Light Duty Vehicle Air Conditioning

Greenhouse Gas Impacts and Potential for Reduction

Final Report

Prepared for:

Northeast States Center for a Clean Air Future

101 Merrimac Street, 10th Floor
Boston, Massachusetts 02114

Prepared by:

Meszler
Engineering Services

906 Hamburg Drive
Abingdon, Maryland 21009
410-569-0599
www.meszler.com

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Notice

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X.1 Introduction

Air conditioning (A/C) systems have become standard equipment on most light duty vehicles sold in the U.S. Once considered optional equipment, A/C systems currently command a market penetration in the U.S. of well over 90 percent of annual sales. [1,2,3] Moreover, global A/C system penetration is rapidly approaching that of the U.S. Current A/C market share in Asia is nearly equal to that of the U.S., while penetration in Western Europe is approaching 80 percent of annual sales (up from only about 10 percent in 1990). With South American market share approaching 50 percent, the total global new vehicle market share for A/C is currently about 75 percent and growing. Clearly, A/C systems should be considered an integral element of new light duty vehicle design and operation.

When viewed from a U.S. market perspective, the transition of A/C systems from optional to standard equipment has important implications in terms of both overall national energy use and national GHG emissions. A/C systems impact vehicular GHG emissions through three distinct mechanisms. Two of these mechanisms are "indirect" in that they are manifest not through any emissions from the A/C system itself, but rather through the system's influence on vehicle fuel consumption. Generally, A/C systems require energy to operate. This energy is typically supplied by the vehicle engine. Except as described below, any difference in engine fuel consumption with and without the A/C system is thus attributable to the presence of the A/C system. The third GHG mechanism is "direct" in that it is manifest through the leakage of refrigerant from the A/C system. Current A/C system refrigerant (HFC-134a) is a GHG. Thus, refrigerant emissions contribute to total A/C GHG loads regardless of indirect impacts on vehicle fuel consumption.

Given the potential for both direct (refrigerant leakage) and indirect (fuel consumption-related) emissions, it is necessary to fully understand the energy consumption impacts of vehicle A/C systems to gain a proper perspective on associated GHG emissions. At the same time it is important to maintain a distinction between A/C energy consumption and overall A/C-related GHG emissions since GHG reduction strategies can affect either or both types of emissions. Energy consumption is responsible for a significant portion of current A/C-related GHG impacts and there is significant effort underway to reduce energy consumption demands. Nevertheless, for reduction strategies in which there is no change in system refrigerant, improving A/C system energy efficiency can address only a portion of total system GHG emissions since indirect emissions will be unaffected. In short, A/C system GHG impacts are only partially correlated with energy consumption. Conversely, A/C system modifications based on substitute refrigerants can have significant impacts on direct GHG emissions, but may have no or negative impacts on system energy demands.

It is also important to recognize that A/C system energy demand cannot reasonably be compared to a zero energy baseline. This is often overlooked in studies that attempt to quantify the fuel consumption impacts of vehicle A/C operation. Zero cooling energy represents an unachievable ideal since vehicle occupants would have no alternative but to lower vehicle windows for cooling in the absence of onboard A/C. Vehicle operation with lowered windows can have a significant impact on vehicle aerodynamics, and thus fuel consumption. At low vehicle speeds, aerodynamic influence on fuel consumption is modest and A/C system energy demands are usually significantly higher than non-A/C "window-down" energy impacts. However, aerodynamic influence on fuel consumption increases with the cube of speed so that "window-down" fuel consumption impacts at higher speeds are substantial and can approach or even exceed A/C system energy demands. Therefore, the actual fuel consumption impact of A/C operation is limited to the *differential* between A/C operation with windows up and no A/C with windows down. Studies citing A/C operational impacts against a "windows-up" background substantially overstate A/C impacts. Such impacts would only be accurate if a zero energy A/C alternative was available.

Nevertheless, as indicated above, A/C systems do impact vehicular GHG emissions through three distinct mechanisms. Two mechanisms impact emissions through increases in vehicle fuel consumption and the third mechanism is emissions (leakage) from the A/C system itself. One of the fuel consumption impacts results from the fact that when operating, A/C systems place an additional load on the vehicle engine, increasing fuel consumption. The second fuel consumption impact is perhaps less obvious and results from the fact that regardless of operational status, A/C systems represent an additional mass that must be transported whenever a vehicle is moved. In effect, the vehicle engine must produce additional energy to "carry the system around," regardless of whether or not it is in use. A typical A/C system weighs on the order of 30-35 pounds. [4] Finally, since current system refrigerant is itself a GHG, any leakage or accidental discharge contributes to overall GHG impacts.

In recognition of these impacts as well as regulatory investigations related to A/C use in both the U.S. and Europe, substantial research into reducing both the direct (leakage) and indirect (fuel consumption-related) impacts of A/C systems has been conducted over the last several years and continues today. Efforts focus on both reducing indirect GHG impacts through the use of lighter and/or more efficient A/C systems and reducing direct GHG impacts through the design of systems with reduced refrigerant leakage rates and/or systems that utilize lower GHG impact refrigerants. At the time of this study, direct emission reduction research is particularly active as the European Union (EU) has proposed regulations limiting direct A/C system emissions in the near term and banning the continued use of current A/C refrigerant (HFC-134a) by 2018.[1] More detailed discussion of the EU rules is presented in Section X.4 below. Although regulatory discussion of controlling indirect A/C emissions is also underway, no limits or design standards are currently in effect. Moreover, as described in Section X.5, current U.S. fuel consumption testing performed to demonstrate compliance with federal Corporate Average Fuel Economy (CAFE) standards captures only a small fraction of indirect A/C GHG emission impacts (i.e., that related to system mass).

[1] Subsequent to the completion of the A/C system assessments described in this report, the European Parliament substantially amended the proposed EU regulations. While these amendments are subject to continuing revision during member state negotiations, the elimination of HFC-134a is currently proposed to be complete by 2014, four years sooner than originally proposed.

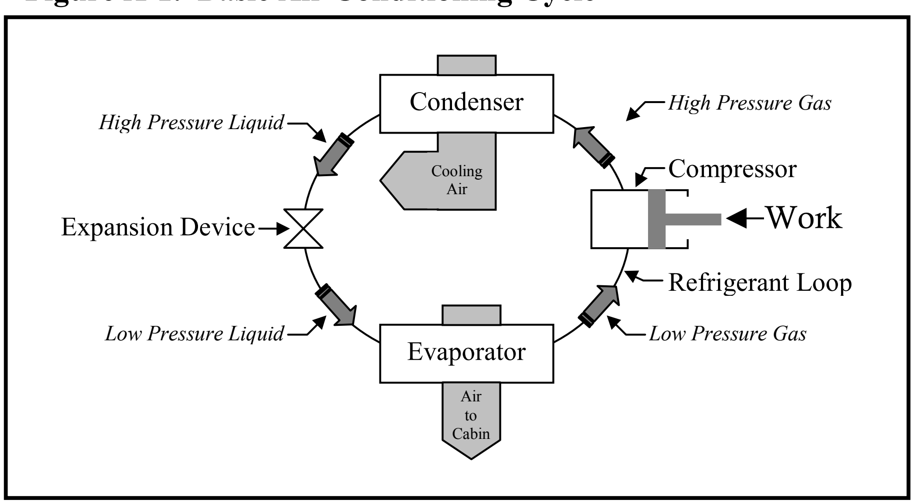
The basic function of a vehicle A/C system must be understood to properly assess ongoing research into reduced impact A/C systems as well as place the potential impact of those efforts, in terms of both costs and benefits, in a proper context relative to other vehicular GHG emissions. Section X.2 is intended to provide the necessary functional overview. Section X.3 then provides a brief overview of the current system refrigerant, HFC-134a, as well as an introduction of potential alternatives (and an overview of their benefits and weaknesses). Section X.4 provides more detailed discussion of the proposed EU A/C system leakage regulations and the potential implications thereof, while Section X.5 discusses the current U.S. regulatory structure. Finally, Sections X.6 and X.7 discuss the GHG reduction potential for direct and indirect A/C-related emissions respectively, while Section X.8 presents a comprehensive look at overall GHG reduction potential.

X.2 Typical A/C System Function

A brief overview of a typical A/C system may assist in understanding both the current GHG impacts of systems and the constraints that affect future reduction potential. Functionally, A/C systems move heat from one location to another. Heat is removed from air being directed into the passenger cabin of the vehicle and deposited into air outside the passenger compartment. A "refrigerant" acts as the transfer medium, temporarily storing the heat between removal from one air mass and deposit into another. While refrigerant properties can influence heat transfer efficiency and overall system cooling capacity, there is no inherent "coolness" to the refrigerant itself. Continuous work must be performed by the A/C system (while it is in operation) to maintain the heat transfer capacity of the refrigerant. It is this work which creates an additional load on the vehicle engine.

All current A/C systems are based on the vapor-compression refrigeration cycle. [5-10] A generalized schematic representation of a typical system is presented as Figure X-1. Such systems take advantage of the fact that significant quantities of heat (per unit mass) can be

Figure X-1. Basic Air Conditioning Cycle



transferred during refrigerant phase changes using ambient air as the external transfer medium. Specifically, heat is released from the refrigerant to air as the refrigerant condenses from a gas to a liquid and heat is released from air bound for the passenger cabin to the refrigerant as the refrigerant evaporates from a liquid to a gas. Repeated over a closed loop system, these actions allow heat to be moved from one air mass to another.

In its most basic form, the system includes only one "active" component, a compressor; used to increase the pressure of gaseous refrigerant. Raising the refrigerant pressure alters its boiling point, allowing the refrigerant to both condense and evaporate in different parts of the same cycle (and provides the motive force necessary to drive refrigerant through the cycle). Without the compressor, no heat transfer is possible. In practice, there are two other active components used to move air across the A/C system heat exchangers (i.e., the condenser and evaporator), but both of these additional components are generally required on vehicles with or without A/C. The radiator fan pulls air across the condenser, which is generally placed directly in front of the radiator. This fan is required to remove heat from engine coolant (in the radiator) regardless of whether or not A/C is present on the vehicle. Similarly, a blower motor moves air across the A/C evaporator, and this motor also performs double duty in that it provides the motive force for all cabin HVAC (heating, ventilation, and air conditioning) functions. Of course, the load induced by the blower motor during A/C function may be incremental as drivers of non-A/C vehicles may or may not operate the blower to force ventilation. Thus, while vehicles will carry the cost of a HVAC blower motor regardless of whether or not A/C is present, the motor may be used more often on an A/C-equipped vehicle.

Many of today's systems also include active electronic control units. While these controls influence the performance of the underlying mechanical functions, basic system function remains dependent on the work performed by the compressor. Accordingly, compression represents the primary additional energy load associated with A/C operation.

The basic vapor-compression refrigeration cycle takes advantage of the thermodynamic properties of the working fluid (i.e., the refrigerant) to move heat from one air mass (i.e., that entering the vehicle cabin) to another (i.e., that not entering the cabin). Heat is transferred from air entering the cabin to the refrigerant as the refrigerant moves through the A/C system evaporator. This heat is retained by the refrigerant as it moves through the compressor and is subsequently transferred to outside air as it moves through the A/C system condenser. The refrigerant is then directed back to the evaporator to begin the cycle anew.

The basic principles that control the efficiency of the cycle are directly related to system design and packaging constraints and the various thermodynamic properties of the refrigerant. A basic understanding of the process and several key aspects of current A/C system design can be gleaned from a "walk through" Figure X-1 above. In undertaking this discussion, it must be recognized that the presented thermodynamic properties and system states are specific to current HFC-134a A/C systems. While all of the near term alternatives to such systems will include similar functional components, not all will operate identically to the two distinct phase change-driven heat exchange processes typical of current systems (i.e., a phase change from gas to liquid in the condenser and a phase change from liquid to gas in the evaporator). For example, carbon dioxide (CO_2) based systems, operate under a single supercritical fluid phase through the heat rejection portion of the cycle.[2] However, phase change is induced as CO_2 passes through the expansion device from the high pressure to the low pressure side of the system, so that function in the evaporator is a two phase process similar to current systems.

Regardless, each refrigerant has specific thermodynamic properties that require consideration in system design, so it is impossible to discuss system characteristics without anchoring such discussion to a specific refrigerant. Specific differences and associated impacts are discussed in detail in the sections that follow. However, to ensure a proper understanding of such issues, it is important to also understand the current baseline against which those alternatives are assessed. Thus, the following discussion and the cited thermodynamic conditions are specific to current HFC-134a A/C systems. Differences for various alternatives will range from modest to substantial, but all will rely on similar functional components.

Examining the system from the low pressure side (i.e., the evaporator side) of the compressor, the refrigerant is in a low pressure, low temperature[3] gaseous state. Generally, the pressure and temperature (for a current HFC-134a system) will be 40-65 pounds per square inch absolute (psia) and 30-60°F respectively.[4] At these pressures, the boiling point of HFC-134a is below ambient (approximately 25-55°F), so the refrigerant will persist in a gaseous state (as required for proper compressor function).

The compressor performs the work necessary to raise the pressure of the refrigerant to 200-350 psia. This also raises the temperature to 180-200°F, so that the refrigerant is transformed into a high pressure, high temperature gaseous state. Here, it is also clear that the temperature of the refrigerant is now well above ambient and will release heat, but it is also clear that the bulk of the temperature gain[5] is due to compression and not due to heat removed from cabin-bound air in the evaporator. However, in raising the refrigerant pressure, the boiling point of the refrigerant is also raised to a point above ambient (to approximately 125-165°F).

The high pressure, high temperature gaseous refrigerant from the compressor is routed through the condenser, which is a high performance heat exchanger. As it passes through the condenser, the refrigerant transfers sufficient heat to the ambient air to reduce its temperature to its boiling point. At this temperature, significant additional heat transfer occurs as the refrigerant condenses

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[2] Some research into CO_2 systems using co-fluids has been performed, which while not strictly phase change driven do operate as dual-state (e.g., absorbed/desorbed) heat exchange systems.

[3] Note, the descriptive term "low" as applied to pressure and temperature through the A/C cycle is relative to pressure and temperature in other portions of the cycle. Thus, the temperature of the refrigerant can be "low" coming out of the evaporator, even though the refrigerant carries a significant quantity of heat that has just been removed from cabin intake air.

[4] Thermodynamic properties are provided as ranges due to the fact that system conditions are dependent on both internal and external (ambient) influences. Internal influences include compressor speed (which is directly controlled by engine speed since the compressor is engine-driven via a belt) and external influences include factors such as ambient temperature, vehicle speed (airflow), etc. Generally more favorable system operation occurs at travel speeds as opposed to idle conditions due to higher engine speeds and improved heat exchange conditions.

[5] The temperature gain is *not* reflective of the total heat being transported since considerable latent heat is absorbed in the evaporator during refrigerant phase change. The thermodynamic principal behind this is discussed in general detail as part of the A/C system evaporator function a few paragraphs below the footnote reference.

from a gaseous to a liquid state. This constant temperature heat transfer is generally denoted as "latent" heat since it represents the quantity of heat that must be absorbed (for evaporation) or removed (for condensation) to promote a phase change, although no associated temperature change occurs. Operating across the refrigerant boiling point allows significant heat transfer to occur over relatively narrow temperature changes. For a typical condenser, heat transfer characteristics break down as follows: [9]

- Heat transfer to boiling point (desuperheating): 65-100 Btu/min
- Heat transfer during condensation: 200-400 Btu/min
- Heat transfer below boiling point (subcooling): 60-120 Btu/min

Clearly, the quantity of heat transferred at the condensation point is a significant influence on overall system performance for a given refrigerant/ambient temperature differential. Following condensation, some additional heat transfer occurs (generally denoted as subcooling) and the refrigerant leaves the condenser as a high pressure, high temperature liquid (190-325 psia and 100-160°F respectively). Under these conditions, the boiling point of the refrigerant remains above ambient (at approximately 120-160°F), so it will persist in a liquid state.

The liquid refrigerant is collected in a "receiver," which is typically packaged with the condenser and acts as a storage device so that the proper amount of refrigerant can be metered to the evaporator.[6] Liquid refrigerant is drawn from the receiver through an expansion device (by the compressor-induced pressure gradient) for delivery to the evaporator. Expansion devices are generally of two types. The simplest, the "orifice tube" has (as its name implies) a small orifice through which liquid refrigerant passes at a rate proportional to the pressure difference between the condenser and the evaporator. In contrast, the "thermal expansion valve" (or TXV) monitors the pressure and temperature at the evaporator outlet and adjusts refrigerant flow so that all refrigerant exiting the evaporator is in a gaseous state. Under either approach, the high pressure liquid on the condenser side of the expansion device passes into a low pressure zone on the evaporator side of the expansion device (the expansion device in conjunction with the compressor maintains the pressure differential by limiting refrigerant flow). In moving from an area of high pressure to an area of low pressure, the pressure and temperature of the refrigerant drop dramatically so that the refrigerant enters the evaporator as a low pressure, low temperature liquid (50-80 psia and 40-60°F respectively). Under these lower pressure conditions, the temperature of the refrigerant exceeds its boiling point, which is reduced to 30-55°F, but is generally well below ambient temperature, so that heat transfer from the ambient air to the refrigerant is facilitated.

The low pressure, low temperature liquid refrigerant is routed through the evaporator, which like the condenser is a high performance heat exchanger. However, whereas the condenser removes heat from the refrigerant by reducing its temperature across its boiling point, the evaporator adds heat to the refrigerant at its boiling point. Heat required to induce the boiling of refrigerant within the evaporator is supplied by ambient air routed across the evaporator and into the vehicle

[6] The receiver also performs other important functions that ensure the integrity of the air conditioning system, but which are not directly related to the heat transfer function. Among these are the separation of liquid refrigerant from any gaseous refrigerant that exits the condenser and the removal of any moisture or dirt from the refrigerant (that could otherwise compromise system performance).

passenger compartment. As heat moves from the ambient air to the refrigerant, the temperature of the air is reduced dramatically and cools the passenger compartment, accomplishing the design goal of the system.

As it leaves the expansion device and enters the evaporator, the liquid refrigerant is already at a temperature that exceeds its boiling point and, as a result, boiling begins immediately. Additional heat from the ambient air promotes a rapid refrigerant phase change from liquid to gas. As was the case in the condenser, significant latent heat (heat absorbed without a change in temperature) must be absorbed to promote the phase change, so that operating across the refrigerant boiling point allows significant heat transfer to occur over relatively narrow temperature changes. In fact, the temperature gradient of the refrigerant across the evaporator will be only a few degrees even through the temperature of air routed into the vehicle cabin will be reduced by 50°F or more. The freezing point of water (32°F) represents a practical lower limit for air cooled over the evaporator since water condensing on the outside of the evaporator will freeze at this temperature and significantly reduce heat transfer.[7] To maintain system temperatures above 32°F, the compressor will either cycle off or the TXV (for systems equipped with such) will reduce the amount of liquid refrigerant entering the evaporator.

Refrigerant from the evaporator is routed back to the compressor where the cycle begins anew. On some systems, especially those using fixed orifice tube expansion devices, there is an accumulator located between the evaporator and the compressor to collect any liquid refrigerant that may have passed through the evaporator without changing state. Any liquid will readily evaporate in the accumulator due to the low boiling point of the refrigerant, but it is critical that refrigerant not enter the compressor in a liquid state to avoid damage to the mechanical components of the compressor.

As indicated in Figure X-1, work must be input via the compressor to perpetuate the A/C system cycle. Typically, compressor load will range from 3 to 5 horsepower (hp) depending on engine speed. It is this load that defines the indirect fuel consumption impacts of the A/C system as well as the potential for indirect GHG emission reductions. The passenger cabin blower motor also consumes power to move ambient air across the evaporator, but the load is much lower; typically on the order of 0.3 hp. Moreover, even without A/C, vehicle occupants may well operate the blower motor to circulate air. Additionally, the radiator fan also consumes power to move ambient air across the A/C condenser, but the same fan would be necessary to move air across the engine coolant radiator even without A/C. Therefore, the overall A/C power demand is generally consistent with that imposed by the compressor alone.

X.3 <u>Current and Alternative Systems and Refrigerants</u>

All current vehicle A/C systems are designed around the refrigerant HFC-134a. Condensers and evaporators are designed and sized to achieve acceptable system heat exchange performance using HFC-134a. Compressors are designed and sized to raise HFC-134a to a pressure where its

[7] Air delivered to the vehicle cabin by the air conditioning system is dehumidified through the condensation of water as the air passes over the cooled evaporator. Contaminants such as dirt and pollen are also greatly reduced by such condensation. This "feature" is also used in conjunction with the onboard vehicle heater to enhance the window demisting capability of vehicles.

boiling point exceeds ambient temperatures. In most cases, systems utilizing alternative refrigerants will require design modifications to accommodate the differing thermodynamic properties of the particular refrigerant. Alternatively, enhancing current system performance or components will incur associated costs and, potentially, system performance compromises. It is critically important to understand each of these design issues in assessing the viability and overall GHG impacts of any movement away from current A/C systems.

Prior to the mid-1990s, vehicle A/C systems were designed around CFC-12 ($C(Cl_2)F_2$, or dichlorodifluoromethane, commonly known as freon). CFC-12 was employed almost universally in vehicle A/C systems for over 50 years. [9] In the 1970s, the linkage between chlorofluorocarbons (CFCs) and stratospheric ozone depletion was established and in response to the Montreal Protocol of 1987 and related U.S. legislation,[8] automakers began phasing out the use of CFC-12 in 1992. By 1995, the phase out was complete and all CFC-12 A/C systems had been replaced with HFC-134a systems. Although the replacement of CFC-12 systems was driven by stratospheric ozone depletion concerns, it had an unanticipated secondary benefit relative to GHG. Not only was the ozone depletion potential (ODP) of HFC-134a zero as compared to CFC-12, but its global warming potential (GWP) was also 85 percent lower. Whereas CFC-12 has a GWP of 8,500, that of HFC-134a is 1,300. Moreover, since the amount of refrigerant in a typical HFC-134a system was also reduced relative to a typical CFC-12 system, direct GHG emissions from A/C systems were reduced by about 90 percent as a result of the phase out of CFC-12.

It is important to recognize that almost any fluid can be made to function as a refrigerant given appropriate system design. Therefore, it is not possible to discuss all potential refrigerant options. However, thermodynamic and other properties render certain substances more effective from a performance, cost, and safety standpoint. For example, as discussed above, HFC-134a (as well as CFCs, other hydrofluorocarbons, and hydrocarbons) possess thermodynamic characteristics such as low normal boiling points and relatively steep boiling point curves that allow for modest compression requirements. HFC-134a was selected as the replacement for CFC-12 not only because of its zero ODP, but also because of its similar thermodynamic properties and the fact that it is not flammable, explosive, poisonous, or corrosive, is odorless, and is generally harmless to materials. It is precisely these benign non-thermodynamic qualities of HFC-134a that may represent the toughest "hurdle" for potential alternative refrigerants.

Chemically, HFC-134a is 1,1,1,2-tetrafluoroethane ($C_2F_4H_2$). Table X-1 presents a list of alternative refrigerants that have been discussed for vehicle application. As indicated above, no list can be comprehensive, but Table X-1 includes those compounds for which modest evaluation or development work has been conducted over the last several years. In addition to CFC-12, which is included to provide a comparative foundation, Table X-1 includes HFC-134a and HFC-152a, as well as the "natural" refrigerant hydrocarbon species, ammonia, carbon dioxide, and air.[9]

[8] Clean Air Act, Title VI, Sections 601 through 618.

[9] "Natural" refrigerants are designated as such in that they occur naturally in the environment, unlike CFCs and HFCs, which are manmade compounds. In general, natural refrigerants have no ozone depletion potential and zero or very low global warming potential.

Table X-1. Basic Environmental Characteristics of Selected Refrigerants [11-14]

Refrigerant	Compound Name	Formula	ODP[a]	GWP[b]
CFC-12	Dichlorodifluoromethane	$C(Cl)_2F_2$	1	8500
HFC-134a	1,1,1,2-Tetrafluoroethane	$C_2F_4H_2$	0	1300
HFC-152a	1,1-Difluoroethane	$C_2F_2H_4$	0	120
HC-290	Propane	C_3H_8	0	20
HC-600a	Isobutane	C_4H_{10}	0	20
HC-C270	Cyclopropane	C_3H_6	0	20
R-744	Carbon Dioxide	CO_2	0	1
R-717	Ammonia	NH_3	0	0
R-729	Air		0	0

[a] ODP is "Ozone Depletion Potential" as defined by the U.S. Environmental Protection Agency in 40 CFR Part 82. The ODP scale is based on CFC-11 (trichlorofluoromethane, $C(Cl)_3F$) so that the indicated ODP of a compound is defined as the ratio of its impact on ozone to the impact of the same mass of CFC-11.

[b] GWP is "Global Warming Potential" as defined by the U.S. Environmental Protection Agency in 40 CFR Part 82. The GWP scale is based on CO_2 (carbon dioxide) so that the indicated GWP of a compound is defined as the ratio of its impact on global warming to the impact of the same mass of CO_2.

As indicated in Table X-1, there is substantial potential for reducing *direct* A/C GHG emissions through refrigerant substitution. For example, HFC-152a has a GWP that is over 90 percent lower than that of HFC-134a. Combined with reduced system charge benefits (i.e., a lesser quantity of refrigerant is required) due to its lower molecular weight, the overall direct GHG reduction potential of HFC-152a is 94 percent. Hydrocarbon refrigerants offer the potential of another 80 percent reduction above and beyond HFC-152a (an over 99 percent reduction potential from HFC-134a). Either CO_2 or air systems would reduce direct GHG potential to zero.[10] However, as stated above, alternative refrigerants generally require associated A/C system design changes and resulting impacts on indirect emissions (i.e., fuel consumption-related emissions) can significantly influence the overall GHG emissions reduction potential of alternative refrigerants. Thus, Sections X.6 through X.8 below examine overall system reduction potential and costs in detail for a range of alternative systems.

[10] Strictly speaking, CO_2 would not reduce direct GWP to zero since it is a GHG. However, CO_2 for A/C system use can be produced directly from CO_2 already generated as waste by other processes, so that the net change in atmospheric CO_2 is zero. Regardless, with a GWP of unity, the direct emissions reduction potential would be 99.9 percent relative to an HFC-134a A/C system even without CO_2 "recycling." At a system charge of about 500 grams, a CO_2 system leaking its complete charge <u>every</u> year would have direct GHG emissions lower than an HFC-134a system leaking at a rate of less than one-half gram per year. At the EU target leakage rate of 20 grams per year (see Section X.4) for HFC-134a, an annual leakage of 500 grams of CO_2 would still reduce direct GHG emissions by 98 percent.

System enhancements are being developed that have the potential to significantly reduce indirect GHG emissions. These enhancements generally focus on the two areas that most affect indirect emissions, system mass and compressor efficiency. Lower system mass reduces the amount of energy necessary to "carry around" the A/C system. More efficient compressors and more efficient compressor control strategies can reduce the amount of energy required to operate the system.

Most current compressors are of internally controlled, fixed displacement design. In a fixed displacement compressor, the volume of compressed refrigerant is solely a function of engine speed (since the compressor is driven by a belt from the crankshaft). To control cooling capacity, a clutch is used to disengage the compressor entirely as needed. Thus, the compressor is either on or off. In most current systems, clutch cycling controls are designed solely to disengage the compressor when the evaporator temperature approaches the freezing point of water. This ensures that water vapor from the ambient air does not freeze on the evaporator surface and inhibit the heat exchange process. Generally, the control mechanism is a pressure or temperature switch in the evaporator discharge line. Regardless, the important point is that the control is independent of actual cooling requirements and, therefore, is generally denoted as "internal" control (or, in the case of pressure switch systems, pneumatic control).

With a fixed displacement compressor, there is generally no way to operate the system in a "partial load" condition. In contrast, variable displacement and variable speed compressors allow the volume of compressed refrigerant to be adjusted according to the cooling load and sufficient energy savings can accrue under certain operating conditions. Variable displacement compressors can be engine-driven in a fashion analogous to fixed displacement compressors, while variable speed compressors generally require an independent electric motor. Both technologies are currently available and are present in the U.S. market in limited volumes. Variable displacement compressors are appearing on an increasing number of small engine and luxury vehicles, while electric compressor market penetration is limited to the 2004 Toyota Prius hybrid electric vehicle due to an electrical system demand that is too large for current conventional vehicle electrical systems.

