970526

Total Environmental Warming Impact (TEWI) Calculations for Alternative Automotive Air-Conditioning Systems

Steven K. Fischer and James R. Sand
Oak Ridge National Lab.

ABSTRACT

The Montreal Protocol phase-out of chlorofluorocarbons (CFCs) has led manufacturers to develop refrigeration and air-conditioning systems that use refrigerants that do not damage stratospheric ozone. Most refrigeration industries have adapted their designs to use hydrochlorofluorocarbon (HCFC) or hydrofluorocarbon (HFC) refrigerants; new automobile air-conditioning systems use HFC-134a. These industries are now being affected by scientific investigations of greenhouse warming and questions about the effects of refrigerants on global warming. Automobile air-conditioning has three separate impacts on global warming; 1) the effects of refrigerant inadvertently released to the atmosphere from accidents, servicing, and leakage; 2) the efficiency of the cooling equipment (due to the emission of CO_2 from burning fuel to power the system); and 3) the emission of CO_2 from burning fuel to transport the system. The Total Equivalent Warming Impact (TEWI) is an index that should be used to compare the global warming effects of alternative air-conditioning systems because it includes these contributions from the refrigerant, cooling efficiency, and weight.

This paper compares the TEWI of current air-conditioning systems using HFC-134a with that of transcritical vapor compression system using carbon dioxide and systems using flammable refrigerants with secondary heat transfer loops. Results are found to depend on both climate and projected efficiency of CO_2 and flammable refrigerant systems. Performance data on manufacturing prototype systems are needed to verify the potential reductions in TEWI. Extensive field testing is also required to determine the performance, reliability, and "serviceability" of each alternative to HFC-134a to establish whether the potential reduction of TEWI can be achieved in a viable consumer product.

INTRODUCTION

The Montreal Protocol phase-out of chlorofluorocarbons (CFCs) has led manufacturers to develop refrigeration and air-conditioning systems that use refrigerants that do not damage stratospheric ozone. The automobile industry responded to these requirements by developing air-conditioning equipment based on HFC-134a that provided comparable or better performance than the preceding generation of air conditioners using CFC-12. Worldwide concerns are now being raised about the impact of using fluorocarbon refrigerants if they are released to the atmosphere because of their global warming potentials (GWPs). When considering the global warming impact of any refrigeration system, however, it is essential to remember that factors other than the GWP of the refrigerant are important, or even dominant for some refrigeration systems.

Automobile air-conditioning has three separate impacts on global warming; 1) the effects of refrigerant inadvertently released to the atmosphere from accidents, servicing, and leakage; 2) the efficiency of the cooling equipment (due to the emission of CO_2 from burning fuel to power the system); and 3) the emission of CO_2 from burning fuel to transport the system. The Total Equivalent Warming Impact (TEWI) is an index that can be used to compare the global warming effects of alternative air-conditioning systems.

ASSUMPTIONS

This paper examines the TEWI of hypothetical automobile air conditioners using HFC-134a, isobutane (HC-600a), propane (HC-290), and carbon dioxide (R-744). The system with HFC-134a is assumed to use the "conventional" components (e.g. belt driven open compressor, direct expansion evaporator, air-cooled condenser) with "representative" component efficiencies. The two air conditioners using hydrocarbons as refrigerants are assumed to be similar to the HFC-134a system with the notable exception of employing a secondary heat transfer loop to isolate the flammable refrigerant from

the passenger compartment. This addition results in a thermodynamic loss because of the additional ΔT imposed by the secondary loop, extra weight (due to the heat exchanger, hoses, and pump), and pumping energy. The air conditioner using CO_2 has been frequently in the literature [1,2,3] and resembles a conventional vapor compression system with the notable exception that the high side heat exchanger operates above the critical point and hence is a gas cooler; there is no condensation from vapor to liquid in the high side gas cooler (above the critical temperature, 31°C).

Previous papers have compared TEWI of alternative technologies by computing energy use from system COP at a single design condition and a fixed number of equivalent full load hours [4,5]. This simplified approach has been criticized as being unrealistic because of the broad range of variables affecting actual system performance and energy use. The analysis described in this paper takes a first step toward expanding upon previous analysis to incorporate information on vehicle speed and regional climate, though there are still simplifying assumptions that could be improved.

