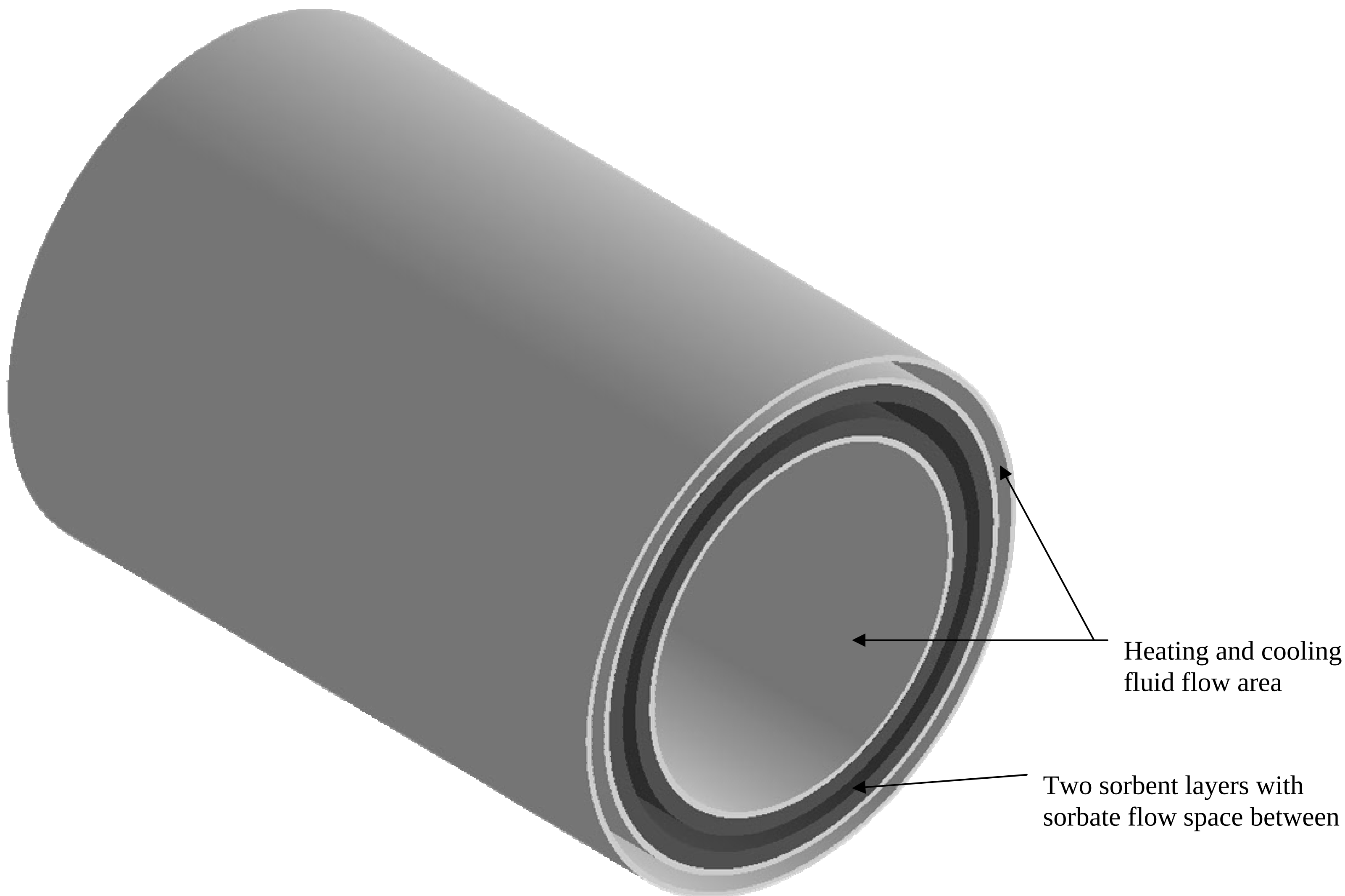


Figure 8. Aluminum Foam with Carbon Sorbent in Smaller Experimental Bed

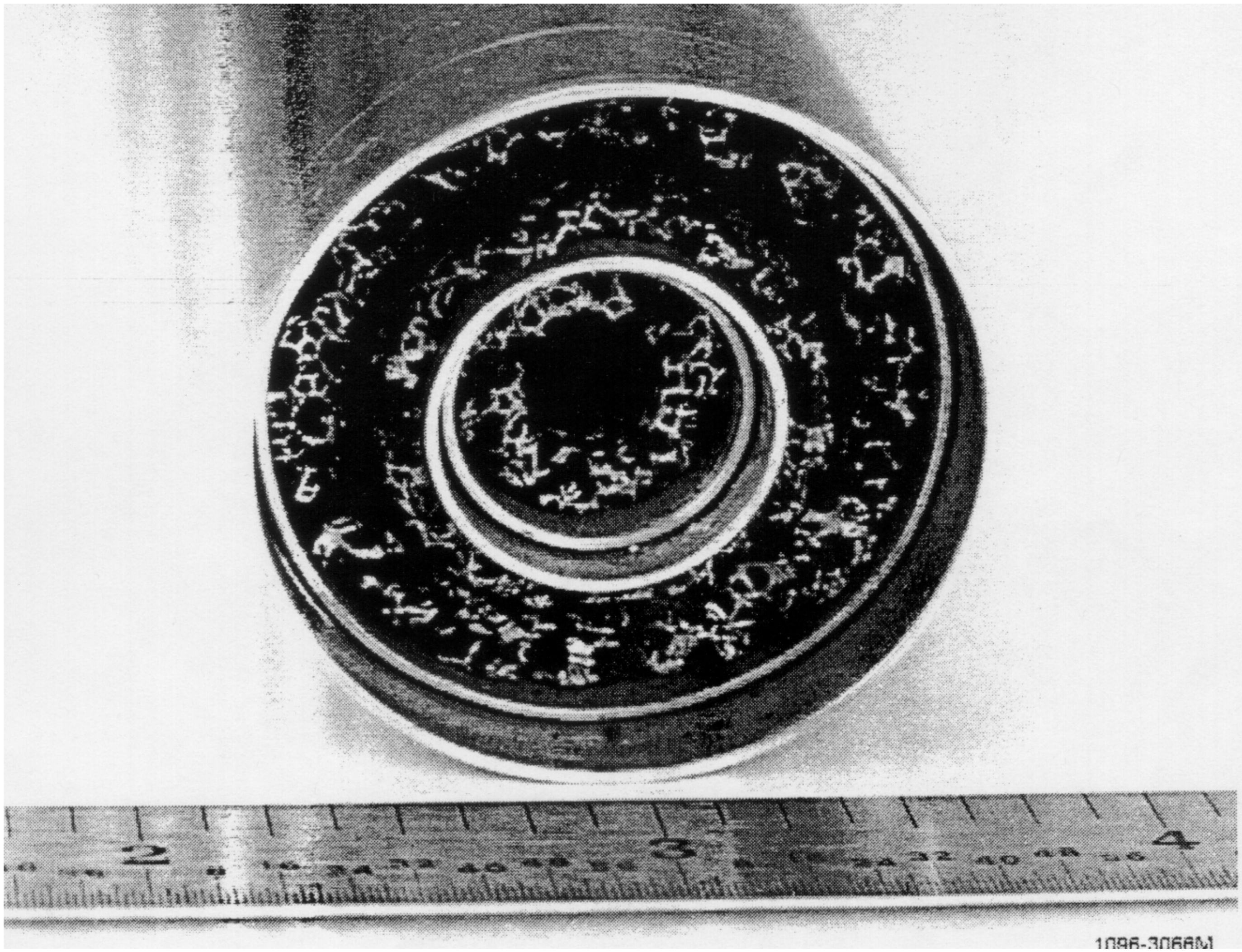


Figure 9. DCL Sorbent Bed End View

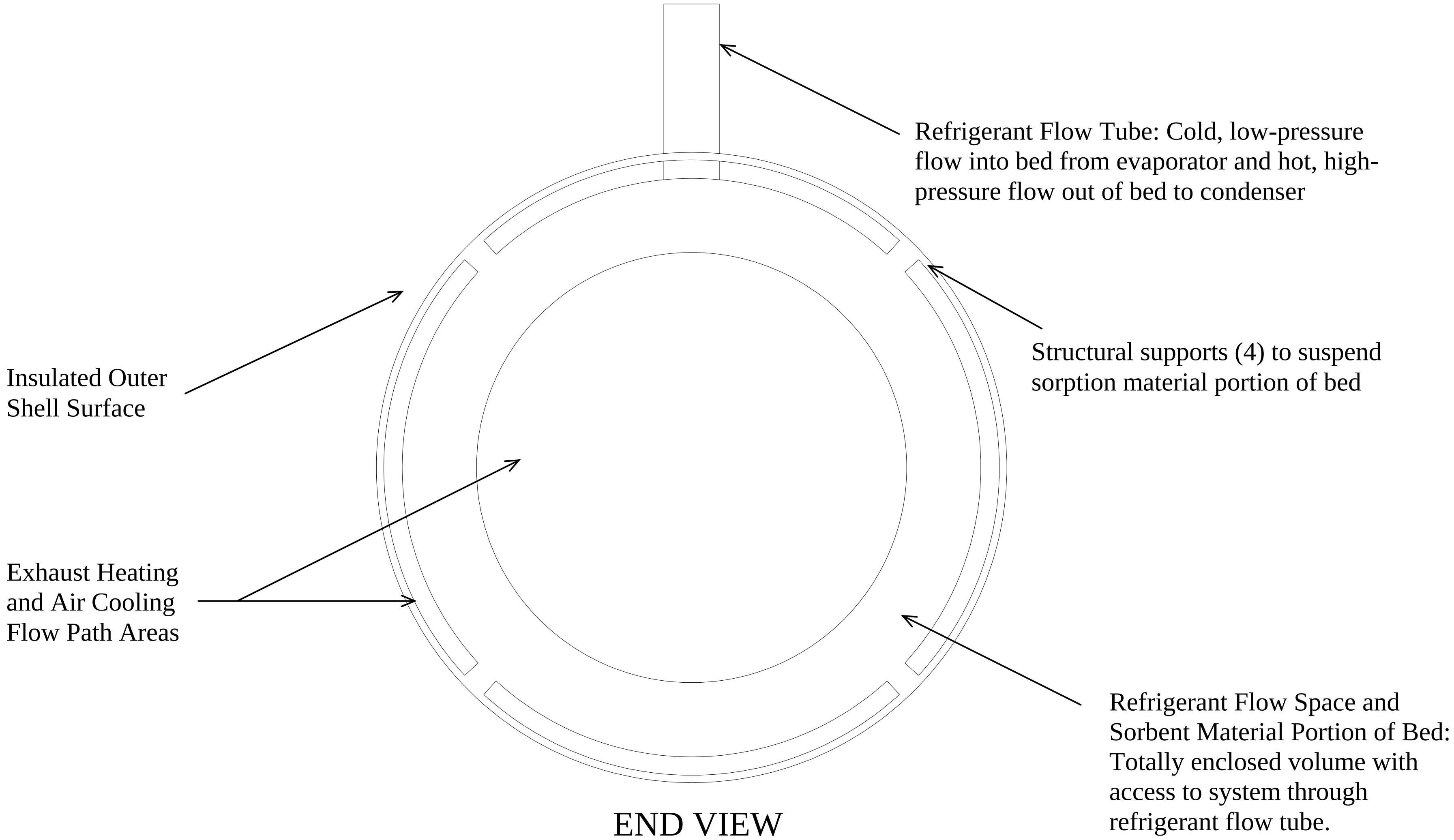
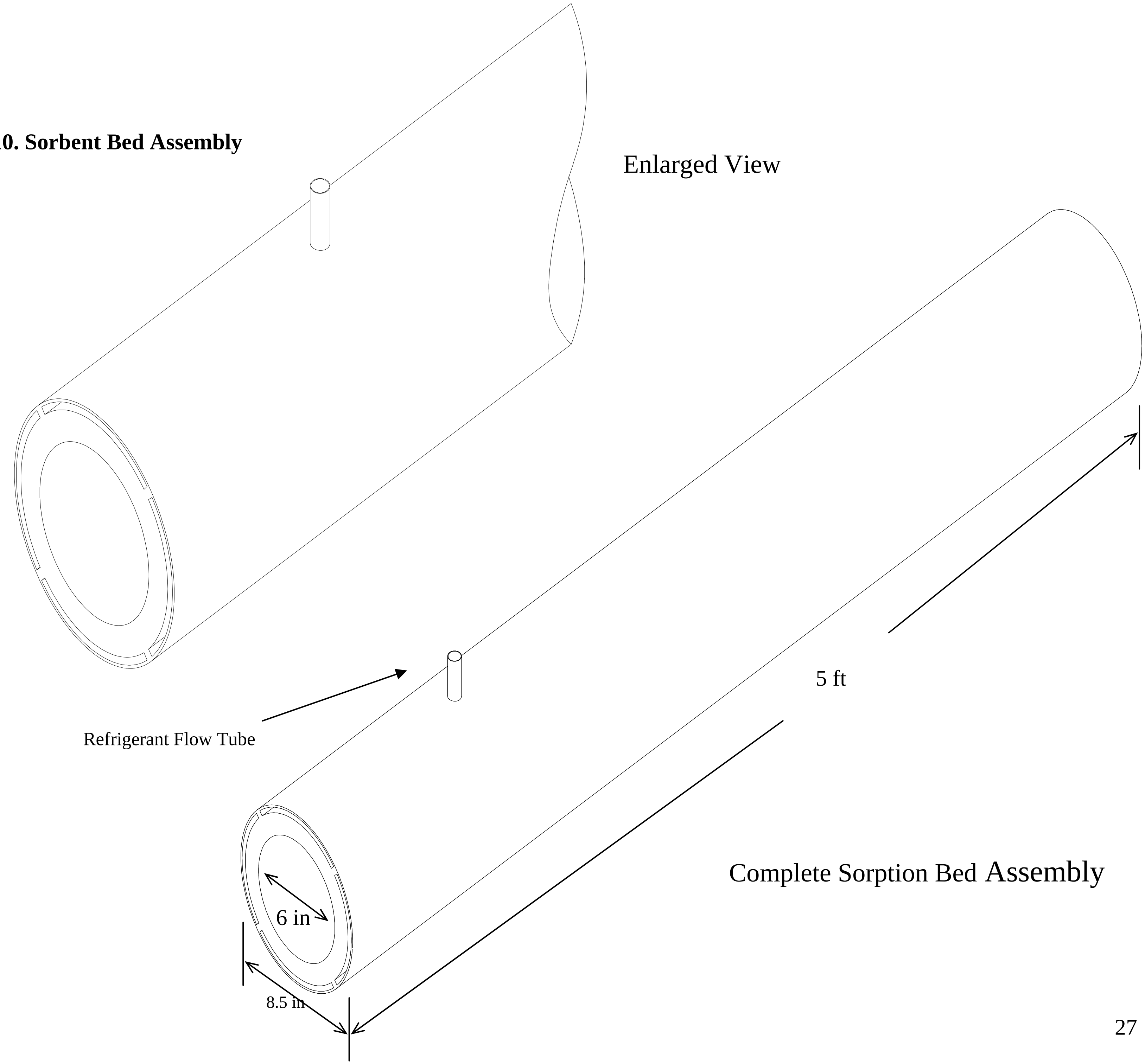


Figure 10. Sorbent Bed Assembly



Another important detail in the bed design is the diameter, but increasing the diameter of each sorbent bed canister has its trade-offs. An increased diameter increases the quantity of sorbent material per unit length of bed, which helps to shorten the bed. A shortened bed length is desirable for most packaging situations. Larger bed diameters result in larger exhaust flow areas that, in turn, will decrease exhaust flow pressure as the flow velocities and associated losses also decrease. It has been determined that the heat transfer area remains about constant as the bed diameter increases due to the overall bed length decrease (with constant sorbent mass), but the resulting heat transfer rate will potentially change (decrease – ref. Heat Transfer to Sorbent Bed section) due to the lower exhaust flow velocity. So, although the larger bed diameters appear to solve some problems, there are other limiting factors. The bad side to increasing the bed diameter is that the total compressor weight and heat rate required also increase directly. These changes result in an increase of the COP as shown in the plots of Figure 11. Figure 11 considers a nominal cooling capacity system. Therefore, the maximum bed diameter is limited by the available exhaust heat.