It is important to recognize that while variable displacement and variable speed compressors allow for the tailoring of A/C system loads to cooling requirements, the efficiency improvements that accrue will only be significant when effective control strategies are also implemented. Such control strategies are designed to recognize cooling demands and adjust compressor function accordingly. Because of the ability to respond to stimuli outside of the basic A/C system loop, such control is generally denoted as "external" control. It is also important to note that adding external control to a fixed displacement compressor may allow for efficiency improvements that approach those of variable displacement or variable speed compressors. In effect, the partial load capabilities of the variable compressors might be approximated by clutch cycling strategies that vary compressor on/off time according to cooling demands. However, clutch cycling results in pressure pulses, and it has not yet been demonstrated that adequate system performance can be maintained for a wide range of operating conditions that approximate the essentially continuous dynamic range of the variable compressor systems. Regardless of whether fixed displacement techniques are ultimately proven, the control strategy is important as the hardware.

In addition to compressor energy demands, substantial work is also ongoing to increase the heat exchange performance of current systems through improved condenser and evaporator designs as well as through strategies such as "suction line heat exchangers." A suction line heat exchanger passes refrigerant exiting the condenser across refrigerant exiting the evaporator to provide additional subcooling to condenser refrigerant prior to its introduction into the evaporator. Overall, it appears that significant efficiency improvements are possible for current HFC-134a systems and the impact of such improvements must be evaluated to determine the true GHG reduction potential of HFC-134a relative to alternative refrigerants. Sections X.6 through X.8 below are intended to provide such an assessment.

Finally, it is also important to recognize that non-performance related issues can greatly influence the viability or cost of alternative A/C system designs. Previous CFC-12 and current HFC-134a refrigerants present few safety concerns. They are non-toxic, non-flammable, and non-explosive. Conversely, most, if not all, of the alternatives currently considered promising present one or more safety issues that must be addressed. For example, HFC-152a is classified as flammable, as are all of the hydrocarbon refrigerants. In contrast, CO_2 (R-744) is non-flammable, but toxic in certain concentrations. Ammonia represents perhaps the worst case safety concern as it is flammable, explosive, and a poison. For this reason, there is little active research into vehicular A/C systems utilizing ammonia. However, risk assessment into HFC-152a, hydrocarbons, and CO_2 is currently being conducted by industry and regulatory interests to better assess the degree to which current system designs may need to be modified. Possible requirements could in-cabin leak monitors and underhood safety relief valves or secondary loop designs, either of which would add cost to alternative system design. For example, with secondary loop designs, the main A/C system loop described previously in Section X.2 would be constructed so that it was entirely outside the passenger cabin.[11] Instead of exchanging heat directly between refrigerant and cabin air across the evaporator, a secondary fluid would be cooled across the main evaporator and directed through an independent loop to an in-cabin heat exchanger where cabin air would be cooled. Thus an additional heat exchanger, secondary fluid, and secondary loop pump would be required.[12] Finally, the critical temperature of CO_2 is sufficiently low (88°F) to require the operating pressure of CO_2 systems to exceed those of current HFC-134a systems by a factor of 7-10. This presents not only potential safety concerns, but issues related to the ability to ensure adequate system charge retention over time.

X.4 <u>European Regulatory Requirements</u>

In August of 2003, the EU formally proposed regulations mandating the phase out of vehicle A/C systems using HFC-134a. [15] In developing these regulations, the EU conducted considerable research into both the direct and indirect emissions potential of current and developing A/C systems. While a concerted effort was taken to consider and reconcile various

[11] All current system components with the exception of the evaporator are currently outside the passenger cabin. To maximize cooling, the evaporator is located behind the dashboard. Thus, leaks in the evaporator portion of the refrigerant loop can affect cabin air quality.

[12] The coolant generally used for secondary loop system research is an ethylene glycol/water (liquid) mixture with a GWP of zero, so that leakage of secondary loop coolant is not an additional GHG issue. Secondary loop impacts on overall A/C system energy impacts are, however, important.

data provided by industry sources on the current state of development of system efficiency improvements as they would affect indirect A/C emissions, the EU was unable to come to a consensus view of how improvements might affect overall system GHG emissions in the near term. [16] Therefore, the EU elected to focus on direct emissions reduction potential and their proposed regulations are structured accordingly.

Beginning in 2005, the EU rules limit the annual leakage rate of any vehicle A/C system using a refrigerant with a GWP greater than 150 to 40 grams for single evaporator systems and 50 grams for dual evaporator systems.[13] As indicated in Table X-1 above, HFC-134a has a GWP of 1300 and all currently evaluated alternatives have GWPs below 150, so practically, the leakage limits apply only to HFC-134a systems. These limits are intended to require little, if any, modification of well designed current systems, but rather force all automakers to use good manufacturing practices to ensure that actual system performance measures up to design standards.

Beginning in 2009, the EU rules, through a system of production credits, limit the number of new HFC-134a systems that can be sold. In 2009, each automaker is allocated credits equal to 80 percent of 2007 production. Between 2010 and 2013, the allocated credits decline to 60, 40, 20, and 10 percent of sales (based on sales two years prior to each credit year). In 2014, *credits* for new HFC-134a systems decline to zero. Credits are both bankable and tradable.

Because credits can be used through 2018, new HFC-134a systems can continue to be introduced five years after the ability to earn credits ceases. Credits greater than the base allocations described above can also be earned through 2014 by introducing either systems using a refrigerant with a GWP of 150 or less or by certifying HFC-134a systems to more stringent leakage standards. Credits for alternative refrigerant systems are earned on a "one earned" for "one sold" basis. For example, sales of 10,000 vehicles with HFC-152a systems would generate 10,000 credits beyond the base allocation. "Tighter" HFC-134a systems must be certified to half the base emission limit (i.e., 20 grams per year for single evaporator systems and 25 grams per year for dual evaporator systems). Credits for these systems are earned on a "one earned" for "two sold" basis. In other words, sales of 10,000 20 gram HFC-134a systems would generate 5,000 credits beyond the base allocation. The rules also provide special provisions for new market entrants and small volume manufacturers.[14]

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[13] Many large vans and sport utility vehicles provide front and rear air conditioning and such systems rely on front and rear evaporators to cool cabin air.

[14] It is important to recognize that subsequent to the completion of the A/C system assessments described in this report, the European Parliament substantially amended the proposed EU regulations. While these amendments are subject to continuing revision during member state negotiations, they significantly alter the implications of the EU requirements. First, the amendments reduce the allowable GWP of mobile A/C systems to 50 beginning in 2011 and require the complete elimination of systems with a GWP above 50 by 2014. This would effectively prohibit both HFC-134a *and HFC-152a*, leaving CO_2 and hydrocarbons as the most likely long term A/C refrigerants in the EU. The credit program that would allow continued sales of higher GWP systems through 2018 has been deleted. Second, the amendments eliminate the specified maximum A/C system refrigerant leakage rates of 20 and 40 grams per year scheduled to begin in 2005, and impose instead a requirement for an unspecified maximum leakage rate for A/C systems with a GWP greater than 150 beginning in 2007. The specific maximum leakage rate and associated test procedure are to be developed. So, while the amendments somewhat relax near term requirements, longer term requirements were made substantially more stringent.

The bottom line is that if adopted as proposed, the EU regulations establish a critical baseline for U.S. regulatory action. The light duty vehicle market has evolved beyond U.S. domination to a true international market. In fact, the EU and North America both comprise about a third of worldwide vehicle sales, with Asia close behind at about 25 percent. [17] Moreover, 5 manufacturers account for over 55 percent, and 12 manufacturers account for over 85 percent of global auto sales. [18] Therefore, it is very likely that requirements for components present on vehicles in all vehicle markets, such as air conditioning, will meet common design standards. This is especially true when one considers the fact that multiple designs will necessitate variable service requirements, including perhaps redundant equipment purchase, etc. As a result, it seems highly likely that the proposed EU regulations, if implemented, will bring about a global end to the use of HFC-134a, either in current or enhanced systems.

From a vehicle manufacturing standpoint, the EU regulations make no distinction between A/C systems based on refrigerants with a GWP of 150 or less.[15] HFC-152a, hydrocarbons, and CO_2 (R-744) are all acceptable replacements and would earn identical credits during the HFC-134a phase out. However, in the supporting discussion for the proposed regulations, the EU states that service and end-of-life refrigerant recycling requirements will not apply to CO_2-based systems. [15] Thus, substantial benefits to the service and scrappage industries would accrue with CO_2 systems, as the need for recycling equipment, at approximately $2,000 per unit, would be eliminated as HFC-134a vehicles are removed from service. It is not clear to what extent this will influence vehicle manufacturer compliance strategies as the viability of CO_2-based systems is not yet certain (as described in more detail in Sections X.6 through X.8 below).

X.5 U.S. Regulatory Requirements

There are currently no explicit GHG emission limits affecting vehicle A/C system design in the U.S. Nevertheless, there are existing rules that have influenced A/C system GHG performance from either a direct (refrigerant leakage) or indirect (increased vehicle fuel consumption) standpoint. However, the impact of these influential rules varies widely and generally cannot be expected to further influence future A/C system design, since associated vehicle manufacturer response is already reflected in current system designs.

U.S. rulemaking affecting A/C system design from an *ozone depletion* standpoint has been in place more than a decade, the net effect of which was the replacement of systems designed around CFC-12 refrigerant with systems designed around HFC-134a. Associated requirements affecting the service industry, in regards to refrigerant recycling and reuse remain in place and affect both remaining CFC-12 A/C systems and HFC-134a A/C systems. [19] Despite the fact that the prohibition of CFC-12 systems was not undertaken from a GHG standpoint, it nevertheless did have a significant impact on the GHG performance of vehicle A/C systems. As indicated in Table X-1 above, the GWP of CFC-12 is 6.5 times that of HFC-134a, so that the switch from CFC-12 systems to HFC-134a systems reduced the GWP of leaking A/C system refrigerant by about 85 percent on an equivalent mass basis. Because HFC-134a A/C system charges (i.e., the amount of refrigerant in the system) are lower than those of CFC-12 systems,

[15] As described in footnote 14, this is no longer true. HFC-152a no longer satisfies the maximum GWP requirements of the EU rules as currently amended, leaving only hydrocarbons and CO_2 as likely replacement candidates for HFC-134a.

GWP reduction associated with refrigerant leakage is actually about 90 percent. So while the ozone depletion rules did not target GHG emissions, they have had an impact. However, from a new vehicle standpoint, the impact is mature and will not affect future refrigerant decision making. Moreover, there are no other rules in effect in the U.S. that will require further consideration of the continued use of HFC-134a (however, the EU rules discussed in Section X.4 above will certainly force such consideration from a market uniformity standpoint even though they have no direct applicability in the U.S.).

Indirect GHG emissions associated with vehicle A/C systems result from system influence on vehicle fuel consumption. Accordingly, rules targeting vehicle fuel consumption can be expected to lead to vehicle manufacturer efforts to minimize indirect A/C impacts. However, even though fuel consumption standards are in effect in the U.S., the linkage between these standards and A/C system efficiency is generally non-existent due to the fact that the associated vehicle testing is performed with the A/C system switched off.

Under the U.S. CAFE program that has been in effect in the U.S. since the mid-1970s, vehicle fuel consumption is measured over two test cycles, one intended to reflect urban driving behavior and one intended to reflect highway driving behavior. Under the urban driving test, vehicle manufacturers must certify that emissions of hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NO_x), and particulate matter (PM) are below applicable U.S. standards. In addition, fuel consumption measured during the urban test is combined with that measured during the highway test to demonstrate compliance with applicable standards.[16]

Prior to 1997, emissions and fuel consumption testing for vehicles with significant penetrations of A/C systems required a 10 percent increase in the load placed on the vehicle during testing in an effort to simulate average annual A/C demands. Thus, there was some attempt to correct both emissions and fuel consumption measurements for A/C operation. However, data collected during this period indicated that the adjustment did a relatively poor job of accurately accounting for A/C system impacts. Moreover, the adjustment was fixed at 10 percent regardless of A/C system design, so it provided no incentive to vehicle manufacturers to maximize system efficiency. In late 1996, the U.S. Environmental Protection Agency (EPA) adopted (as part of the Supplemental Federal Test Procedure, or SFTP) a test cycle independent of the city and highway cycles to directly capture A/C impacts. [20] As part of the same rulemaking, the EPA eliminated the requirement for the 10 percent city and highway cycle load adjustment.

Cycle SC03 of the SFTP was adopted specifically to assess vehicle A/C system impacts on emissions.[17] While specific emission standards were established only for HC, CO, and NO_x, vehicle manufacturers must also measure vehicle CO_2, so fuel consumption can be calculated on

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[16] The urban cycle is officially designated as the Urban Dynamometer Driving Schedule (UDDS or LA4) and is incorporated into the Federal Test Procedure (FTP or FTP75). The highway cycle is officially designated as the Highway Fuel Economy Test (HFET or HWFET).

[17] SC03, an acronym for Start Cycle 03, is a 594 second driving cycle originally developed to capture driving behavior immediately after vehicle startup. However, it is also used to assess vehicle A/C system impacts. The SC03 test is conducted under test conditions that simulate an ambient temperature of 95°F, a humidity of 100 grains of water per pound of dry air (approximately 40 percent relative humidity), a solar load of 850 watts per square meter, and (variable) airflow across the A/C condenser as would be expected in actual driving.

the basis of emitted carbon. However, manufacturers are only required to perform the SC03 cycle with the vehicle A/C system on. There is no requirement for a comparative A/C off test, and SC03 results cannot be compared to those from either the FTP or HFET due to driving cycle and test procedure differences. So while the A/C system operational impacts on HC, CO, and NO_x can be compared to applicable SFTP standards, there is no direct assessment of the incremental impacts of A/C system operation.

Of course, any of these standardized tests could be augmented to include a direct assessment of incremental A/C system impacts, either in terms of fuel consumption or CO_2 emissions. An A/C-off replication of the SC03 test or an A/C-on replication of the FTP or HFET tests would allow for the direct comparison of A/C-on versus A/C-off CO_2 emissions. Given that the SC03 cycle already includes requirements for appropriate ambient conditions for A/C-on testing, it would seem the more appropriate candidate for comparative testing. However, it should be recognized that *no* simple on/off testing will accurately simulate A/C system impacts under all conditions and in all areas of the U.S. Actual ambient conditions during A/C operation will vary both geographically and across time. Conditions on one day will not be the same as conditions on another, so that average A/C system operating conditions will comprise a bounded continuum. A population or time weighted average set of conditions could be developed, but even that average will vary from actual conditions in any one area (if derived over a large geography) or will be geographically constrained in its applicability.

Instead, it is likely that the accurate definition of a generalized test cycle will require testing under a series of varying ambient conditions. For example, varying ambient temperature and humidity conditions. Measurements over a series of tests would then be weighted according to the fraction of time systems actually operated at (or near) each specific set of conditions. This is especially important as more efficient A/C technologies, such as variable displacement compressors, become more prevalent in the market, since efficiency advantages relative to conventional technology (e.g., fixed displacement compressors) will vary with ambient conditions.

Such considerations are discussed further in Section X.7 below. What is most important to recognize is that vehicle manufacturers have little incentive *from a fuel consumption standpoint* to implement more efficient A/C systems under current U.S. regulatory requirements. Compliance with all requirements that hold manufacturers to a fuel consumption limit is demonstrated with the A/C system switched off. It should be recognized, however, that since A/C system mass does affect vehicle fuel consumption regardless of whether or not the system is operating, the mass-related impact of A/C systems on fuel consumption *may* be considered during current U.S. fuel consumption testing. Consideration is limited by the fact that light duty vehicles are tested in weight increments of 125 pounds up to 4,000 pounds loaded vehicle weight (vehicle curb weight plus 300 pounds) and in weight increments of 250 pounds for heavier vehicles. [21] Given that typical A/C system mass is on the order of 30-35 pounds, it is unlikely that a large percentage of vehicles are shifted from one test weight to another as the result of A/C system presence. Nevertheless, some fraction of vehicle testing will reflect higher emissions due to A/C system mass. Regardless, as described in detail in Section X-7 below, this impact, relative to the indirect fuel consumption impacts of an operating A/C system, is relatively minor under most circumstances.

Finally, it should also be recognized that manufacturers may have incentive to implement A/C system efficiency improvements independent of vehicle testing requirements. For example, the incremental engine load associated with A/C system fixed displacement compressor clutch cycling is sufficiently large on small output engines to cause perceptible and perhaps unacceptable driveability concerns for some vehicles and customers. Thus manufacturers have been moving toward variable displacement compressors in the small engine market to provide more acceptable driveability. Generally, such a switch also yields reductions in indirect A/C system GHG emissions. Nevertheless, beyond driveability improvements and perhaps goodwill, there is no incentive for vehicle manufacturers to maximize A/C system efficiency under the current U.S. regulatory structure.

X.6 Direct GHG Emissions from Vehicle A/C Systems

Due to the fact that HFC-134a, the refrigerant used in current vehicle A/C systems, is a GHG, any refrigerant emissions from A/C systems contribute directly to the overall GHG emissions of vehicles. Although A/C systems are designed to be closed for the life of a vehicle, refrigerant emissions can occur through several mechanisms. Continuous refrigerant emissions occur through "regular" leakage at component joints and the compressor shaft seal, as well as through hose permeation. "Irregular" leakage can also occur as the result of system damage through accidents, etc. The evacuation of refrigerant either during A/C system or other vehicle service can also lead to either significant or minor leakage depending on the use and efficiency of refrigerant recycling equipment. Finally, evacuation of refrigerant at the end of vehicle life also results in leakage at a rate dependent on the use and efficiency of refrigerant recycling equipment.

Pre-Sale Emissions. Refrigerant emissions also occur upstream of the vehicle manufacturing process, through losses during refrigerant manufacturing, distribution, and storage. In addition, some refrigerant is lost during the initial filling of the vehicle A/C system. Such emissions are generally recognized to be small relative to in-service leakage, estimated at perhaps 20-25 grams of HFC-134a per vehicle lifetime. [22]

Regular Emissions. Estimates of the rate of regular HFC-134a leakage span a relatively wide range of 10-130 grams per year, but most estimates are in the 50-60 gram per year range. [11,22-30] From late November 2002 through January of 2003, measurements of the mass of vehicle A/C system refrigerant were taken for 276 light duty vehicles in the EU. [30] Vehicles up to seven years old were selected so that only "current generation" A/C systems were included.[18] The vehicles reflect a quasi-random sample obtained through a network of 19 auto dealerships. The sample is quasi-random in that any vehicles with A/C system damage or a history of A/C system service were excluded. In addition, the sample is stratified to span the climatic variability of the EU as well as appropriately reflect the EU distribution of vehicle makes and ages. The study also attempts to control for as many influences on estimated

[18] Vehicle A/C systems introduced immediately following the phase out of CFC-12 may have non-representative leakage rates due to "first generation" learning influences.

emission rates as possible.[19] By comparing measured charge mass to initial fill specifications, a regular HFC-134a leakage rate of 52 grams per year was estimated, with uncertainty placing the true estimate in the range of 48-57 grams per year.

Several additional insights can be gleaned from the EU testing. First, although testing was conducted across a range of climates by including vehicles from Portugal, Germany, and Sweden, no statistical difference in the estimated regular emission rate between areas was found. Second, 12 vehicle makes were represented in the sample and although there is variation in the mean leakage rates, there is no statistical difference between the rates of 10 of the associated manufacturers (all are within a range of 38-59 grams per year on average). However, one manufacturer exhibits a statistically low leakage rate of 29 grams per year, while a second exhibits a statistically high leakage rate of 82 grams per year. Third, differences between orifice tube and TXV systems, as well as between automatic and manual control systems were also statistically insignificant. Leakage rate did, however, vary with initial charge size. Systems with initial charges below about 750 grams leaked at a rate of 44 (±5) grams per year, while larger systems leaked at a rate of 67 (±9) grams per year. Differences for vehicles with engines above and below about 2 liters displacement were insignificant, as were differences between gasoline and diesel vehicles. Finally, no differences were observed between leakage rates for vehicles with mileage accumulation rates above and below about 12,700 miles per year.

The EU data represent the most robust assessment of regular A/C refrigerant emission rates performed to date. Of course, there is concern in extrapolating these data to the U.S. due to differences in fleet makeup and more extreme variations in climate. Although the EU study found no statistical differences across tested climates, the climatological variation between test sites is less than observed in the U.S. However, the measured leakage rates are consistent with estimates developed by other researchers and the variation in test climates does represent a wide range of U.S. climates, so there is no obvious basis for adjustment. Fleet makeup differences are also a potential concern. For example, significant fractions of EU vehicles are manufactured by Volkswagen, Opel, Fiat, Peugeot, Renault, and Citroen (about 50 percent of the EU fleet in total), whereas the fraction of U.S. sales by these same manufacturers is small, with only Volkswagen having a market presence. However, Ford, Chrysler, BMW, Mercedes, Volvo, Saab, Audi, Honda, Mazda, Mitsubishi, Nissan, Toyota, and Suzuki vehicles are also represented in the EU test data at approximately 50 percent of the test sample and these same manufacturers have significant U.S. market presence. Given that the EU study shows statistically insignificant leakage rate differences across all but two manufacturers, it seems reasonable to assume that emission rate differences in the U.S. will be minor. Moreover, when the EU corrected for differences between the sample distribution and actual EU fleet distribution, the estimated emission rate changed only from 52 grams per year (for the sample) to 54 grams per year (for the fleet corrected sample). Considered in conjunction with the fact that the 52 gram emission rate is consistent with the leakage rates estimated by other researchers, there is no basis to assume any

[19] Issues such as incomplete system evacuation (i.e., refrigerant left either in the vehicle or in recovery equipment hoses) and uncertain initial charge mass (i.e., some manufacturers specify a charge interval such as 0.6-0.7 kilograms rather than a precise charge) can affect estimated leakage rates.

significant differences in U.S. leakage rates.[20] Therefore, this study assumes the regular leakage rate for current HFC-134a systems to be approximately 50 grams per year.

It should be noted that the assumption of 50 grams per year could underestimate actual leakage rates over at least the near term. As indicated above, the EU testing did find a direct correlation between initial fill size and leakage rate. To the extent that larger vehicles such as SUVs are likely to have higher capacity A/C systems due either to interior size or dual evaporator support capability, any continuing shift toward such vehicles could shift emission rates higher accordingly. Unfortunately, there is no easy way to determine the A/C system charge across the range of vehicle models currently sold in the U.S. to produce an accurate estimate of how system charge varies with vehicle class and size and the requisite surveys are beyond the scope of this study. However, given the proposed EU leakage rules, it is expected that any estimation error due to this deficiency will be minor.

It is perhaps also worthy of note that relative to the proposed EU leakage standards, only three of the 12 distinct manufacturers (or related manufacturers) had average leakage rates below 40 grams per year. Eight others had leakage rates between 40 and 60 grams per year. Based on the insensitivity of the EU data to vehicle age, it appears that the 40 grams standard will require at least some tightening of current system designs or manufacturing tolerances. However, it is also apparent that some current systems can comply. HFC-134a systems designed to obtain the EU's "advanced system" credits under the proposed 20 gram per year standard will require significant tightening.

Irregular Emissions. By definition, precise estimates of the rate of irregular HFC-134a leakage are also difficult to develop. Various researchers have estimated per vehicle-equivalent irregular emission rates of 7-14 grams per year. [22,25,29] As was the case with regular emission rate estimates, the most robust of the irregular emission rate estimates were derived in an EU-sponsored study that examined vehicle repair records from nine German service garages. [29] In the EU study, irregular A/C service was distinguished from normal A/C service using a refrigerant charge loss cutoff of ±40 percent of initial charge. A/C repairs in which the volume of lost refrigerant exceeded 40 percent of the initial system charge were presumed to be irregular service events on the premise that A/C system performance at a 40 percent or less loss level would still be acceptable in most cases and not indicative of an "irregular" discharge.[21] By comparing the measured system losses of such vehicles[22] to the number of (A/C equipped) customers at each study service garage, a nominal irregular emission rate of 14 grams per A/C-equipped vehicle per year was estimated.

[20] Because the vehicle manufacturer data are coded for proprietary reasons in the EU study, it is not possible to analyze or reweight the data to better reflect actual U.S. fleet makeup.

[21] Only vehicles up to seven years old were considered since they were viewed as representative of "current generation" HFC-134a systems. At a regular loss rate of about 50 grams per year, the oldest such vehicles could be expected to have lost about 350 grams, or about 41 percent of their initial charge (based on a 856 gram initial charge for EU study vehicles) due to normal leakage.

[22] A total of 480 irregular leakage vehicles were identified.

It might be argued that irregular leakage would generally lead to complete or near complete refrigerant loss and that, therefore, the use of a 40 percent charge cutoff might overestimate irregular emissions. However, it is expected that any such overestimation is insignificant for two reasons. First, of the A/C systems opened in the EU study, only 23 percent had refrigerant loss rates between 40 and 100 percent. In contrast, 38 percent exhibited a 100 percent refrigerant loss and these systems accounted for just under 75 percent of the total refrigerant lost by all irregular loss systems. Combined, such systems exhibited a loss of just under 85 percent of initial charge.[23] Second, the study also estimated the average regular HFC-134a emission rate through vehicles exhibiting less than 40 percent total leakage.[24] The resulting estimated regular emission rate of 52 grams compares very well with the regular emission rate estimated through a later EU study [30] as well as with the 50 gram per year rate assumed in this study. Therefore, it appears likely that the vehicles exhibiting greater than 40 percent charge loss are not accounted for in the regular emission rate, so their emissions should be accounted for either through a higher regular emission rate or as a component of the irregular emission rate. Both the EU study and this study assume the latter.

Uncertainty also arises with regard to the applicability of the EU-estimated irregular HFC-134a leakage rate to the U.S. given the potential for differences in fleet makeup and accident behavior. The EU study approach should minimize the affects of fleet makeup since the estimated vehicle emission rate is based on data specific to each surveyed repair garage. The vehicles at each service garage are treated as a distinct population, so that estimated statistics are not EU-specific, but rather garage specific. Therefore, unless a particular service garage is more or less prone to vehicle accident or malfunction driven repairs, associated statistics should be consistent with other services garages subject to similar repair rates. Comparable statistics on overall vehicle accident behavior in the EU and U.S. are not readily available, but statistics on the number of accidents involving personal injury are available from the United Nations Economic Commission for Europe (UNECE). For 1998, the UNECE indicates the accident rate for the U.S. to be about 9.9 accidents per 1,000 vehicles. [31] Corresponding statistics for EU countries range from 1.6 to 10.3, with rates for Germany between 7.5 and 7.9. Assuming overall accident rates are reasonably well correlated with injury accident rates, it appears that an EU-based irregular leakage rate may slightly underestimate the corresponding U.S. rate. However, given the overall uncertainty in the emission rate estimate, the reported difference between German and U.S. accident rates (20-25 percent) does not appear unreasonably large. Thus, the EU estimated irregular emission rate for current HFC-134a A/C systems of 14 grams per vehicle per year has been assumed without change in this study.

Service Emissions. In addition to regular and irregular leakage, refrigerant emissions also occur during system service due to imperfect recycling techniques. A loss of 4-13 percent of total system charge is generally estimated as appropriate for current recycling systems and techniques. [11,22,25,26] Losses for service facilities employing good recycling techniques appear to be on the low end of the range, at approximately 6 percent. Assuming an average initial charge of 750

[23] The primary cause of irregular leakage was condenser leaks as might be expected given the frontal location and high surface area of the component. In total, 42 percent of irregularly leaking vehicles were identified as having condenser leaks. Hose leaks (17 percent) and evaporator leaks (17 percent) were the next leading failure causes.