Several basic assumptions affect the calculation of TEWI irrespective of system configuration or vehicle air-conditioning load. These include:

- 57 liters gasoline / 100 kg incremental weight / 10,000 km,
- 2.32 kg CO_2 released / liter fuel consumed,
- 0.243 kg CO_2 released / kWh heat energy into the engine,
- 25% engine efficiency, and
- 85% of vehicle operating time at engine speeds above 16.1 km/h (10 mph), 15% of time at idle conditions

Other assumptions are independent of the climate, but vary with the choice of refrigerant. They include the evaporating temperature, refrigerant GWP, system weight, and the high side refrigerant temperature. These are summarized in Table 1. The analysis performed in this paper assumes that there is a temperature difference, the approach ΔT, between the ambient air temperature and the high-side refrigerant temperature. Two sets of values have been chosen that depend only on compressor shaft speed; two ΔT's are used for the subcritical systems (HFC-134a, HC-600a, HC-290) and two smaller ΔT's for the transcritical system [2].

Other parameters, such as the design capacity and driving distance, vary between Europe and the U.S. These are summarized in Table 2. Typical meteorological binned weather data has been used for cities in four different geographic regions in the U.S. [6] and typical meteorological year binned data were averaged for several cities in each of five European countries for use in the energy input calculations. These data are shown in Figures 1 and 2.

Siewert published a distribution for vehicle speed during 50,000 miles (80,500 km) of use [7]. These data indicate 15% of vehicle use is at or below 16 kph (10 mph) and that 85% is at higher speeds; consequently it is assumed that 15% of compressor operation is at a low shaft speed and 85% is at a high speed. Siewert's data are also used to estimate the number of hours of vehicle use per year. These results are summarized in Table 3.

Figure 3 shows the dependence of air-conditioner operation on average daily temperature [8]. A compromise in accuracy was made by applying this correlation to binned hourly weather data instead of average daily temperature; future work should employ a correlation of on-time with ambient temperature instead of average daily temperature.

System COPs were computed for each of the refrigerants using a commercial software package designed for solving thermodynamic problems [9]. The subcritical systems were evaluated using calculated refrigerant properties at:

- the compressor inlet,
- the compressor exit,
- the condenser exit
- the evaporator inlet, and
- saturated vapor leaving the evaporator.

The analysis of the supercritical CO_2 system includes the use of intermediate heat exchange between high and low-side, although this does not provide useful cooling, and pressures that give the maximum COP. The calculated COPs are shown in Figs. 4 and 5.

METHODOLOGY

The global warming effects of the energy to operate and transport the air conditioner and from emissions of refrigerant are calculated separately. Energy input is computed from annual cooling provided divided by the temperature dependent COPs. The number of annual vehicle operating hours in Table 3 are combined with the temperature distributions in Figs. 2 and 3 to estimate hours of vehicle operation as functions of the ambient temperature (the temperature distributions are first converted from hours/year at each temperature to percentage of year). The derived distribution of hours of vehicle operation as a function of temperature are then combined with the compressor on-time in Fig. 3 and design capacity to calculate air-conditioner cooling output as functions of ambient temperature. Highway speed and idle condition energy inputs are computed at each temperature by dividing the cooling output by the corresponding COP and summed over the year. The average annual energy input is computed as 85% of the highway speed a/c energy use and 15% of the idle condition energy use. Air conditioner energy input, in kWh, is converted to lifetime CO_2 emissions using the assumed engine efficiency (25%), 0.243 kg CO_2 /kWh input, and the assumed lifetime.

Fuel consumption for transporting the weight of the air conditioning system is computed using 57 liters/100 kg/10,000 km for incremental fuel use for weight increases. This value is combined with the assumed vehicle use for the U.S. and Europe and the air-conditioner lifetime. Lifetime fuel use for the weight of the air conditioner is converted to CO_2 emissions using 2.32 kg CO_2/liter of gasoline.

Refrigerant emissions are based on the assumed lifetimes of the air conditioners and average refrigerant losses of 55 grams/year [10,11] and the assumption that 95% of the refrigerant charge is recovered after the useful lifetime of the system. No practical experience is available for leakage from either the hydrocarbon or carbon dioxide air conditioners, but the effect on TEWI is insignificant because of the extremely low GWPs of these gases.