The analytical model created evaluates the mass and volume of each element of the sorbent bed assembly. The elements are the sorbent, the aluminum foam (holding the sorbent), and the canister structure (tubing, aluminum or stainless steel). Stainless steel tubing for direct exhaust heating is primarily considered but an aluminum structure/tubing is discussed with respect to certain larger R134A system concept layouts. The model also determines the heat required to temperature cycle the bed structure and the associated heat into the refrigerant. The model helps to compare different design options and evaluate performance. The output data of the analytical model's evaluation of different bed designs, the two different refrigerants (R717 and R134A) and specific cooling capacities are located in the appendix B. The significant information and results of the analytical model's output data is discussed in the body of this report.

The bed layout designs considered for this project's evaluation are similar to a design presented by Jack Jones in a paper at the ASME International Heat Pump Conference in 1994[5]. The similarities are with respect to nominal diameter, carbon holding capacity and bed volume. Jones based an 8" OD bed design on success of a smaller scale test with 1.1 lb of carbon in a 2.65 inch diameter bed, 23 inches long. The total bed mass computed per the analytical model used in this paper gives much larger bed structure masses because the direct exhaust heating beds are more robust with stainless steel versus the aluminum used in Jones' water heated and cooled bed. This paper's analytical model predicts a bed's metallic mass to be more than 2 times that predicted by Jones (which was that the masses of sorbent and aluminum hardware were equal) even when an all aluminum bed design is considered. An all aluminum design could be used for certain heat transfer fluids but not in the case of direct exhaust heating to the beds due to corrosion from moisture in the exhaust gases. The larger mass used in the analysis of the bed designs will help add to the design's heat requirement, thus adding confidence of the system's feasibility with potentially improved compressor designs.

The main sorbent bed design constraints are as follows:
1. Provide proper sorbent material quantity and adequate maximum and minimum temperature extremes to produce required refrigerant flow rate. This applies to the compressor volume and sorbent material mass, and the heat transfer performance.
2. Minimize exhaust backpressure to that which is below the engine manufacturers' maximum allowable. Cummins large diesel engines' (M11 series) exhaust maximum allowable backpressure, created by the piping and silencer, is 40 inches of water [9]. The specified normally acceptable inside diameter of the exhaust pipe is 5 to 6 inches.

Figure 11. Effects of Sorbent Bed Diameters on Compressor Length, Mass and Heat

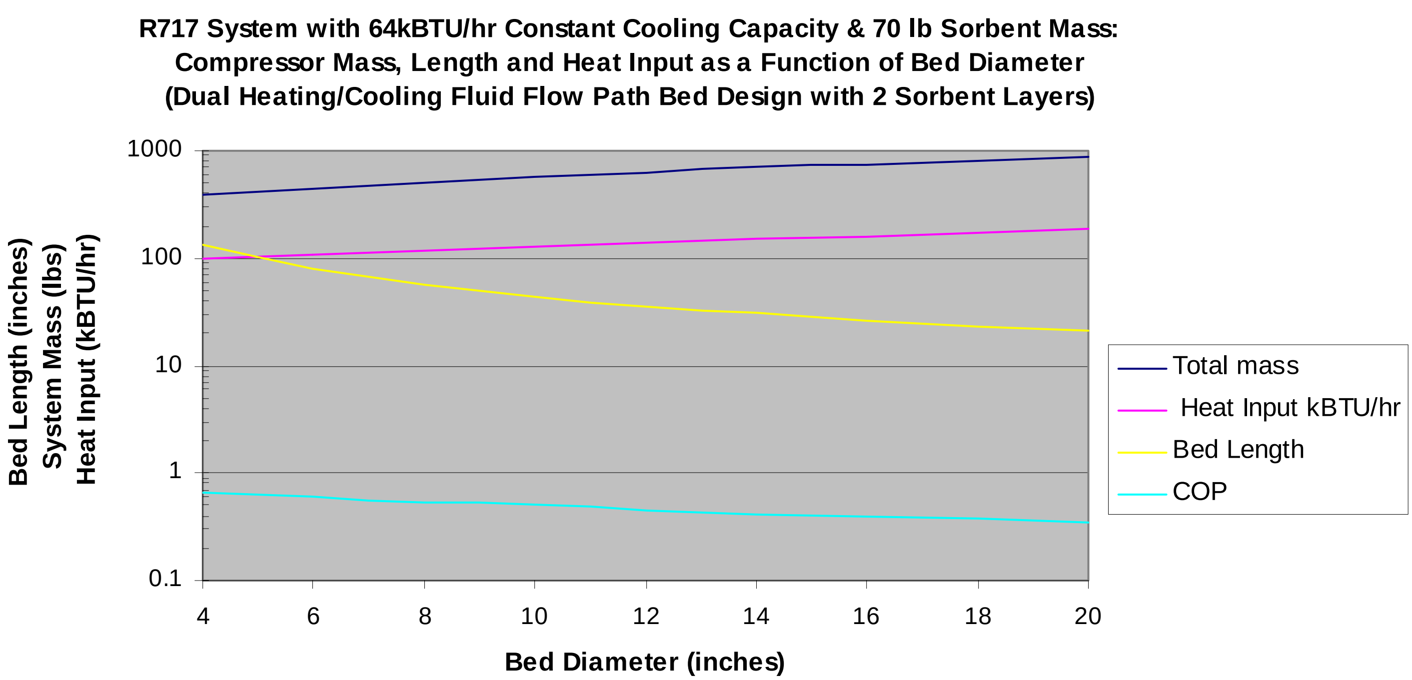
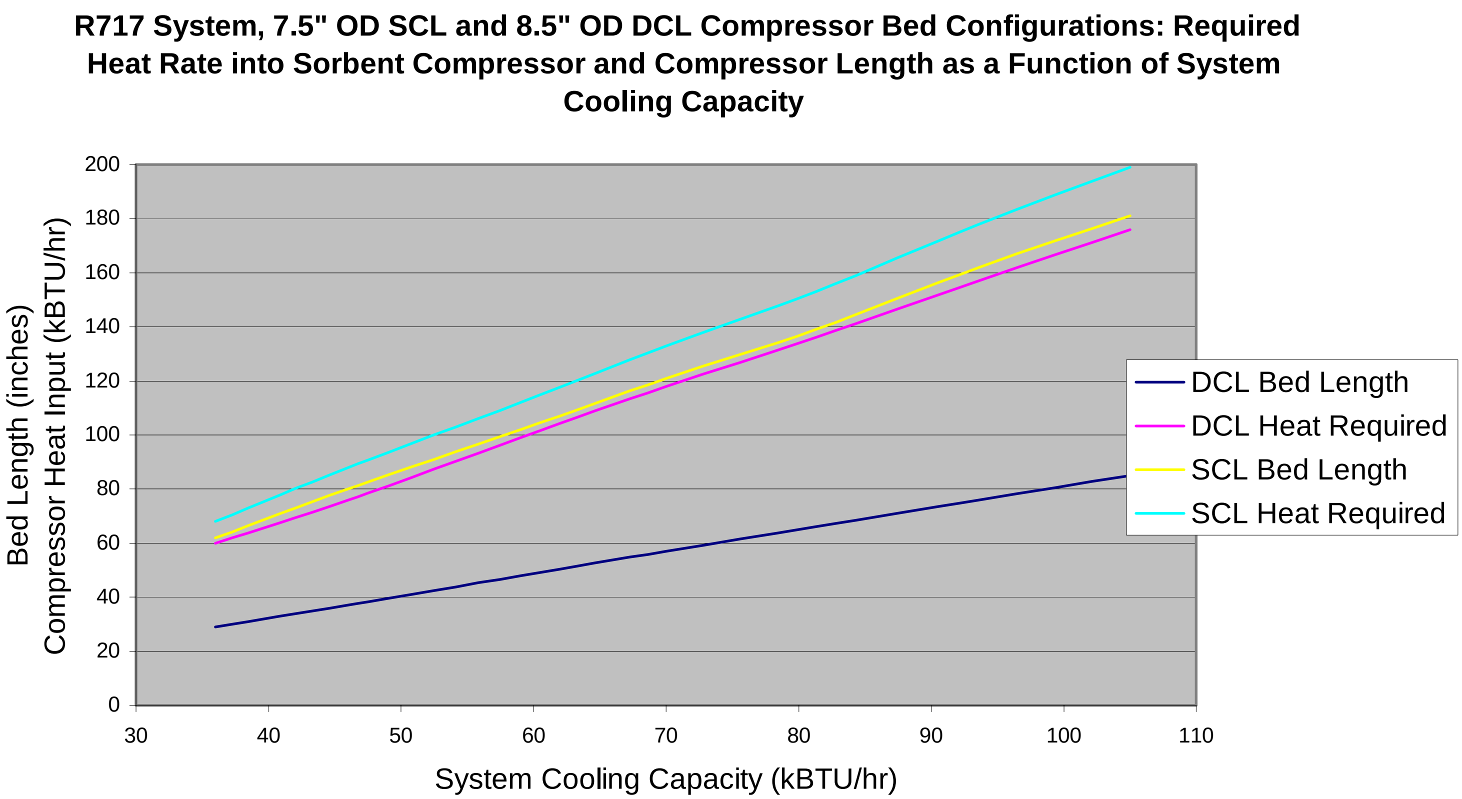


Figure 12. Bed Length and Heat Required for Both Bed Design Options



Sorbent Compressor Assembly Design

As stated in the above Sorbent Bed Layout Cross-Sectional Design section, the sorbent beds utilize direct exhaust heating into the beds. The direct exhaust heating helps to reduce system size and complexity. The cooling is accomplished by forced ambient airflow through the same flow path as that used during the exhaust-heating phase. A compressor assembly design option is shown in Figure 13, and a heating and cooling fluid flow schematic is shown in Figure 14. Each bed could have its own dedicated ambient air fan or one fan could have a manifold allowing it to accommodate more that one bed. Per a heat transfer analysis (appendix C), an airflow rate in the order of 2400 standard cubic feet per minute (SCFM) per bed could perform the cooling required in the time required.

Figure 12 shows plots of the four bed compressor's required bed length and required heat input for a R717 system considering each of the two bed design options (SCL single flow and DCL dual flow).

Figure 13. Direct Exhaust Heating / Ambient Air Cooling Manifold with Four Sorbent Bed