[24] A total of 216 regular leakage vehicles were identified.

grams and a normal leakage rate of 50 grams per year, vehicles will generally require A/C system service twice during their lifetime.[25] At a 6 percent loss rate, this equates to about 45 grams per service or 90 grams per lifetime.

End-of-Life Emissions. Finally, imperfect recovery at the end of a vehicle's life also contributes to refrigerant emissions. A loss of 10-30 percent of remaining charge has been assumed to be reflective of such emissions, given current recycling systems and techniques. [22,25,26] Losses given good recycling techniques appear to be on the low end of the range at approximately 15 percent. Assuming an average initial charge of 750 grams, a normal leakage rate of 50 grams per year, and two years normal leakage since the last system service, vehicles will generally have about 650 grams of refrigerant remaining in the A/C system at the end of their life. At a 15 percent loss rate, this equates to about 98 grams per vehicle per lifetime.

It should be recognized that the assumed service and end-of-life loss rates are based on good recycling and recovery practices. To the extent that actual practice is either non-existent (i.e., direct atmospheric venting) or poorly performed, refrigerant emissions will be substantially higher. For example, losses associated with atmospheric venting during two service trips during a vehicle lifetime and one end-of-life loss might be 2,500 grams of refrigerant or more. Obviously, this is a significant increase over the 188 grams assumed in this study. Appropriate regulations (in place) and enforcement are, therefore, required to ensure that actual emissions approach those assumed.

Based on the assumptions described, Table X-2 presents a summary of the estimated direct refrigerant emission rates for current HFC-134a A/C systems. As indicated, the estimated total annual direct emission rate per vehicle is 81.3 grams, which over an estimated 12 year lifetime results in total direct emissions of 976 grams. To put this level of emissions in context, it is perhaps appropriate to compare it to the certification tailpipe emissions of CO_2 for a typical light duty vehicle. With a GWP of 1300, 976 grams of HFC-134a is equivalent to 1268.8 kg (1.4 tons) of CO_2. The amount of tailpipe CO_2 emitted is dependent on vehicle fuel economy, but as an example, a 20 mile per gallon (mpg) vehicle[26] will emit just under 66,339 kg (73.1 tons) of CO_2 over a 150,000 mile lifetime. Thus, relative to a 20 mpg vehicle, current direct HFC-134a emissions increase effective CO_2 emissions by just under 2 percent. With no service or end-of-life recycling or recovery, the impact could rise to over 6 percent. Of course, for vehicles with higher fuel economy the relative importance of direct HFC-134a emissions would be larger and vice versa.

Figure X-2 presents a graphical depiction of the relationship between direct HFC-134a emissions and vehicle fuel economy. As indicated, at the estimated average lifetime HFC-134a emission rate of 976 grams, direct refrigerant leakage will increase lifetime vehicle CO_2 equivalent tailpipe emissions by about a half a percent for the lowest fuel economy vehicles and by as much

[25] This service rate is generally applicable independent of initial system charge as manufacturers, in recognition of regular leakage, typically overfill systems with refrigerant to provide a minimum five year period before service is required.

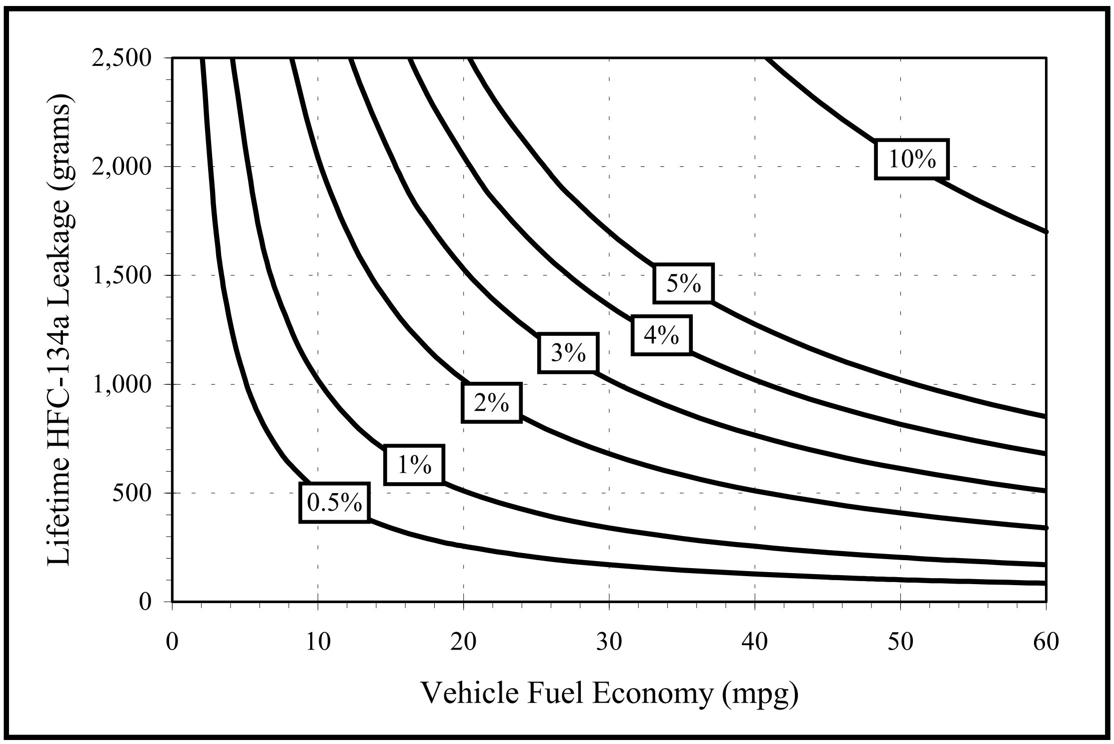
[26] 20 mpg is approximately equal to the current combined light duty vehicle (car and light truck) in-use fuel economy in the U.S. (assuming in-use fuel economy is about 85 percent of the combined certification fuel economy of about 24 mpg).

Table X-2. Baseline Direct Emission Rate for Current HFC-134a A/C Systems

Emission Type	Emission Rate	Lifetime Rate[a]
Pre-Sale HFC-134a Emissions	20 grams	20 grams
Regular HFC-134a Emissions	50 grams/year	600 grams
Irregular HFC-134a Emissions	14 grams/year	168 grams
HFC-134a Service Emissions	45 grams/service	90 grams
End-of-Life HFC-134a Emissions	98 grams	98 grams
Total HFC-134a Emission Rate	81 grams/year[a]	976 grams

[a] Assuming a 12 year vehicle life.

Figure X-2. Direct HFC-134a Emissions Versus Vehicle Fuel Economy



The presented curves depict the equivalent increase in lifetime vehicle CO_2 emissions for a given mass of direct lifetime HFC-134a emissions and vehicle fuel economy. A vehicle lifetime is assumed to be 12 years and 150,000 miles and CO_2 emissions per gallon of gasoline are assumed to be 19.5 pounds.

as 5 percent for the most fuel efficient vehicles. Perhaps also of interest is a brief consideration of direct A/C emissions impact prior to the introduction of HFC-134a. Through the early to mid 1990s, A/C system refrigerant was CFC-12, system charges were substantially larger, and service recycling and recovery was not required. With a GWP 6.5 times and a leakage rate perhaps three times that of HFC-134a, direct emissions for a CFC-12 system were about 20 times as large on a CO_2 equivalent basis. For a 20 mpg vehicle, direct emissions from a CFC-12 system would have been about 40 percent of tailpipe CO_2 emissions. So, even though HFC-134a leakage can be significant, it is about 95 percent less significant than was the case with CFC-12 systems.

There are two basic approaches to further reducing the impact of direct A/C system refrigerant emissions. Under the first approach, structural improvements can be implement to reduce the rate of refrigerant leakage. This approach will impact the regular emissions component of the overall leakage rate, which is estimated to be about 60 percent of the total rate as shown in Table X-2 above. Under the second approach, systems using alternative refrigerants can be introduced in place of HFC-134a. As alluded to in Table X-1 above, there are a number of potential alternative refrigerants offering substantially reduced GWP relative to direct emissions of HFC-134a. Generally, these alternatives span three GWP ranges. HFC-152a is a hydrofluorocarbon refrigerant with half the fluorine of HFC-134a, and a GWP that is 91 percent lower. Hydrocarbons are also effective refrigerants, and their associated GWP is about 98 percent lower than HFC-134a and 80 percent lower than HFC-152a. Finally, zero or near-zero GWP refrigerants include air, CO_2, and ammonia. It is important to recognize that the universe of potential A/C system refrigerants is finitely large, so that all potential alternatives cannot possibly be evaluated in detail. However, those alternative refrigerants for which significant vehicle A/C system research has been performed span a GWP range between zero and current system GWP and, as a result, provide a robust indication of the overall GHG reduction potential of alternative systems.

Most of the ongoing research related to vehicle A/C systems is devoted to HFC-134a system enhancement, HFC-152a, and CO_2. Some work related to hydrocarbons, especially propane, has also been performed. In contrast, research into vehicle applications of either ammonia systems or the air cycle is considerably more limited. Although some *preliminary* theoretical research has indicated that both the air cycle and ammonia systems[27] should be superior to the use of CO_2 as a refrigerant, [9,32] almost all current research into zero (or near zero) GWP vehicle A/C is directed toward CO_2 systems. Both the air cycle and ammonia are viable, as A/C systems based on both are currently in use in non-vehicular applications. The air cycle is used extensively in aircraft applications, where air compression is important for cabin pressurization, etc. and air cycle cooling can be efficiently combined with other compression applications. Ammonia is used in a number of commercial applications. However, ammonia systems are generally sealed

[27] Air-based A/C systems can be either open or closed. In a closed system, confined air serves as a refrigerant in a dual heat exchanger system analogous to current vehicle HFC-134a systems. In an open system, there is no refrigerant per se. Air to be cooled is compressed, passed over a single heat exchanger, and re-expanded before being directed into the passenger cabin. Cooling results from the heat exchange/expansion process. Accordingly, open air cycle systems would represent a departure from current vapor compression systems. Ammonia systems would be closed vapor compression systems analogous to current HFC-134a systems.

due to the toxicity of ammonia and sealed systems are very difficult to implement in vehicle engine compartment applications. Therefore, among the zero (or near zero) GWP refrigerants, this study includes complete emission impact estimates only for CO_2. To the extent that air cycle or ammonia refrigerants are of interest, direct leakage impacts will be similar to those of CO_2 systems. Similarly, there are a wide range of potential hydrocarbon refrigerants, so that detailed analysis of the entire class is resource intensive. However, since all possess the same GWP and generally similar thermodynamic properties, impact estimates for a single hydrocarbon refrigerant can be extrapolated to the entire class. In this study, the potential impacts of hydrocarbon refrigerants are evaluated through an analysis of propane, but estimates for other potential hydrocarbon refrigerants will be similar.

Under this approach, direct refrigerant emissions were investigated for six alternative vehicle A/C systems: current HFC-134a, two levels of enhanced HFC-134a, HFC-152a, propane, and CO_2. From a direct emissions perspective, enhanced HFC-134a systems are based on the proposed EU levels of refrigerant leakage of 40 and 20 grams per year, which reflect regular leakage reductions of about 20 and 60 percent relative to current HFC-134a systems. All alternatives except CO_2 assume an *effective* refrigerant recovery and recycling program in both the service and end-of-life industries. The absence of such a program will lead to emission rates considerably larger than those assumed in this study.

Table X-3 presents the resulting direct refrigerant emission estimates as well as additional associated assumptions. As indicated, reduction in direct GHG emissions due to A/C refrigerant leakage can range from modest for enhanced HFC-134a systems to near 100 percent for propane and CO_2 systems. However, as discussed above and in detail in Section X.7 below, the total GHG impact of all A/C systems must consider both direct and indirect impacts. Accordingly, the tabulated direct emission reduction potentials should be viewed only as partial indicators of overall GHG impacts. These potentials combined with impacts on indirect GHG emissions will determine overall GHG reduction potential. Moreover, these estimates say nothing about either feasibility issues or cost, which are addressed in detail in Section X.8 below.

X.7 Indirect GHG Emissions from Vehicle A/C Systems

Section X.6 above presented estimates for the direct GHG emissions associated with current and alternative vehicle A/C systems due to the leakage of refrigerant. However, A/C systems also affect overall vehicle GHG emissions through an impact on vehicle fuel consumption. These indirect impacts result from two specific influences. First, additional fuel is consumed to “carry around” the mass of the A/C system, regardless of whether or not the system is operating. Thus, this mass-based influence affects vehicle fuel consumption on a continuous basis. Second, when the A/C system is in operation, the compressor and blower motor draw power from the engine, increasing vehicle fuel consumption. To the extent that the impact of these influences varies from that of current HFC-134a systems, changes in indirect GHG emissions can affect the overall GHG potential of alternative A/C systems.

Mass-Based Impacts. Energy required to carry the mass of the A/C system can be estimated using a relationship between vehicle fuel consumption and mass that has been developed though the theoretical analysis of vehicle energy losses and confirmed through decades of vehicle

Table X-3. Direct Emission Estimates for Current and Alternative A/C Systems

	Current HFC-134a System	40 gram HFC-134a System	20 gram HFC-134a System	20 gram HFC-152a System	20 gram Propane System	Porous CO_2 System	50 gram CO_2 System
Initial System Charge (grams)	750	700	600	500	400	650	650
Vehicle Useful Life (years)	12	12	12	12	12	12	12
Recharge Interval (years)	5	5	5	5	5	1	5
Lifetime Recharge Events	2	2	2	2	2	12	2
Regular Leakage Rate (grams/year)	50	40	20	20	20	650	50
Lifetime Pre-Sale Emissions (grams)	20	20	20	20	20	20	20
Lifetime Regular Emissions (grams)	600	480	240	240	240	7,800	600
Lifetime Irregular Emissions (grams)	168	157	134	112	90	146	146
Lifetime Service Emissions (grams)	90	84	72	60	48	468	1,378
End-of-Life Emissions (grams)	98	91	78	65	52	650	650
Total Lifetime Emissions (grams)	976	832	544	497	450	9,084	2,794
GWP (per unit mass)	1,300	1,300	1,300	120	20	1	1
Lifetime CO_2 Equivalent Emissions (kg)	1,269	1,082	707	59.6	9.0	9.1	2.8
CO_2 Equivalent Emissions (kg/year)	106	90	59	5.0	0.8	0.8	0.2
Percent Change from Current HFC-134a	Baseline	-14.8%	-44.3%	-95.3%	-99.3%	-99.3%	-99.8%

Notes: (1) The initial system charge for current HFC-134a systems reflects a typical charge for light duty vehicles. However, charges for some larger vehicles and dual evaporator systems can be over a kilogram.

(2) The initial system charge for all alternative systems is set to provide the indicated system recharge interval for a system with a capacity similar to the current HFC-134a system. The HFC-152a and propane systems are assumed to leak at the same rate as the tightest HFC-134a system (i.e., the same sealing technology applied to the tightest HFC-134a system will function similarly for HFC-152a and propane). Two CO_2 systems are evaluated, one providing essentially no effective refrigerant confinement and one providing confinement similar to current HFC-134a systems. The "porous" system assumes full refrigerant discharge annually and is included to illustrate the GHG risk of ineffective design, not to indicate an acceptable consumer alternative. The "50 gram" system should be viewed as the marketable alternative. It is perhaps also worthy of note that research systems will have lower initial charges than indicated in the table. For example, a typical CO_2 system charge is 450 grams (relative to a 750 gram HFC-134a system). The evaluated systems have been "overcharged" to provide sufficient cooling capacity throughout the intended service interval.

(3) Since there are no alternative systems currently marketed from which to draw information on in-use leakage behavior, the following assumptions were made. Pre-service losses are the same as those of current HFC-134a systems. For HFC-152a and propane this should be reasonably accurate. For CO_2, losses could be higher, but even an error of 2 orders of magnitude has virtually no impact on the potential GHG reduction relative to current HFC-134a systems. Irregular emission rates are set proportional to those of current HFC-134a systems based on the ratio of initial system charges. Per-service emission rates are set proportional to those of current HFC-134a systems based on the ratio of initial system charges, with total service emissions corresponding to the number of service events over the life of the alternative system. Note that even though the "porous" CO_2 system loses its entire refrigerant charge each year, it still has service emissions due to losses during recharging. The "50 gram" CO_2 system has higher service losses because the entire system is presumed to be evacuated directly to the atmosphere during servicing (i.e., no recovery or recycling is assumed). Thus CO_2 loses represent "worst case" losses, while those of all other systems represent losses with an effective recovery program in place. End-of-life emission rates are set proportional to those of current HFC-134a systems based on the ratio of initial system charges, with the exception of CO_2 systems where an entire system charge is assumed to be vented to the atmosphere.

testing. Before examining these estimates, it is perhaps worth repeating that even though the A/C system is installed on vehicles during the certification emissions and fuel consumption testing that precedes all U.S. vehicle sales, the mass of the system is generally not sufficient to influence emission testing results on a scale commensurate with system mass. This is due to the fact that, as described in more detail in Section X.5 above, vehicles are tested in weight increments of 125 pounds up to 4,000 pounds loaded vehicle weight and 250 pounds for heavier vehicles. Since typical A/C system mass is on the order of 30-35 pounds, [4,11,32] its impact on certification fuel consumption is therefore equivalent to either zero, 125, or 250 pounds depending on the specific vehicle being tested and the proximity of the vehicle to the boundary of a test weight bin. In short, the level of mass resolution during certification testing is not sufficient to isolate A/C system mass influences.

A widely accepted rule-of-thumb in the auto industry states that vehicles, over a distance of 10,000 miles, will consume fuel at an approximate rate of 10 gallons per 100 pounds (or 0.00001 gallons per mile per pound). Based on this relation, each pound mass requires the consumption of 1.5 gallons of fuel over a 150,000 mile lifetime.[28] Previous A/C research publications have interpreted this rule-of-thumb as appropriate for determining the incremental fuel consumption due to the additional weight of vehicle A/C systems [23,33]. However, while the rule of thumb can provide a reasonably accurate representation (±15 percent) of *total* vehicle fuel consumption, the error associated with the *incremental* portion of estimated fuel consumption is quite large, overestimating fuel consumption impacts by about 75 percent.

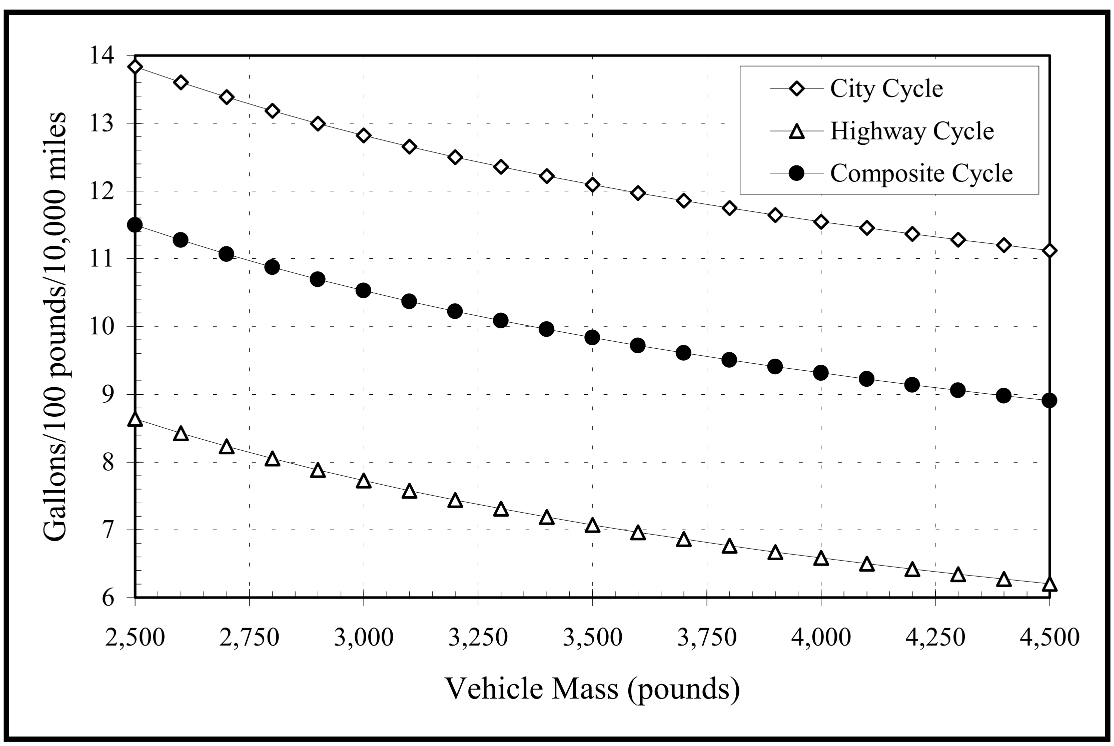
Because the driving cycles associated with U.S. fuel economy certification testing represent well defined vehicle speed versus time profiles, the theoretical fuel consumption of a vehicle over the cycles can be estimated from the forces acting on the vehicle as it is operated according to the varying cycle speeds. The requisite equations were first developed by Sovran and Bohn, General Motors researchers, in the early 1980s. [34] These equations, which allow the mathematical estimation of the energy required to overcome rolling resistance, aerodynamic drag, and vehicle inertia over the required driving cycles can be applied to a given vehicle on the basis of its mass, drag coefficient, frontal area, and tire rolling resistance coefficients. Total energy consumed over the certification cycles is a function of this "tractive" energy combined with energy lost in the driveline, energy used to power accessories, energy used during idle and braking periods, and the energy conversion efficiency of the vehicle engine. These additional energy loads represent about 15 percent of total energy demand over the highway certification cycle and 35 percent of total energy demand over the city certification cycle. [35] It is perhaps interesting to note that the *tractive* energy demand over the city certification cycle is only modestly greater (about 7 percent) than that of the highway cycle on a per-mile basis, but because the city cycle includes significantly more braking and idling (over 40 percent of the total cycle time versus less than 10 percent for the highway cycle), total energy demand is much higher over the city cycle. Fuel consumption demand difference across the cycles is further distinguished by the fact that engine operation during the highway cycle generally occurs under more efficient energy conversion conditions, so that energy input requirements for a given energy demand are less than those of the city cycle.

[28] Thus, a 3,500 pound vehicle would consume 5,250 gallons of fuel over a 150,000 mile lifetime, for a net fuel economy of 28.6 miles per gallon; a value quite consistent with current vehicle certification fuel consumption.

Typical energy demands for the U.S. fuel economy certification cycle can be easily estimated for a variety of vehicle masses, holding other parameters of influence constant. Figure X-3 presents the results of such an exercise. It is clear that for the composite, or combined, certification cycle, the estimated fuel consumption rate does match the rule-of-thumb rate (10 gallons per 100 pounds per 10,000 miles) quite nicely. Over the city cycle, energy requirements are higher due to the added non-tractive demands, while energy requirements over the highway cycle are lower for exactly the opposite reason. It is also clear that the relation between mass and the rate of fuel consumption is not constant, so that incremental fuel consumption estimates derived using an assumed constant fuel consumption rate will be incorrect.

The degree to which such estimates will be in error can easily be determined by comparing the incremental consumption derived using the theoretical calculations and that derived using the rule-of-thumb consumption rate. Under the latter, a 100 pound increment would add 150 gallons to the lifetime (assumed to be 150,000 miles) fuel consumption of a base vehicle. However, the theoretical calculations show the actual increment to be 85 gallons. On an overall fuel consumption basis, this difference is reasonably small (generally within 15 percent of the total

Figure X-3. Vehicle Fuel Consumption by Mass



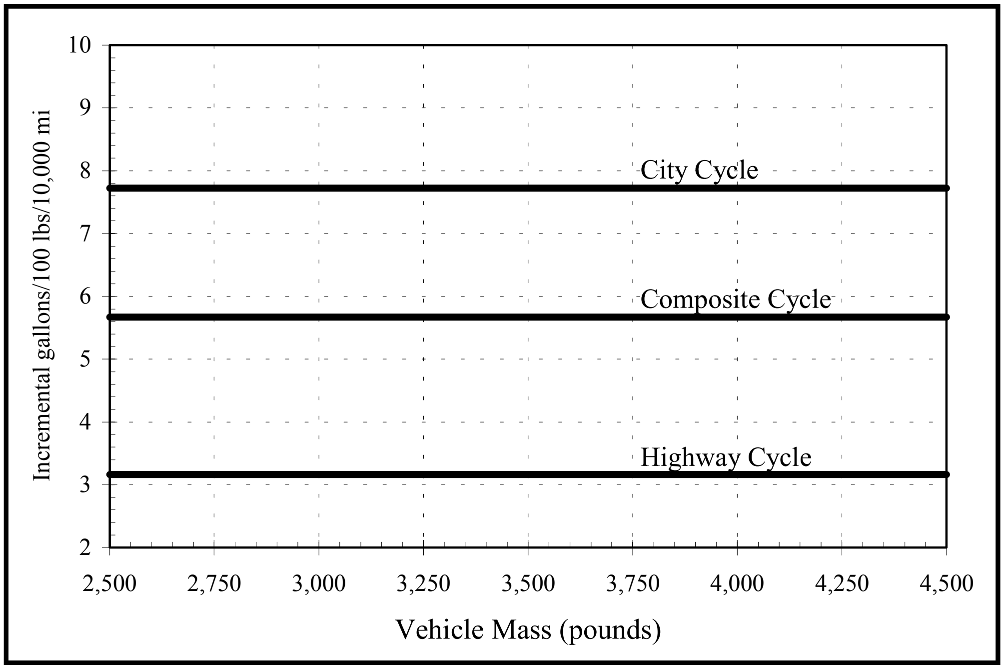
Note: The estimated data assume the following vehicle characteristics (which are held constant across all evaluated masses): coefficient of drag = 0.33, vehicle frontal area = 2.1 square meters, and rolling resistance coefficient (zero order) = 0.011 (all higher order rolling resistance coefficients = 0) consumption), but it is clear that the incremental consumption has been overestimated by approximately 75 percent. This overestimate derives from the fact that a significant portion of the vehicle fuel consumption is *not* a function of mass. While the rolling resistance and inertial components of tractive energy requirements are directly related to vehicle mass, the aerodynamic drag portion is not. Moreover, consumption during idling and accessory load are either unrelated or only weakly related to mass. Non-mass components of energy consumption comprise between 40 and 50 percent of the city cycle (primarily due to vehicle idling loads although aerodynamic drag also contributes over 15 percent of the tractive energy requirements) and between 50 and 60 percent of the highway cycle (primarily due to aerodynamic drag which contributes almost 50 percent of the tractive energy requirements).

Finally, using the theoretical relations, it is possible to correctly estimate the *incremental* rate of change in fuel demand due to an *incremental* change in vehicle mass, which is the parameter of interest in determining A/C system mass impacts. This rate of change is independent of initial vehicle mass (since it excludes the non-mass related energy demands) and is estimated to be about 5.7 gallons per 100 pounds per 10,000 miles over the composite fuel economy certification cycle. Corresponding estimates for the city and highway portions of the cycle are 7.7 and 3.2 gallons per 100 pounds per 10,000 miles respectively. Figure X-4 presents a graphical depiction of the estimates. Using the estimate of 5.7 gallons per 100 pounds per 10,000 miles, it is possible to accurately estimate the mass-related impacts of current and alternative A/C systems.

Current A/C system mass is on the order of 30-35 pounds. [4,33] For this study, a value of 33 pounds is assumed on the basis of detailed (component-by-component) A/C system mass data presented by Modine Manufacturing Company at the 2003 SAE Automotive Alternate Refrigerant Systems Symposium (AARSS). [4] This value was selected on the basis of several factors. First, it is consistent with system mass estimates assumed by other researchers. Second, it was derived by the direct weighing of the components and system charge for a current HFC-134a A/C system removed from a 2002 Jeep Liberty (3.7 liter V6, 4WD). Third, the system charge of 730 grams of HFC-134a is consistent with the "typical" 750 gram change assumed for current HFC-134a systems throughout this study. Fourth, comparative mass data for a replacement CO_2 A/C system installed in the same vehicle are also provided by Modine. Thus a direct comparison of mass for "equivalent" HFC-134a and CO_2 A/C systems is available. Table X-4 presents a breakdown of the system mass data for the two systems.