RESULTS

The results of the analyses are shown in Figures 6 and 7 for the four regions of the U.S. and the five European countries. There is a separate bar for each of the refrigerants in each of the countries and regions; each bar has a segment for CO_2 emissions resulting from power input to the compressor, belt/clutch assembly, and fan; a segment corresponding to fuel consumption for transporting the weight of the air conditioner assembly, and a segment for the direct global warming effect of refrigerant emissions. Each bar for the transcritical CO_2 system (R-744) also contains a segment resulting from unknowns in the operating efficiency. The first segment for energy use for CO_2 is determined using small approach temperatures for the gas cooler (i.e. 17°C idle and 3°C highway) which is believed achievable by at least one manufacturer. The second segment is additional energy required using the same approach temperatures assumed for the subcritical systems. The segments of each bar due to weight are nearly the same for all of the systems and direct effects of refrigerant emissions are only visible on the bars for HFC-134a. The direct effects are so small for the hydrocarbon refrigerants and transcritical system that they do not show up on these drawings.

In regions of the U.S. with high cooling demands, the hydrocarbon systems have TEWI comparable to HFC-134a systems. Recent information about prototype testing indicates that the transcritical system could have steady state COPs similar to or better than the current COPs for HFC-134a [12, 13]. If this is the case, the transcritical system would have a TEWI lower than any of the other systems; although efficiency of systems using HFC-134a can be improved if it is optimized to reduce energy use.

CONCLUSION

Several conclusions can be drawn from the results in Figs. 6 and 7. First, there may or may not be a lower TEWI using a flammable refrigerant depending on the system size and annual cooling load. There is no significant reduction in TEWI for the hydrocarbon refrigerants in the U.S. The CO_2 system has a comparable to higher TEWI than HFC-134a in the U.S. under the current assumptions for COPs and though higher than the hydrocarbons, it is not significantly higher using the low values of the approach temperatures.

References

1. J. Pettersen and G. Skaugen, *"Operation of Trans-critical CO_2 Vapor Compression Circuits in Vehicle Air Conditioning,"* Refrigeration Science and Technology Proceedings: New Applications of Natural Working Fluids in Refrigeration and Air Conditioning, Hannover, Germany, May 10-13, 1994, ISSN 0151-1637, pp. 495-506.
2. J. Pettersen, *"An Efficient New Automobile Air-Conditioning System Based on CO_2 Vapor Compression,"* ASHRAE Transactions (1994) Vol. 100, Pt. 2, pp. 657-665.
3. G. Lorentzen and J. Pettersen, *"A New, Efficient and Environmentally Benign System for Car Air Conditioning,"* International Journal of Refrigeration, Vol 16, No. 1, 1993, pp. 4-12.
4. S. Fischer, P. Hughes, and P. Fairchild, Energy and Global Warming Impacts of CFC Alternative Technologies, AFEAS Report, December 1991.
5. S. Fischer, J. Tomlinson, and P. Hughes, Energy and Global Warming Impacts of Not-In-Kind and Next Generation CFC and HCFC Alternatives, AFEAS Report, 1994.
6. Engineering Weather Data, AFM 88-29, TM 5-785, NAVFAC P-89, July 1978.
7. H. Siewert, *"Automotive Air-Conditioning Compressors,"* ASHRAE Transactions (1983), Vol. 89, Part 1B, pp. 622-629.
8. Personal communication from H. Fernqvist of Volvo Car Corporation to J. Sand of Oak Ridge National Laboratory, May 31, 1996.
9. EES: Engineering Equation Solver: Users Manual, F-Chart Software, Middleton, Wisconsin.
10. Personal communication from K. Hara, Japan Industrial Conference for Ozone Layer Protection, to J. Sand, Oak Ridge National Laboratory, June 17, 1996.
11. Personal communication from J. Köhler, IPEK, to V. Baxter, Oak Ridge National Laboratory, May 29, 1996.
12. Personal communication from H. Sugi, Nippondenso Co. Ltd., to S. Fischer, Oak Ridge National Laboratory, September 19, 1996.

13. Personal communication from Dwayne Taylor, Nippondenso America, to S. Fischer, Oak Ridge National Laboratory, May 16, 1996.