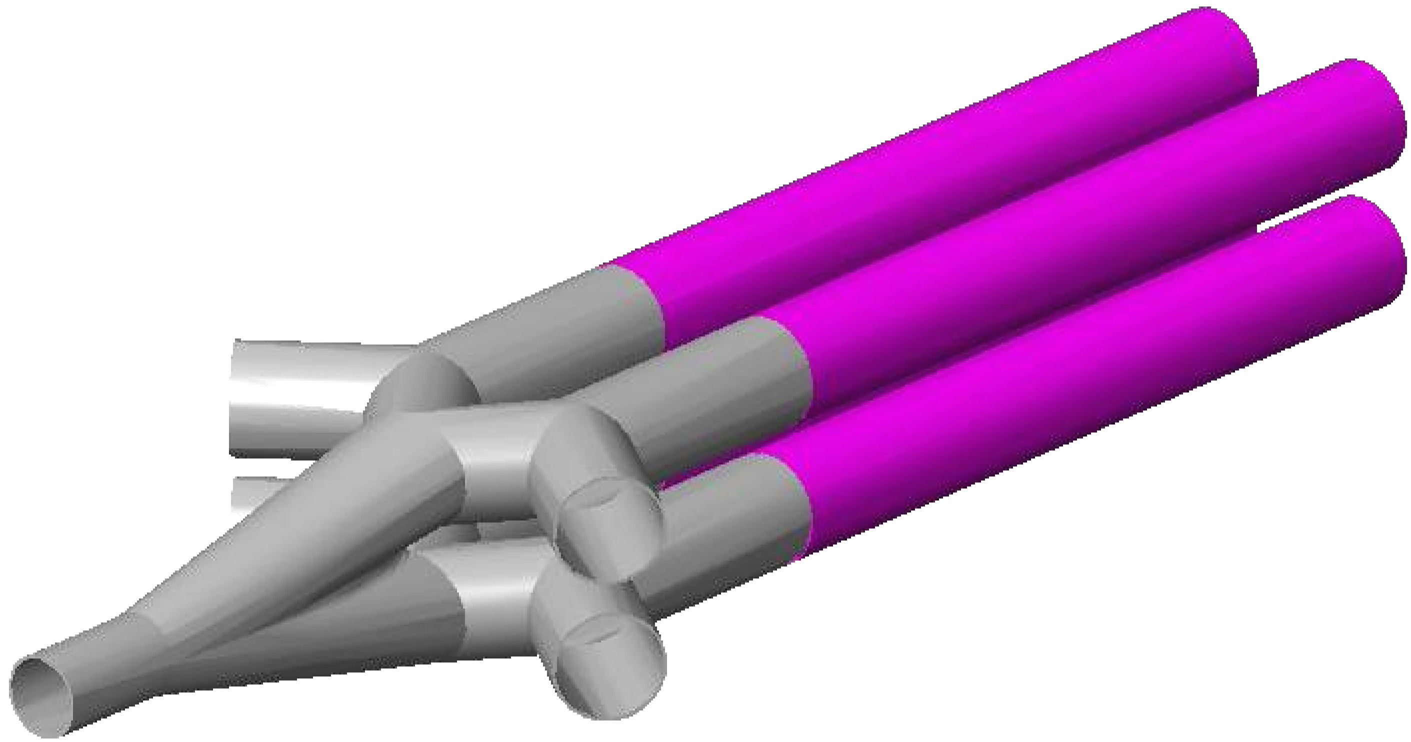
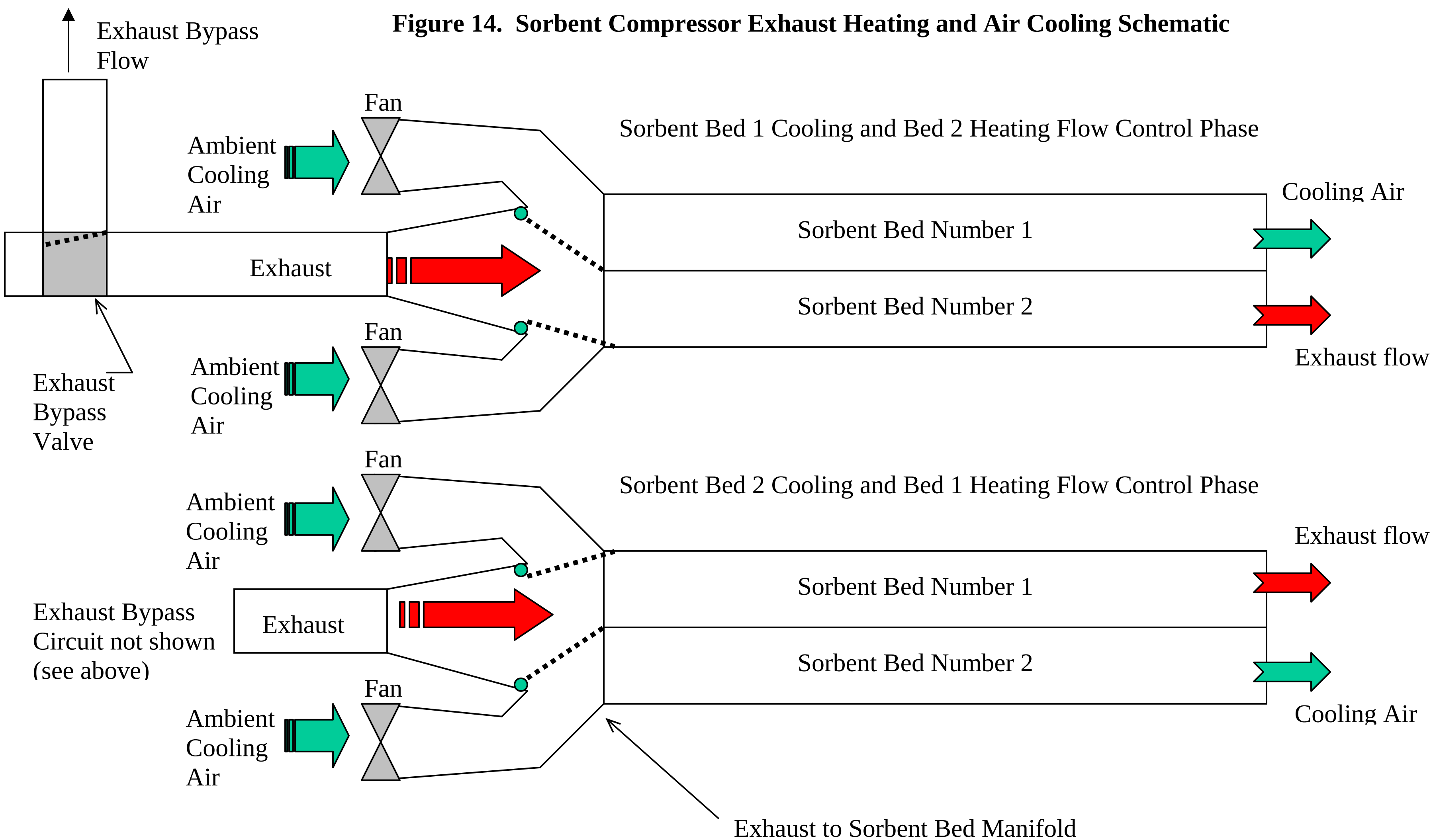


Figure 14. Sorbent Compressor Exhaust Heating and Air Cooling Schematic



Sorbent Compressor Heating and Cooling Flow Paths and Flow Directional Valves

Considering the two beds shown in the Sorbent Compressor Assembly, the exhaust heating flow and/or the ambient air cooling flow would be directed by large, robust high-temperature flapper valves indicated by the dashed line in the manifold.

System Packaging, Size and Layout

The system packaging and layout does have various trade-offs for the different potentials. This section will consider a variety of options, will provide rational for the potentials chosen, and will elaborate upon those potential layouts.

As a rule of thumb, the refrigeration system on a tractor-trailer will be either all on the tractor or all on the trailer such that there are no refrigerant hoses that will be required to be connected between the tractor and trailer. This design rule will eliminate the potential of disturbing the integrity of the closed refrigerant fluid system (i.e.; air ingestion or refrigerant leakage). This design rule does result in the sorbent compressor being collocated with the refrigeration system.

Figure 15 provides relative size information for the conventional tractor-trailer and bus systems before introducing the relative sizes of adsorption systems.

Sorbent Bed Compressor and Refrigeration System on the Trailer

If we first consider the sorbent compressor and refrigeration system on the trailer, then the tractor's exhaust heat will need to be routed to the sorbent compressor on the trailer. This can be done one of two ways. The first, and most obvious, is to route the tractor exhaust to the trailer and then through the sorbent beds. This is the most effective and direct use of the exhaust heat, but the potential for too long of exhaust and too high of exhaust back pressure is a problem. There are bigger problems with the changing direction of tractor relative to trailer (during turns). The only way for the exhaust to be routed to the trailer is if the tractor-trailer exhaust interface were at the pivot ("king pin" - trailer attach point to the tractor). This would call for a custom design tractor-trailer hitch with the 6 to 8 inch exhaust duct routed though the middle. The tractor portion of the exhaust could be routed back to the "king pin" and then extend up into the trailer where an exhaust duct on the trailer could be attached with a V-band clamp and gasket arrangement. The trailer's portion of the duct would be routed to the front or bottom of the trailer depending on where the sorbent bed is mounted. The short portion of the exhaust duct in the trailer would require a heat-insulating shroud. When the tractor and trailer are disconnected, then a blanking plate would be installed onto the tractor's exhaust duct outlet at the pivot/trailer attach point. With the blanking plate installed, the tractor exhaust is expelled out the conventional exhaust pipe by the switching of a large exhaust bypass valve from sorbent compressor heating mode to bypass mode. On either the tractor or trailer, at the pivot point, the exhaust duct would require a swivel that could consist of two sealing surfaces that may rotate upon each other and are pressed together by a spring force.

Since the exhaust duct interface at the pivot appears to be a complex major redesign, another option of getting the heat to the sorbent compressor on the trailer is now discussed. The only other option of getting the tractor's exhaust heat to the trailer is by way of utilizing a pumped working fluid that would extract the heat from the exhaust by means of a heat exchanger on the tractor. The working fluid would be routed by insulated flex hoses including disconnecting couplings that connect to the trailer and then to the sorbent compressor. This poses a few problems. First, it requires an extra working fluid closed loop subsystem which is not only required to heat the bed, but is also required to cool the sorbent bed; this complicates the subsystem design compared to simpler concept of direct exhaust heating and forced ambient air cooling. The working fluid subsystem would therefore require an ambient air radiator and a valving arrangement to accommodate the heating and cooling requirements of each sorbent bed. Although this working fluid subsystem would allow for the more complicated heat regeneration concept to be used, the valving quantity, mass and control can get excessive. Also, there would be heat losses associated, and further complications of selecting a special working fluid that could withstand the high temperature desired for the bed to be exposed. If the working fluid were at all toxic, then the interconnections to the trailer would have potential operational issues.

Although the sorbent compressor and refrigeration system on the trailer has its design complications, it appears to be the only layout scheme suited for the high cooling capacity/large sorbent compressor systems using R134A versus the higher performing toxic R717 (ammonia). As has been determined, the R134A sorbent compressor sizes for systems with cooling capacity larger than 30,000 BTU/hr are so large that they could only be stored on the trailer. And when the beds get too long, exhaust back pressure issues favor a bed heating / cooling heat transfer working fluid to be used. In the following, we discuss the pros and cons of having the sorbent bed compressor and refrigeration system collocated on the tractor.

Sorbent Bed Compressor and Refrigeration System on the Tractor

Locating the sorbent compressor on the tractor looks attractive mainly because of its close proximity to the tractor exhaust system, which minimizes exhaust system length, and the associated back pressure and heat losses that come with it. But now, with the sorbent compressor and refrigeration system on the tractor, we are faced with the need to deliver the refrigeration effect to the trailer. This is more easily done than delivering the exhaust heat to the trailer. In this case we would still use a working fluid, but now it will only be required to be cold and a simple glycol water mix could be utilized. The working fluid would be chilled by the evaporator of the refrigeration system on the tractor and then pumped to and through a cooling coil located inside the trailer. The chilled working fluid subsystem would be rather simple with a safe ‘glycol or antifreeze and water mix’, minimal valves, an electric pump, an expansion tank (bellows type), insulated flexible hoses, and trailer interface disconnect couplings. As mentioned in the previous section, for systems with cooling capacities of 64,000 BTU/hr and larger, the R134A system appears to have too large of sorbent compressor size to fit on the tractor, but an R717 system of the same cooling capacity could fit on the available space of a tractor.

The available packaging space on the tractor is fairly plentiful, especially with the large stretch wheelbases that are becoming more popular. In many instances, extra stretch trucks with custom extended wheel bases are being built for various reasons such as added room for sleepers and added comfort of the ride associated with the longer wheel base. These stretch tractor configurations do help to accommodate the option of tractor mounted sorbent compressor and refrigeration system. Reference Figures 16 and 18 for a diagram and schematic of the tractor mounted system.

When the compressor is to be mounted on the trailer, the use of a trailer's cargo volume is not the baseline packaging design; a preferred design mounts the compressor beneath the trailer as is discussed. Mounting the compressor under the trailer or on the roof of a bus requires the compressor assembly to have each bed side by side for a flatter layout.

The mounting of the adsorption compressor below the trailer's main floor will not call for significant or any retrofitting, and could potentially be just an add-on. The add-on mounting structure will be similar to the method presently used to mount the conventional diesel-powered refrigeration system's fuel tank. Secondary mounting structure will be fastened to the trailer's primary structure and reach downward to support / suspend the adsorption compressor unit to the trailer's underside. The typical primary structure of a large trailer's underside is 4 inch I-Beams of 80,000-pound yield high-strength steel on 12-inch centers.
A picture of the sizes of adsorption A/C units in relation to bus sizes is shown in Figure 17.

. **Figure 15. Relative Sizes of Conventional Systems to 40' Bus or 50' Trailer**