As indicated in Table X-4, the major components of the current HFC-134a and replacement CO_2 system have essentially identical mass (in total), but the CO_2 system requires more robust plumbing to operate efficiently under the higher system pressures typical of CO_2 systems. Overall, the CO_2 system has a mass about 0.7 pounds greater than the HFC-134a system. It should be noted that this differential is considerably less than differentials estimated by other researchers, which typically range from 7 to 22 pounds. [25,33] However, it is not clear that such alternative estimates consider that the mass savings associated with a smaller compressor, smaller evaporator, and lower refrigerant charge can offset the increased gas cooler mass required in CO_2 systems. Other researchers [e.g., 23] have assumed similar system masses for HFC-134a and CO_2 systems.

Figure X-4. Incremental Vehicle Fuel Consumption per Change in Mass



Note: The estimated data assume the following vehicle characteristics (which are held constant across all evaluated masses): coefficient of drag = 0.33, vehicle frontal area = 2.1 square meters, and rolling resistance coefficient (zero order) = 0.011 (all higher order rolling resistance coefficients = 0)

Table X-4. Mass Comparison of Current HFC-134a and Replacement CO_2 A/C System for a 2002 Jeep Liberty [4]

Component	Supplier	R-744		HFC-134a		Delta
		(kg)	(lbs)	(kg)	(lbs)	
Compressor	LuK	7.07	15.59	7.47	16.47	-5.4%
Hard Lines	Modine	0.28	0.62	0.30	0.66	-6.7%
Flexible Suction Hose	Goodyear	0.93	2.05	0.61	1.34	+52.5%
Flexible Discharge Hose	Parker					
Fixed Orifice Tube	ITW	N/A	N/A	N/A	N/A	
Suction Line Heat Exchanger/Accumulator	Modine	1.90	4.19	1.22	2.69	+55.7%
Pressure Transducer	TI	N/A	N/A	N/A	N/A	
Control System	Modine	N/A	N/A	N/A	N/A	
Evaporator	Modine	1.81	3.99	2.09	4.61	-13.4%
Gas Cooler/Condenser	Modine	3.01	6.64	2.54	5.60	+18.5%
Refrigerant Charge	Modine	0.26	0.57	0.73	1.61	-64.4%
Total System Mass		**15.26**	**33.64**	**14.96**	**32.98**	**+2.0%**
Mass of Compressor, Heat Exchangers, and Charge		14.05	30.97	14.05	30.97	0.0%
Fraction of Mass from Major Components		92%	92%	94%	94%	

Based on additional supporting performance data collected during the AARSS, an effective mass differential between current HFC-134a and alternative CO_2 systems of 0.7 pounds is considered representative. Figure X-5 presents a summary of test ride data collected during the AARSS, showing estimated "comfort indices" for 12 evaluated A/C systems. [36] Six of the systems were CO_2-based, including vehicle "B," which is the Jeep Liberty system developed by Modine. Three systems were current HFC-134a production systems, which provide a benchmark for a qualitative comparative assessment. As indicated in Figure X-5, the Jeep CO_2 system (as well as several other CO_2 systems) performed at least as well as all three of the current HFC-134a production systems. Therefore, it seems reasonable to assume that the Jeep CO_2 system is fairly representative of production CO_2 system potential.[29]

In general, mass information for comparable A/C systems using other refrigerants is limited by the development nature of current research. Based on theoretical calculations, previous research has estimated the potential mass of alternative A/C systems relative to current HFC-134a systems as follows: [11]

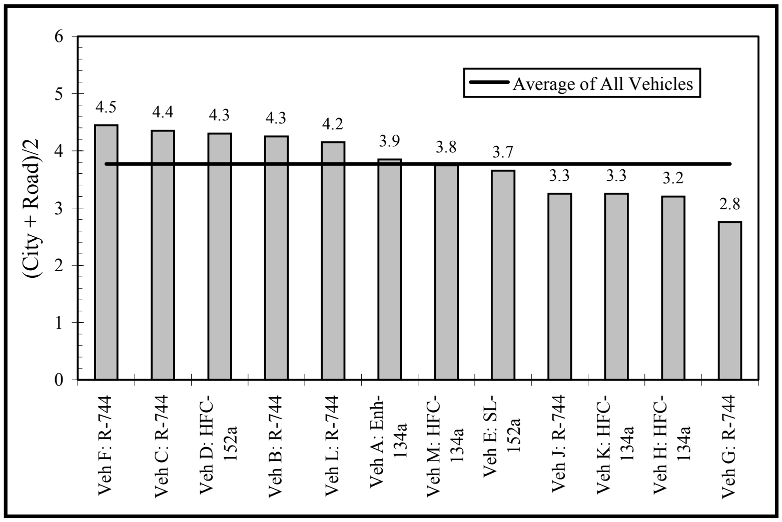
Enhanced HFC-134a	8 percent mass reduction from current HFC-134a
HFC-152a	8 percent mass reduction from current HFC-134a
R-290 (Propane)	8 percent mass reduction from current HFC-134a
R-717 (Ammonia)	8 percent mass reduction from current HFC-134a
R-729 (Air)	16 percent mass reduction from current HFC-134a

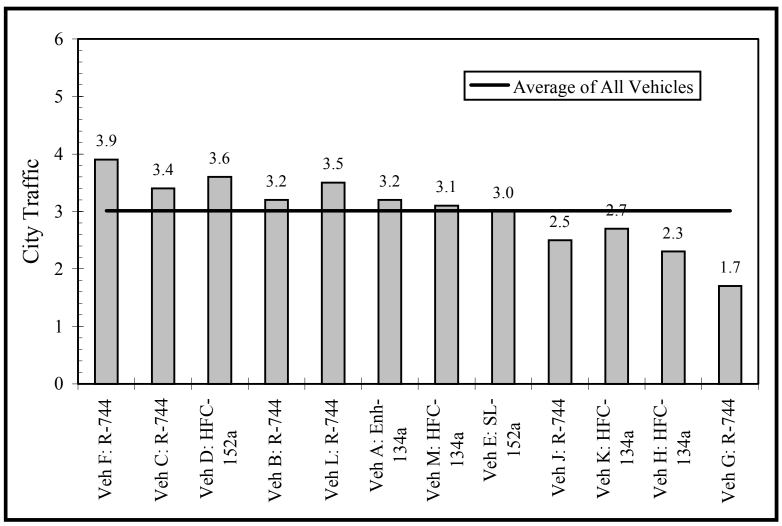
It should be recognized that the same research also estimated a 50 percent increase in the mass of CO_2 systems relative to current HFC-134a. However, given the fact that all of the other systems except the R-729 system are quite similar to current HFC-134a systems, it is expected that the estimates for these systems are more robust than were those for CO_2.

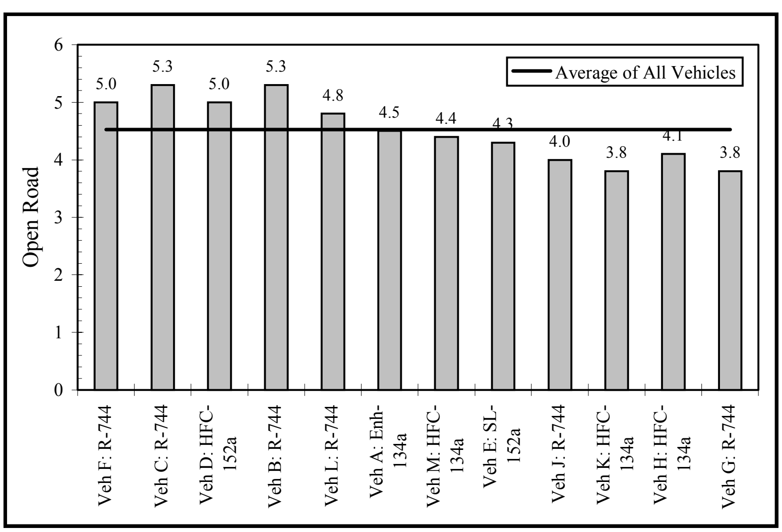
Additionally, secondary loop systems may also increase overall system mass if required to alleviate safety concerns with refrigerants such as HFC-152a, propane, ammonia, or CO_2. As is the case with the base systems, data on secondary loop impacts on overall system mass is limited. Generally, secondary loop systems consist of all of the components of the base system, plus an additional heat exchanger, a small pump to move coolant through the secondary loop, the secondary loop plumbing, the secondary loop coolant, and a secondary loop coolant reservoir. Arthur D. Little, in a study performed for the Alliance for Responsible Atmospheric Policy cites an estimated weight penalty for secondary loop systems of 10-15 percent relative to a conventional system. [38] Based on the 33 pound HFC-134a system weight itemized in Table X-4 above, this would equate to an additional mass for a secondary loop system of only 3 to 5 pounds, which seems quite optimistic for the required components.

[29] It is important, however, to recognize that the systems developed for the AARSS are generally not production systems (with the exception of the base HFC-134a systems). As stated in the Symposium Summary, "In general most demonstration vehicles provided a comparable level of comfort to production HFC-134a vehicles. Again, as in prior years, the system performance varied due to system airflow and panel outlet air temperature. As noted in prior years several demonstration systems were operating without adequate evaporator freeze protection resulting in potential core icing conditions." [36]

Figure X-5. Alternative A/C System Comfort Index: 2003 SAE Alternate Refrigerants Symposium [36]







The Arthur D. Little study cites two Delphi research papers as the source of the mass impact estimate, but a review of both papers for this study reveals no specific information on secondary loop system mass. [12, 39] It is possible that Arthur D. Little performed supplemental investigation into the Delphi research, but that is not apparent from the study report. An alternative, albeit crude, estimate of system mass impacts can be derived through the expected mass of the additional individual components. Based on the weight of current vehicle fuel pumps and small (i.e., 3 to 5 gallon per minute) pumps, it seems likely that the mass for a small secondary loop pump will be approximately 2 pounds. The additional mass of a secondary loop heat exchanger should be approximately equal to the mass of current system heat exchangers, or about 4 to 5 pounds. Finally, secondary loop coolant lines, reservoir, and the secondary coolant itself are likely to add an additional 2 pounds or so, resulting in a total secondary loop mass penalty of 8 to 9 pounds, a total mass impact about 25 percent greater than current single loop systems. Given the apparent optimism of the Arthur D. Little mass impacts and in the absence of alternative data, this study assumes an 8 pound incremental mass impact for secondary loop systems.

Using these data in conjunction with the base A/C system weight data presented in Table X-4 and the incremental fuel consumption for changes in weight as presented in Figure X-4, the mass component of the indirect A/C system GHG impacts can be calculated. Table X-5 presents the resulting emission estimates assuming a useful vehicle life of 150,000 miles. As indicated, the energy required to "carry" a current HFC-134a system is equal to an increase in lifetime fuel consumption of about 28 gallons, which is equivalent to an additional 250 kilograms of CO_2. This represents an emission rate approximately 20 percent as large as the estimated direct (i.e., emission rate of 1,269 kilograms (see Table X-3). However, for additional perspective on overall significance, it is perhaps helpful to recognize that increasing overall fuel efficiency from 25 to 26 mpg would save 231 gallons of gasoline over 150,000 miles, or 8 times as much as reducing A/C system mass to zero. Of course, as vehicle fuel efficiency increases the significance of the A/C system mass impact also increases, and mass represents only one of three A/C system GHG impacts. For example, assuming 19.5 pounds of CO_2 to be equivalent to one gallon of gasoline, the direct GHG impacts of a current HFC-134a A/C system are equivalent to the additional consumption of about 143 gallons of gasoline. Thus together, the mass plus direct emission impacts are about 75 percent as large as the impact of the example one mpg fuel efficiency increase.

Relative to current HFC-134a systems, the most promising alternative system designs (i.e., enhanced HFC-134a, HFC-152a, propane, and CO_2) are estimated to result in A/C system mass impacts ranging from a 9 percent CO_2 emissions reduction to a 27 percent CO_2 emissions increase. Without secondary loops, both HFC-152a and propane systems would reduce mass impacts by about 9 percent, equivalent to the reduction expected with enhanced HFC-134a systems. With secondary loops, HFC-152a and propane systems would generate increases of about 15 percent. Without a secondary loop, CO_2 impacts would be about 3 percent higher than current HFC-134a systems, while secondary loop impacts would increase the relative impact to about 27 percent. This worst case impact is equivalent to the additional consumption of about 8 gallons of fuel over a vehicle lifetime (150,000 miles).

Table X-5. Indirect Mass-Based Emission Estimates for Current and Alternative A/C Systems

A/C System Type	System Weight (pounds)	Incremental Lifetime Fuel Consumption[a] (gallons)	Incremental Lifetime CO_2 Emissions[b] (kg)	Average Annual CO_2 Emissions[c] (kg/year)	Percent Change from Current HFC-134a
Current HFC-134a	33	28.2	249.6	20.8	Baseline
Enhanced HFC-134a	30	25.7	226.9	18.9	-9.1%
HFC-152a	30	25.7	226.9	18.9	-9.1%
HFC-152a with Secondary Loop	38	32.5	287.4	23.9	+15.2%
Propane (R-290)	30	25.7	226.9	18.9	-9.1%
Propane with Secondary Loop	38	32.5	287.4	23.9	+15.2%
CO_2 (R-744)	34	29.1	257.1	21.4	3.0%
CO_2 with Secondary Loop	42	35.9	317.6	26.5	+27.3%
Ammonia (R-717)	30	25.7	226.9	18.9	-9.1%
Ammonia with Secondary Loop	38	32.5	287.4	23.9	+15.2%
Air (R-729)	28	23.9	211.8	17.6	-15.2%

a. Assuming 5.7 gallons per additional 100 pounds per 10,000 miles over 150,000 lifetime miles.
b. Assuming 19.5 pounds of CO_2 per gasoline gallon consumed.
c. Assuming 12 year average useful life.

Of course, given the development status of most of the alternative A/C systems, there is significant uncertainty regarding the precision of the estimated incremental system masses. However, using the data presented above, this uncertainty can be translated into potential GHG impacts through the recognition that each incremental pound mass equates to the incremental consumption of about 0.9 gallons of gasoline, or an incremental 7.6 kilograms (16.7 pounds) of CO_2 emissions over a vehicle lifetime (150,000 miles). Thus, the values presented in Table X-5 can be easily adjusted as alternative system estimates become more robust.[30]

[30] In this regard, it should be noted that subsequent to the completion of this analysis, the details of the various A/C systems included in the SAE Alternate Refrigerant Cooperative Research Program (ARCRP) were released. The ARCRP was initiated to evaluate the energy efficiency of various mobile A/C systems. As tested in the ARCRP, the mass differential between a comparable enhanced HFC-134a and CO_2 system was about 8 pounds, substantially greater than that assumed in this study. However, both systems had masses less than assumed in this study, at about 24 pounds for HFC-134a (versus 30 in this study) and 32 pounds for CO_2 (versus 34 in this study). The ARCRP also included a secondary loop system, with an incremental mass estimated to be about 17 pounds (versus 8 in this study). If mass-based emissions are recalculated using these estimates, the emission rates presented in Table X-5 would be reduced by about 45.6 kg (3.8 kg/year) for HFC-134a and non-secondary loop HFC-152a and propane systems, while emissions for a non-secondary loop CO_2 system would be reduced by about 15.2 kg (1.3 kg/year). Emissions from secondary loop HFC-152a and propane systems would increase by about 22.8 kg (1.9 kg/year), while secondary loop CO_2 emissions would increase by about 53.2 kg (4.4 kg/year). Since the author has not had an opportunity to critically review the systems included in the ARCRP, it is not possible to more fully evaluate the mass differentials. Regardless, these data are likely as valid as any other data reviewed for this study, and indicate that estimated emissions are accurate only to within about ±20 percent.

Energy-Based Impacts. In addition to the indirect GHG impacts associated with A/C system mass, indirect GHG emission impacts also accrue due to A/C system energy demands. However, unlike direct and indirect mass-related emission impacts, which essentially accrue more or less continuously over a vehicle lifetime, A/C energy demands accrue only when the system is in use. Since A/C system usage will vary with climate, it is not possible to quantify an energy use GHG impact that applies to all areas. Areas with significant cooling demands will promote higher A/C system energy use and higher indirect GHG emissions. Areas with low cooling demands may have overall A/C-related GHG impacts that only modestly exceed those quantified above for direct and indirect mass-based emissions. Thus, indirect energy-based GHG impacts must be viewed as a range across specific geographic areas. Of course, energy use also varies temporally in accordance with annual climate cycles. However, for purposes of overall energy use, these annual cycles can be viewed on an average annual basis so that long term GHG impact estimation is not affected.

It is also important to recognize that energy-based indirect emissions can be affected in two principal ways as viewed from an alternative A/C system perspective. First, the energy demand of current HFC-134a A/C systems is affected by system design, principally through the ability of the system to adjust energy demand to meet a given cooling demand. Most current systems in the U.S. rely on internally (freeze point) controlled fixed displacement compressors (FDCs) that provide a constant flow of refrigerant for any given demand. System control is generally limited to cycling the compressor on and off to maintain evaporator surface temperature above the point at which condensed water from the ambient air would freeze and inhibit system performance. The resulting inability to continuously adjust refrigerant flow to meet cooling demand is relatively inefficient. Alternative system designs are available and can be used to promote increased A/C efficiency. For example, externally controlled variable displacement compressors (VDCs) can be utilized to provide a dynamic system response, so that the flow of refrigerant (and the associated compressor energy demand) can be varied in accordance with cooling demand.[31] This can result in a significant energy demand reduction for most cooling demands, so that the overall energy demand of current HFC-134a systems can be reduced through system redesign, without any modification of refrigerant. Additional enhancements such as improved heat exchangers, the addition of suction line heat exchangers that further cool refrigerant leaving the condenser, and enhanced air management strategies can also provide energy demand reductions relative to current systems. Since such approaches are essentially independent of the system refrigerant, their potential impacts are investigated relative to current HFC-134a system energy demands.

The potential modification of current A/C systems to utilize an alternative refrigerant leads to the second area of potential impact on overall A/C system energy demand. To the extent that

[31] It is important to note that similar efficiency gains may also be possible through the incorporation of external controls in FDC-based systems. Such control could rely on rapid clutch cycling to provide the dynamic system response currently lacking from most A/C designs. It is not the intent of this study to specify a particular technology path that might be employed by any or all manufacturers, but rather to illustrate the potential efficiency improvements that are possible through system redesign. Therefore, while this study bases its conclusions on an A/C system utilizing an externally controlled VDC, it should not be assumed that all solutions providing similar efficiency improvement will rely on this same technology.

alternative refrigerant systems are more or less efficient than HFC-134a systems, overall system energy demand for a given cooling demand will be affected accordingly. Thus, the energy efficiency implications of alternative systems must be assessed to fully evaluate potential GHG impacts. However, since the basic system design enhancements discussed above for improving the energy efficiency of current HFC-134a systems (e.g., externally controlled VDC versus internally controlled FDC, suction line heat exchanger, improved air management strategies) also apply to potential alternative refrigerant systems, the consistency of basic design strategies across systems is critical to the development of comparable GHG impact estimates. Therefore, all alternative system impact estimates in this study assume the same basic design components as an HFC-134a system enhanced to include the energy reduction strategies discussed above and defined in more detail below. In other words, the energy demand impacts for each alternative refrigerant system are determined relative to an enhanced HFC-134a system so that the impacts of the refrigerant switch are isolated.

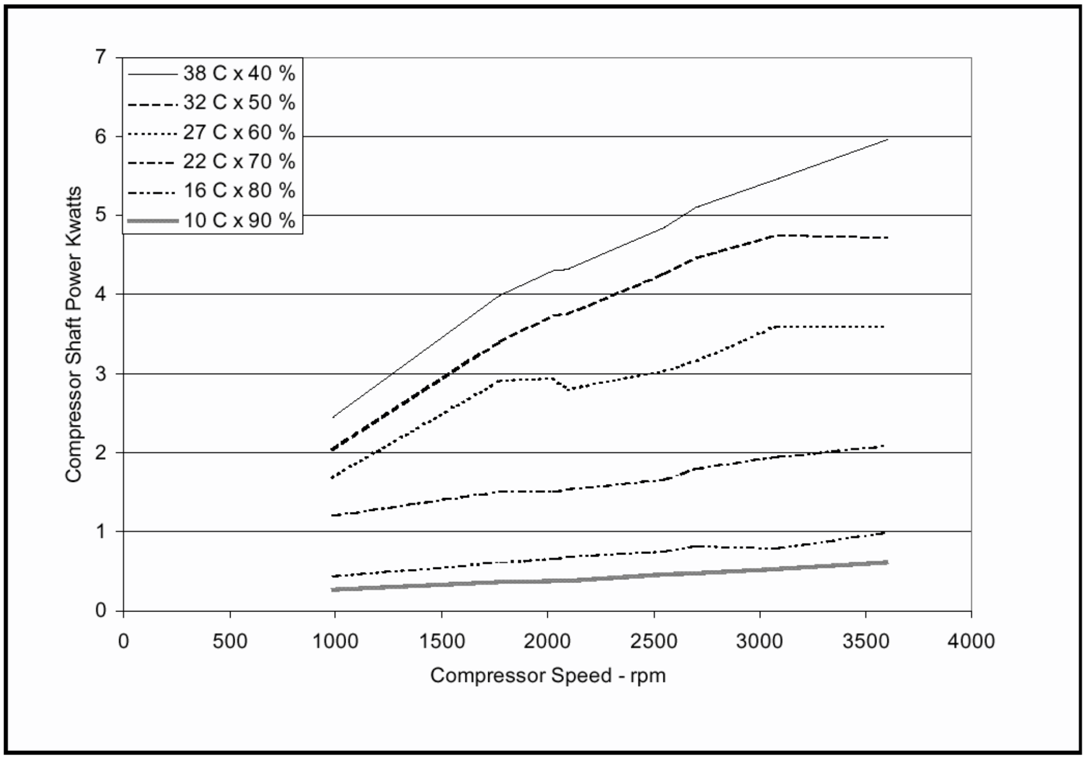
As stated above, an appropriate set of evaluation conditions must first be defined in order to estimate the energy-based impacts of current A/C systems and the potential influences of alternative systems. Compressor (and therefore A/C system) loads vary with a number of operating and design factors. Among these are cooling demand (which is a function of solar radiation, ambient temperature, and humidity), compressor displacement (which is generally a function of vehicle size), compressor shaft speed (which varies with engine speed), airflow (which varies with vehicle speed), and system operating mode (fresh air mode versus recirculation mode). Figure X-6, developed by W.O. Forrest at Delphi, illustrates the typical magnitude of such variation for an A/C system utilizing a freeze point (i.e., internally) controlled FDC of 210 cubic centimeters (cc) displacement. [40]

Factors such as window glazing, interior color, cabin insulation, variations in occupant comfort, and a host of other influences can also affect cooling demand. However, these latter factors, while affecting overall cooling demand, are also generally independent of A/C system design. Thus, such factors can be treated as invariant in comparing the potential impacts of alternative A/C systems. Nevertheless, it should be recognized that additional energy demand reductions can be achieved for A/C systems of any design through optimization of these independent cooling demand influences.

For purposes of this study, a vehicle miles of travel (VMT) weighted average set of evaluation conditions has been developed to compare alternative A/C system energy consumption. However, it should be recognized that impacts in most areas will vary somewhat from those estimated in accordance with local conditions. In an effort to characterize the practical extent of this variation, the study also includes impact estimates for areas with below and above average cooling demands. In all cases, typical vehicle evaluation parameters are utilized since the expected variation in average vehicle characteristics is assumed to be minor relative to variation in average cooling demand.

The National Renewable Energy Laboratory (NREL) has preformed considerable analysis in an effort to quantify typical vehicle cooling demands in the U.S. [41,42,43] As part of this work, NREL has developed "Typical Meteorological Year" (TMY) data for 239 cities throughout the U.S. TMY reflect expected hour-by-hour meteorological data for each city based on thirty years

Figure X-6. Illustrative Variation in Power Demand for a Pneumatically Controlled 210 cc FDC A/C System (Forrest-2002 [40])



Note: The power demand figures include the effect of compressor "on/off" cycling as required to maintain efficient evaporator function.

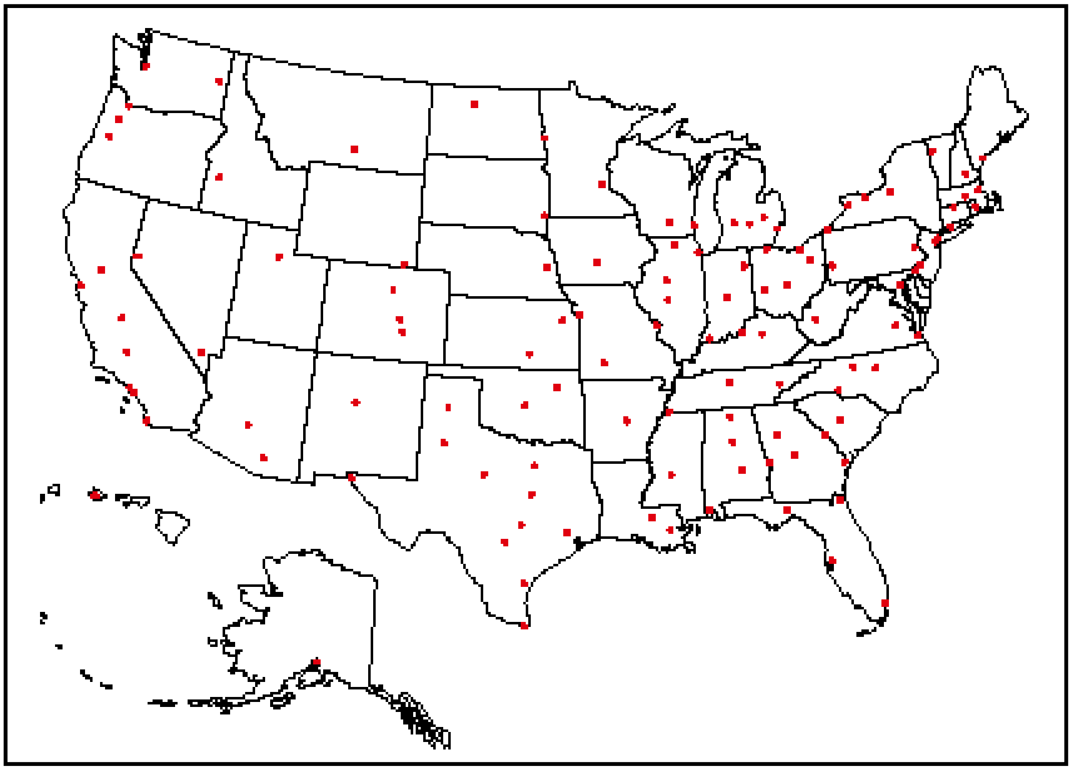
(1961-1990) of data collected by the National Weather Service. NREL subjected data for a geographically diverse subset of 116 of these cities, as presented in Figure X-7, to detailed "thermal comfort" analysis to estimate the frequency and conditions associated with typical vehicle A/C usage.[32] Under this thermal comfort analysis, NREL estimates the fraction of hours during the TMY that A/C is activated (for either cooling or demisting) in each city. These estimates are then combined with variation in trip behavior by time-of-day and time-of-year to generate overall VMT-weighted A/C usage estimates for each city. For states in which multiple cities were analyzed, overall statewide VMT-weighted A/C usage and associated meteorological conditions were developed by aggregating the individual city data in accordance with population.[33] Table X-6 and Figure X-8 summarize the resulting estimates.

As indicated in Table X-6, the NREL data estimate state-specific VMT-weighted average temperature and relative humidity during A/C operation to range from 63-86ºF (17-30ºC) and

[32] The 116 cities were generally selected on the basis of population (greater than 100,000 people), but also include at least one city from each state.

[33] Readers interested in more detail regarding the NREL analysis are referred to references 41, 42, and 43.