Table 1. Assumed Parameters for Alternative Refrigerants

System Parameter	HFC-134a	HC-600a	HC-290	R-744
Evaporator				
type	direct expansion	secondary loop	secondary loop	direct expansion
temperature	4.4°C	-1.1°C	-1.1°C	4.4°C
secondary fluid pump	N/A	47 W	47 W	N/A
Condenser/Gas Cooler				
approach ΔT (idle)	25°C	25°C	25°C	17°C
approach ΔT (highway)	15°C	15°C	15°C	6°C and 3°C
Compressor				
efficiency (idle)	65%	65%	65%	70%
efficiency (highway)	60%	60%	60%	65%
belt/clutch efficiency	95%	95%	95%	95%
Evaporator Blower				
power	250 W	250 W	250 W	250 W
alternator efficiency	55%	55%	55%	55%
Refrigerant GWP	1300	11	11	1
System Parameters				
capacity	7.0 kW	7.0 kW	7.0 kW	7.0 kW
weight	13.6 kg	16.3 kg	16.3 kg	13.6 kg

Table 2. Assumed Geographic Parameters

System Parameter	U.S.	Europe
Capacity	7.0 kW	7.0 kW
Charge (HFC-134a)	1100 g	700 g
Emissions (g HFC-134a/year)	55 g	55 g
Equipment Lifetime	11 years	10 years
Vehicle Use		
km / year	16,580	13,200
hours / year	249	198
Weather Data	Southeast (Miami) Northeast (Long Island) Midwest (Chicago) Southwest (Phoenix)	United Kingdom Germany Greece Italy Spain

Table 3. Vehicle Usage in the U.S. and Europe

Vehicle Speed		Vehicle Use[1]		Pro-Rated Annual U.S. Usage		Pro-Rated Annual European Usage		Fraction of Total
		miles	hours	kilometers	hours	kilometers	hours	
Idle	Idle	0	150	0	31	0	25	12.4%
10 mph	16 kph	350	35	116	7	70	6	2.9%
20 mph	32 kph	3,500	175	1,160	36	720	29	14.5%
40 mph	64 kph	14,000	340	4,640	70	2,880	56	28.1%
60 mph	97 kph	25,000	420	8,290	87	5,150	69	34.7%
80 mph	129 kph	7,150	90	2,370	19	1,480	15	7.4%
Total		50,000	1,210	16,580	249	13,200	200	100%

[1] H. G. Siewert, *"Automotive Air-Conditioning Compressors,"* ASHRAE Transactions, 1983, Part 1B.

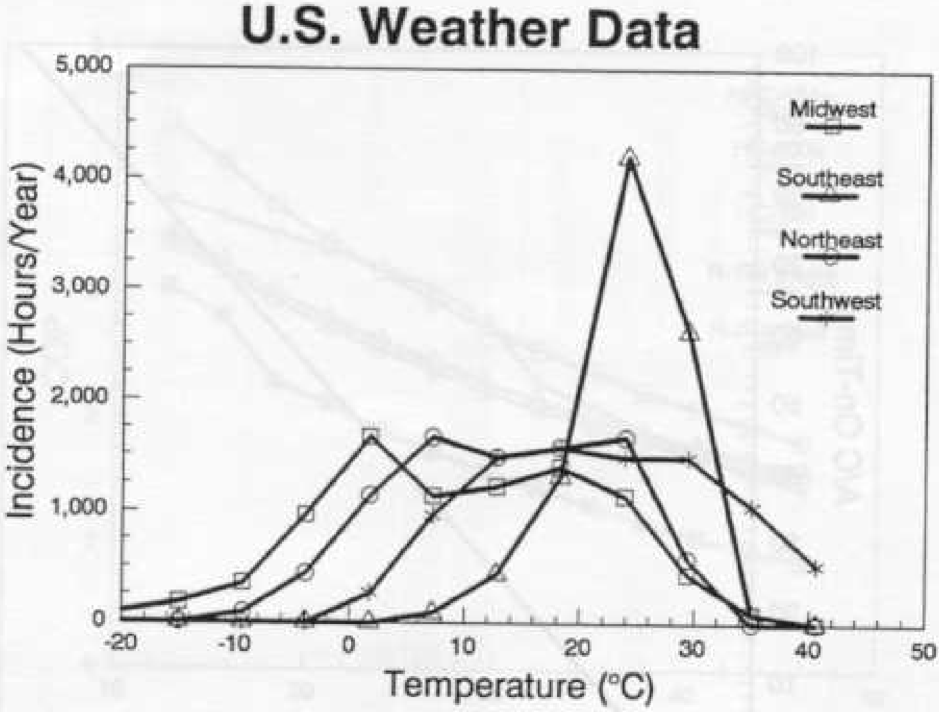

Fig. 1. Ambient temperature distribution for the U.S.