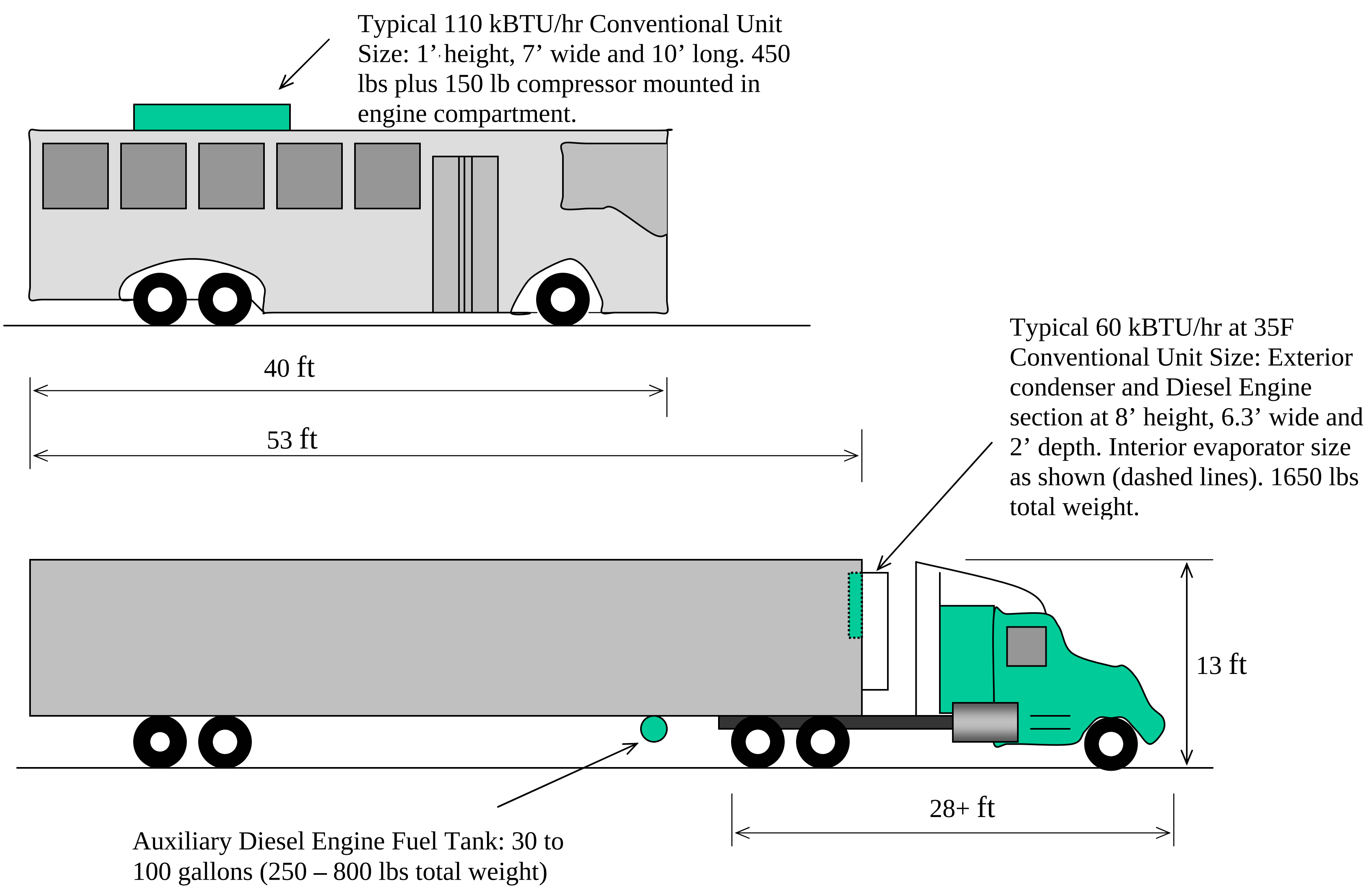
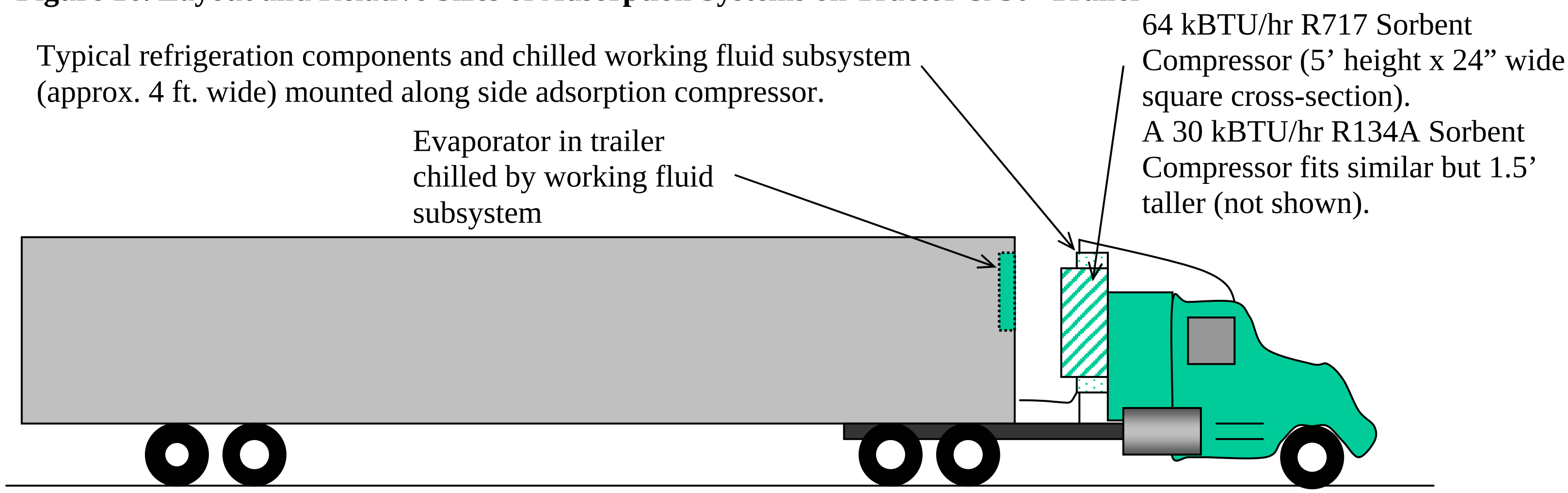
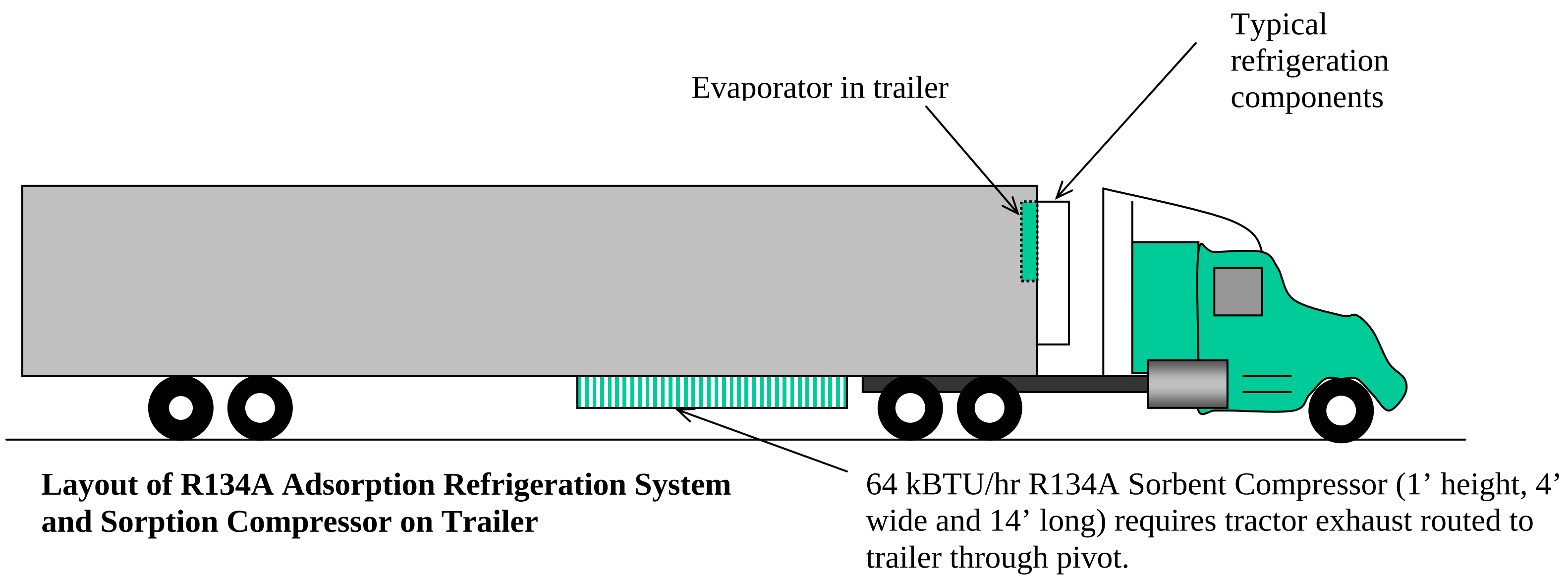


Figure 16. Layout and Relative Sizes of Adsorption Systems on Tractor & 50' Trailer



Layout of Adsorption Refrigeration System and Sorption Compressor on Tractor



Layout of R134A Adsorption Refrigeration System and Sorption Compressor on Trailer

Figure 17. Relative Sizes and Layout of Adsorption Systems on 40' Buses

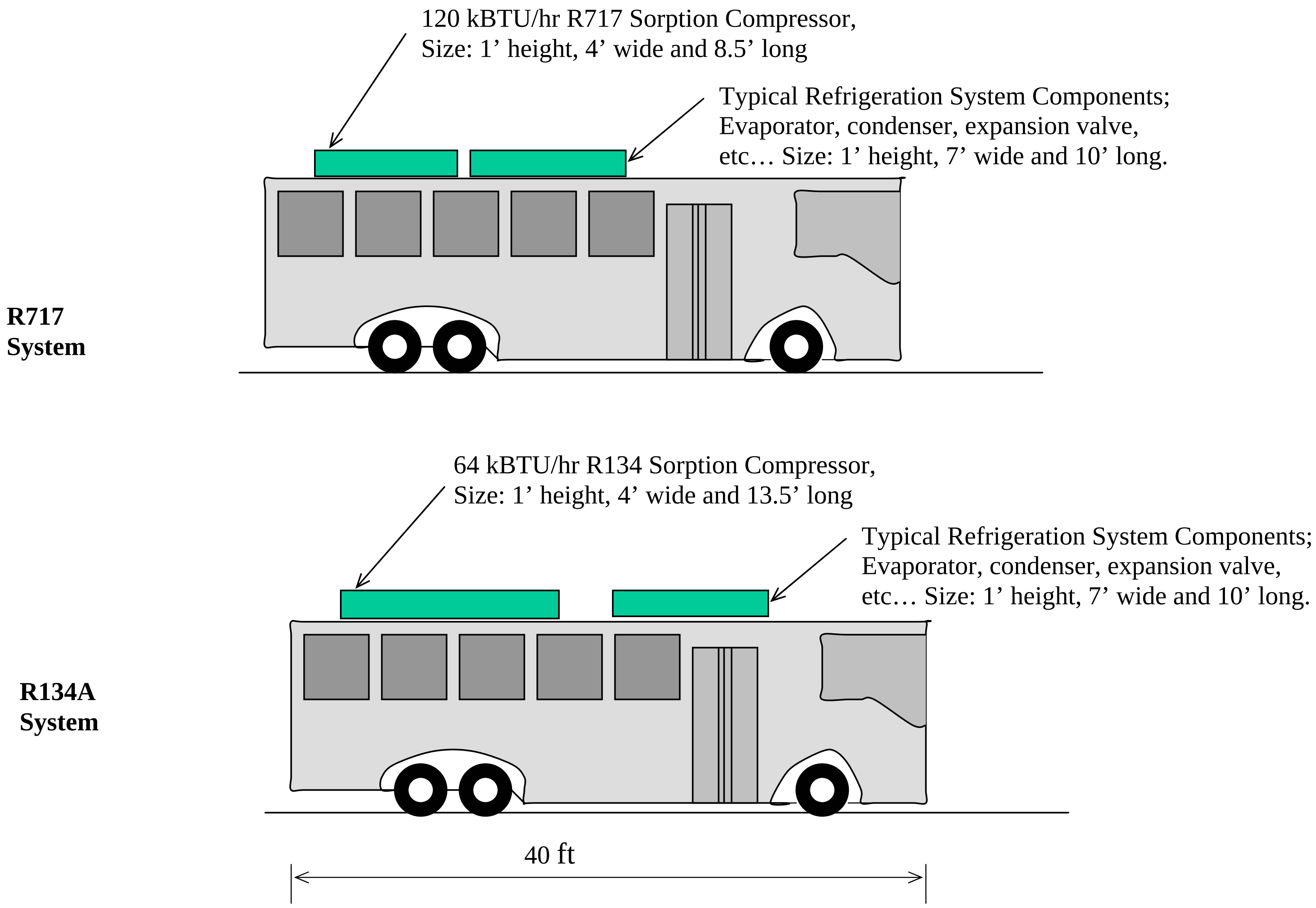
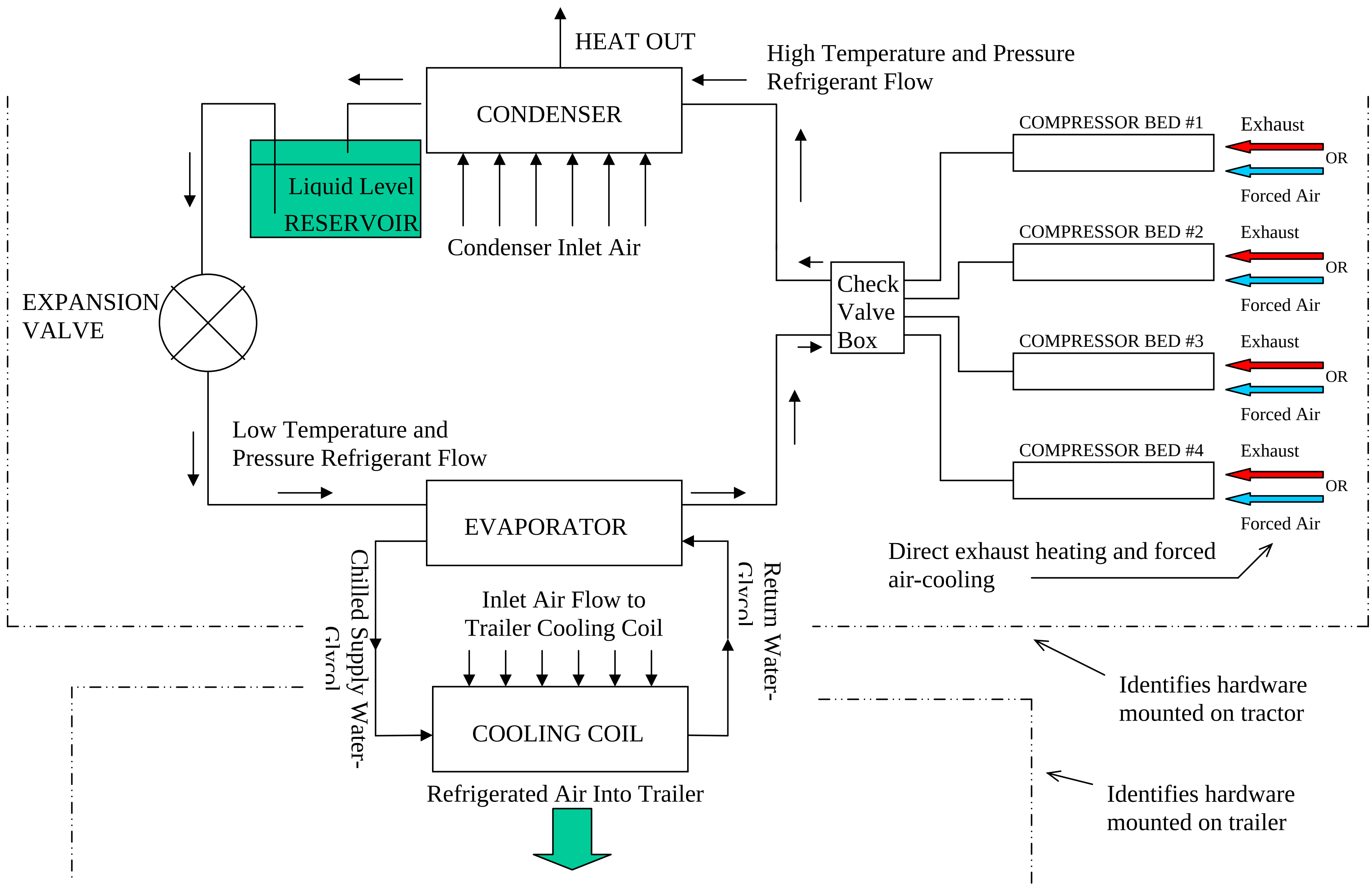


Figure 18. Layout Schematic – Compressor Mounted on Tractor



System Weight

The comparison of adsorption compressor refrigeration system weight to the mechanical compressor refrigeration system's weight has many interesting points. In this design evaluation, we will only consider the weight differences in the refrigeration system based on differences associated with the substitution of design elements related to an adsorption compressor (to replace the typical elements related to a mechanical compressor). Typical baseline weight of other components within the typical refrigeration system, such as evaporator and condenser, will be used as a baseline for this trade study effort.