Figure X-7. Cities Used to Estimate Expected A/C Usage and Associated Meteorological Conditions (NREL-2002 [41])



31-81 percent respectively. Weighting the individual state data by VMT, produces estimates of U.S. average conditions during A/C usage of 77ºF (25ºC) and 69 percent relative humidity. [44] Because both temperature and humidity affect the amount of energy that must be removed (as heat) from ambient air to produce a desired level of cooling, the conversion of these data to specific enthalpy provides a more robust measure of average cooling demand for a given area. It should also be noted that solar load also affects A/C system usage, and local solar load data are considered explicitly by NREL for each of the 116 cities in determining the fraction of VMT that is accumulated with the A/C system activated (as indicated in Table X-6).

Specific enthalpy indicates the energy per unit mass, which for ambient air can be calculated from temperature and humidity data at a given atmospheric pressure. Since the NREL data available for this study did not include pressure data, pressures were estimated for each of the 116 cities in the NREL dataset on the basis of elevation and standard U.S. atmospheric pressures. Specifically, elevation and population data for each of the 116 cities were obtained from geographic and census datasets. [45,46,47,48] State-specific elevation data were calculated as the population-weighted elevation of the individual cities represented in each state. A U.S. average elevation estimate was also developed by VMT-weighting the state estimates in a fashion identical to that used to produce U.S. average temperature and humidity estimates. Thus, both individual state and U.S. average elevation estimates are consistent with the geographic basis of the NREL temperature and humidity data.

Table X-6. Frequency of Use and Typical Conditions during A/C System Operation

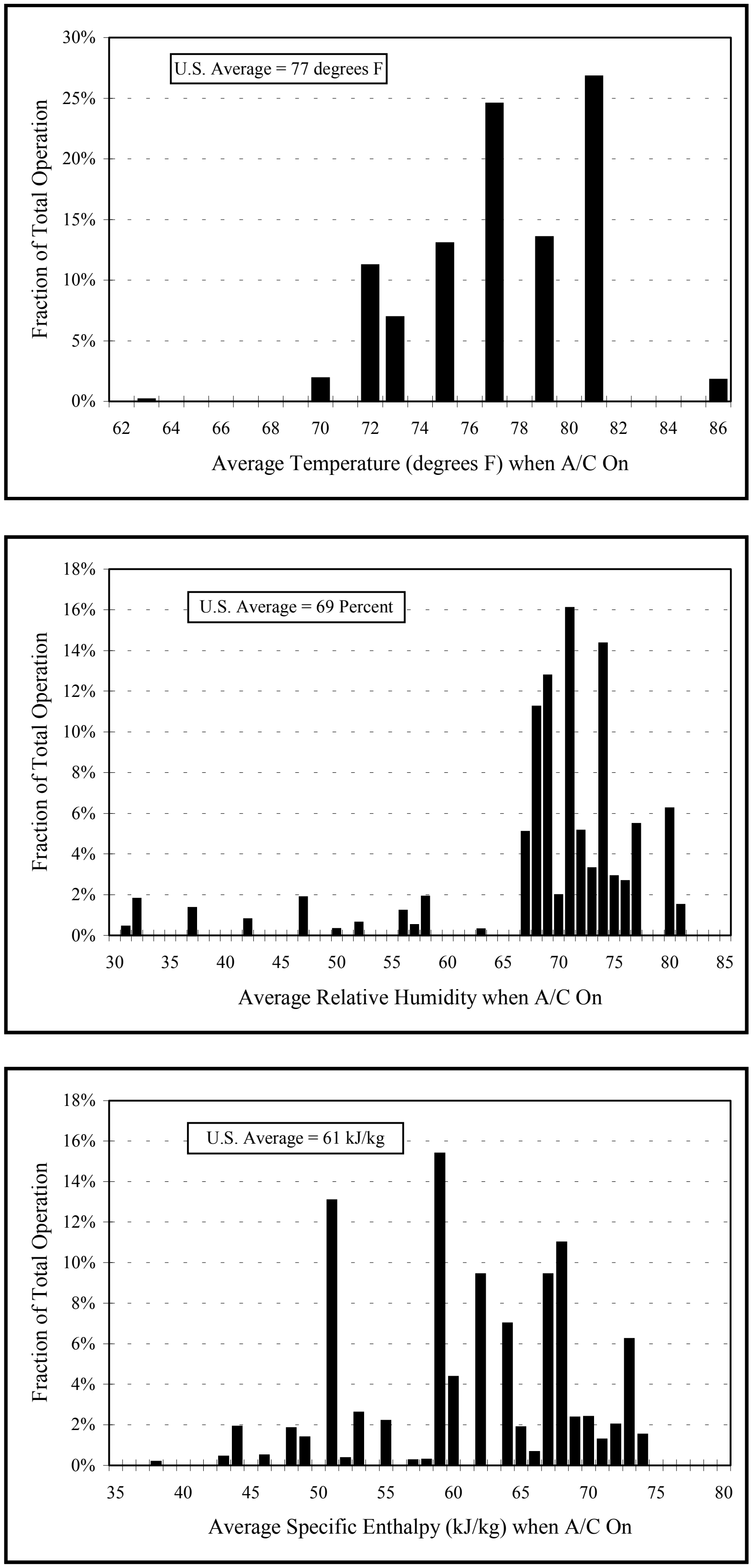
State	2002 Annual VMT (million miles)	Number of Cities Represented	Percent of VMT with A/C On	Average Conditions when A/C Operating			
				Temperature (degrees C)	Temperature (degrees F)	Relative Humidity (percent)	Specific Enthalpy (kJ/kg)
United States	2,855,756	116	34	25	77.0	69	60.9
Louisiana	43,295	2	50	27	80.6	81	73.8
Florida	178,367	4	57	27	80.6	80	73.2
Alabama	57,515	4	43	27	80.6	76	71.5
Mississippi	36,429	1	52	27	80.6	75	70.6
Tennessee	68,229	3	40	27	80.6	74	70.5
Hawaii	8,886	1	69	27	80.6	73	69.0
Arkansas	30,080	1	43	27	80.6	72	68.8
Kansas	28,443	2	33	27	80.6	69	68.5
Oklahoma	45,731	2	40	27	80.6	69	68.2
South Carolina	47,290	1	50	27	80.6	70	67.7
Texas	221,026	11	49	27	80.6	69	67.6
Georgia	108,321	5	41	26	78.8	74	67.3
Missouri	68,163	3	33	26	78.8	74	67.2
North Carolina	92,894	3	38	26	78.8	74	67.1
Nebraska	18,719	1	26	25	77.0	76	65.5
Maryland	57,249	1	28	26	78.8	71	64.6
Virginia	77,450	2	31	25	77.0	77	64.3
Kentucky	46,841	2	30	25	77.0	75	64.1
Indiana	72,523	3	32	25	77.0	74	63.7
Iowa	30,847	1	26	25	77.0	71	62.4
Illinois	105,401	4	25	25	77.0	71	62.0
Delaware	8,875	1	28	25	77.0	72	61.7
West Virginia	20,005	1	27	24	75.2	77	61.7
Pennsylvania	104,476	3	26	25	77.0	71	61.6
New Jersey	69,942	2	23	25	77.0	69	60.2
Minnesota	54,562	1	20	24	75.2	73	60.0
Ohio	107,861	5	25	24	75.2	72	59.5
New York	133,057	4	21	25	77.0	67	59.3
Connecticut	31,205	2	21	24	75.2	73	59.1
Michigan	100,144	4	24	24	75.2	71	58.8
South Dakota	8,499	1	23	25	77.0	63	58.7
Wisconsin	58,746	2	17	23	73.4	77	58.6
Rhode Island	8,142	1	17	24	75.2	71	58.0
North Dakota	7,336	2	19	24	75.2	67	57.4
Massachusetts	53,266	2	16	23	73.4	71	54.9
Vermont	9,677	1	18	23	73.4	70	54.8
New Mexico	22,789	1	32	26	78.8	42	53.5
Arizona	51,334	2	58	30	86.0	32	52.9
Montana	10,395	1	18	25	77.0	47	51.7
Colorado	43,545	3	21	24	75.2	47	51.3
Wyoming	9,007	1	15	23	73.4	50	51.1
California	320,942	7	29	22	71.6	68	50.8
Utah	24,564	1	26	26	78.8	37	49.2
Maine	14,727	1	37	23	73.4	57	48.6
Nevada	17,966	2	41	23	73.4	52	48.5
Oregon	34,578	3	42	23	73.4	56	48.2
Idaho	14,167	1	23	25	77.0	37	45.7
Washington	54,776	2	25	21	69.8	58	44.4
New Hampshire	12,578	1	19	26	78.8	31	42.8
Alaska	4,896	1	6	17	62.6	67	37.6

Notes: (1) VMT data are from the Federal Highway Administration, *Highway Statistics 2002*. [44]

(2) Data for the number of cities from which state data are developed, percent of VMT with A/C operating, average temperature and average humidity with A/C operating are from the National Renewable Energy Laboratory. [41,42]

(3) Data on the specific (ambient) enthalpy with A/C operating are calculated from average temperature and humidity data, in conjunction with calculated standard atmospheric pressure for the population-weighted elevation of represented cities.

Figure X-8. Average Ambient A/C Operating Conditions



Ambient pressure estimates were developed from the elevation estimates using U.S. standard atmospheric data (e.g., 1013.25 millibar (mbar) pressure at sea level). [49] Resulting pressure estimates range from 810-1013 mbar for elevations ranging from 7-6067 feet. The U.S. average estimate is 990 mbar for an elevation of 656 feet. These estimates were used in conjunction with the NREL temperature and humidity data to estimate the average enthalpy of ambient air during A/C system operation. [49,50] The resulting estimates are presented in Table X-6 (which is sorted by decreasing enthalpy) and Figure X-8. State-specific average enthalpies during A/C system use range from about 38-74 kilojoules (kJ) per kilogram (kg), with the U.S. average estimated at 61 kJ/kg.

It can be seen from Table X-6 that the highest enthalpy estimates are generally for states in the southeastern and southern midsection portions of the country, where high temperature conditions are combined with high humidity. While higher enthalpy is indicative of a higher cooling load on the A/C system, overall system impacts also depend on the overall frequency of system use. Table X-7 redisplays the A/C system usage rates from Table X-6, but sorted in order of decreasing frequency. Figure X-9 presents a frequency distribution of usage rates in the U.S. As indicated in Table X-7, states in the desert southwest join those in the southeastern and southern midsection portions of the country in demonstrating high A/C system usage rates. It is perhaps worthy of note that a several of the NREL estimated usage rates appear incongruous with associated temperature and humidity data and A/C activation rates in neighboring states. Specifically, both Oregon and Maine exhibit above average A/C system usage rates for near-minimum average enthalpies, especially when considered relative to neighboring states. Although below average, Indiana also displays incongruous behavior relative to its neighbors. Higher than typical demisting demand may explain some of the incongruity for both Oregon and Maine, but the data required to confirm this are unavailable for this study. Regardless, none of these states exhibit sufficient influence to significantly affect the overall U.S. average data used for estimating A/C system energy consumption in this study.

As shown in Figure X-6 above, A/C compressor power consumption varies with ambient conditions. Using the NREL data to reflect average ambient operating conditions, average compressor power consumption can be estimated. Essentially, this involves interpolating between the various consumption curves presented in Figure X-6 to arrive at a typical average consumption rate. Using the average U.S. atmospheric pressure estimate derived above (990 mbar), the temperature and humidity test conditions defined in Figure X-6 are approximately equivalent to the enthalpy data presented in Table X-8. From these data, it is clear that tests conducted at 27ºC (80.6ºF) and 60 percent relative humidity differ from U.S. average A/C operating conditions by only about 2 percent. Moreover, high average A/C use conditions typical of the southeastern U.S. (Louisiana in particular) are approximately midway between A/C system tests conducted at 32ºC (89.6ºF) and 50 percent relative humidity and 38ºC (100.4ºF) and 40 percent relative humidity. Finally, A/C system tests conducted at 16ºC (60.8ºF) and 80 percent relative humidity are within 4 percent of the low average A/C usage conditions encountered in Alaska.

Because weather extremes in the U.S. can vary from average by a significant extent, it is important to evaluate the potential impact of such extremes. Temperatures in southwestern states

Table X-7. Frequency of A/C Use by State

Percent of VMT with A/C On	State(s)	Percent of VMT with A/C On	State(s)
69	Hawaii	30	Kentucky
58	Arizona	29	California
57	Florida	28	Delaware, District of Columbia, Maryland
52	Mississippi	27	West Virginia
50	Louisiana, South Carolina	26	Iowa, Nebraska, Pennsylvania, Utah
49	Texas	25	Illinois, Ohio, Washington
43	Alabama, Arkansas	24	Michigan
42	Oregon	23	Idaho, New Jersey, South Dakota
41	Georgia, Nevada	21	Colorado, Connecticut, New York
40	Oklahoma, Tennessee	20	Minnesota
38	North Carolina	19	New Hampshire, North Dakota
37	Maine	18	Montana, Vermont
34	United States VMT-Weighted Average	17	Rhode Island, Wisconsin
33	Kansas, Missouri	16	Massachusetts
32	Indiana, New Mexico	15	Wyoming
31	Virginia	6	Alaska

Figure X-9. Distribution of A/C System Usage Rates

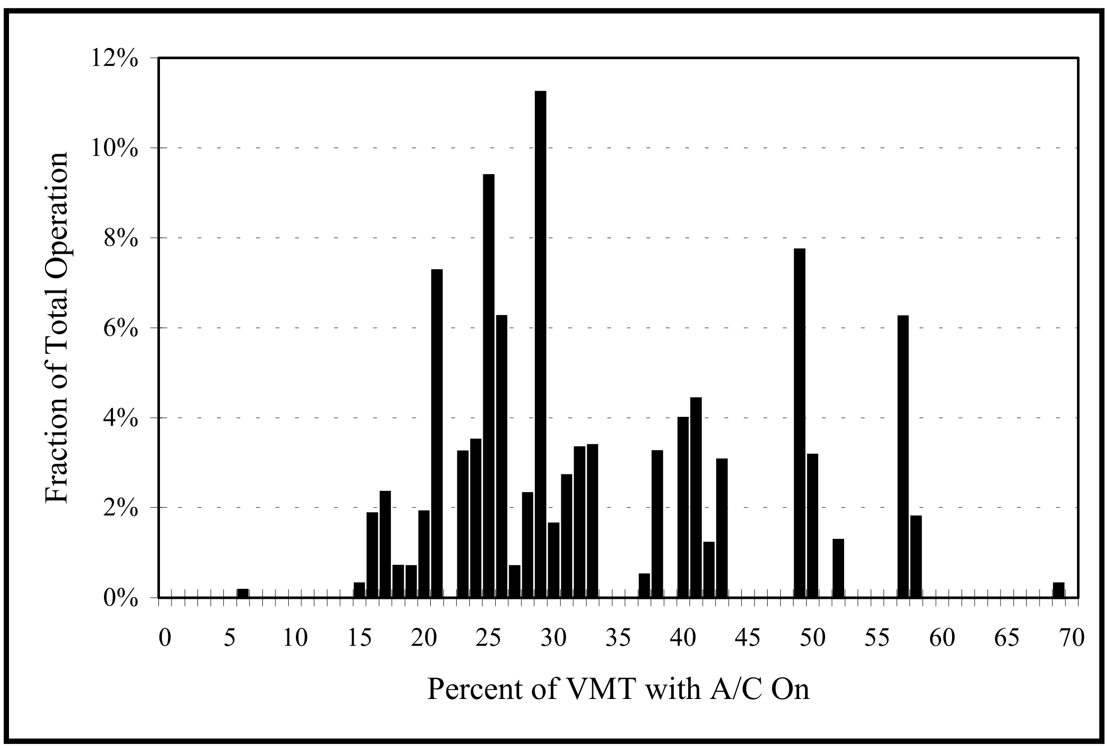


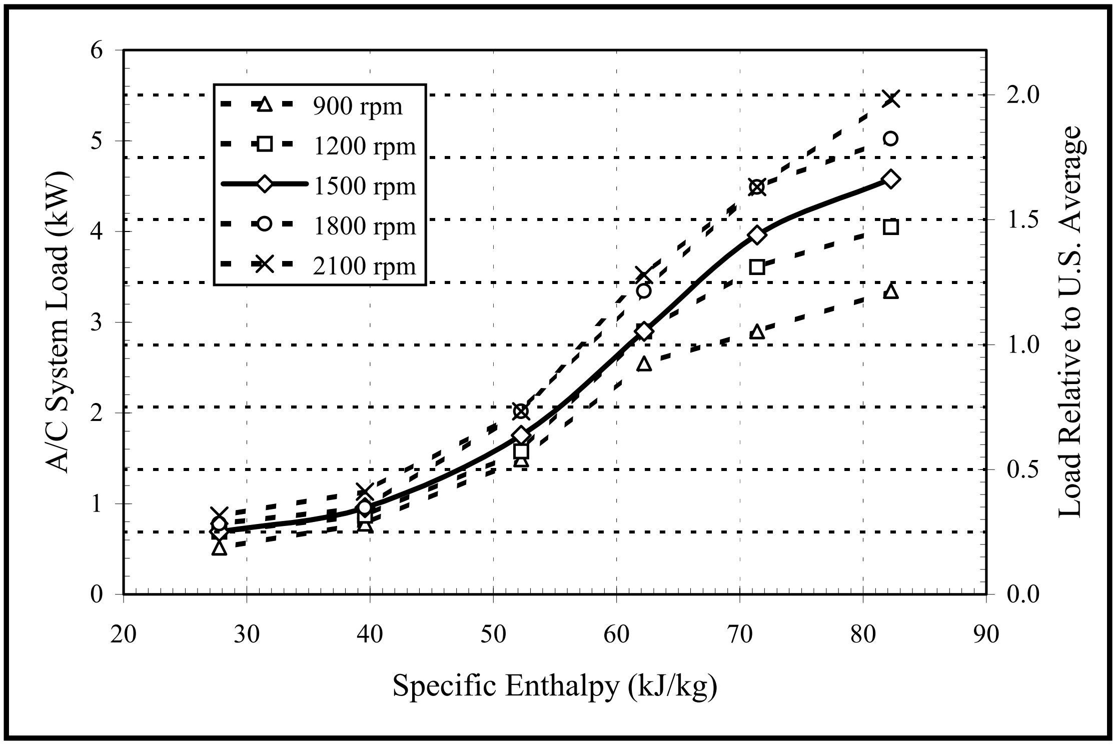
Table X-8. Approximate Enthalpies of Typical A/C System Test Conditions

Test Conditions	Temperature (degrees C)	Temperature (degrees F)	Relative Humidity (percent)	Specific Enthalpy (kJ/kg)
Figure X-6 -- Condition 1	38	100.4	40	82.3
Figure X-6 -- Condition 2	32	89.6	50	71.4
Figure X-6 -- Condition 3	27	80.6	60	62.2
Figure X-6 -- Condition 4	22	71.6	70	52.3
Figure X-6 -- Condition 5	16	60.8	80	39.6
Figure X-6 -- Condition 6	10	50.0	90	27.8
Comparable Ambient Conditions				
U.S. Average A/C Conditions	25	77.0	69	60.9
U.S. High Average (Louisiana)	27	80.6	81	73.8
U.S. Low Average (Alaska)	17	62.6	67	37.6
U.S. High Temperature	45	113.0	20	77.0
SC03 Cycle Conditions	35	95.0	40	72.3

such as Arizona can climb above 110ºF fairly often in the summer months. However, during these excursions, humidity levels are generally limited to about 20 percent or lower. As indicated in Table X-8, the approximate enthalpy for 45ºC (113ºF) and 20 percent relative humidity is about 77 kJ/kg, or "only" about 4 percent higher than the high average A/C use conditions representative of the southeastern U.S. Additionally, as shown in Table X-8, the approximate enthalpy of emissions testing performed over the SC03 air conditioning cycle is also approximately equal to the high average U.S. conditions. Thus, it appears that the range associated with average A/C usage conditions across the U.S. is quite robust in its ability to reflect the broad range of A/C operating conditions. Of course, by definition, there are instances when operating conditions will fall outside the included range (e.g., above average enthalpy conditions in Louisiana), but such excursions will constitute only a small fraction of U.S. VMT since the range between the low and high average conditions is broad enough to capture both above and below average excursions in most states. Finally, to the extent that A/C system power demand is linear (or near linear) with enthalpy, then the use of average ambient conditions to reflect both above and below average conditions will lead to only minor error. To investigate the degree of such linearity, the data presented in Figure X-6 were examined by plotting A/C power demand against enthalpy. Figure X-10 presents the resulting curves.[34] While it is clear that the relations are generally logistic, the rate of change in slope is modest, so that an assumption of linearity for enthalpies between 40 and 80 kJ/kg is reasonably accurate.

[34] It should be noted that the presented data have been converted from compressor shaft power to total A/C system load as described further in the discussion that follows. As a result, the plotted power demand values do not exactly match those presented in Figure X-5.

Figure X-10. Internally Controlled A/C Power Demand Versus Ambient Enthalpy for Various Engine Speeds



Using these data, A/C system energy use and associated GHG impacts can be estimated. However, since A/C system design varies across vehicles, it is first necessary to develop an appropriate characterization of a representative A/C system. As described above, two basic compressor designs and two basic control strategies are available, generally denoted as either fixed displacement or variable displacement and internal control or external control. Both compressor designs and control strategies are present in the current A/C market. In the U.S., internally controlled FDC systems are dominant, but VDC system market share is increasing outside the U.S. As will be described in more detail below, externally controlled VDC systems can offer substantial efficiency advantages over typical internally controlled FDC systems.[35] However, since A/C system efficiency is currently not considered in determining vehicle fuel consumption under the current U.S. regulatory structure, vehicle manufacturers have little incentive for utilizing VDC systems for efficiency reasons. Of course, this is also the case for the EU, where VDC systems are more prevalent.

[35] It is possible that externally controlled FDC systems may be able to offer efficiency improvements similar to externally controlled VDC systems, but research in this area is less advanced than research on externally controlled VDC systems. In effect, the control strategy may be more important than the compressor technology. Given the availability of research on the demonstrated efficiency of externally controlled VDC systems, this study relies on an externally controlled VDC system configuration for determining potential A/C efficiency improvements. Should continuing research on FDC systems demonstrate similar potential, it is possible that advanced A/C systems may continue to include FDC designs.

The driving factor behind greater VDC system penetration outside the U.S. is not reduced fuel consumption, but performance "pleasability." The clutch cycling operation associated with FDC systems places a substantial intermittent "on/off" load on the vehicle engine. For smaller engines, the cycling impact can be both perceptibly dramatic (in terms of changing engine speed) and quite significant in terms of its impact on power available for driving maneuvers. Both the variability and magnitude of A/C system load are lessened dramatically with VDC systems under most operating conditions. Average engine displacement in the EU is less than 2 liters, as opposed to nearly 3.5 liters in the U.S. [51] In fact, only about 6 percent of U.S. light duty vehicle sales include engines under 2 liters displacement. Since larger engines are impacted to a lesser degree by compressor clutch cycling, the performance-based demand for VDC systems is substantially less in the U.S. Therefore, while worldwide production of VDC systems is increasing substantially, most of those systems are being built for non-U.S. applications. As a result, the typical current U.S. A/C system is assumed to be of internally controlled FDC design in this study.

Compressor displacement also varies considerably across the U.S. market. The smallest cars can utilize compressors with displacements as small as 90 cc, while larger light trucks can rely on compressor displacements over 200 cc. For this study, it is assumed that typical compressor sizes range from about 150 cc for a compact car to 210 cc for a large SUV, so that average compressor displacement is about 180 cc given current vehicle sales shares. Since A/C system load is approximately proportional to compressor displacement, the use of a single average compressor displacement should produce accurate fleet average GHG impact estimates. [40]

As indicated in Figure X-6, compressor power demand also varies with compressor speed, which in turn varies with engine speed. Since the compressor is driven by a belt from the engine crankshaft, compressor speed is a function of engine speed and the ratio of the crankshaft pulley diameter to the compressor drive pulley diameter. For this study, a pulley ratio of 1.67 (engine-to-compressor) is assumed, so that engine speed is approximately equal to 0.6 times a given compressor speed. [40] In addition, some energy is lost through inefficiencies in the transmission of power from the engine to the compressor. Typical belt losses are 2-4 percent. [40] An average value of 3 percent is used for this study.

Finally, the total A/C system power also includes the blower motor that draws air across the evaporator and into the passenger cabin. Theoretically, it might also include the radiator fan used to draw air across the condenser, but that fan is required for proper radiator function and is not considered to be an added A/C load in this study. Estimates of blower motor power demand range from 0.1 to 0.5 kilowatts (kW), with typical estimates centering at about 0.25 kW. [9,12,23,42] Accordingly, a value of 0.25 kW is assumed for this study.

Taking all these factors into consideration allows the A/C compressor shaft power curves previously presented in Figure X-6 to be converted into total A/C system load curves by engine speed as follows:

$$\text{Engine Speed (rpm)} = \text{Compressor Speed (rpm)} \times 0.6$$

$$\begin{aligned}\text{System Power (kW)} &= \text{Compressor Shaft Power (kW)} \times \text{Displacement Ratio} \\ &\quad \times \left(\frac{1}{\text{Belt Efficiency}}\right) \\ &\quad + \text{Blower Power (kW)} \\ &= \text{Compressor Shaft Power (kW)} \times \left(\frac{180}{210}\right) \times \left(\frac{1}{0.97}\right) + 0.25\end{aligned}$$

The A/C system loads presented in Figure X-10 above reflect the results of this conversion, so estimated power demands for current HFC-134a systems can be read directly from that figure. However, as is also indicated by the series of curves presented in Figure X-10, power demand varies with engine speed, with high speed demands generally being from 40-70 percent greater than low speed (i.e., idle speed) demands. Since the U.S. fuel consumption estimates against which A/C energy demand will be evaluated are based on the two driving cycles that comprise the CAFE testing protocol (i.e., the UDDS and the HFET cycles), it is appropriate to consider A/C energy demand for the engine speeds generally observed over those same cycles. However, even over the same driving cycle, engine speeds can vary due to differences in engine design, transmission and differential gear ratios, transmission shift schedules, and tire size.

To evaluate the magnitude of this expected variation, the average engine speeds were estimated for the five class-specific design vehicles documented in Section XX of the study report. The specific evaluation characteristics of each of the five vehicles, as well as the resulting cycle average speed estimates are presented in Table X-9. Using the indicated tire diameters and axle ratios, the vehicle speed for any given engine speed and gear selection can be calculated. From these calculations, the engine speed at each shift point can be estimated from the assumed shift schedule, which, in the absence of specific vehicle information, is based on the "default" EPA shift schedule.[36] As shown in the table, the calculated shift speeds appear to be quite reasonable, generally occurring around 2000 rpm (revolutions per minute), so it is expected that the default shift schedule is a reasonable approximation of actual shift patterns. Moreover, it is expected that actual shift schedules are more aggressive (quicker shifts) and since cycle average engine speed is relatively insensitive to quicker shifting due to the narrow speed range that is "available" below the cycle average, it is expected that any error associated with the use of the default schedules will be quite modest.

Based on the estimated shift points, second-by-second vehicle speeds defined in the UDDS and HFET driving cycles can be used in conjunction with the estimated engine/vehicle speed map to produce estimates of the cycle average engine speeds, as indicated in Table X-9. As shown, the average engine speed over the composite of the two cycles is within about 5 percent of 1500 rpm for all five vehicles. Given this similarity, A/C system performance at an average engine speed of 1500 rpm should produce an estimate of typical A/C system impact on vehicle GHG emissions that is consistent with the other (i.e., non-A/C) impact estimates developed in this

[36] For emissions and fuel consumption certification testing purposes, the EPA provides vehicle manufacturers with several options for determining the shift schedules of manual transmission vehicles. Once of these options is a set schedule under which upshifts occur at 15 mph, 25 mph, 40 mph, and each five mph increment thereafter. This is referred to as the "default" shift schedule in this study.