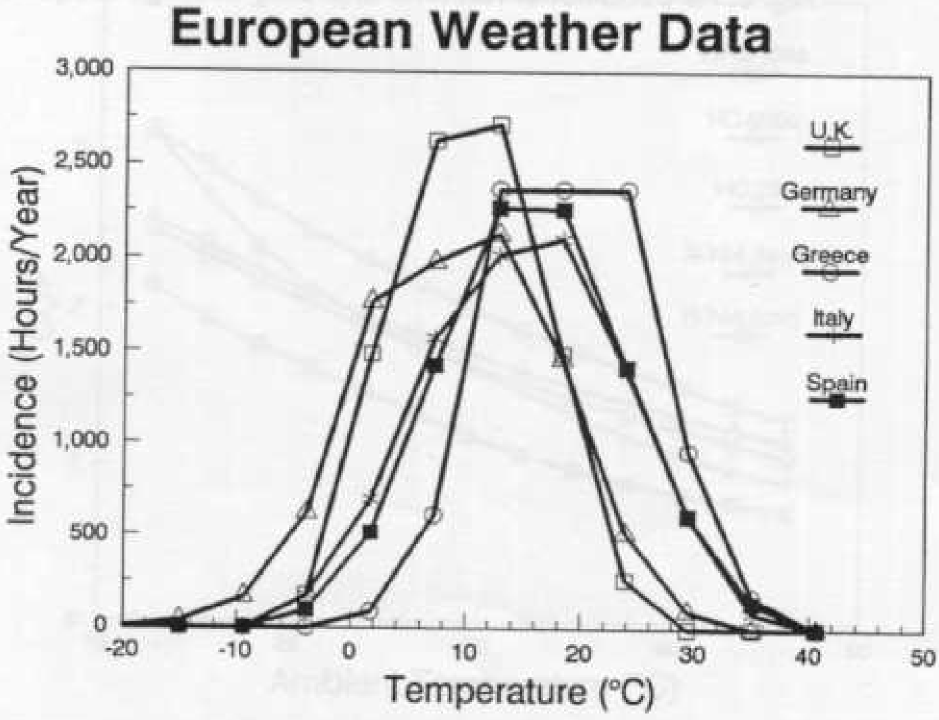

Fig. 2. Ambient temperature distribution for Europe.