One big weight trade off which is favorable to the adsorption system is the complete removal of or reduction of the mechanical compressor diesel engine fuel and fuel tank. The refrigerated trailer diesel fuel and tank, for the diesel driven mechanical compressor, is typically stored in a large cylindrical tank from 30 to 120 gallons in size, which is suspended under the trailer (under mount tank). A moderate sized 50-gallon fuel tank weighs approximately 35 lbs and holds up to 350 lbs of fuel. The larger 120-gallon tank can hold up to 850 lbs of fuel. Not only can the fuel weight be reduced or eliminated, but the safety hazard of traveling with the flammable fuel can also be reduced or eliminated.

A typical 25 to 35 h.p. large compressor and clutch can weigh from 115 to 150 lbs (reference the Thermo King 29 h.p. models X426 and X430, and 35 h.p. S391). In certain design applications this weight can be removed because the mechanical compressor is completely eliminated. Complete roof mounted or rear window mounted conventional A/C systems for buses can weigh from 350 to 700 lbs, not including the remotely located 115+ lb compressor (an auxiliary diesel engine to drive the compressor is typically not required on the bus systems). Almost all of the 350 to 700 lb roof mounted hardware (condenser, evaporator and such) would still be required for the adsorption compression system, but the mechanical compressor weight has potential for elimination. In the case of the conventional system on a large refrigerated trailer, which typically utilizes a 25 to 35 h.p. diesel engine to drive the compressor, the system weight can be in the order of 1600 lbs (reference Thermo King Super II-190 at 1495 lbs and the SB-200 or SB-300 at 1635 lbs). In the case of a refrigerated trailer operating with an adsorption compression system, without any back-up mechanical compressor and auxiliary diesel compressor drive engine, the system weight without consideration of the sorbent compressor weight (cold end only) could be in the 350 to 650 lb range. But the system weight saved by the elimination of the mechanical compressor and diesel drive engine is more or less regained with the added sorbent compressor assembly weight. A tabulated summary of system component weights and a chart comparing refrigerated trailer total system weights is provided. The adsorption system weight data is for the higher performing DCL bed designs. Considering an R134A system, weight can exceed 4,000 lbs for a 120,000 BTU/hr larger cooling capacity system (required of large buses). Only the reasonably sized R134A system mass for cooling capacities of 64,000 and 30,000 BTU/hr are shown in the comparison chart and tables. Note that CSW is nomenclature for ‘conventional system weight’ and ASW is ‘adsorption system weight’.

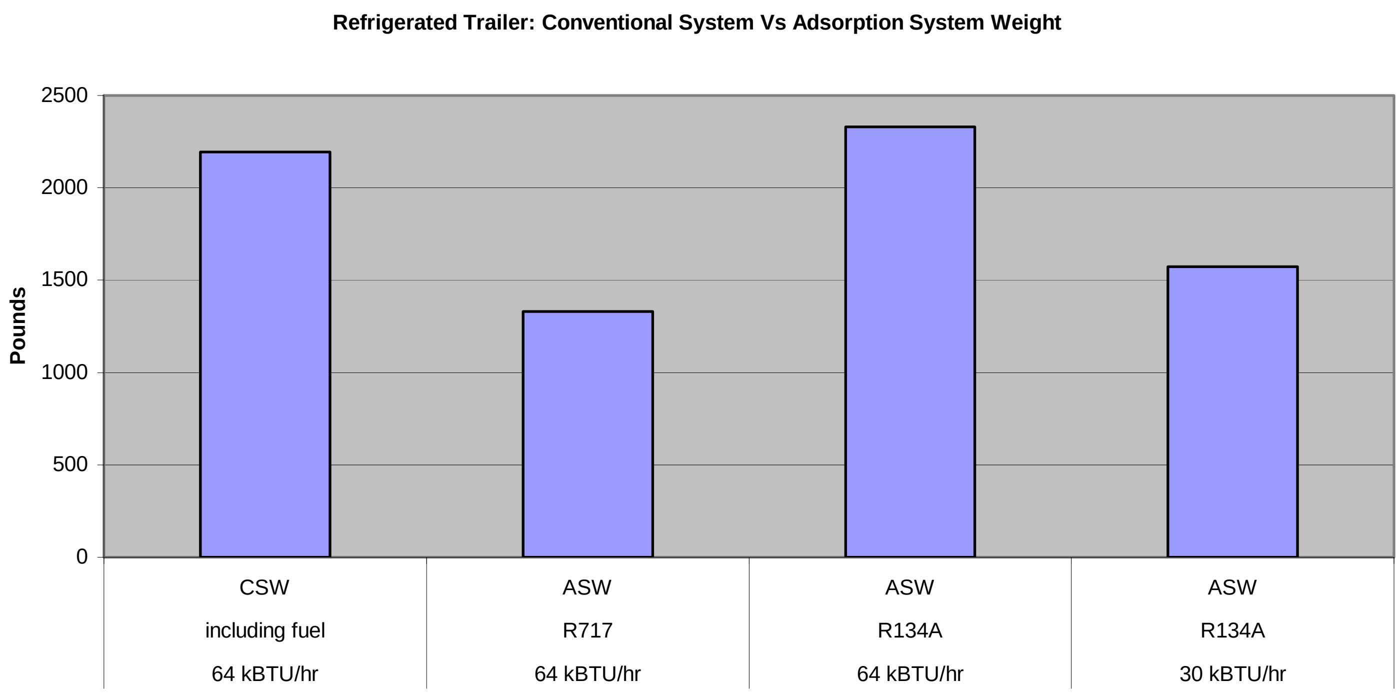


Table 6. Refrigerated Trailer System Weights Summary

Component Description	Conventional System (lbs)	R717 64 kBTU/hr Adsorption System (lbs)	R134A 64 kBTU/hr Adsorption System (lbs)	R134A 30 kBTU/hr Adsorption System (lbs)
Typ. Ref. Components w/o Mech. Compressor	600	600	600	600
Mech. Comp.	150	N/R	N/R	N/R
Aux. Diesel Engine	650	N/R	N/R	N/R
Diesel Fuel Tank	45	N/R	N/R	N/R
100 gal. Fuel	750	N/R	N/R	N/R
Sorbent Bed Assembly	N/R	480	1480	722
Sorbent Bed Manifold incl Fans & Valves	N/R	250	250	250
Total System Weights	**2195**	**1330**	**2330**	**1572**

Table 7. Bus System Weights Summary

Component Description	Conventional System (lbs)	R717 120 kBTU/hr Adsorption System (lbs)	R134A 64 kBTU/hr Adsorption System (lbs)	R134A 30 kBTU/hr Adsorption System (lbs)
Typ. Ref. Components w/o Mech. Compressor	600	600	600	600
Mech. Comp.	150	N/R	N/R	N/R
25 gal. Fuel	180	N/R	N/R	N/R
Sorbent Bed Assembly	N/R	918	1480	722
Sorbent Bed Manifold incl Fans & Valves	N/R	250	250	250
Total System Weights	**930**	**1768**	**2330**	**1572**

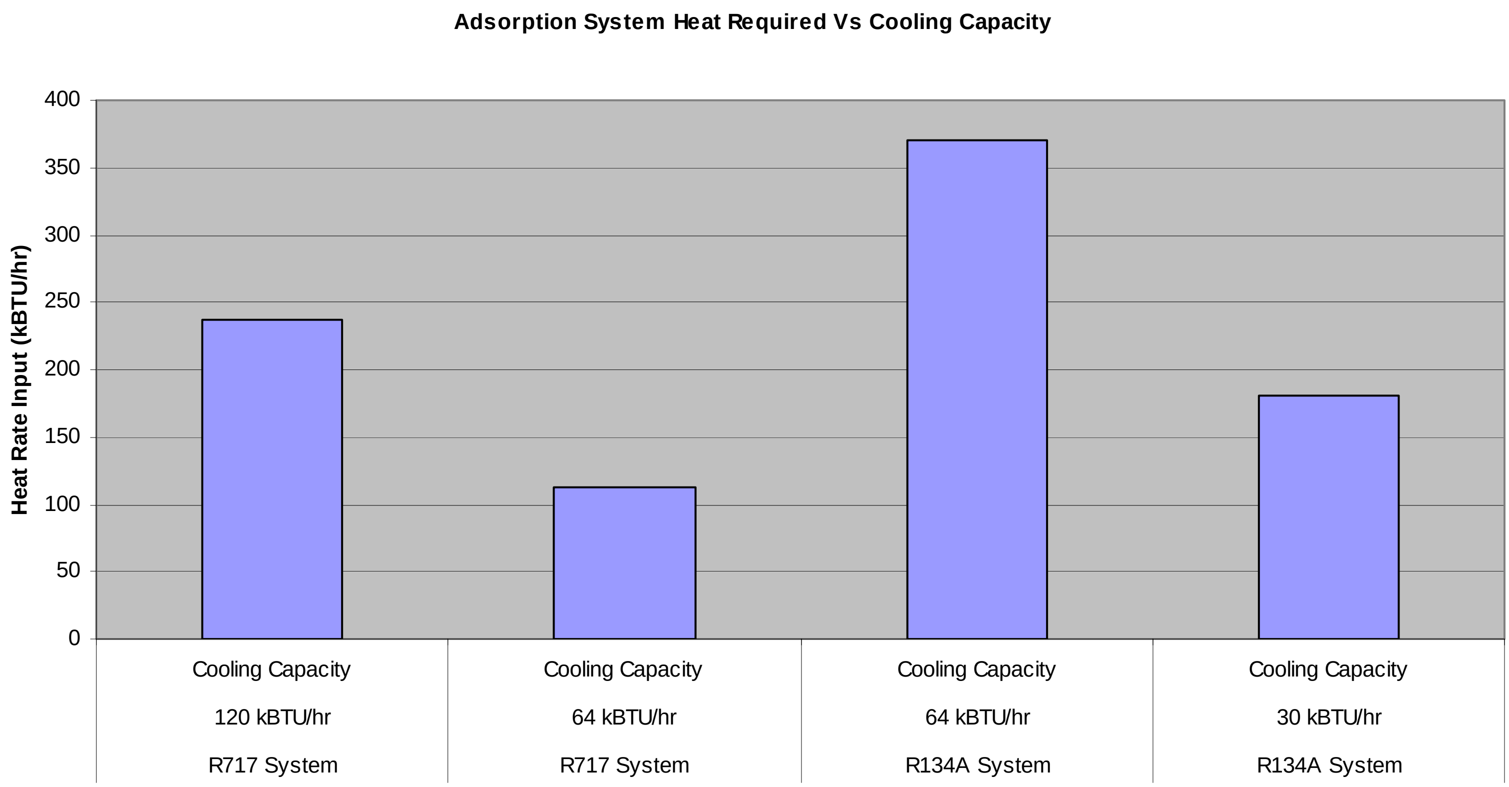
In summary, the truck trailer complete adsorption refrigeration system consisting of the typical refrigeration components and sorbent compressor would range from 1300 to 2300 lbs. The R717 adsorption system could save up to 800 lbs compared to the conventional trailer systems. The high output (120 kBTU/hr) R717 adsorption system for buses can have an increased weight over conventional bus systems by as much as 800 lbs. The R134A adsorption system weight is comparable to existing mechanical compression systems used for refrigerated trailers but it is at least 2.5 times heavier than mechanical compression systems used on large busses. Note that the bus system does not require the auxiliary diesel engine to drive the compressor, but instead, taps the required mechanical compressor shaft power off of the main engine (saving the weight of an auxiliary engine).

Refrigerated Trailer Payload Weight

Large diesel truck and trailer maximum loads are 80,000 lbs with the proper axle ratings. The truck/tractor and large 48+ ft trailer weight of 30,000 lb result in only a 50,000 lb payload. The R717 adsorption compressor refrigeration system, without any mechanical compressor, can potentially provide an increased payload from 1 to 2% by elimination of the mechanical compressor, auxiliary diesel engine and fuel tank. The weight of the adsorption compressor mounted on the trailer can also more evenly distribute weight across the sub floor of the trailer versus the localized/concentrated nature of the mechanical compressor, its auxiliary diesel engine, and the associated fuel tank. The even weight distribution can help from exceeding any one axle's limit.

Heat Input to Adsorption Compressor

This section could be considered the most important with respect to the feasibility of an adsorption compression system. Although such design issues as system weight and volume effect the attractiveness of implementation, the input heat required and the heat available can make or break any potential application. Fortunately, it has been determined that the large diesel engine exhaust heat rate does have the capability to operate sorbent bed compressors of the size needed for system cooling capacities required. The total heat required per a given cooling capacity based on a 1.0 COP was reported for an R717 system in the "Adsorption Compressor Basic Requirements" part of the "Performance Parameters" section. But, when the detailed bed designs of this section are modeled and analyzed, a lower COP in the order of 0.6 results for R717 systems (as expected without heat regeneration) and 0.2 for R134A systems. Evaluations of heat required to temperature cycle the compressor hardware mass and associated refrigerant mass flow for compressor detailed designs considered are shown in the following graph. A sorbent bed cycling temperature change of 300°F (100 – 400°F) was used in the evaluations; the analysis details for various bed designs are located in "Appendix B. - Sorbent Bed Design Evaluations". The system required heat rates of the following graph are for the higher performing DCL sorbent bed design.


Adsorption System Heat Required Vs Cooling Capacity
Heat Rate Input (kBTU/hr)
0
50
100
150
200
250
300
350
400
Cooling Capacity 120 kBTU/hr R717 System
Cooling Capacity 64 kBTU/hr R717 System
Cooling Capacity 64 kBTU/hr R134A System
Cooling Capacity 30 kBTU/hr R134A System

Per the values listed in Table 5 of available heat from exhaust, 135,000 BTU/hr to 290,000 BTU/hr are nominal values with less than 500 kBTU/hr maximum from the larger horsepower diesel engines at maximum operating conditions. Comparing these available exhaust heat values to the required input heat of the above graph, the lower performing R134A system design is feasible with the lower cooling capacity system and is marginal at the 64,000 BTU/hr system size. The higher-performing R717 system shows to be feasible at even the highest cooling capacity output of 120,000 BTU/hr. These heat input values are determined using the stainless steel construction design required of a direct exhaust heating concept. If the sorbent bed was heated and cooled by use of a working fluid (which extracts heat from exhaust source and transfers the heat to the bed) compatible with a lower density material such as aluminum, then the required heat input can be reduced. An evaluation of a 64,000 BTU/hr cooling capacity R134A system with sorbent beds constructed of aluminum shows a heat input of 222,000 BTU/hr (ref Appendix B). Although this would require the auxiliary 'Heating and Cooling Fluid Subsystem' with an ambient radiator, it does provide feasibility of the larger cooling capacity R134A system. As mentioned earlier, a heating and cooling heat transfer fluid subsystem would also allow a 'heat regeneration system' to be utilized, but the added hardware volume, weight and valve control might offset the benefits of the reduced heat input required.