Table X-9. Estimation of CAFE Cycle Engine Speed (Parameters Used and Results)

Parameter		Cavalier	Taurus	Sierra	Tacoma	Town & Country
Engine		2.2L L4 DOHC	3.0L V6 DOHC	5.3L V8 OHV	3.4L V6 DOHC	3.3L V6 OHV
Transmission		A4	A4	A4	A4	A4
Gear Ratios	1st	2.96	2.77	2.48	2.80	2.84
	2nd	1.63	1.54	1.48	1.53	1.57
	3rd	1.00	1.00	1.00	1.00	1.00
	4th	0.68	0.69	0.75	0.71	0.69
	Axle	3.63	3.98	4.10	4.10	3.62
Tire Specifications		P195/70R14	P215/60R16	P235/75R16	P225/75R15	P215/70R15
Tire Circumference (inches)		77.7	82.2	93.9	88.9	84.4
Assumed Idle Speed (rpm)		900	900	900	900	900
Estimated Shift Speeds (rpm) Assuming upshifts at 15, 25, and 40 mph.	1st to 2nd	2190	2130	1720	2050	1930
	2nd to 3rd	2010	1970	1710	1860	1780
	3rd to 4th	1970	2040	1840	1952	1810
Estimated Cycle Average Engine Speed (rpm)	UDDS	1406	1413	1279	1359	1290
	HFET	1697	1781	1729	1730	1580
	Composite	1537	1579	1482	1526	1421
Estimated Cycle Average Engine Speed (rpm) Range and Simple Average	UDDS	range: 1279-1413 average: 1349 variation: - 70/+64 (-5%/+5%)				
	HFET	range: 1580-1781 average: 1703 variation: -123/+78 (-7%/+5%)				
	Composite	range: 1421-1579 average: 1509 variation: - 88/+70 (-6%/+5%)				

study. Thus, the A/C system power demand curve depicted in Figure X-10 for 1500 rpm is used to derive the following energy-related A/C system GHG impact estimates.

It can be easily observed from Figure X-10, that the estimated A/C system power demand at an engine speed of 1500 rpm and enthalpy conditions typical during U.S. average A/C use is approximately 2.75 kW (3.7 hp). For low average and high average U.S. A/C operating conditions (38 and 74 kJ/kg enthalpy respectively), estimated A/C system power demand is 0.9 kW (1.2 hp) and 4.1 kW (5.5 hp) respectively. These data are converted into GHG impact estimates as follows:

1. The additional input energy requirement is estimated by dividing the additional A/C-based energy demand (at the engine) by engine work efficiency. Various studies have demonstrated gasoline engine work efficiency values that range from about 25 percent to over 30 percent. [e.g., 11,52] However, most research

indicates a nominal value of about 28 percent and that value has been assumed for this study.

2. The additional input energy requirement is converted to a gasoline consumption rate (gallons per hour) by dividing the energy requirement by the quantity of energy in a gallon of gasoline. Although the energy content of gasoline varies, a lower heating value of 115,000 Btu per gallon is typical, and that value has been used for this study. [53]

3. The gasoline consumption rate (gallons per hour) is converted to gallons per mile by dividing gallons per hour by the average travel speed during A/C operation. Since all fuel consumption data in this study are defined on the basis of the composite CAFE driving cycle (UDDS+HFET), the composite average speed of those cycles, 32.5 mph, has been used for this study.

4. Additional annual fuel consumption is calculated by multiplying the gallon per mile consumption increase associated with A/C system operation by the average annual vehicle mileage and the fraction of that mileage accumulated with the A/C system activated. For this study, average annual mileage is assumed to be 150,000 lifetime miles divided by a 12 year useful life, or 12,500 miles per year, consistent with the assumptions used to estimate direct A/C system GHG emissions as described in Section X.6 above. The A/C operating fraction was estimated above as 34 percent of VMT for average U.S. conditions and 6 percent and 50 percent for low and high average conditions respectively.

5. The additional annual fuel consumption is converted to CO_2 emissions equivalent by multiplying by 19.5 pounds of CO_2 per additional gallon consumed.

6. A supplemental calculation indicating the approximate fleetwide change in fuel consumption is also performed on the basis of the observed (unadjusted) fleet average fuel consumption during CAFE testing for 2002 model year vehicles. This calculation is for illustrative and comparative purposes only and does not influence the estimated incremental CO_2 emission rate.

Table X-10 presents the resulting GHG impact estimates for a current HFC-134a system of internally controlled FDC design. As indicated, U.S. average A/C operating conditions promote the consumption of an additional 38 gallons of gasoline annually on a per-vehicle basis, with low and high average condition impacts ranging from 2 to 84 gallons. This equates to the additional emissions of 338 kg of CO_2 annually for U.S. average conditions and 20-739 kg of CO_2 annually for low and high average conditions. Impacts of this magnitude indicate that vehicle fuel consumption increases by 7-30 percent (21 percent on average) with the A/C system operating, and annual average fuel consumption increases by 0.4-16 percent (7 percent on average) considering both operating and non-operating periods.

Table X-10. Estimation of Energy-Based A/C Impacts for Current HFC-134a Systems

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	2.75 (3.69)	4.1 (5.5)	0.91 (1.22)
Incremental Input Energy Demand with A/C On (kW (hp))	9.83 (13.18)	14.65 (19.64)	3.25 (4.36)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0090	0.0134	0.0030
Average Annual Incremental Fuel Demand (gal/year)	38.1	83.6	2.2
Additional CO_2 Induced (kg/year)	337.3	739.3	19.7
2002 Baseline Fuel Economy (CAFE, mpg)	23.7		
2002 Baseline Fuel Consumption (CAFE, gal/mile)	0.0422		
Changes to Fuel Economy and Fuel Consumption with A/C System On			
2002 Fuel Economy (CAFE, mpg)	19.5	18.0	22.1
Change in 2002 Fuel Economy (mpg)	-4.2 (-17.5%)	-5.7 (-24.1%)	-1.6 (-6.6%)
2002 Fuel Consumption (CAFE, gal/mile)	0.0512	0.0556	0.0452
Change in 2002 Fuel Consumption (gal/mile)	+0.0090 (+21.3%)	+0.0134 (+31.7%)	+0.0030 (+7.0%)
Average Annual Changes to Fuel Economy and Fuel Consumption			
2002 Fuel Economy (CAFE, mpg)	22.1	20.5	23.6
Change in 2002 Fuel Economy (mpg)	-1.6 (-6.7%)	-3.2 (-13.7%)	-0.1 (-0.4%)
2002 Fuel Consumption (CAFE, gal/mile)	0.0452	0.0489	0.0424
Change in 2002 Fuel Consumption (gal/mile)	+0.0031 (+7.2%)	+0.0067 (+15.8%)	+0.0002 (+0.4%)

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Given the number of assumptions and the analytical nature of several components of the energy consumption calculation, it is perhaps appropriate to compare the fuel consumption impact estimates to actual vehicle test data. It is, of course, important to recognize that not all parameters in any testing program can be expected to exactly match those utilized in this study, but such a comparison can nevertheless provide important insight into the macroscale accuracy of derived impact estimates. In a detailed study of the effects of air conditioning on measured vehicle fuel economy sponsored by the Coordinating Research Council (CRC), the aggregate A/C system fuel economy impact of 12 vehicles tested over the Universal Cycle at a specific enthalpy of approximately 27.3 Btu/pound ranged from about 2.5 to 5 mpg. [54] As indicated in Table X-10, this compares to an estimated impact for average U.S. operating conditions of 4.2 mpg. Enthalpy of 27.3 Btu/pound is equal to about 63.5 kJ/kg, so test conditions in the CRC study are similar, but a bit more aggressive than those assumed for average U.S. condition in this study. However, the base fuel economy of the CRC test vehicles is stated to be 17.4 mpg, which is substantially lower than the 23.7 mpg base used in this study. As a result, a greater impact on fuel consumption is required in the CRC program to produce an mpg impact equivalent to that of this study, so that the impacts estimated in this study are closer to the lower end of the CRC study impact range. However, the specific design parameters of the A/C systems (e.g., compressor displacements) tested in the CRC study are unknown, so it is not possible to conclude with certainty that the estimated impacts are directly comparable. Although the high end range of CRC study impacts appear to be somewhat higher than impacts estimated in this study, the differences are sufficiently small so that differences in underlying assumptions (in this study) and test parameters (in the CRC study) are likely to provide appropriate insight into the rationale for any observed impact differences.[37]

Before addressing alternative A/C system impacts, it is perhaps important to restate that the fuel consumption (and GHG impacts) associated with A/C operation are frequently misportrayed as incremental to vehicle operation without A/C. While it can be argued that this is exactly what the comparative metrics show, the impracticality of such an interpretation can be best understood by considering the likelihood that a vehicle operator (without A/C) would operate a vehicle without maximum ventilation in ambient conditions that would otherwise dictate the use of A/C. Under a realistic interpretation, such an operator would certainly open windows to provide ventilation. Thus, while fuel consumption over the CAFE cycle (or any driving cycle) would be unchanged (as it is for A/C systems today), real world fuel consumption would increase due to the aerodynamic impacts of the open vehicle windows. Given this reality, it is inaccurate to describe the full impacts of A/C system operation as incremental to non-A/C vehicle operation as has been done in numerous studies. The estimated fuel consumption impacts of A/C systems do serve as valuable metrics for comparing alternative systems, but should be considered in the context of a non-zero baseline in determining impacts relative to non-A/C vehicle operation.

While a detailed analysis of the impacts of "window open" operation on vehicle fuel consumption is beyond the scope of this study, a brief discussion of the potential magnitude of such impacts is beneficial in placing the estimates presented in this study in an appropriate context. In an aerodynamic drag study of 14 light duty vehicles, operation with windows down was estimated to reduce composite CAFE cycle fuel economy by 3 to 7 percent. [55] From the data presented in Table X-10, it can be seen that is about 20 to 40 percent of the estimated energy consumption impact of A/C systems for average U.S. operating conditions. However, as indicated above, average vehicle speed over the composite CAFE cycle is only about 30 mph and since energy required to overcome aerodynamic drag increases with the cube of velocity, window down impacts will be substantially larger at higher speeds. For example, the same aerodynamic drag study indicated window down fuel economy impacts to be 4 to 10 percent over the highway portion (48 mph average speed) of the CAFE cycle. Finally, the study compared A/C system impacts to windows down impacts for 3 vehicles operating at 30, 50, 70,

[37] It is perhaps worth noting that the CRC A/C study has been dismissed as inaccurate by some commenters. However, the author has seen no specific rationale for this dismissal and believes that the CRC findings are generally consistent with other limited datasets investigating A/C impacts. For example, a comparison of FTP75 emissions performed by the California Air Resources Board for four vehicles with and without the A/C system in operation found fuel economy impacts ranging from 3-8 mpg. As might be expected, the larger fuel economy impacts were observed for vehicles with a relatively high base fuel economy (25 mpg and greater). If these same data are considered in terms of fuel consumption (gallons per mile), variability is considerably reduced with an average impact of 0.01 ±0.002 gallons per mile. As indicated in Table X-10, the baseline A/C system fuel consumption impact estimated in this study for average U.S. operating conditions is 0.009 gallons per mile.

and 80 mph. A/C system impacts were 4 to 8 percentage points greater at 30 mph, but that differential declined to 0 to 3 percentage points at 70 mph. Clearly, the effect of windows down operation must be considered before ascribing specific incremental fuel use solely to A/C systems. Such an exercise is not conducted in this study given its focus on comparative assessments across alternative A/C systems. Nevertheless, readers should take caution in assuming that A/C power demand is solely responsible for the incremental fuel consumption associated with cabin cooling.

The obvious initial alternative to current HFC-134a systems is an enhanced HFC-134a system that takes advantage of available efficiency improving technologies. Specifically, substitution of an appropriately sized externally controlled VDC in place of an internally controlled FDC can provide substantial efficiency improvement. [12,40,56,57,58] Externally controlled VDC system benefits are derived by tailoring compressor function to the specific cooling demands of the passenger cabin and maintaining a constant reduced level of cooling in place of the "full on/full off" performance characteristics of an internally controlled FDC system. As cooling demands increase, the benefits of the externally controlled VDC system (relative to an internally controlled FDC system) approach zero since the system is forced to operate at full displacement (like an internally controlled FDC system). However, for a wide range of operating modes, the benefits are substantial.[38]

Researchers have also demonstrated the benefits of externally controlled VDC systems in reducing the need for cooled air reheating. [12,40,56,57] However, this benefit primarily accrues to vehicles with automatic climate control systems. Because A/C systems are designed to provide adequate cooling under worst case ambient and engine operating (i.e., idle) conditions, they generally overcool air under milder ambient conditions. Since there is no ability to control cooling capacity with an internally controlled FDC system, vehicles with automatic climate control systems achieve a desired cabin air temperature under widely ranging ambient and operating conditions by passing overcooled air across the vehicle heater core. Thus, energy beyond that actually required to meet passenger cabin cooling demands is expended. In contrast, externally controlled VDC systems can achieve dynamic cooling performance by adjusted refrigerant flow through the A/C system to minimize the necessity for air reheat. However, in vehicles without automatic climate control, the output of the A/C system is controlled either by manually switching the system on/off or by adjusting the blower speed in accordance with cabin occupant desires. Since manually controlled A/C systems continue to dominate the U.S. market, the efficiency benefits of reheat reduction are not considered in this study.[39]

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[38] Once again, it is important to note that the benefits of an externally controlled VDC may be approximated by an externally controlled FDC. However, research demonstrating this capacity in conjunction with simultaneous A/C system stability has yet to be performed. Nevertheless, the important points to remember are that there may be multiple technology paths to a given level of efficiency and that the introduction of external A/C system controls is as, or more, critical than the specific compressor selection.

[39] This does not mean that benefits associated with reduced series reheat will not accrue, simply that no associated credit is taken in this study. In effect, the benefits for the enhanced A/C system assumed in this study *relative to a baseline system in which series reheat is significant* will be greater than estimated in this study (not because the enhanced system is more efficient, but because the baseline system is less efficient). For example, while this study estimates a power consumption reduction of about 55 percent for the enhanced A/C system, other researchers estimate a reduction of 75 percent for the exact same system. Although benefits are not ascribed in this study, reduced series reheat will be an integral component of a high efficiency enhanced A/C system.

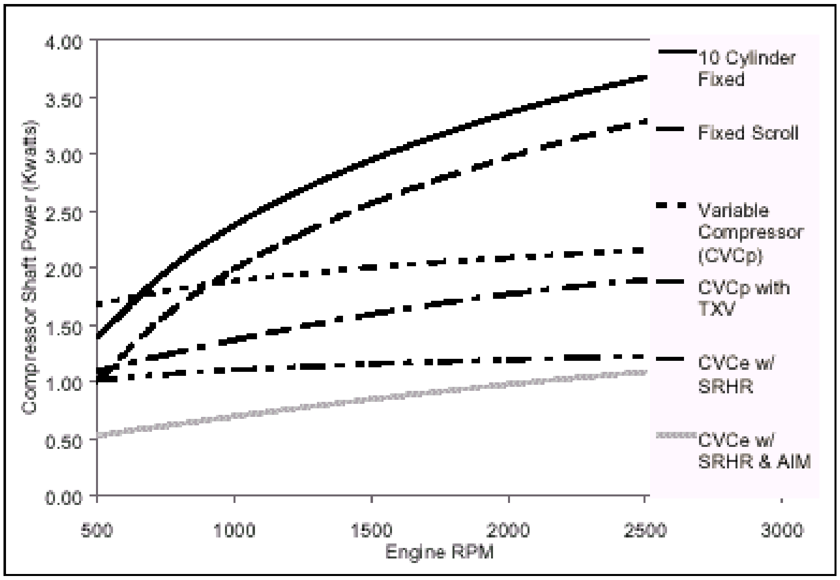
Nevertheless, there are air management strategies that can be used to improve the efficiency of manually (and automatically) controlled A/C systems. [12,40,56,57] Most A/C systems currently offer two user-selectable modes of operation. Under the “normal” (or “outside air”) mode, the A/C system directs a continuous stream of cooled ambient air through the passenger cabin, exhausting displaced air back into the ambient environment. In this mode, the system is continually cooling air from ambient conditions and the cooled air is discharged after a single pass through the passenger cabin. In contrast, in “recirculation” (or “maximum”) mode, the A/C system recycles displaced (and previously cooled) air exiting the passenger cabin back into the A/C intake air stream. Since this air has been previously cooled, it generally requires less heat removal than ambient air to attain a “recooled” state. Thus, less work is required of the A/C system. Air intake management strategies that automate the optimum use of recirculated air can substantially reduce the energy demands of A/C systems.[40]

Figures X-11 and X-12 present an overview of the potential benefits of an externally controlled VDC system utilizing automatic air management strategies. [40,56] In Figure X-11, as developed by W.O. Forrest and M.S. Bhatti at Delphi, the benefits of internally and externally controlled VDC system configurations both alone and in combination with reduced reheat and air intake management strategies are presented for a single set of ambient conditions that are approximately equal to U.S. average A/C operating conditions. Figure X-12, also developed by Forrest at Delphi, illustrates how the compressor power requirements of an externally controlled VDC system in combination with reduced reheat and air intake management strategies vary with ambient conditions. These power requirements can be compared to those presented in Figure X-6 above for a similarly sized internally controlled FDC system. It is important to recognize that these figures are illustrative only in that they are based on systems somewhat larger than that assumed for an average A/C system in this study. Figures X-6 and X-12 are based on a 210 cc displacement FDC system, while Figure X-11 is based on a 215 cc displacement FDC system. This study assumes a 180 cc average FDC system displacement.

To accurately consider the impacts of an enhanced externally controlled VDC-based system employing an air intake management strategy to force cooled air recirculation, the relations presented in Figures X-11 and X-12 were adjusted to reflect a 180 cc base FDC system as well as to factor out the benefits of a reduced reheating strategy (since, as described in footnote 39, the benefits associated with such a strategy were not credited in this study). As indicated in Figure X-11, the efficiency benefits of air intake management are expressed incremental to those of reduced reheating. To factor out the reduced reheating efficiency benefits, the ratio of power demand with both reduced reheat and air intake management to power demand with only reduced reheat was applied to power demand without either the reduced reheat or air intake management strategies. Similarly, the base system displacement adjustment was accomplished by normalizing the power demands for the externally controlled VDC system to those of the corresponding base (internally controlled) FDC system and treating the normalized power demand as representative of the efficiency benefits associated with a non-specific baseline

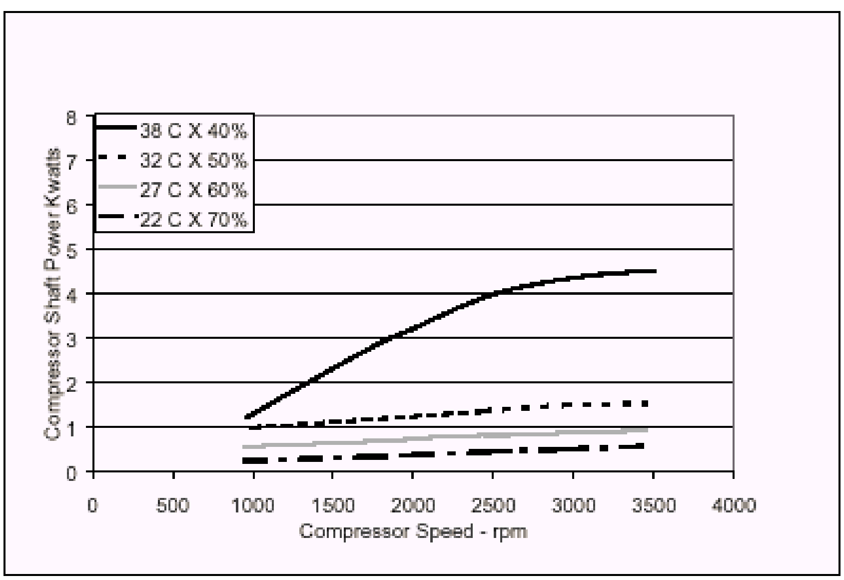
[40] To maintain healthful air quality, such systems do not use 100 percent recirculated air. The implemented system determines the optimum mix of ambient and recirculated air to minimize cooling demand while maintaining a healthful air supply to the passenger cabin.

Figure X-11. Power Demand for Various A/C System Technologies Relative to a 215 cc Internally Controlled FDC System (Forrest-2002 [56])



Note: The power demand figures for FDC technology include the effect of compressor “on/off” cycling as required to maintain efficient evaporator function at ambient conditions of 26.7ºC (80.1ºF) and 60 percent relative humidity (approximately 61 kJ/kg specific enthalpy). “CVC” stands for Compact Variable Compressor, “p” stands for pneumatically controlled, “e” stands for externally controlled, “SHSR” stands for Series Reheat Reduction Strategy, “AIM” stands for Air Inlet Mixture (i.e., forced recirculation), and “TXV” stands for Thermal Expansion Valve.

Figure X-12. Ambient Impacts on Power for an Externally Controlled VDC System with Enhanced Air Management (Forrest-2002 [40])



system replacement. The resulting efficiency benefit ratios are presented in Figure X-13. As indicated, efficiency improvements of 50-60 percent are available over a wide range of engine speeds.

The potential improvements presented in Figure X-13 are specific to certain ambient operating conditions (26.7ºC, or 80.1ºF, and 60 percent relative humidity, or approximately 61 kJ/kg specific enthalpy). Since benefits vary with ambient conditions as indicated in Figure X-12, the relationships presented in that figure were also normalized and applied to the externally controlled VDC system power demands at an engine speed of 1500 rpm to derive a curve for power demand across a range of ambient conditions. Demand at 1500 rpm was selected as most appropriate for a determination of average U.S. impacts based on an analysis of typical engine speeds over the CAFE driving cycles, as described previously above. Figure X-14 presents the resulting power demand curves plotted in conjunction with the 1500 rpm power consumption curve for the base internally controlled FDC system, as previously presented in Figure X-10 above. As might be expected, the power demand curves show the greatest benefit of externally controlled VDC systems to be in the mid range of typical ambient conditions encountered during A/C system operation. At ambient conditions inducing higher cooling demand, the operating displacement of the externally controlled VDC system approaches that of an internally controlled FDC system and benefits decline. Benefits also decline at low ambient, low cooling demand conditions because the "on" time for an internally controlled FDC clutch cycling system is small, thereby reducing the level of "excess" power demand available for reduction. Using the presented relationships, the specific efficiency improvements for a externally controlled VDC-based A/C system employing an enhanced air intake management strategy can be estimated for average, high average, and low average U.S. A/C operating conditions.

The energy-based GHG impacts for an HFC-134a system enhanced to include an externally controlled VDC in combination with air intake management (i.e., forced air recirculation) are estimated using calculations identical to those described above for a current HFC-134a system using an internally controlled FDC. The only change is the substitution of lower system power demands for average, high average, and low average U.S. operating conditions. For the baseline internally controlled FDC system, those demands were 2.75 kW, 4.10 kW, and 0.91 kW respectively. For the enhanced externally controlled HFC-134a system, the corresponding demands drop to 1.24 kW, 2.28 kW, and 0.51 kW. Blower motor power demand is assumed to remain unchanged at 0.25 kW, as is power transmission efficiency at 97 percent. Under these assumptions, overall incremental CO_2 due to A/C system operation is reduced by 44-55 percent, or 9-328 kg per year (186 on average) as presented in Table X-11.

Externally controlled VDC utilization and improved intake air management can increase the performance of both HFC-134a and alternative A/C systems. Therefore, as stated above, this study assumes that both strategies will be applied to all evaluated alternatives so that the benefits of enhanced HFC-134a systems are not overstated relative to those of alternative refrigerants. However, readers should also be careful to compare alternative refrigerant system impacts to those of an enhanced HFC-134a system (not the baseline internally controlled HFC-134a system) to avoid erroneous conclusions about the potential performance of HFC-134a.

Figure X-13. Normalized Externally Controlled VDC System Power for Restricted Ambient Conditions (26.7ºC/60% Rel. Humidity)

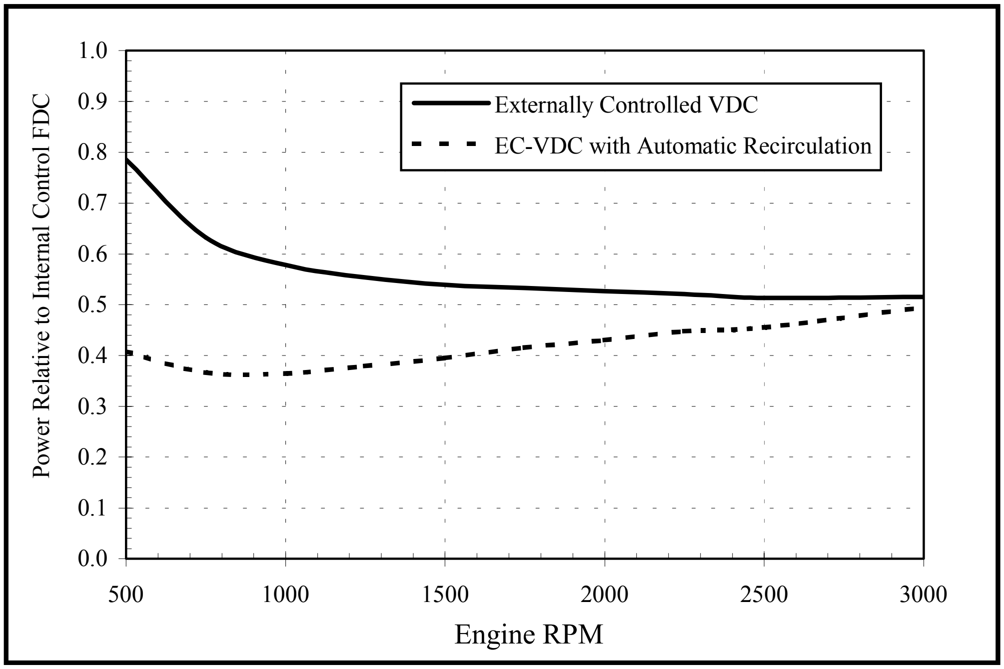


Figure X-14. Externally Controlled VDC System Power Demand Versus Ambient Enthalpy for a 1500 rpm Engine Speed

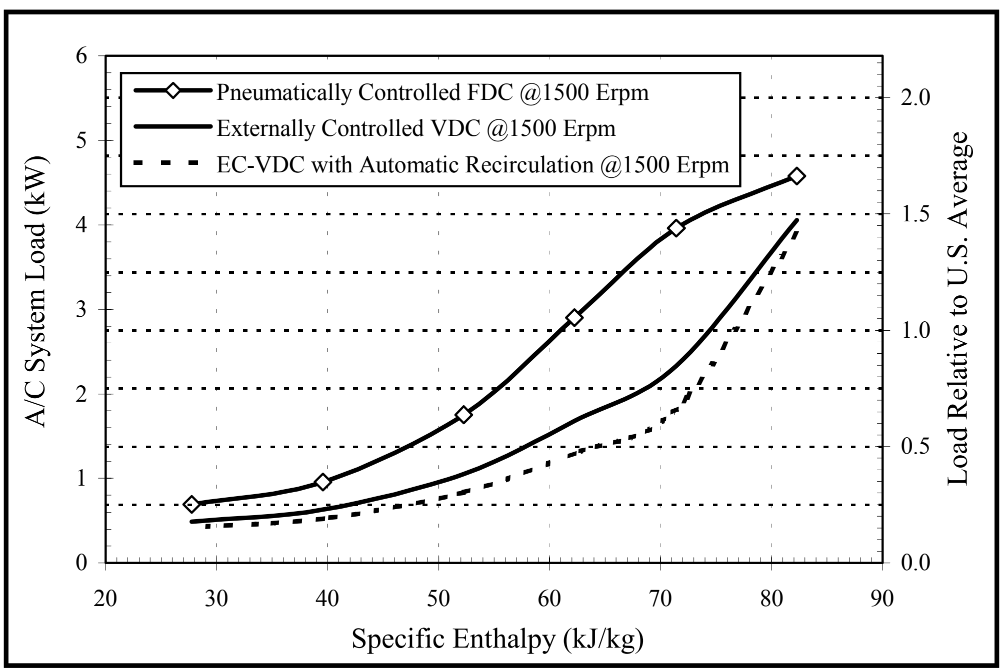


Table X-11. Estimation of Energy-Based A/C Impacts for an Enhanced HFC-134a System Utilizing a VDC in Conjunction with Forced Air Recirculation

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.24 (1.66)	2.28 (3.06)	0.51 (0.68)
Incremental Input Energy Demand with A/C On (kW (hp))	4.42 (5.93)	8.15 (10.93)	1.81 (2.43)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0040	0.0074	0.0017
Average Annual Incremental Fuel Demand (gal/year)	17.2	46.5	1.2
Additional CO_2 Induced (kg/year)	151.7	411.3	11.0
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-185.6	-328.0	-8.7
Change in Added CO_2 from Base HFC-134a System (pct)	-55%	-44%	-44%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

HFC-152a has been shown to have physical, thermodynamic, and transport properties similar to those of HFC-134a, while the corresponding properties for hydrocarbon refrigerants are generally also similar or superior to the hydrofluorocarbon refrigerants. [12] Like HFC-134a, the critical temperature of both HFC-152a and propane is above 200ºF, while all three exhibit normal boiling points below -10ºF. Both HFC-152a and propane also exhibit latent heat, specific heat, and thermal conductivity properties similar to and in most cases superior to those of HFC-134a.