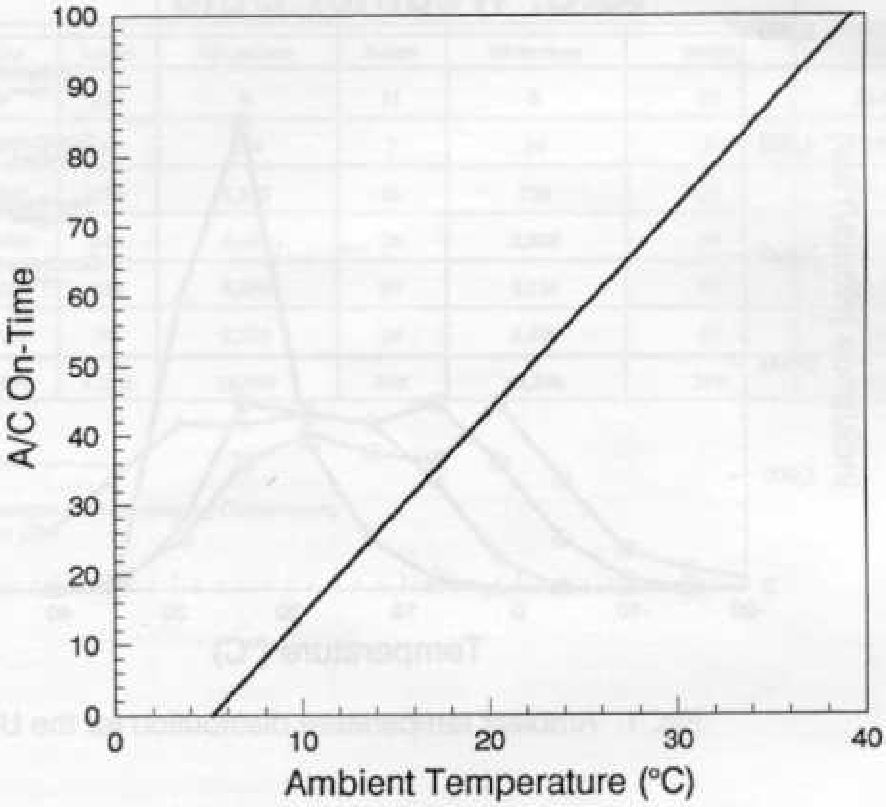

Fig. 3. Air conditioner run time and daily average temperature.

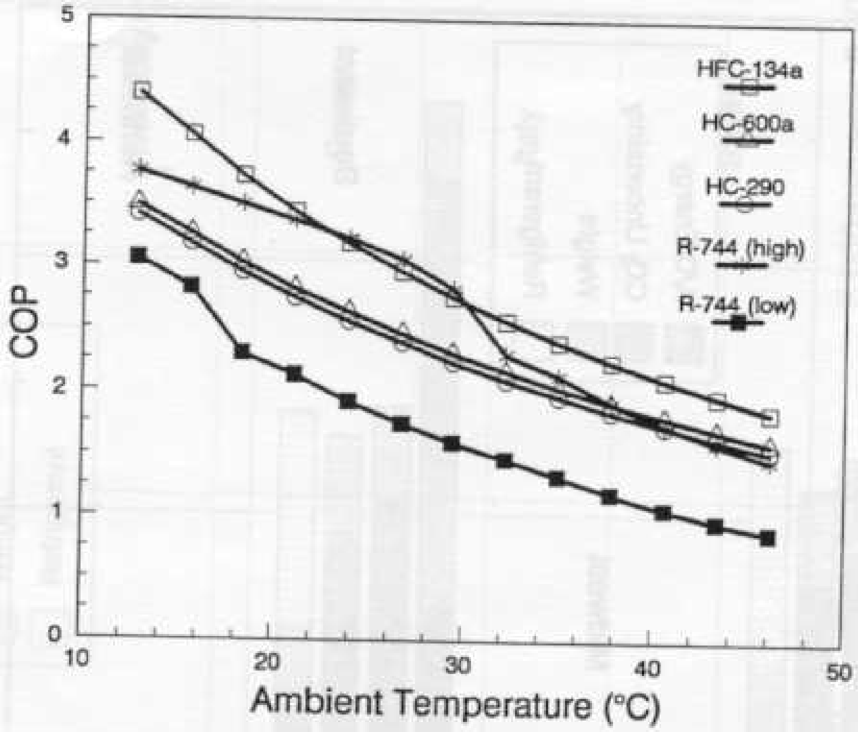

Fig. 4. System COP at highway driving speeds.