The above energy balance of available exhaust heat and sorbent bed required heat does indicate reasonable feasibility. Yet another evaluation was considered as to the potential forced convective heat transfer quantities with respect to the exhaust direct heating and ambient air-cooling flows applied to the bed designs (ref Appendix C). Per evaluations of bed designs in Appendix B, a forced convection heat transfer coefficient ranging from 12 to 24 BTU/hr/sqft/F is required. Using exhaust flow conditions in the case of heating and a 2400 SCFM forced ambient airflow for cooling, the heat into and out of the sorbent beds match the requirements of the convective heat transfer coefficient ranging from 25 to 39 BTU/hr/sqft/F. It is worth noting that although a larger bed diameter does reduce exhaust flow backpressure and allows more sorbent mass density per unit length of bed, it has a negative effect for heat transfer. The lower flow velocities associated with the larger flow area lower the film conductance and heat transfer rates. The thought of adding a flow volume void or solid to reduce flow cross-sectional area is an option. But a material that can withstand the direct exhaust temperatures would also have an undesirable heat content to be temperature cycled.

Conclusions and Recommendations

Throughout this research effort, no obvious or notable showstoppers were determined with respect to the feasibility of an adsorption system to be utilized for A/C on a bus or for a refrigerated trailer. In fact, most all the evaluations of this report were conservative, thereby giving additional margin to the feasibility of an adsorption system. Specifically, the system size, weight, and heat balance are all within reason of a feasible design. Of course the R717 system has a higher performance and therefore reduced size, weight and required heat input of that of a R134A system, but the hazard associated with an R717 ammonia system in a transit vehicle increases desire for the user friendly R134A system.

Continued efforts to implement an adsorption refrigeration system in large transit vehicle applications could prove to provide multiple benefits. One major benefit reported is the elimination of diesel exhaust emissions associated with the power required to drive the conventional mechanical compressor, and the resulting savings of fuel and natural resources. Also of note are the safety benefits of the potential removal of the auxiliary diesel fuel tank mounted on refrigerated trailers. A typical futuristic issue is conservation and preservation, which the fuel savings and eliminated emissions also support. Modularity is seen as a feature of the future; the direct exhaust heating sorbent compressor assembly is a modular unit versus the conventional mechanical compressor, diesel drive engine and fuel tank associated with the refrigerated trailer. Modularity is associated with ease of maintenance and operation. Even the complicated subsystems of the conventional compressor's diesel drive engine (subsystems such as fuel, oil, cooling, electrical, exhaust, etc, which include a large part count) make the sorbent compressor a relatively simple unit in comparison. If this new refrigeration system design concept is taken up to a hardware prototype/demonstration design with successful results, a new industry could materialize. This technology could then flow into other vehicle applications such as use on airplanes (APU exhaust heat), locomotives, lower acoustic signals desired on submarines, or even future space craft.

It is recommended that a 'direct exhaust heating and forced ambient air cooling' sorbent bed compressor prototype hardware unit be built and tested to better verify the system performance with respect to cooling capacity and heat rates. A working model that could provide proven performance data could potentially increase the 'technology readiness level' reducing risk and stimulating further development and ultimately, implementation.

The Specific Cooling Power (SPC) of the sorbent material was determined for two practical sorbent bed assembly configurations (the higher performing DCL design and the lower performing SCL design). A detailed analysis was also performed on the performance for two practical bed structure materials. One is robust stainless steel and the other material aluminum. It was determined that the lower performing R134A system (compared to R717, ammonia) is limited to a cooling capacity of less than 60,000 BTU/hr with a stainless steel construction (due to excessive weight and heat input required beyond that of available exhaust heat). An R134A system with a cooling capacity greater than

60,000 BTU/hr is required to be an aluminum structure which also requires a special heating and cooling working fluid compatible with aluminum (versus the direct exhaust heating simpler stainless steel design).

However, there are more questions we need to ask in order to assess the feasibility of the adsorption refrigeration system concept utilizing waste exhaust heat to provide cooling for refrigerated trucks and transit buses. These design questions shall be considered for future research work in this area and are listed below.

Design questions such as:

1. Determining the details of a control system required for heating the sorbent beds and operating the potential exhaust bypass valve(s).
2. Detail design of support structure to suspend sorption material portion of bed.
3. Detail design and operations required of the packaging of the sorbent material. Address design qualification tests such as vibration and shock, and thermal cycle tests.
4. Design considerations to install filters or screens to remove chances of the sorbent material to migrate into critical components such as the systems required check valves and / or other components that particulate could cause malfunction or internal valve leakage.
5. Extra effort is required to identify the more detailed cost and safety benefits offered by the sorption system beyond the general qualitative and briefly determined quantitative statements of fuel savings and reductions of exhaust emissions (see Application – Environmental Impact section of this report for some details of reduction in emissions).
6. Performing a thorough Failure Modes and Effects Analysis and Hazards Analysis would be an effective tool for the practical design of the unit.

References

1. “Adsorption Air Conditioning for Containerships and Vehicles”, Craig Christy, Dave Fusco and Reza Toossi; METRANS contract number 65A0047 June 2001

1.1 J.W. McBain, *The Sorption of Gases and Vapours by Solids*. Routledge, London, 1932, Chap. 5

2. Dow Chemical Company – A Guild to Glycols

3. Keenan and Keyes, *Thermodynamic Properties of Steam*, Wiley 1936

4. GenCorp Aerojet’s Final Report No. 10875, Advanced Rail Vehicle Air Conditioning Project, Final Report, Advanced Transit Products Development Program, Los Angeles Standard Light Rail Vehicle – November 1996.

5. Jack A. Jones, “Carbon/Ammonia Regenerative Adsorption Heat Pump” presented at ASME International Heat Pump Conference, New Orleans, Louisiana, January 1994.

6. Li Zhi Zhang & Ling Wang; Performance Estimation of an Adsorption Cooling System for Automobile Waste Heat Recovery 26 April 1997 – Applied Thermal Engineering Vol. 17 No.12 pp. 1127-1139

7. Edward F. Obert, Internal Combustion Engines and Air Pollution 1973

8. Wabash National Corporation:
www.wabashnational.com/products/features/solarguard.htm

9. Cummins Engine Company, Inc.; Shop Manual, M11 Series Engines 1994 Bulletin No 3666075-00 printed 11/94 (Section E, pages E-4)

10. Piping Engineering 2nd edition, published by Tube Turns Division of Chemetron Corporation 1971

Appendices

A. Definitions

A-1. Coefficient of Performance of an Adsorption Compression System

To facilitate a comparison of the performance between the conventional mechanical compressor refrigeration cycle system and the potentially upcoming adsorption compressor refrigeration system, we generalize the COP equation. The denominator of the COP equation shall be considered as the system's required energy input, as opposed to the limiting application of work input.

To support this project with adsorption compression, the COP is redefined into a more generalized form and is as follows

$$COP = Q_L / E_{input}$$

Specifically, in a mechanical compressor system

$$COP = |Q_L| / W$$ as per convention.

And in a sorption compression system

$$COP = |Q_L| / Q_{compressor}$$ per the generalized definition.

A-2. Total Mass of Sorbent Material in Experimental Systems Compressor

The experimental adsorption compression refrigeration system has four sorbent beds. Each bed has six 1.5 inch diameter x 36 inch long aluminum canisters. Each canister's interior is filled with three annular shaped aluminum foam sections packed with the sorbent carbon material. Each of the aluminum foam sections has a different mean radius such that there is an inner, mid and outer and each runs the length of the 36 inch long canister.

Each carbon impregnated sorbent canister aluminum section is referred to as a foam section. Each has an aluminum foam structure at 8% of the density of solid aluminum (0.1 lb/cu.), giving an aluminum foam section density of 0.008 lb/cu.in. There are three carbon impregnated aluminum foam layers each consisting of eight 4.5 inches long sections in each 36-inch long canister. A 4.5 inch-long section of the mid foam layer was investigated closely and has dimensions of 0.926 inch outside diameter and a 0.715 inch inside diameter. The net volume of each 4.5 inch long mid section is approximately 1.22 cu.in, giving each section an aluminum mass of 0.01 lbs (using the 8% aluminum foam density provided by Aerojet). The carbon mass per 4.5 inch-long mid section is 0.01 lbs per an actual weight measurement, performed once the carbon was removed from the foam. This gives a carbon density of 0.008 lbs/cu.in Therefore the mass of aluminum foam is approximately equal to the mass of sorbent carbon integral to the foam. The total mass of carbon in each mid layer per 36 inch long canister is 0.08 lbs

We now determine the total mass of aluminum foam in all three layers of each 36-inch long sorbent canister. The inner layer has an OD of 0.575 inches and an ID of 0.290 inches giving a total aluminum mass of 0.06 lbs in which the sorbent carbon mass is considered to be 0.06 lbs per canister. The outer layer has an OD of 1.348 inches and an ID of 1.040 inches, giving a total aluminum mass of 0.15 lbs in which the sorbent carbon mass is considered to be 0.15 lbs per canister. Summing up all tree carbon layers per canister, the total mass of carbon per canister is 0.3 lbs.

There are 6 sorbent canisters (36 inch tubes) per bed and 4 beds per the complete adsorption compressor. That totals 24 sorbent canisters, each 36 inches long, in the adsorption compressor. The adsorption compressor has a total mass of carbon of 7 pounds. During the refurbishment of the phase I effort, one of the canisters (tubes) was determined to leak and was removed from the number 2 bed. Therefore the compressor tested had 23 canisters, which totals 6.7 lbs of carbon total.

Notes and an unofficial report obtained from Aerojet indicated a carbon mass of approximately 2 lbs per bed. Therefore, this information indicates 8 lbs of sorbent material in all four beds. This does correlate well with the 7.44 lbs determined with the above investigation.
NOTE / SUMMARY: Above investigation determined the carbon load density on the experimental sorbent bed to be 0.008 lbs/cu.in. This is half of that determined in the investigation discussed below. Error in the above evaluation could be due to a reduced / inaccurate weight of carbon caused by a failure to remove all carbon from the aluminum foam prior to weighing.

A second and different evaluation of the experiment's carbon load determined approximately 0.016 lb/cu.in carbon load density. The data used and excel spread sheet output is shown in appendix A-3. The 0.016 lb/cuin compares well with 0.013 lb/cuin predicted in a JPL paper (ref Jones[6]).

A-3. Carbon Mass per Volume in Experimental System – Actual Carbon Density

Experimental System Compressor Carbon Density
(given Al and Al Foam density, cross sectional dimensions and length, and mass of each inner, outer mid sections; mass of carbon per Al Foam Volume (density) is determined.)

Carbon sorbent outer OD	A1 OD =	1.31	in	Outer Foam w/ Al Tube at OD, M =	0.89	lb
Carbonsorbent outer ID	A1 ID =	1.03	in			
Carbon sorbent mid OD	A2 OD =	0.91	in	Mid Foam w/ Al Tube at OD, M =	0.37	lb
Carbon sorbent mid ID	A2 ID =	0.72	in			
Carbon sorbent inner OD	A3 OD =	0.58	in	Inner Foam w/ Al Tube at OD, M =	0.32	lb
Carbon sorbent inner ID	A3 ID =	0.28	in			

Length=	35	in	Outer Sorbent Volume, V1 =	17.945	cu in
No.Tubes =	1		Mid Sorbent Volume, V2 =	8.509	cu in
Outer area, A1 =	0.51	sq in	Inner Sorbent Volume, V3 =	7.089	cu in
Mid area, A2 =	0.24	sq in			
Inner area, A3 =	0.20	sq in	Outer Alum Foam Mass, M1 =	0.144	lbs
Aluminum density =	0.1	lb/cuin	Mid Alum Foam Mass,M2 =	0.068	lbs
Alum foam density =	0.01	lb/cuin	Inner Alum Foam Mass, M3 =	0.057	lbs
			Total Alum Foam Mass =	0.27	lbs
Alum tubing wall thickness =	0.03	in			
Outer Al tubing 1 mass =	0.360	lb	Outer Sorbent Mass =	0.387	lbs
Mid Al tubing 2 mass =	0.198	lb	Mid Sorbent Mass =	0.104	lbs
Inner Al tubing 3 mass =	0.159	lb	Inner Sorbent Mass =	0.104	lbs
Al outer shell mass =	0.41				
			Total Sorbent Mass =	0.59	lbs
Total sorbent, foam, tubing & shell =	1.99	lbs			

			Sorbent density 1 =	0.0215	lb/cuin
Total Sorbent per Bed =	3.57	lb	Sorbent density 2 =	0.0122	lb/cuin
Total Sorbent per 4bed =	14.26949	lb	Sorbent density 3 =	0.0147	lb/cuin
			Ave Sorbent Density =	0.0161	lb/cuin

B. Sorbent Bed Design Evaluation

Each of the following data sheets provides detailed data on the specific bed design option (DCL or SCL), refrigerant used (R717 or R134A), and cooling capacity. Each evaluation considers a four (4) bed adsorption compressor basic design. The detailed data consists of the required bed length to meet the desired carbon mass load corresponding to the cooling capacity specified, the independent mass of each type of material which makeup the bed design (steel, 8% aluminum foam, and carbon sorbent material). This includes the four bed total mass, approximate value of heat required to cycle the bed from 100°F to 400°F, the heat transfer surface area including the required forced convection heat transfer coefficient h, and the approximate dimensions and volume. The Dual Flow design option is the Dual Carbon Layer (DCL) bed design. The Single Flow design option is the Single Carbon Layer (SCL) bed design option.