Thus, A/C systems utilizing HFC-152a or propane refrigerants should be capable of transferring heat more efficiently than similar HFC-134a systems. Existing research appears to confirm this, as indicated in Figures X-15 and X-16 which present a summary of available test and simulation data for both alternative refrigerants. [12,40,59,60]

As shown in Figure X-15, there is considerable scatter in the data that has been collected on the performance of HFC-152a systems. Nevertheless, the data consistently show improved efficiency and it appears that this improvement increases with ambient enthalpy. Given the degree of scatter for the various operating conditions evaluated, an “average” estimate of relative system performance was developed using a simple linear regression. The regression predicts about an 11 percent efficiency improvement for enthalpies in the 80 kJ/kg range declining to about 5 percent in the 60 kJ/kg range. Because available evaluation data were limited to the lower end of this range, the regression was not extrapolated but instead held constant for low enthalpy conditions. Using this approach, the estimated power demand reductions (relative to an HFC-134a system) for average, high average, and low average U.S. A/C operating conditions are 5 percent, 8 percent, and 5 percent respectively.

Figure X-15. HFC-152a System Power Demand Relative to HFC-134a System Power Demand

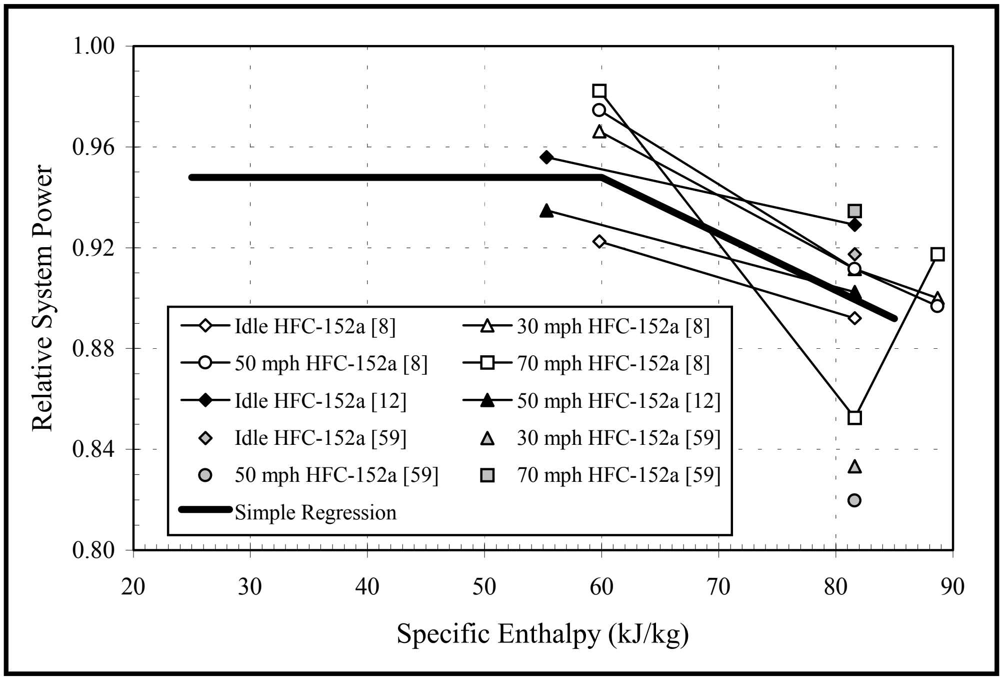


Figure X-16. Propane System Power Demand Relative to HFC-134a System Power Demand

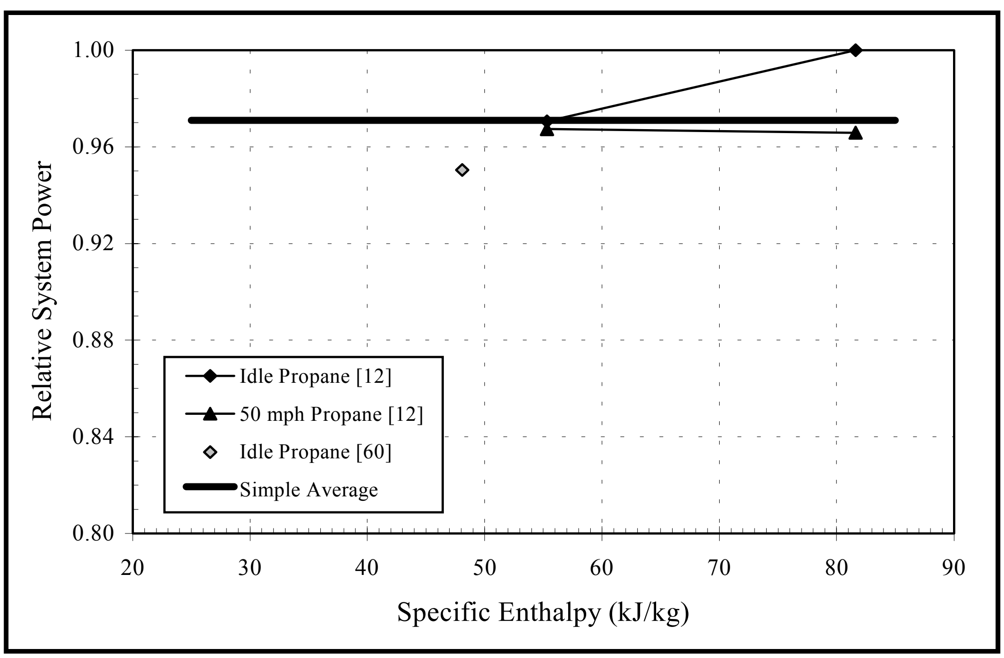


Figure X-16 presents similar evaluation data for propane systems. [12,60] As indicated, there is limited available data, and like data for HFC-152a there is considerable associated scatter. As with the data for HFC-152a, a similar simple regression analysis was conducted, but the estimated regression slope is not significant. Therefore, a simple arithmetic average was selected as the most appropriate approach to estimating power demand reduction for propane systems. As indicated in Figure X-16, the arithmetic average reduction relative to HFC-134a systems is 3 percent (across the entire range of ambient enthalpies). Using these data, the energy-based impacts of HFC-152a and propane systems were estimated using the same approach described above for HFC-134a systems. Tables X-12 and X-13 present the resulting impact estimates.

The energy demand impacts of CO_2-based A/C system have been subject to considerable debate. Early theoretical calculations predicted poor performance for CO_2 systems relative to HFC-134a. [e.g., 11,33] Those calculations were predicated on a series of thermodynamic properties of CO_2 (primarily its low critical temperature) that implied poor performance relative to HFC-134a. However, considerable research has demonstrated that the theoretical concerns are substantially overstated and that in practice CO_2 systems can perform better than HFC-134a systems for a wide range of operating conditions. [e.g., 9,61,62,63] This apparent discrepancy results from the fact that CO_2 possesses better heat transfer properties than HFC-134a, and CO_2 systems operate at reduced pressure ratios resulting in higher compressor efficiency.[41]

For this study, recent research conducted at the University of Illinois was used to estimate CO_2 system energy demand. [63] This research was selected specifically because it compares the performance of a CO_2 system to an enhanced HFC-134a system of equivalent compressor and evaporator design and size. Previous research has often been based on systems that were not comparable from a design standpoint, biasing results toward either HFC-134a or CO_2. It is fairly well accepted that CO_2 systems perform well at low ambient temperatures due to advantageous heat transfer properties, but that this performance advantage declines as temperature increases, so that performance at very high ambient conditions (near or above about 45ºC) can be less efficient than that of HFC-134a systems. The recent University of Illinois work is consistent with these expectations, as indicated in Figure X-17, which presents the relative efficiencies of equivalent CO_2 and HFC-134a systems.[42]

As shown, CO_2 systems possess distinct efficiency advantages at low ambient enthalpies. As one might expect, the magnitude of the efficiency advantage is dependent on compressor speed, and the University of Illinois research includes data for two specific speeds, 900 and 2500 rpm. For convenience, a regression line that includes data from both speeds is also included in Figure X-17. The ratio between compressor speed and engine speed is dictated by the ratio of the respective drive pulley circumferences and is generally determined through the optimization of system performance for a given vehicle. Compressor capacity is directly related to speed, while

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[41] Pressure ratio is compressor output pressure over compressor input pressure and does not imply that the working pressure for CO_2 systems is lower than that of HFC-134a. As is discussed in Section X.8, the working pressure in CO_2 systems is substantially higher than that of HFC-134a systems and this is a concern in regard to the marketability of CO_2 systems.

[42] Reference 63, used to construct Figure X-17, does not provide data on the assumed ambient humidity associated with the A/C system test conditions. However, those data were located in a second reference, [64] so the development of accurate specific enthalpy data as presented in Figure X-17 was possible.

Table X-12. Estimation of Energy-Based A/C Impacts for an HFC-152a System without Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.19 (1.60)	2.12 (2.84)	0.50 (0.67)
Incremental Input Energy Demand with A/C On (kW (hp))	4.26 (5.71)	7.56 (10.14)	1.78 (2.39)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0039	0.0069	0.0016
Average Annual Incremental Fuel Demand (gal/year)	16.5	43.2	1.2
Additional CO_2 Induced (kg/year)	146.2	381.7	10.8
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-191.1	-357.6	-8.9
Change in Added CO_2 from Base HFC-134a System (pct)	-57%	-48%	-45%

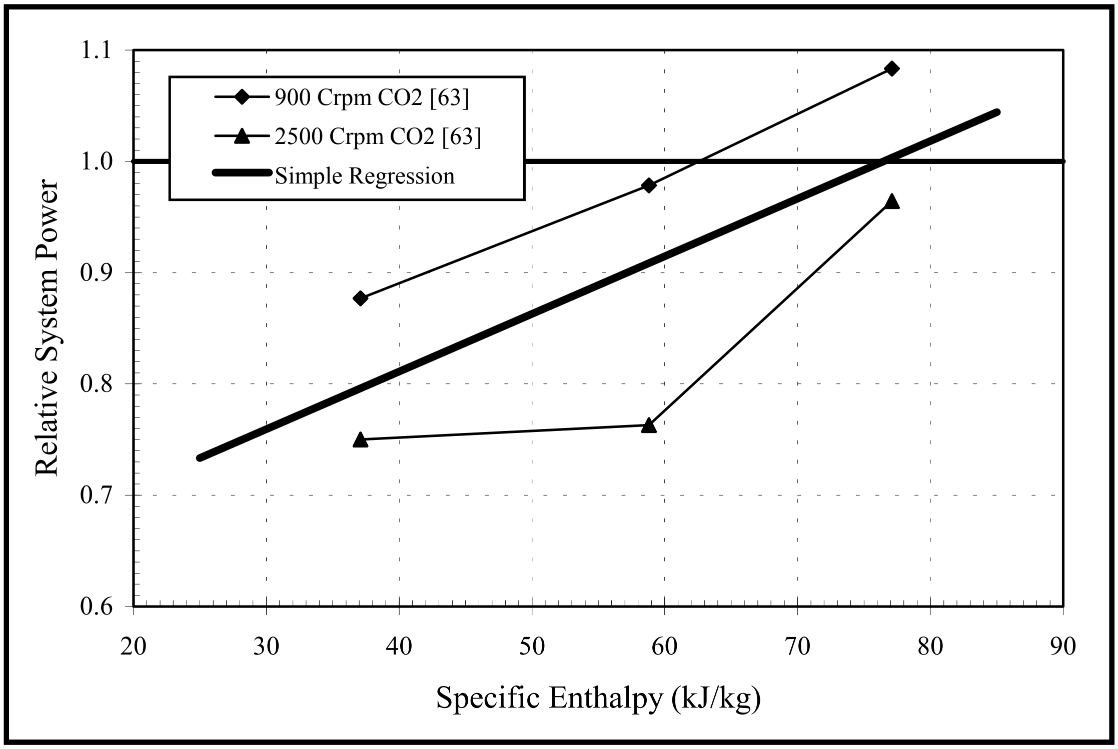
Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Table X-13. Estimation of Energy-Based A/C Impacts for a Propane System without Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.21 (1.62)	2.22 (2.98)	0.50 (0.67)
Incremental Input Energy Demand with A/C On (kW (hp))	4.31 (5.79)	7.94 (10.64)	1.78 (2.39)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0039	0.0072	0.0016
Average Annual Incremental Fuel Demand (gal/year)	16.7	45.3	1.2
Additional CO_2 Induced (kg/year)	148.1	400.6	10.8
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-189.2	-338.7	-8.9
Change in Added CO_2 from Base HFC-134a System (pct)	-56%	-46%	-45%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Figure X-17. CO_2 System Power Demand Relative to HFC-134a System Power Demand



compressor efficiency is indirectly related to speed. Balancing these relations with required cooling performance is a responsibility of vehicle designers. Actual pulley circumference ratios (engine to compressor) range from slightly less than one to nearly two, so it is not possible to isolate a single ratio that applies to all vehicles. However, if it is assumed that most future designers will attempt to design systems for operation in the region of maximum compressor efficiency (i.e., lower compressor speed, pulley ratios nearer to unity), then typical compressor speeds will approach typical engine speeds. On the basis of such an assumption, the average efficiency improvements of a CO_2 system over the CAFE cycle (at an approximate average engine speed of 1500 rpm, as previously described) should approach the average of the improvements noted at 900 and 2500 rpm. In other words, benefits should be similar to those depicted in the simple regression line of Figure X-17. On this basis, the estimated power demand reductions for average, high average, and low average U.S. A/C operating conditions are 8 percent, 1 percent, and 20 percent respectively. The associated energy-based impacts of CO_2 systems, estimated using the same approach described above for HFC-134a systems, are presented in Table X-14.[43]

[43] It should be noted that subsequent to the completion of the A/C portion of this study, summary data from the SAE Alternate Refrigerant Cooperative Research Program (ARCRP) were released. The released summary data imply an average CO_2 efficiency benefit of about 7 percent relative to an enhanced HFC-134a system, quite consistent with the 8 percent benefit assumed in this study.

Table X-14. Estimation of Energy-Based A/C Impacts for a CO_2 System without Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.16 (1.56)	2.26 (3.03)	0.46 (0.62)
Incremental Input Energy Demand with A/C On (kW (hp))	4.15 (5.57)	8.07 (10.82)	1.65 (2.21)
Fraction of VMT with A/C Operating	0.34	0.5	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0038	0.0074	0.0015
Average Annual Incremental Fuel Demand (gal/year)	16.1	46.0	1.1
Additional CO_2 Induced (kg/year)	142.6	407.3	10.0
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-194.7	-332.0	-9.7
Change in Added CO_2 from Base HFC-134a System (pct)	-58%	-45%	-49%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

HFC-152a, propane (as well as other hydrocarbon-based systems), and CO_2 A/C systems all present certain safety risks beyond those of HFC-134a systems. HFC-152a and propane are considered to be flammable (albeit HFC-152a to a lesser extent than propane), while CO_2 is considered to be toxic due to physiological effects at moderately high concentrations and its ability to asphyxiate at sufficiently high concentrations. A risk assessment for all three systems is currently being conducted through a cooperative government/industry process to determine the extent of imposed risk. Based on the results of that assessment, it is possible that one or more of these refrigerants may be restricted from use in traditional A/C systems where the system evaporator is located inside the passenger cabin (behind the instrument panel). Alternatively, the basic system design could remain unchanged, but added safety equipment could be required to detect and respond to refrigerant leaks (e.g., by automatically lowering vehicle windows). At this time, it is unclear how the risk assessment process will conclude.

Based on the current uncertainty regarding the level of risk imposed by HFC-152a, propane, and CO_2 systems, it is appropriate to examine the impacts of potential system designer response to possible risk constraints. One particular response, the use of secondary loop A/C systems, could significantly impact the energy-demand estimates for each of the systems as presented above. In a secondary loop system, the primary system refrigeration loop is analogous to current A/C system designs, except that: (1) it is located entirely outside the passenger cabin, and (2) the evaporator is coupled to a secondary fluid refrigeration loop rather than cabin-bound air ducts. The secondary loop fluid, generally a water/glycol mixture, transfers heat from passenger cabin-bound air to the primary refrigerant. Thus, the secondary loop system isolates the passenger cabin from the primary refrigerant, but also adds complexity and incremental energy demand. The incremental energy demand is due to both a pump required to move the secondary fluid through its circuit and the incremental efficiency losses associated with an additional heat exchange operation.

Research has been performed to evaluate the energy impacts associated with secondary loop systems. [12,39] Figure X-18 presents a summary of the associated data. As indicated, the data is somewhat limited, but sufficiently consistent so that incremental energy demand on the order of 10-25 percent seems likely. As with other study data, the secondary loop data were subjected to simple regression analysis in an effort to develop the most representative relationship between energy impacts and system operating conditions. However, the estimated regression slope is not significant and, therefore, a simple arithmetic average was selected as the most appropriate approach to estimating incremental system power demand. As indicated in Figure X-18, the arithmetic average incremental demand is 18 percent. Using this estimate, the energy-based impacts of HFC-152a, propane, and CO_2 as presented above can be adjusted to determine energy demands in instances where such systems are employed with a secondary loop. Tables X-15, X-16, and X-17 present the resulting adjusted estimates.[44]

No empirical data are available for the energy demand impacts of either ammonia or air cycle A/C systems in vehicle applications. However, theoretical work suggests that ammonia systems would consume energy at about the same rate as HFC-152a systems. [11] This same work suggests that the air cycle would be about 10 percent more efficient than CO_2, but the energy demand estimates for CO_2 are in wide disagreement with subsequent empirical data. [11,32] Since both systems appear to be garnering only mild interest (ammonia due to toxicity issues, air due to significant system design differences), it seems reasonable to assume energy demands similar to those for other systems subjected to more detailed testing. Therefore, readers interested in considering either in more detail might ascribe an initial estimate of the energy impacts for ammonia systems equal to those of HFC-125a systems and an initial energy impact estimate for air cycle systems approximately equal to those of CO_2 systems. Of course, substantially more detailed analysis should be conducted before either is considered as a "competitive" alternative to HFC-134a.

Table X-18 presents a summary of the energy-based GHG impacts for each of the evaluated alternatives. As indicated, significant benefits can be derived relative to current HFC-134a systems, *regardless of refrigerant*. Under average A/C operating conditions, the lowest CO_2 emissions can be expected from non-secondary loop CO_2 systems, but the difference between enhanced HFC-134a, HFC-152a, propane, and CO_2 is only about 6 percent. Under high average A/C operating conditions, a similar spread of 7 percent is estimated, but HFC-152a demonstrates the lowest emission rate. CO_2 systems demonstrate the lowest emission rate at low average A/C operating conditions, but in a practical sense, none of the systems consume significant energy under these conditions and emissions are dominated by mass and refrigerant leakage influences. For secondary loop systems, the relationship between CO_2, HFC-152a, and propane is unchanged from the relationship for non-secondary loop systems. However, enhanced

[44] As was the case for CO_2 system impacts, summary data on secondary loop system performance from the SAE Alternate Refrigerant Cooperative Research Program (ARCRP) were released subsequent to the completion of the A/C portion of this study. The released summary data imply a secondary loop efficiency disbenefit of about 18-20 percent relative to an enhanced HFC-134a system, quite consistent with the 18 percent disbenefit assumed in this study.

Figure X-18. Incremental Secondary Loop Power Demand Relative to Single Loop System Power Demand

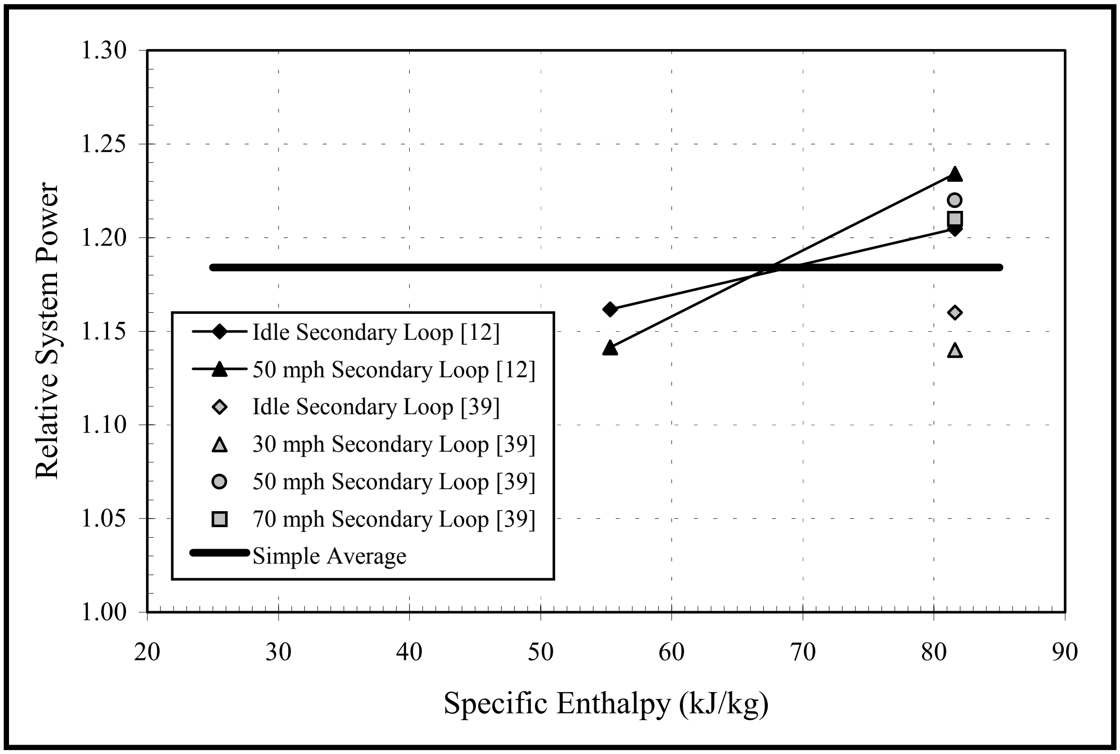


Table X-15. Estimation of Energy-Based A/C Impacts for an HFC-152a System with Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.36 (1.82)	2.46 (3.30)	0.54 (0.72)
Incremental Input Energy Demand with A/C On (kW (hp))	4.85 (6.50)	8.79 (11.79)	1.92 (2.57)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0044	0.0080	0.0018
Average Annual Incremental Fuel Demand (gal/year)	18.8	50.1	1.3
Additional CO_2 Induced (kg/year)	166.4	443.6	11.6
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-170.9	-295.7	-8.1
Change in Added CO_2 from Base HFC-134a System (pct)	-51%	-40%	-41%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Table X-16. Estimation of Energy-Based A/C Impacts for a Propane System with Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.39 (1.86)	2.58 (3.46)	0.55 (0.74)
Incremental Input Energy Demand with A/C On (kW (hp))	4.95 (6.64)	9.21 (12.36)	1.97 (2.64)
Fraction of VMT with A/C Operating	0.34	0.50	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0045	0.0084	0.0018
Average Annual Incremental Fuel Demand (gal/year)	19.2	52.6	1.3
Additional CO_2 Induced (kg/year)	170.0	465.1	11.9
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-167.3	-274.2	-7.7
Change in Added CO_2 from Base HFC-134a System (pct)	-50%	-37%	-39%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Table X-17. Estimation of Energy-Based A/C Impacts for a CO_2 System with Secondary Loop

Parameter	Average Conditions	High Average Conditions	Low Average Conditions
A/C System Power Demand (kW (hp))	1.33 (1.78)	2.63 (3.53)	0.5 (0.67)
Incremental Input Energy Demand with A/C On (kW (hp))	4.74 (6.36)	9.4 (12.61)	1.78 (2.39)
Fraction of VMT with A/C Operating	0.34	0.5	0.06
Incremental Fuel Demand with A/C On (gal/mile)	0.0043	0.0086	0.0016
Average Annual Incremental Fuel Demand (gal/year)	18.4	53.6	1.2
Additional CO_2 Induced (kg/year)	162.7	474.5	10.8
Change in Added CO_2 from Base HFC-134a System (kg/yr)	-174.6	-264.8	-8.9
Change in Added CO_2 from Base HFC-134a System (pct)	-52%	-36%	-45%

Notes: (1) The brake work efficiency of engines is assumed to be 28 percent.
(2) Gasoline energy content is assumed to be 115,000 Btu per gallon.
(3) Average annual vehicle use is assumed to be 12,500 miles (150,000 lifetime miles divided by a 12 year useful life).
(4) Average travel speed is assumed to be 32.5 mph, the average speed for the composite CAFE cycle (UDDS+HFET).

Table X-18. Summary of Energy-Based A/C Impacts for Alternative A/C Systems

A/C System Type	Added CO_2 Per Year (kg)			Percent Change from Baseline		
	Average Conditions	High Average Conditions	Low Average Conditions	Average Conditions	High Average Conditions	Low Average Conditions
Baseline HFC-134a	337.3	739.3	19.7	-0%	-0%	-0%
Enhanced HFC-134a	151.7	411.3	11.0	-55%	-44%	-44%
HFC-152a	146.2	381.7	10.8	-57%	-48%	-45%
HFC-152a with Secondary Loop	166.4	443.6	11.6	-51%	-40%	-41%
Propane (R-290)	148.1	400.6	10.8	-56%	-46%	-45%
Propane with Secondary Loop	170.0	465.1	11.9	-50%	-37%	-39%
CO_2 (R-744)	142.6	407.3	10.0	-63%	-48%	-51%
CO_2 with Secondary Loop	162.7	474.5	10.8	-57%	-39%	-48%

Note: The enhanced HFC-134a system and all non-baseline alternatives include an externally controlled VDC in conjunction with an enhanced air management strategy including forced air recirculation.

HFC-134a (without a secondary loop, since none is required) emits about 7 percent less CO_2 than the lowest emitting secondary loop system under both average and high average A/C operating conditions. By far the greatest energy demand reduction is associated with the move to externally controlled variable displacement compressors and advanced intake air management. The impacts of these strategies, estimated at a 55 percent CO_2 reduction under average A/C operating conditions and a 44 percent reduction under either high or low average operating conditions, far exceeds the impacts associated with refrigerant substitution.

Section X.8 presents a summary discussion that combines the GHG impact estimates for all three A/C system influences: direct refrigerant emissions, mass-based indirect emissions, and energy-based indirect emissions. These three influences in combination describe the overall GHG reduction potential of modifications to vehicle A/C system design.

X.8 <u>Summary of Potential GHG Emission Reduction, Cost, and Other Issues of Influence</u>

As described in the preceding sections, there are a number of factors that affect both the magnitude of GHG emissions associated with vehicle A/C systems, and the magnitude of potential reductions associated with system modifications. Included are factors related to the design of the system itself, but perhaps less obvious are factors such as the ambient conditions in which the vehicle is operated. Table X-19 presents the estimated CO_2-equivalent emissions for the various A/C system alternatives evaluated in this study. Figures X-19 through X-21 graphically depict the emission estimates, sorted in order of increasing GHG reduction potential, for vehicles operating under average, high average, and low average U.S. operating conditions.