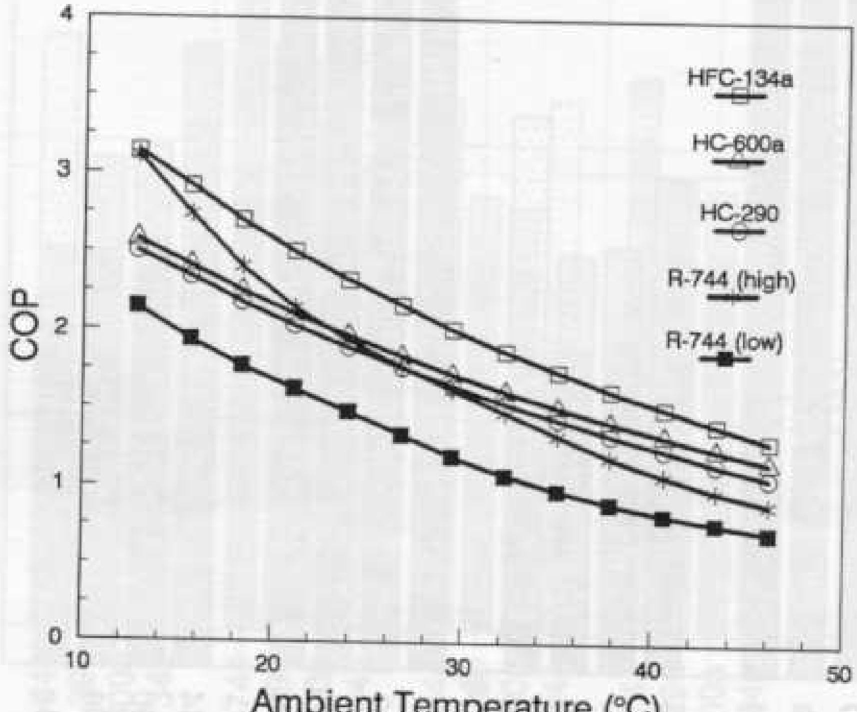

Fig. 5. System COP at idle conditions.

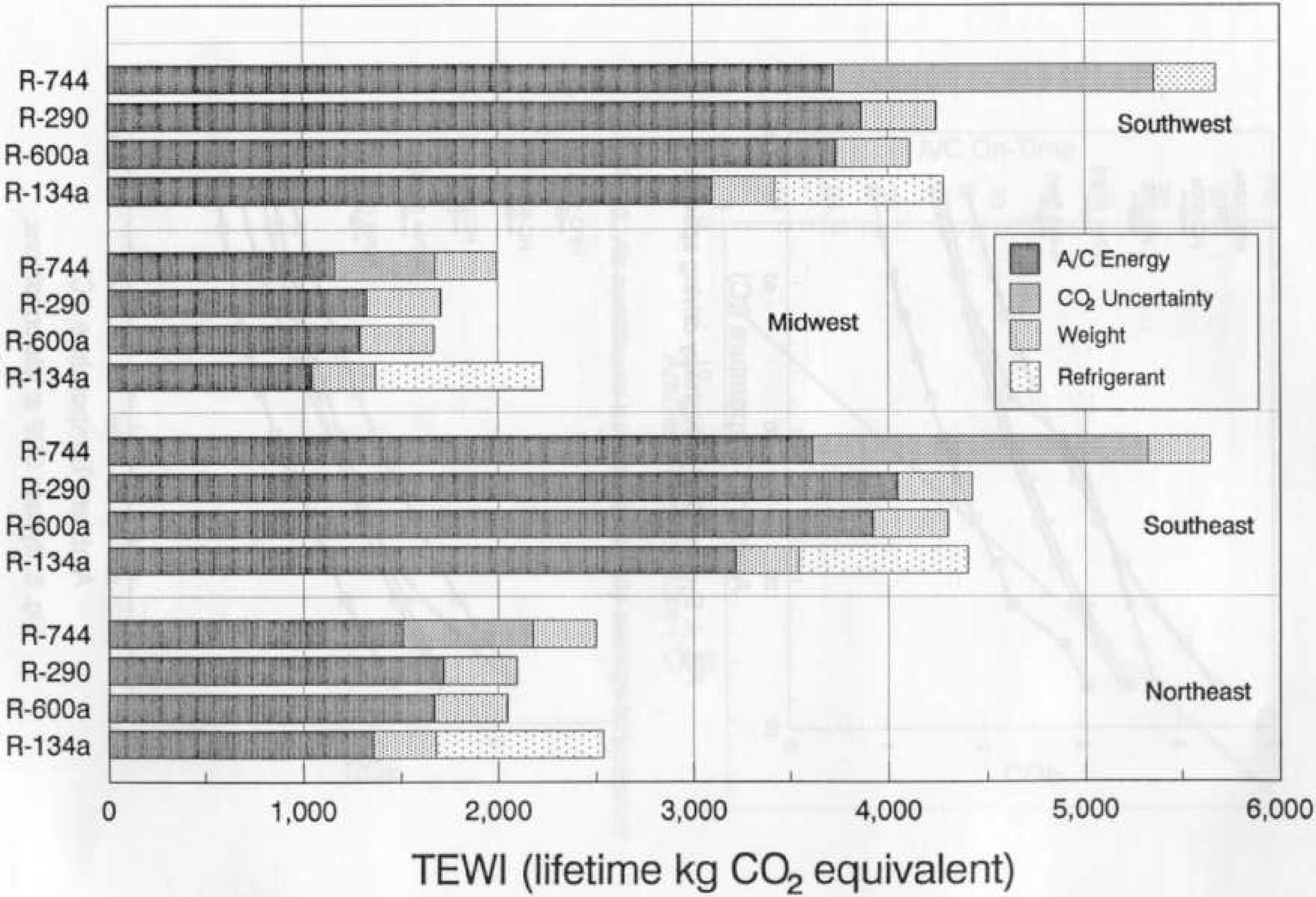

Fig. 6. TEWI of automobile air-conditioning systems in the U.S.