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.
SS Design Option 2 **Dual Flow - DCL**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts (inner & outer flow paths for both exhaust heating and ambient air cooling of bed)(two sorbent layers)
Bus AC using **R717** with **120,000 BTU/hr** Cooling Capacity. **Sorbent Maximized.**

Sorbat flow space-void gap = 0.125 in   Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD	**A1 OD =**	8	in	Outer sorbent cross-sec. area, A1 =	10.40 sq in
Carbonsorbent outer ID	A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	4284.9 cu in
Carbon sorbent inner OD	A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85 sq in
Carbon sorbent inner ID	A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	3646.8 cu in
Outer heating fluid flow space / gap =		0.25	in		
Outer SS Shell Diameter	OS D =	8.5	in	Alum foam density =	0.008 lb/cuin
Bed Length=		103	in	Outer Alum Foam Mass, M1 =	34.3 lbs
Number of Beds =		4		Mid Alum Foam Mass,M2 =	29.2 lbs
				Total Alum Foam Mass =	63.5 lbs
Stainless steel (SS) density =		0.25	lb/cuin		
SS tubing wall thickness =		0.1	in	Sorbent mass density=	0.016 lb/cuin
corrosion allowance		0.03	in	Outer Sorbent Mass =	68.6 lbs
				Inner Sorbent Mass =	58.3 lbs
Total heating/cooling				**Total sorbent mass =**	126.9 lbs
heat transfer fluid surface area =		62.9	sq ft		
Total heating/cooling heat transfer fluid				Sorbent Outer SS tubing 1 mass =	262.0 lb
flow cross-sectional area per bed =		30.32	sqin	Sorbent Inner SS tubing 2 mass =	190.8 lb
				Outer Shell tubing mass =	274.9 lb
Total sorbent, alum. foam				**Total SS tubing mass =**	727.6 lb
and SS tubing & shell Mass =		**917.99**	**lbs**		

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R717=0.5 BTU/lb/F

Total heating power required using a 12 minute
complete cycle time and 200 F temperature change

Bed Hardware Heating = 115235 BTU/hr
Refrigerant Heating = 26651 BTU/hr
Total Heat Rate Required = 141886 BTU/hr

Req'd forced convection coef (deltaT=100), h = 15.043 BTU/hr/sqft/F

Total heating power required using a 12 minute
complete cycle and 300 F temperature change

Bed Hardware Heating = 172853 BTU/hr
Refrigerant Heating = 39976 BTU/hr
Total Heat Rate Required = 212829 BTU/hr

Req'd forced convection coef (deltaT=100), h = 22.565 BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Height= 10.7 inches   Width= 42.8   Cross-section Area, A = 458.0 sqin
Length = 103 inches
Total Volume = 27.3 sq ft

Adsorption Compressor mass, volume, heat transfer coef., and bed heat required for SCL.
SS Design Option 1 **Single Flow**: Heating/Cooling Fluid Flow Through Central Exhaust Duct
(one central flow path for exhaust heating and ambient cooling forced air flow)(one sorbent layer)
Bus AC using **R717** with **120,000 BTU/hr** Cooling Capacity. **Sorbent Maximized**.

Sorbat flow space - void gap = 0.125 in   Sorbent Thickness = 0.438 in

(one sorbent layer A1 on outside surface of inner tube with refrigerant flow space between sorbent layer and outer shell; exhaust flow thru inner area of inner tube)

Exhaust Inner Flow Pipe ID	ID =	6.0	in	sorbent cross-sectional area, A1 =	9.12	sq in
Carbon sorbent ID	A1 ID =	6.19	in	Sorbent Volume, V1 =	7841	cu in
Carbon sorbent OD	A1 OD =	7.07	in			
			in			
Outer SS Shell Diameter	OS ID =	7.3173	in	Alum foam density =	0.008	lb/cuin
	Bed Length=	215	in	**Alum Foam Mass =**	62.7	lbs
	Number of Beds =	4				
	Stainless steel (SS) density =	0.25	lb/cuin	Sorbent mass density=	0.016	lb/cuin
	SS tubing wall thickness =	0.096	in	**Sorbent Mass =**	125.5	lbs
	corrosion allowance	0.03	in			
	Total heating/cooling			Inner SS tubing 2 mass =	393.5	lb
	heat transfer fluid surface area =	56.258	sq ft	Outer Shell tubing mass =	472.4	lb
	Total heating/cooling heat transfer fluid			**Total SS tubing mass =**	865.9	lb
	flow cross-sectional area =	28.26	sqin			

Total sorbent, alum. foam and SS tubing & outershell Mass = 1054 lbs

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R717=0.5 BTU/lb/F

Total heating power required using a 12 minute total cycle time
with 200 F temperature cycling

Bed Hardware Heat Rate Requirement =	131507	BTU/hr
Refrigerant Heat Rate Requirement =	26345	BTU/hr
Total Heat Rate Required =	157852	BTU/hr

Req'd forced convection coef (deltaT=100),
h = 15.584 BTU/hr/sqft/F

Total heating power required using a 12 minute total cycle time
with 300 F temperature cycling

Bed Hardware Heat Rate Requirement =	197260	BTU/hr
Refrigerant Heat Rate Requirement =	39518	BTU/hr
Total Heat Rate Required =	236778	BTU/hr

Req'd forced convection coef (deltaT=100),
h = 23.375 BTU/hr/sqft/F

Compressor Cross-sectional Area & Volume

Height= 9.509 inches   Width= 38.034   Cross-section Area, A = 361.6 sqin

Length = 215 inches

Total Volume = 45.0 sq ft

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.
SS Design Option 2 **Dual Flow**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts
(inner & outer flow paths for both exhaust heating and ambient air cooling of bed)(two sorbent layers)
Refrigerated Trailer using **R717** with **64,000 BTU/hr** Cooling Capacity. **Sorbent Maximized.**

Sorbat flow space-void gap = 0.125 in   Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD	**A1 OD =**	8	in	Outer sorbent cross-sec. area, A1 =	10.40	sq in
Carbonsorbent outer ID	A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	2246.4	cu in
Carbon sorbent inner OD	A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85	sq in
Carbon sorbent inner ID	A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	1911.9	cu in
Outer heating fluid flow space / gap =		0.25	in			
Outer SS Shell Diameter	OS ID =	8.5	in	Alum foam density =	0.008	lb/cuin
	Bed Length=	54	in	Outer Alum Foam Mass, M1 =	18.0	lbs
	Number of Beds =	4		Mid Alum Foam Mass,M2 =	15.3	lbs
				Total Alum Foam Mass =	33.3	lbs
Stainless steel (SS) density =		0.25	lb/cuin			
SS tubing wall thickness =		0.1	in	Sorbent mass density=	0.016	lb/cuin
corrosion allowance		0.03	in	Outer Sorbent Mass =	35.9	lbs
				Inner Sorbent Mass =	30.6	lbs
Total heating/cooling				**Total sorbent mass =**	66.5	lbs
heat transfer fluid surface area =		33.0	sq ft			
Total heating/cooling heat transfer fluid				Sorbent Outer SS tubing 1 mass =	137.3	lb
flow cross-sectional area per bed =		30.32	sqin	Sorbent Inner SS tubing 2 mass =	100.0	lb
				Outer Shell tubing mass =	144.1	lb
Total sorbent, alum. foam				**Total SS tubing mass =**	381.5	lb
and SS tubing & shell Mass =		**481.28**	**lbs**			

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R717=0.5 BTU/lb/F

Total heating power required using a 12 minute complete cycle time and 200 F temperature change

Bed Hardware Heating =	60415	BTU/hr
Refrigerant Heating =	13972	BTU/hr
Total Heat Rate Required =	74387	BTU/hr

Req'd forced convection coef (deltaT=150), h = 15.043 BTU/hr/sqft/F

Total heating power required using a 12 minute complete cycle and 300 F temperature change

Bed Hardware Heating =	90622	BTU/hr
Refrigerant Heating =	20958	BTU/hr
Total Heat Rate Required =	111580	BTU/hr

Req'd forced convection coef (deltaT=150), h = 22.565 BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Cross-section Area, A =	458.0	sqin
Cross-section dimensions =	21.4	inches on each side (square cross-section))
Length =	54	inches
Total Volume =	14.3	sq ft

SCL

Adsorption Compressor mass, volume, heat transfer coef., and bed heat required.
SS Design Option 1 **Single Flow**: Heating/Cooling Fluid Flow Through Central Exhaust Duct
(one central flow path for heating exhaust and ambient cooling forced air flow)(one sorbent layer)
Refrigerated Trailer using **R717** with **64,000 BTU/hr** Cooling Capacity. **Sorbent maximized**.

Sorbat flow space - void gap = 0.125 in   Sorbent Thickness = 0.438 in

(one sorbent layer A1 on outside surface of inner tube with refrigerant flow space between sorbent layer and outer shell; exhaust flow thru inner area of inner tube)

Exhaust Inner Flow Pipe ID	ID =	6.0	in	sorbent cross-sectional area, A1 =	9.12	sq in
Carbon sorbent ID	A1 ID =	6.19	in	Sorbent Volume, V1 =	4194	cu in
Carbon sorbent OD	A1 OD =	7.07	in			
			in			
Outer SS Shell Diameter	OS ID =	7.3173	in	Alum foam density =	0.008	lb/cuin
	Bed Length=	115	in	**Alum Foam Mass =**	33.6	lbs
	Number of Beds =	4				
	Stainless steel (SS) density =	0.25	lb/cuin	Sorbent mass density=	0.016	lb/cuin
	SS tubing wall thickness =	0.096	in	**Sorbent Mass =**	67.1	lbs
	corrosion allowance	0.03	in			
	Total heating/cooling heat transfer fluid surface area =	30.092	sq ft	Inner SS tubing 2 mass =	210.5	lb
				Outer Shell tubing mass =	252.7	lb
	Total heating/cooling heat transfer fluid flow cross-sectional area =	28.26	sqin	**Total SS tubing mass =**	463.1	lb

Total sorbent, alum. foam and SS tubing & outershell Mass = 563.80 lbs

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R717=0.5 BTU/lb/F

Total heating power required using a 12 minute total cycle time
with 200 F temperature cycling

Bed Hardware Heat Rate Requirement =	70341	BTU/hr
Refrigerant Heat Rate Requirement =	14092	BTU/hr
Total Heat Rate Required =	84432	BTU/hr

Req'd forced convection coef (deltaT=100),
h = 15.584 BTU/hr/sqft/F

Total heating power required using a 12 minute total cycle time
with 300 F temperature cycling

Bed Hardware Heat Rate Requirement =	105511	BTU/hr
Refrigerant Heat Rate Requirement =	21138	BTU/hr
Total Heat Rate Required =	126649	BTU/hr

Req'd forced convection coef (deltaT=100), h = 23.375 BTU/hr/sqft/F

Compressor Cross-sectional Area & Volume

Cross-section Area, A = 276.7 sqin
Cross-section dimensions = 16.6 inches on each side (square cross-section)
Length = 115 inches
Total Volume = 18.4 cu ft

R134A 64 kBTU/hr

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.

SS Design Option 2 **Dual Flow - DCL**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts
(both inner & outer flow paths for exhaust heating and ambient air cooling of bed)

Refrigerated Trailer using **R134A** with **64,000 BTU/hr** Cooling Capacity. **Maximumized Sorbent Capacity.**

Sorbat flow space-void gap = 0.125 in Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD	**A1 OD =**	8	in	Outer sorbent cross-sec. area, A1 =	10.40	sq in
Carbonsorbent outer ID	A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	6905.7	cu in
Carbon sorbent inner OD	A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85	sq in
Carbon sorbent inner ID	A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	5877.4	cu in
Outer heating fluid flow space / gap =		0.25	in			
Outer SS Shell Diameter	OS D =	8.5	in	Alum foam density =	0.008	lb/cuin
Bed Length=		166	in	Outer Alum Foam Mass, M1 =	55.2	lbs
Number of Beds =		4		Mid Alum Foam Mass,M2 =	47.0	lbs
				Total Alum Foam Mass =	102.3	lbs
Stainless steel (SS) density =		0.25	lb/cuin			
SS tubing wall thickness =		0.1	in	Sorbent mass density=	0.016	lb/cuin
corrosion allowance		0.03	in	Outer Sorbent Mass =	110.5	lbs
				Inner Sorbent Mass =	94.0	lbs
Total heating/cooling heat transfer fluid surface area =		101.3	sq ft	**Total sorbent mass =**	204.5	lbs
Total heating/cooling heat transfer fluid flow cross-sectional area per bed =		30.32	sqin	Sorbent Outer SS tubing 1 mass =	422.2	lb
				Sorbent Inner SS tubing 2 mass =	307.4	lb
				Outer Shell tubing mass =	443.1	lb
Total sorbent, alum. foam and SS tubing & shell Mass =		**1479.48**	**lbs**	**Total SS tubing mass =**	1172.7	lb

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R134A=0.32 BTU/lb/F

Total heating power required using a 12 minute
complete cycle time and 200 F temperature change

Bed Hardware Heating = 185719 BTU/hr

(ref spec. heat)x(ref mass flow)x(deltaT) Refrigerant Heating = 60083 BTU/hr

Total Heat Rate Required = 245802 BTU/hr

Req'd forced convection coef (deltaT=100), h = 12.128 BTU/hr/sqft/F

Total heating power required using a 12 minute
complete cycle and 300 F temperature change

Bed Hardware Heating = 278578 BTU/hr

Refrigerant Heating = 90124 BTU/hr

Total Heat Rate Required = 368703 BTU/hr

Req'd forced convection coef (deltaT=100), h = 18.192 BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Cross-section Area, A = 361.0 sqin
Cross-section dimensions = 19.0 on each side (square cross-section))
Length = 166 inches
Total Volume = 34.7 sq ft

R134 30KBTU/hr

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.

SS Design Option 2 **Dual Flow - DCL**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts
(both inner & outer flow paths for exhaust heating and ambient air cooling of bed)

Refrigerated Trailer using **R134A** with **30,000 BTU/hr** Cooling Capacity. **Maximumized Sorbent Capacity.**

Sorbat flow space-void gap = 0.125 in Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD A1 OD =	**8**	in	Outer sorbent cross-sec. area, A1 =	10.40	sq in
Carbonsorbent outer ID A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	3369.7	cu in
Carbon sorbent inner OD A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85	sq in
Carbon sorbent inner ID A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	2867.9	cu in
Outer heating fluid flow space / gap =	0.25	in			
Outer SS Shell Diameter OS D =	8.5	in	Alum foam density =	0.008	lb/cuin
Bed Length=	81	in	Outer Alum Foam Mass, M1 =	27.0	lbs
Number of Beds =	4		Mid Alum Foam Mass,M2 =	22.9	lbs
			Total Alum Foam Mass =	49.9	lbs
Stainless steel (SS) density =	0.25	lb/cuin			
SS tubing wall thickness =	0.1	in	Sorbent mass density=	0.016	lb/cuin
corrosion allowance	0.03	in	Outer Sorbent Mass =	53.9	lbs
			Inner Sorbent Mass =	45.9	lbs
Total heating/cooling			**Total sorbent mass =**	99.8	lbs
heat transfer fluid surface area =	49.4	sq ft			
Total heating/cooling heat transfer fluid			Sorbent Outer SS tubing 1 mass =	206.0	lb
flow cross-sectional area per bed =	30.32	sqin	Sorbent Inner SS tubing 2 mass =	150.0	lb
			Outer Shell tubing mass =	216.2	lb
Total sorbent, alum. foam			**Total SS tubing mass =**	572.2	lb
and SS tubing & shell Mass =	**721.92**	**lbs**			

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; SS=0.12 BTU/lb/F; Carbon=0.17 BTU/lb/F, R134A=0.32 BTU/lb/F

Total heating power required using a 12 minute
complete cycle time and 200 F temperature change

	Bed Hardware Heating =	90622	BTU/hr
(ref spec. heat)x(ref mass flow)x(deltaT)	Refrigerant Heating =	29318	BTU/hr
	Total Heat Rate Required =	119939	BTU/hr

Req'd forced convection coef (deltaT=100), h = 12.128 BTU/hr/sqft/F

Total heating power required using a 12 minute
complete cycle and 300 F temperature change

Bed Hardware Heating =	135933	BTU/hr
Refrigerant Heating =	43976	BTU/hr
Total Heat Rate Required =	179909	BTU/hr

Req'd forced convection coef (deltaT=100), h = 18.192 BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Cross-section Area, A =	361.0	sqin
Cross-section dimensions =	19.0	on each side (square cross-section))
Length =	81	inches
Total Volume =	16.9	sq ft

R134A 64 kBTU/hr Aluminum Sorbent Beds

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.