Several important insights can be derived from the presented data. In almost all cases, significant reductions (on a percentage basis) are possible regardless of the refrigerant considered. The only exception is in low average usage areas, where emissions from current systems are dominated by leakage and mass-based influences. In average and high average usage areas, energy-based influences dominate overall system emissions and, as a result, system performance enhancements related to the use of externally controlled variable displacement compressors coupled with an effective air intake management strategy provide substantial GHG reduction benefits independent of the system refrigerant. With the exception of low average A/C usage areas, system mass impacts tend to be minor. In low average usage areas, however, mass impacts are the dominant emissions influence for all refrigerants except HFC-134a, where the high CO_2 equivalency of refrigerant leakage overwhelms indirect emissions influences. Note that to provide a point of context for the absolute emission levels, Figures X-19 through X-21 also present an estimate of the annual CO_2 emissions associated with general vehicle operation. For example, a vehicle that achieves an average fuel consumption of 20 mpg produces about 5,500 kg of CO_2 annually, or about 12 times as much CO_2 as current HFC-134a systems under average U.S. A/C operating conditions.

Estimating the cost of the A/C system alternatives is somewhat difficult given the research nature of most of the evaluated options. Nevertheless, previous researchers have produced some estimates. [15,65] However, in many cases, biases have been demonstrated in favor of one option or another and, as a result, it is sometimes difficult to rationalize the various available cost estimates. For example, the cost of an incremental safety system may be applied to one refrigerant and not another, or the costs of safety considerations overlooked entirely. In this study, an attempt is made to correct for these inconsistencies so that presented costs may differ somewhat from those presented in the cited references. Basically, the following cost assumptions are employed in this study:[45]

- The cost associated with the upgrade of current system components to a system that includes an externally controlled variable displacement compressor with electronic controls is estimated to be $40. This cost applies to all evaluated A/C alternatives.

- HFC-152a and propane systems accrue no additional component costs, except as related to safety as noted below.

- CO_2 systems accrue an additional $20 cost associated with the upgrade of system hoses and components for higher pressure operating conditions, so that the total system component cost for CO_2 systems is estimated to be $60.

- For non-secondary loop designs, HFC-152a, propane, and CO_2 systems are assumed to require additional safety equipment, including in-cabin leak sensors and engine compartment evacuation valves. An additional cost of $22.50 is assumed for all three alternatives.

[45] In accordance with the costing approach for the larger study of which this study is a component, all presented costs reflect incremental variable costs to the vehicle manufacturer (relative to current HFC-134a systems) in the 2009-2015 timeframe under a high volume production situation.

Table X-19. Estimated A/C System GHG Emissions

A/C System Type	CO_2 or CO_2-Equivalent Emissions (kg/year)				Percent Change from Baseline
	Direct Emissions	Indirect Mass-Based Emissions	Indirect Energy-Based Emissions	Total Emissions	
U.S. Average Operating Conditions					
Baseline HFC-134a	106.0	20.8	337.3	464.1	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	90.0	18.9	151.7	260.6	-44%
Enhanced HFC-134a (20 gram/yr leakage)	59.0	18.9	151.7	229.6	-51%
HFC-152a (20 gram/yr leakage)	5.0	18.9	146.2	170.1	-63%
HFC-152a with SL (20 gram/yr leakage)	5.0	23.9	166.4	195.3	-58%
Propane (20 gram/yr leakage)	0.8	18.9	148.1	167.8	-64%
Propane with SL (20 gram/yr leakage)	0.8	23.9	170.0	194.7	-58%
CO_2 (porous)	0.8	21.4	142.6	164.8	-64%
CO_2 with SL (porous)	0.8	26.5	162.7	190.0	-59%
CO_2 (50 gram/yr leakage)	0.2	21.4	142.6	164.2	-65%
CO_2 with SL (50 gram/yr leakage)	0.2	26.5	162.7	189.4	-59%
High U.S. Average Operating Conditions					
Baseline HFC-134a	106.0	20.8	739.3	866.1	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	90.0	18.9	411.3	520.2	-40%
Enhanced HFC-134a (20 gram/yr leakage)	59.0	18.9	411.3	489.2	-44%
HFC-152a (20 gram/yr leakage)	5.0	18.9	381.7	405.6	-53%
HFC-152a with SL (20 gram/yr leakage)	5.0	23.9	443.6	472.5	-45%
Propane (20 gram/yr leakage)	0.8	18.9	400.6	420.3	-51%
Propane with SL (20 gram/yr leakage)	0.8	23.9	465.1	489.8	-43%
CO_2 (porous)	0.8	21.4	407.3	429.5	-50%
CO_2 with SL (porous)	0.8	26.5	474.5	501.8	-42%
CO_2 (50 gram/yr leakage)	0.2	21.4	407.3	428.9	-50%
CO_2 with SL (50 gram/yr leakage)	0.2	26.5	474.5	501.2	-42%
Low U.S. Average Operating Conditions					
Baseline HFC-134a	106.0	20.8	19.7	146.5	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	90.0	18.9	11.0	119.9	-18%
Enhanced HFC-134a (20 gram/yr leakage)	59.0	18.9	11.0	88.9	-39%
HFC-152a (20 gram/yr leakage)	5.0	18.9	10.8	34.7	-76%
HFC-152a with SL (20 gram/yr leakage)	5.0	23.9	11.6	40.5	-72%
Propane (20 gram/yr leakage)	0.8	18.9	10.8	30.5	-79%
Propane with SL (20 gram/yr leakage)	0.8	23.9	11.9	36.6	-75%
CO_2 (porous)	0.8	21.4	10.0	32.2	-78%
CO_2 with SL (porous)	0.8	26.5	10.8	38.1	-74%
CO_2 (50 gram/yr leakage)	0.2	21.4	10.0	31.6	-78%
CO_2 with SL (50 gram/yr leakage)	0.2	26.5	10.8	37.5	-74%

Note: SL signifies "Secondary Loop" and the indicated gram per year leakage defines the system design standard. Porous indicates a CO_2 system that is recharged annually and is included only to illustrate the insensitivity of CO_2 emissions performance to leakage rate.

Figure X-19. Potential A/C System GHG Reductions for Average U.S. Operating Conditions

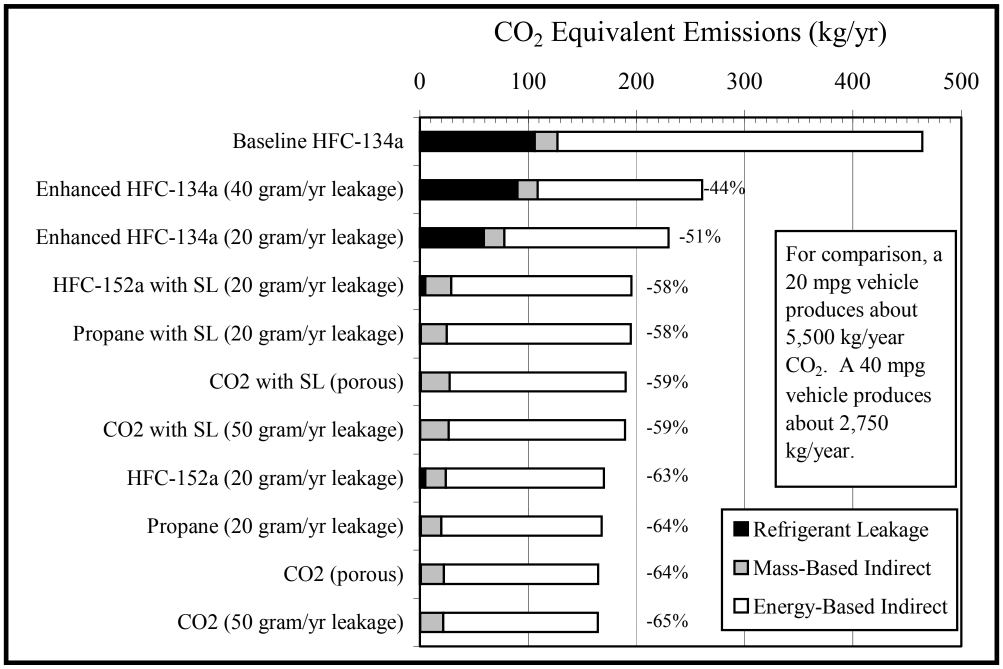


Figure X-20. Potential A/C System GHG Reductions for High Average U.S. Operating Conditions

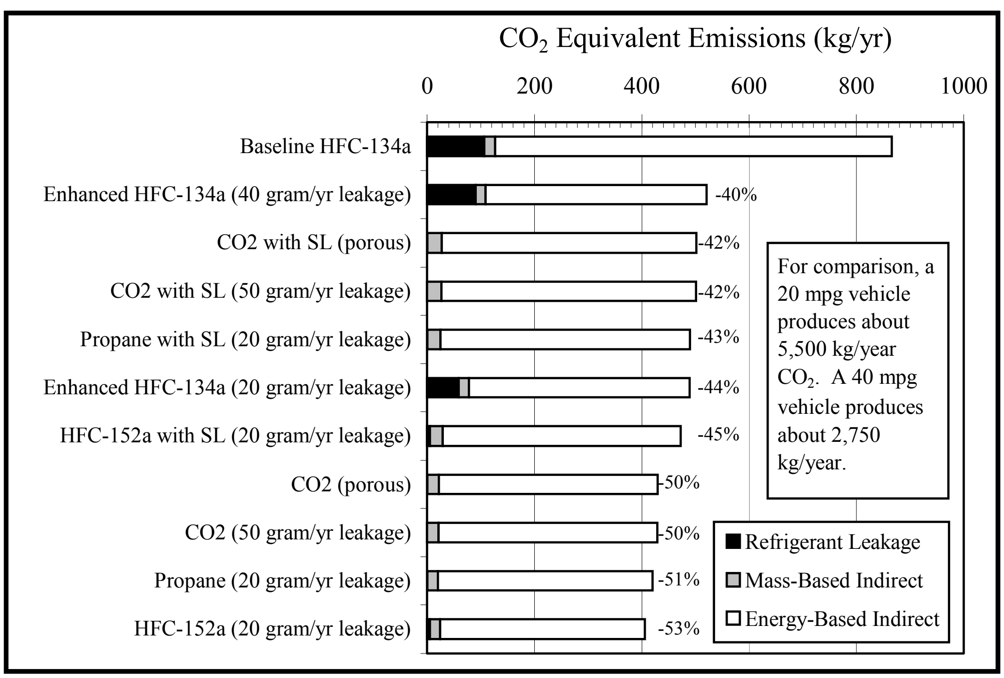
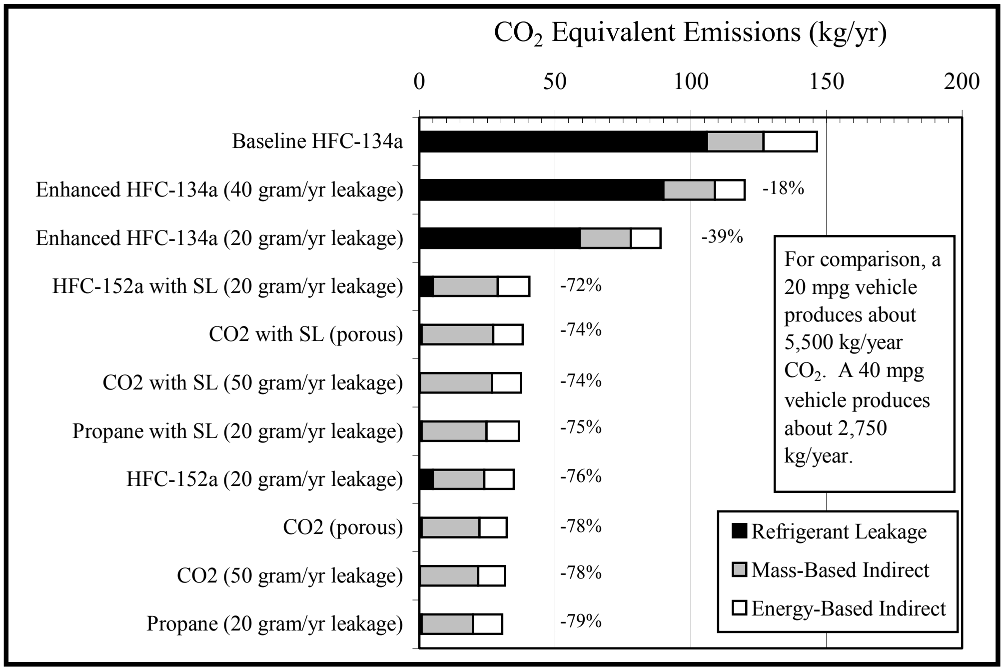


Figure X-21. Potential A/C System GHG Reductions for Low Average U.S. Operating Conditions



- For secondary loop designs, the incremental cost of a secondary loop is assumed to be \$50 and this estimate is independent of the primary loop refrigerant. Thus HFC-152a, propane, and CO_2 systems with secondary loops all reflect this incremental cost. However, the added cost of the safety equipment described above (\$22.50) is subtracted out of secondary loop systems, so that the net incremental cost is \$27.50.

Based on these estimates, both the total incremental system cost for each A/C option as well as the cost per ton of associated CO_2 (or CO_2 equivalent) reductions can be estimated from the GHG emission estimates presented in Table X-19. Table X-20 presents the cost and cost effectiveness estimates. For convenience in comparing these data to other CO_2 control strategies, cost effectiveness estimates are presented both in terms of CO_2 and carbon. As indicated, for average U.S. A/C usage conditions, the cost effectiveness of the various A/C alternatives ranges from -\$8 to -\$43 per ton CO_2 (-\$31 to -\$158 per ton carbon), with the 40 gram enhanced HFC-134a system being the most cost effective.[46] However, the enhanced HFC-134a system provides only about 70 percent of the benefits of non-secondary loop HFC-152a, propane, or CO_2 systems. Relative to secondary loop systems, the 40 gram enhanced HFC-134a system provides about 75 percent of the CO_2 reduction benefits. In high average usage areas, the cost per ton drops even further due to the proportionally higher emission

[46] A negative cost per ton indicates that savings due to reduced fuel use over the lifetime of a vehicle exceed initial component costs.

Table X-20. Estimated A/C System Cost and Cost Effectiveness

A/C System Type	Total Equivalent CO_2 (kg/year)	Total Equivalent CO_2 (kg/life)	Total CO_2 Reduction (kg/life)	Incremental System Cost ($)	Cost Benefit ($/ton CO_2) [see Note 2]	Cost Benefit ($/ton C) [see Note 2]
U.S. Average Operating Conditions						
Baseline HFC-134a	464.1	5,569.2	Baseline	0.00	Baseline	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	260.6	3,127.2	2,442.0	40.00	-43.09	-158.01
Enhanced HFC-134a (20 gram/yr leakage)	229.6	2,755.2	2,814.0	40.00	-37.40	-137.12
HFC-152a (20 gram/yr leakage)	170.1	2,041.2	3,528.0	62.50	-25.22	-92.47
HFC-152a with SL (20 gram/yr leakage)	195.3	2,343.6	3,225.6	90.00	-13.95	-51.16
Propane (20 gram/yr leakage)	167.8	2,013.6	3,555.6	62.50	-24.62	-90.27
Propane with SL (20 gram/yr leakage)	194.7	2,336.4	3,232.8	90.00	-13.08	-47.96
CO_2 (porous)	164.8	1,977.6	3,591.6	82.50	-19.95	-73.16
CO_2 with SL (porous)	190.0	2,280.0	3,289.2	110.00	-8.42	-30.87
CO_2 (50 gram/yr leakage)	164.2	1,970.4	3,598.8	82.50	-19.91	-73.01
CO_2 with SL (50 gram/yr leakage)	189.4	2,272.8	3,296.4	110.00	-8.40	-30.80
High U.S. Average Operating Conditions						
Baseline HFC-134a	866.1	10,393.2	Baseline	0.00	Baseline	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	520.2	6,242.4	4,150.8	40.00	-51.25	-187.91
Enhanced HFC-134a (20 gram/yr leakage)	489.2	5,870.4	4,522.8	40.00	-47.03	-172.45
HFC-152a (20 gram/yr leakage)	405.6	4,867.2	5,526.0	62.50	-38.84	-142.42
HFC-152a with SL (20 gram/yr leakage)	472.5	5,670.0	4,723.2	90.00	-29.47	-108.06
Propane (20 gram/yr leakage)	420.3	5,043.6	5,349.6	62.50	-37.46	-137.34
Propane with SL (20 gram/yr leakage)	489.8	5,877.6	4,515.6	90.00	-27.23	-99.86
CO_2 (porous)	429.5	5,154.0	5,239.2	82.50	-33.46	-122.68
CO_2 with SL (porous)	501.8	6,021.6	4,371.6	110.00	-21.91	-80.33
CO_2 (50 gram/yr leakage)	428.9	5,146.8	5,246.4	82.50	-33.41	-122.51
CO_2 with SL (50 gram/yr leakage)	501.2	6,014.4	4,378.8	110.00	-21.87	-80.20
Low U.S. Average Operating Conditions						
Baseline HFC-134a	146.5	1,758.0	Baseline	0.00	Baseline	Baseline
Enhanced HFC-134a (40 gram/yr leakage)	119.9	1,438.8	319.2	40.00	88.62	324.93
Enhanced HFC-134a (20 gram/yr leakage)	88.9	1,066.8	691.2	40.00	40.92	150.06
HFC-152a (20 gram/yr leakage)	34.7	416.4	1,341.6	62.50	36.19	132.68
HFC-152a with SL (20 gram/yr leakage)	40.5	486.0	1,272.0	90.00	61.22	224.48
Propane (20 gram/yr leakage)	30.5	366.0	1,392.0	62.50	34.88	127.88
Propane with SL (20 gram/yr leakage)	36.6	439.2	1,318.8	90.00	59.22	217.14
CO_2 (porous)	32.2	386.4	1,371.6	82.50	49.56	181.72
CO_2 with SL (porous)	38.1	457.2	1,300.8	110.00	74.86	274.48
CO_2 (50 gram/yr leakage)	31.6	379.2	1,378.8	82.50	49.30	180.77
CO_2 with SL (50 gram/yr leakage)	37.5	450.0	1,308.0	110.00	74.45	272.97

Notes: (1) SL signifies "Secondary Loop" and the indicated gram per year leakage defines the system design standard. Porous indicates a CO_2 system that is recharged annually and is included only to illustrate the insensitivity of CO_2 emissions performance to leakage rate. Lifetime emissions are based on a 12 year estimated life.

(2) Cost benefit calculations include accrued fuel savings due to indirect emission reductions, estimated using the following assumptions: 19.5 pounds CO_2 per gallon of gasoline, 12 years/150,000 miles vehicle life, 4.5 percent annual decline in travel, 12 years of fuel savings, $1.50 per gallon fuel price, and a 12 percent annual discount rate for future savings.

reductions associated with greater A/C system usage. Conversely, in low average usage areas, the cost per ton rises dramatically due to low system usage rates.

In considering overall A/C system GHG reduction potential, there are several other issues that should be recognized. Each issue, however, could demand the focus of significant investigation beyond the scope of this study. The following list presents a basic overview of a variety of these issues, and where appropriate indicates where estimates produced in this study might be subject to uncertainty based on future related developments.

- *Basic* research into A/C approaches that differ fundamentally from the approaches evaluated in this study has been conducted. For example, basic theoretical work related to systems such as metal hydride heat pumps, absorption cycles, heat pipes, and turbocharger-driven compression has been reported. However, none of these approaches has achieved a level of development that allows an accurate determination of practical feasibility, impact, or cost to be estimated in any way that would allow for reasonable comparison to more advanced alternatives. For this reason, such options are not considered in this study.

- Similarly, there are a number of vehicle design parameters unrelated to the A/C system itself that affect A/C energy demand. For example, cabin design, window glazing, interior color, instrument panel design, cabin ventilation, and myriad other parameters can all affect occupant comfort and, therefore, A/C demand. Although worthy of investigation, the analysis of the cost effectiveness of such design strategies relative to the cost of more efficient A/C systems is beyond the scope of this study.

- The feasibility of onboard diagnostic (OBD) systems to detect reduced refrigerant charge levels has been demonstrated and such systems could be used to enforce mandated refrigerant leakage rates if imposed. Such systems would add costs beyond those assumed in this study, but could be considered as part of a regulatory program that, for example, might allow the continued use of HFC-134a with appropriate low level OBD detection safeguards.

- The ability to CO_2 systems to retain a specified charge over time has not yet been demonstrated in use. CO_2 systems operate at much higher pressures than current vapor compression systems and this poses a significant leakage challenge for CO_2 system designers. Typical operating pressures for CO_2 systems are about 500-700 psia on the low pressure side and 1400-1900 psia on the high pressure side, as compared to 40-65 psia on the low pressure side and 200-350 psia on the high pressure side for HFC-134a systems. Thus, CO_2 systems must be able to prevent refrigerant leakage while operating at pressures 7-10 times greater than those of current systems. While prototype systems have been developed and tested, some uncertainty remains about the ability of CO_2 systems to perform adequately in consumer use. It should be noted that this primarily a marketability issue, and is not important from an emissions perspective. To demonstrate this fact, this study included a "porous" CO_2 system that required a complete refrigerant refill annually and, as presented above, the emissions performance of this system is virtually identical to a "tight" CO_2 system. However, the cooling performance is the critical factor from a consumer perspective and a porous system is not a viable A/C alternative. Based on progress to date, it seems likely that adequate system design can be achieved, but the issue should not be dismissed at this point in system development.

- The ability of HFC-152a, propane, and CO_2 systems to meet acceptable safety requirements is currently being evaluated. Both HFC-152a and propane represent potential flammability concerns, while CO_2 presents toxic concerns related to mental acuity and asphyxiation. Possible responses include the installation of safety systems, which have already been demonstrated, [e.g., 59] or the use of secondary loop A/C systems. This study includes the effect of both approaches on estimated system performance and cost, but it in unclear whether an ongoing government/industry risk assessment will conclude that either approach is adequate to address all safety issues.

- Electric A/C compressors may offer further efficiency advantages over current belt driven systems. These advantages primarily result from the ability to control compressor speed independent of engine speed, which becomes especially important on hybrid electric vehicles (HEV) and conventional gasoline vehicles utilizing engine-off at idle technology. Adapting A/C system capacity to cooling demands through compressor speed variation, as opposed to the compressor displacement variation approach employed in VDC systems, offers volumetric efficiency advantages that result in additional reductions in system energy-demands. Some of the volumetric efficiency gain is lost to reduced power transmission efficiency (electric drive efficiencies are typically in the 65 percent range, whereas belt drive efficiency generally exceeds 95 percent), but it appears that the net gain could be positive. However, current 12 volt electric systems are not adequate to handle the additional power demands imposed by electric compressors. Migration of vehicles to 42 volt systems, as is being discussed for a variety of power consumption reasons other than A/C, would enable electric compressor use. However, given the uncertainty of a 42 volt future in the 2009-2015 timeframe of this study, electric compressors have not been considered as a large market A/C option. Nevertheless, electric A/C systems can be expected to achieve a modest level of market penetration in accordance with HEV, 42 volt, and engine-off at idle technology.

- As described above, the current A/C system technology for this study was assumed to utilize pneumatically (or internally) controlled fixed displacement compressor technology. However, it should be recognized that there are current variable displacement compressor systems in the U.S. market, and that the externally controlled VDC market share can be expected to increase through the 2009-2015 timeframe. In the absence of regulatory consideration of A/C system impacts on fuel consumption, vehicle manufacturers have little incentive to switch to VDC systems for fuel consumption benefits. However, in some cases, manufacturers do have an incentive to switch to external control VDC systems to improve vehicle driveability and "feel." The incremental engine load associated with FDC clutch cycling is sufficiently large on small output engines to cause perceptible and perhaps unacceptable driveability concerns for some vehicles and customers. Thus, some manufacturers have been moving toward VDC systems in the small engine market to improve customer acceptance. In markets dominated by small displacement engines such as Europe and Japan, the market share of VDC systems is growing dramatically, but this same level of growth is not expected in the U.S. Average engine displacement in the EU is less than 2 liters, as opposed to nearly 3.5 liters in the U.S., with only about 6 percent of U.S. light duty vehicle sales reflecting engines under 2 liters displacement. Since larger engines respond less dramatically to compressor clutch cycling, the performance-based incentive for VDC systems is substantially less in the U.S. For example, even if VDC systems were installed on *all* light duty vehicles with displacements of 2 liters or less, the total U.S. new vehicle market share of VDC systems would be less than 15 percent through the 2009-2015 timeframe. If application was extended to *all* engines of 2.5 liters or less, the market incentive would double to about 30 percent, further increasing to about 40 percent at 3 liters or less. Given continuing dominance in the U.S. market, a pneumatically controlled FDC-based system was utilized as the baseline A/C system technology in this study. Nevertheless, it should be recognized that some fraction of the U.S. market already incorporates VDC technology and that this presence will impose a modest error on total fleetwide emission impacts derived from the study-assumed baseline conditions.

- Some research cites the ability of CO_2 systems to also perform in reverse (as heat pumps) and provide vehicle heating benefits as a major benefit of moving to CO_2 systems. Such ability would be most important in high efficiency vehicle applications with low waste heat availability. This study has not considered such benefits for two primary reasons. First, it is not clear that operating the A/C compressor to support heat pump operation is the most efficient approach to supplying vehicle heat. Clearly such operation would increase A/C system usage rates dramatically. Second, the ability of a CO_2 system to operate as a heat pump is not unique. Existing residential and commercial applications, as well as recent vehicle research, demonstrates that vapor compression cycle refrigerants possess the same ability. [e.g., 66]

- In accordance with the costing methods for other portions of this study, alternative A/C system costs include only the estimated high volume variable costs of components and do not consider the fixed costs associated with system introduction (e.g., engineering, and any incremental production, manufacturing, or assembly plant costs). For A/C systems, fixed costs to the vehicle manufacturer should be modest since the systems are generally purchased from suppliers and not manufactured by the vehicle manufacturer. Nevertheless, this approach can result in the omission of barriers to marketability and should be considered accordingly. Similarly, costs external to initial vehicle manufacture have also not been considered. This primarily involves service industry costs such as the cost of replacing recycling equipment, diagnostic tools, etc., but can also include additional consumer costs if service and maintenance differences exist across system alternatives. These omitted costs can vary considerably across the various systems. For example, with the exception of the potential replacement of the system desiccant to ensure compatibility, HFC-152a can effectively be used in current HFC-134a systems (i.e., it is virtually a "drop-in" replacement). With HFC-152a and propane, service equipment will need to be upgraded or replaced and both HFC-134a and alternative refrigerant equipment will need to be maintained during the switchover period. In the case of CO_2 systems, the service industry should incur no need for recycling equipment, but will require leak detectors and appropriate high pressure service equipment. However, if service intervals are shortened for CO_2 systems, consumers would incur additional costs.

- As stated in Section X.6, this study assumes effective refrigerant recycling in both vehicle service and end-of-life disposal practices. If actual practices are ineffective, direct refrigerant emission impacts will be up to 3.5 times greater than indicated in that section, and total GHG impacts (direct plus indirect) will increase by about 50 percent for U.S. average operating conditions. The potential impacts of enforcing refrigerant recycling should be considered in the overall evaluation of alternative refrigerants. The negative effects of ineffective recycling practices decline in step with the GWP of A/C system refrigerant, so that the potential negative impacts of HFC-152a, propane, and CO_2 systems are significantly less than those for HFC-134a systems. CO_2 systems reflect a lower bound risk since direct emissions from CO_2 systems have a zero net GWP, allowing the complete elimination of refrigerant recycling requirements.

- Finally, the study also does not consider emissions resulting from the energy used to produce refrigerants. Refrigerant leakage prior to vehicle charging is considered in the estimation of direct emissions, but energy used to power manufacturing and distribution equipment is not considered. Generally, all alternatives will require energy to produce and it may be that energy required to produce natural fluids such as CO_2 will be somewhat less than that required to produce HFCs, but such analysis is beyond the scope of this study.

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