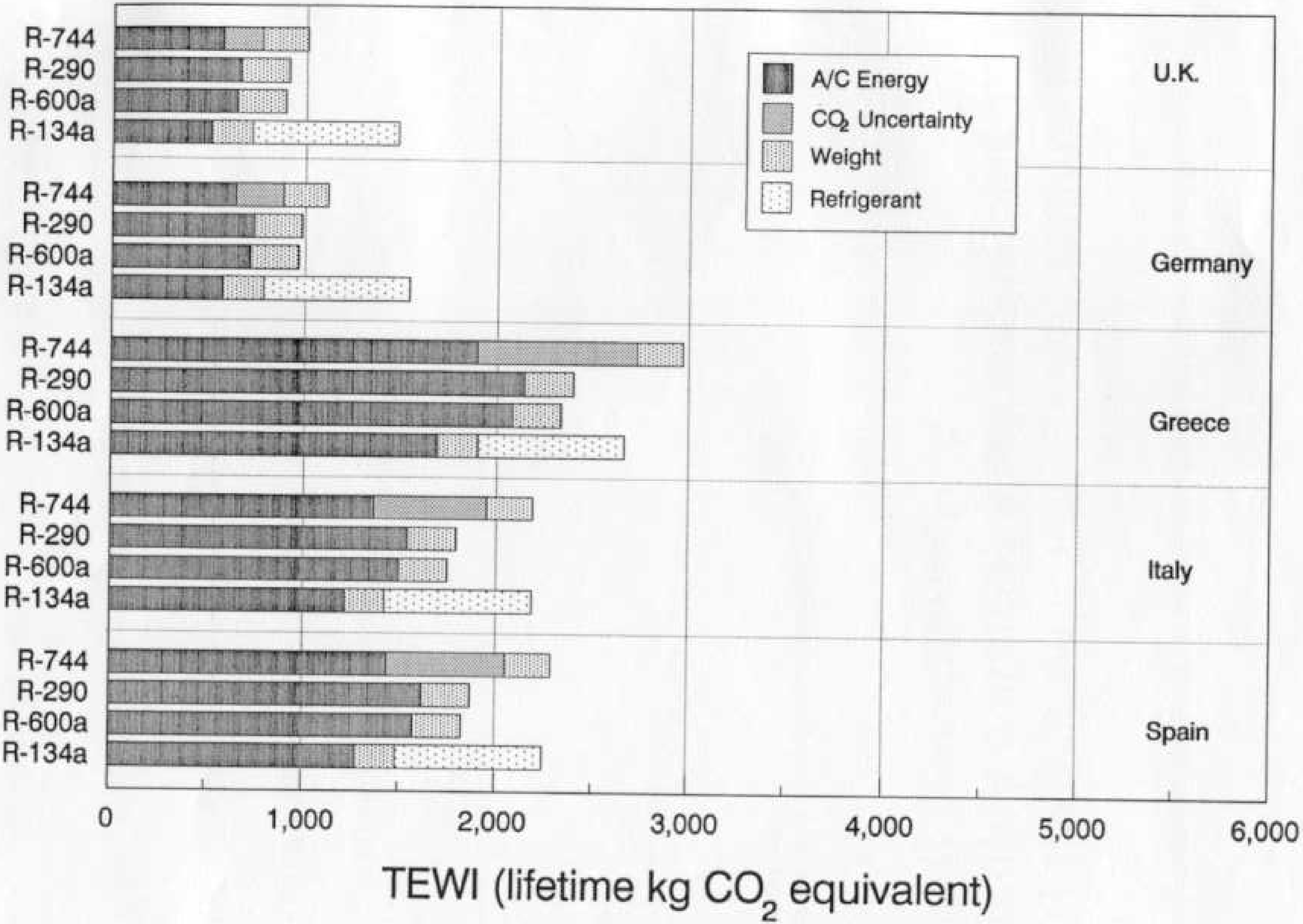

Fig. 7. TEWI of automobile air-conditioning systems in Europe.