Alum. Design Option 2 **Dual Flow - DCL**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts
(both inner & outer flow paths for exhaust heating and ambient air cooling of bed)

Refrigerated Trailer using **R134A** with **64,000 BTU/hr** Cooling Capacity. **Maximumized Sorbent Capacity.**

Sorbat flow space-void gap = 0.125 in   Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD	**A1 OD =**	8	in	Outer sorbent cross-sec. area, A1 =	10.40	sq in
Carbonsorbent outer ID	A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	6739.3	cu in
Carbon sorbent inner OD	A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85	sq in
Carbon sorbent inner ID	A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	5735.8	cu in
Outer heating fluid flow space / gap =		0.25	in			
Outer Al Shell Diameter	OS D =	8.5	in	Alum foam density =	0.008	lb/cuin
Bed Length=		162	in	Outer Alum Foam Mass, M1 =	53.9	lbs
Number of Beds =		4		Mid Alum Foam Mass,M2 =	45.9	lbs
				Total Alum Foam Mass =	99.8	lbs
Aluminum density =		0.1	lb/cuin			
Al tubing wall thickness =		0.1	in	Sorbent mass density=	0.016	lb/cuin
corrosion allowance		0.03	in	Outer Sorbent Mass =	107.8	lbs
				Inner Sorbent Mass =	91.8	lbs
Total heating/cooling				**Total sorbent mass =**	199.6	lbs
heat transfer fluid surface area =		98.9	sq ft			
Total heating/cooling heat transfer fluid				Sorbent Outer Al tubing 1 mass =	164.8	lb
flow cross-sectional area per bed =		30.32	sqin	Sorbent Inner Al tubing 2 mass =	120.0	lb
				Outer Shell tubing mass =	173.0	lb
Total sorbent, alum. foam				**Total Al tubing mass =**	457.8	lb
and Al tubing & shell Mass =		**757.17**	**lbs**			

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; Carbon=0.17 BTU/lb/F, R134A=0.32 BTU/lb/F

Total heating power required using a 12 minute
complete cycle time and 200 F temperature change

	Bed Hardware Heating =	98845	BTU/hr
(ref spec. heat)x(ref mass flow)x(deltaT)	Refrigerant Heating =	58635	BTU/hr
	Total Heat Rate Required =	157480	BTU/hr
Req'd forced convection coef (deltaT=100),	h =	7.962	BTU/hr/sqft/F

Total heating power required using a 12 minute
complete cycle and 300 F temperature change

	Bed Hardware Heating =	148267	BTU/hr
	Refrigerant Heating =	87953	BTU/hr
	Total Heat Rate Required =	236220	BTU/hr
Req'd forced convection coef (deltaT=100),	h =	11.943	BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Cross-section Area, A =	361.0	sqin
Cross-section dimensions =	19.0	on each side (square cross-section))
Length =	162	inches
Total Volume =	33.8	sq ft

R134A 30 kBTU/hr Aluminum Sorbent Beds.

Adsorption Compressor mass,volume, heat transfer coef., and bed heat required.

Alum. Design Option 2 **Dual Flow - DCL**: Heating/Cooling Fluid Flow Through Concentric Annular Ducts
(both inner & outer flow paths for exhaust heating and ambient air cooling of bed)

Refrigerated Trailer using **R134A** with **30,000 BTU/hr** Cooling Capacity. **Maximumized Sorbent Capacity.**

Sorbat flow space-void gap = 0.125 in   Sorbent Thickness = 0.438 in x 2

(one sorbent layer A1 on inside surface of outer tube & one sorbent layer A2 on outside surface of inner tube with refrigerant flow space between; exhaust flow thru inner area of inner tube and between outer tube and an outer shell)

Carbon sorbent outer OD A1 OD =	8	in	Outer sorbent cross-sec. area, A1 =	10.40	sq in
Carbonsorbent outer ID A1 ID =	7.124	in	Outer Sorbent Volume, V1 =	3369.7	cu in
Carbon sorbent inner OD A2 OD =	6.874	in	Inner sorbent cross-sec. area, A2 =	8.85	sq in
Carbon sorbent inner ID A2 ID =	5.998	in	Inner Sorbent Volume, V2 =	2867.9	cu in
Outer heating fluid flow space / gap =	0.25	in			
Outer Al Shell Diameter OS D =	8.5	in	Alum foam density =	0.008	lb/cuin
Bed Length=	81	in	Outer Alum Foam Mass, M1 =	27.0	lbs
Number of Beds =	4		Mid Alum Foam Mass,M2 =	22.9	lbs
			Total Alum Foam Mass =	49.9	lbs
Aluminum density =	0.1	lb/cuin			
Al tubing wall thickness =	0.1	in	Sorbent mass density=	0.016	lb/cuin
corrosion allowance	0.03	in	Outer Sorbent Mass =	53.9	lbs
			Inner Sorbent Mass =	45.9	lbs
Total heating/cooling			**Total sorbent mass =**	99.8	lbs
heat transfer fluid surface area =	49.4	sq ft			
Total heating/cooling heat transfer fluid			Sorbent Outer Al tubing 1 mass =	82.4	lb
flow cross-sectional area per bed =	30.32	sqin	Sorbent Inner Al tubing 2 mass =	60.0	lb
			Outer Shell tubing mass =	86.5	lb
Total sorbent, alum. foam			**Total Al tubing mass =**	228.9	lb
and Al tubing & shell Mass =	**378.59**	**lbs**			

Heat Rate Required

Specific Heat: Alum=0.1BU/lb/F; Carbon=0.17 BTU/lb/F, R134A=0.32 BTU/lb/F

Total heating power required using a 12 minute
complete cycle time and 200 F temperature change

	Bed Hardware Heating =	49422	BTU/hr
(ref spec. heat)x(ref mass flow)x(deltaT)	Refrigerant Heating =	29318	BTU/hr
	Total Heat Rate Required =	78740	BTU/hr

Req'd forced convection coef (deltaT=100), h = 7.962 BTU/hr/sqft/F

Total heating power required using a 12 minute
complete cycle and 300 F temperature change

Bed Hardware Heating =	74134	BTU/hr
Refrigerant Heating =	43976	BTU/hr
Total Heat Rate Required =	118110	BTU/hr

Req'd forced convection coef (deltaT=100), h = 11.943 BTU/hr/sqft/F

Four Bed Compressor Cross-sectional Area & Volume

Cross-section Area, A =	361.0	sqin
Cross-section dimensions =	19.0	on each side (square cross-section))
Length =	81	inches
Total Volume =	16.9	sq ft

C. Heat Transfer Analysis

Exhaust Flow Heat Transfer to Center Exhaust Flow Tube

List of analysis parameters:

Pressure=	30	psia (approx 2 ATM)	
Temperature =	400	F	
Diameter, d =	6	inch	
Conductivity of air @ 400F, K=	0.0223	BTU/(hr ft F)	(ref J.P. Holman, "Heat Exchangers" fig.1-4)
Viscosity of air @ 400F, u =	0.0622	lbm/(hr ft)	(ref Max Jakob, "Elements of Heat Transfer and Insulation" table II-2)
Specific Heat of air @ 400F, Cp =	0.245	BTU/(lbm F)	(ref Max Jakob, "Elements of Heat Transfer and Insulation" table II-2)

Dimensionless Fluid Flow & Heat Transfer Numbers:

Reynolds number, Re = 630000

Prandtl number, Pr = 0.6834

Nusselt number, Nu = 894.53 for heating, Nu = 0.023*(Re**0.8)(Pr**0.3)

(ref. Frank P. Incropera, et all; Fundamentals of Heat and Mass Transfer 1985 Ch8)

Considering Nu = hd/k , where h is the forced convection constant, and rearranging gives h=Nuk/d

h = 39.896 Btu/(hr sqft F)

Heat transfer per unit length, q/L = h Pi d (Tex - Tw),
where Tex is exhaust temperature and Tw is mean temperature of tube wall.

q/L = 9395.5 Btu/(hr ft) where Tex= 400 F and Tw = 250 F

Exhaust Heat Transfer to Annulus of DCL Design

Outer shell surface insulated.
List of analysis parameters:

Pressure=	30	psia (approx 2 ATM)	
Temperature =	400	F	
Inner annulus wall OD =	8	inch	
Hyd. Diameter, Dh =	0.5	inch	Dh =Do - Di = 8.5 - 8.0
Conductivity of ex. air @ 400F, K=	0.0223	BTU/(hr ft F)	(ref J.P. Holman, "Heat Exchangers" fig.1-4)
Viscosity of ex. air @ 400F, u =	0.0622	lbm/(hr ft)	(ref Max Jakob, "Elements of Heat Transfer and Insulation" table II-2)
Specific Heat of ex. air @ 400F, Cp =	0.245	BTU/(lbm F)	(ref Max Jakob, "Elements of Heat Transfer and Insulation" table II-2)

Dimensionless Fluid Flow & Heat Transfer Numbers:

Reynolds number, Re = 27616 high annulus mass flow

Prandtl number, Pr = 0.6834

Nusselt number, Nu = 73.291 for heating, Nu = 0.023*(Re**0.8)(Pr**0.3)

(ref. Frank P. Incropera, et all; Fundamentals of Heat and Mass Transfer 1985 Ch8)

Considering Nu = hd/k , where h is the forced convection constant, and rearranging gives h=Nuk/d

h = 39.22546 Btu/(hr sqft F)

Heat transfer per unit length, q/L = h Pi d (Tex - Tw),
where Tex is exhaust temperature and Tw is mean temperature of the wall.

q/L = 12316.79 Btu/(hr ft) where Tex= 400 F and Tw = 250 F

Reynolds number, Re = 7408 lower annulus mass flow

Prandtl number, Pr = 0.6834

Nusselt number, Nu = 25.579

h = 13.69 Btu/(hr sqft F)

q/L = 4298.6 Btu/(hr ft)

D. Rough Order of Magnitude Sizing from Experimental System

Simple Size Scaling of the Experimental System – Rough Estimate

The experimental system's size is scaled to larger sizes required for truck trailer and bus applications. Each experimental sorbent compressor bed has six 36 inch-long sorbent bed canisters. One bed's overall dimensions are 5"x7"x36". Four beds make-up the total adsorption compressor, resulting in a total size of 10"x14"x36", which is a volume of approximately 3 cu.ft. The refrigeration output of this experimental unit was approximately 13,000 Btu/hr with a heat input rate of 10,000 Btu/hr. Control of the bed temperature cycling was improved during phase II, which most likely helped the system performance compared to phase I efforts. This experimental data is used to extrapolate to size a system comparable to that used on a large truck refrigerated container/trailer that has been determined to require a refrigeration system capable of producing 46,000 Btu/hr cooling at 35°F. A sorbent compressor size of approximately 3 to 4 times that of the 3 cu.ft. compressor used in our experiment would be required. That equates to an adsorption compressor size of 12 cu.ft. A large refrigerated container, at 53 feet, typically has a volume of 5000 to 6000 cu.ft. of which an adsorption compressor volume of 12 cu.ft. is less than 1% of the usable volume.

The sorbent mass required would be 57 lbs. This rough evaluation compares well with the detailed evaluation section.

Using the experimental data and scaling up, the heat required could be in the order of 40,000 Btu/hr. It has been determined in the Performance Parameters section that a large diesel truck motor exhaust could provide 250,000 BTU/hr from a 225 h.p. engine and as much or more than 450,000 Btu/hr for the larger h.p. engines. A large truck's cooling water could provide approximately 158,400 Btu/hr but at a much lower temperature of 180 F.

This above simple scaling is preceded by a more detailed evaluation in the "System Design" section. The detailed evaluation also gives some credibility to the accuracy of the simple scaling exercise above.

Sorbent Compressor Weight

A simple evaluation of the weight of a sorbent compressor is based on scaling of the experimental unit's performance to a larger unit size. A more detailed design is also considered in a later discussion and its associated weight determined.

Considering the adsorption compressor bed system design of the system at CSULB, each of the four sorbent beds including the supporting structure weigh approximately 15 lbs and contains a sorbent weight of approximately 3.5 lbs. The total weight of all four beds is 60 lbs and total sorbent material is approximately 14 lbs. Considering the CSULB system's output was in the order of 13,500 BTU/hr, and a refrigerated trailer requiring a 46,000 BTU/hr system, an adsorption compressor 4 times the size and weight of the experimental system at CSULB would be required. A sorbent compressor in the order of 250 lbs would be a minimum compressor weight. With the addition of a more robust outer shell metallic structure to withstand exhaust heat loads, the weight could increase by two or three fold (stainless steal density = 3 x aluminum density) giving a 500 to 750 lb sorbent compressor. The adsorption compressor on a trailer or bus would require exhaust ducting that can add an extra 150 lbs; considering a 6 inch diameter stainless steal duct with at least a 1/8 inch thick wall and 10 to 15 feet long. The truck's refrigerated trailer using direct exhaust heating will require a flexible stainless steal duct to interface with the trucks exhaust. Additional large flapper valves, truck to trailer exhaust coupling, sorbent bed cooling fans and supporting structure could also increase weight by approximately 50 to 100 lbs. Summing weights of all adsorption compressor components totals a minimum of 700 lbs and a maximum of 900 lbs.