Simulation of an Automotive Heat Pump

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ABSTRACT

A computer simulation model for steady-state operation of a mobile heat pump was used to study the effectiveness and performance of heat pumps for automotive use. In this study, simulations are performed using an automotive heat pump modeling program. The primary concern is the effect on performance of cycle reversing using components currently employed in automotive air-conditioning systems. Second, simulations are run using both R-12 and R-134a in order to gauge their relative effectiveness in both cooling and heating modes. Results are presented to show relative performance between refrigerants and between cooling and heating modes.

INTRODUCTION

While mobile air conditioning (MAC) is still an optional item, its market share has reached 90% in passenger cars and 70% in light trucks in North America. MAC has the highest parasitic power consumption of all the accessory systems in a car, and yet the basic MAC system has remained almost unchanged. For efficient MAC system design, computer models are needed for performing parametric study.

For electric vehicles to be a commercial success, air conditioning is a necessary item. In regions that have mild weather, winter heating with mobile heat pumps is desirable because it will eliminate the need for supplemental heating equipment. Modeling must be done to assess the viability of mobile heat pump applications; however, no models were previously available for mobile application.

This study presents an analysis of using a heat pump for providing heating and cooling to the passenger compartment of automobiles, through simulations using both R-12 and R-134a, with a mobile heat pump model. This model (Kyle 1993) was derived from a residential heat pump model (Fischer and Rice 1983). Presently the model supports R-12, R-22, R-114, R-502, and R-134a and is easily adapted to support others. The main purpose of the research is to evaluate the effectiveness of currently available and employed automotive air-conditioning components as they perform in a heating mode. Additionally, comparisons are made and conclusions are drawn as to the relative performances of R-12 and R-134a.

The automotive heat pump model (AHPM) was originally verified on a component-by-component level. To complement the existing validation, an additional validation run was done with this version of the AHPM. It compares data gathered from a laboratory test of an automotive air conditioner to AHPM simulation data generated for the modeled system.

OVERVIEW OF MODEL

This heat pump computer simulation was performed using a slightly modified version of the AHPM. The AHPM is designed to simulate many types of automotive air-conditioning and heat pump arrangements. It includes two compressor simulation models, a map-based variable-speed model, and an explicit efficiency single-speed model. The map-based model allows one to simulate an arrangement around actual compressor performance, creating a close approximation of an actual system, provided the data contributing to the map are accurate and realistic. The explicit efficiency model simulates compressor characteristics based on user-specified values for isentropic and volumetric efficiencies. Both finned-tube and plate-fin evaporators are supported, as are finned-tube condensers. Flow control is handled through either an explicitly defined expansion device—capillary tube, thermostatic expansion valve, or short tube orifice—or an explicitly defined level of condenser exit subcooling. Modular programming design facilitates the easy addition of new or modified component simulators.

Component modules are grouped together into two main iterative loops. The high-side loop combines the compressor, condenser, and expansion modules and uses the current evaporator exit state as a starting point to iterate until mass flow rates through the compressor and expansion device match. The low-side loop consists of the evaporator, which, based on current refrigerant inlet and outlet states and current mass flow, iterates through inlet air temperatures until a match is made with specified and calculated evaporator exit superheat. The high- and low-side loops are contained in boxes in Figure 1. The outermost loop

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returns the computed evaporator inlet air temperature—based on the most recent low-side loop calculation—and, if necessary, adjusts the estimate of the evaporator exit saturation temperature. This is repeated until a match is made between the user-specified evaporator inlet air temperature and the latest computed value for the same from the low-side loop.

The diagram in Figure 1 shows general flow for all models possible with the AHPM and in bold lines shows the path this simulation uses. Following through the steps of Figure 1, this simulation uses specified condenser subcooling and does not explicitly specify an expansion device. The model reacts by allowing for an infinitely variable opening that is later translated into an appropriately sized expansion device. Along with the subcooling expansion device module, a finned-tube condenser and an explicit efficiency compressor are modeled to complete the high-side group. Low-side calculations use the finned-tube evaporator model so that continuity can be maintained between cooling and heating modes, as the condenser must be finned-tube.

OVERVIEW OF SIMULATION

This simulation is done to compare the effectiveness of an automotive heat pump system in both cooling and heating modes using either R-12 or R-134a. The diagram in Figure 2 is of the simulated setup. In the cooling mode, the ambient air temperature (air passing through the condenser) is varied from 80°F (26.7°C) to 120°F (48.9°C) in 5°F (2.8°C) increments. Likewise, in the heating mode, the ambient air temperature (air passing through the evaporator) is varied between 30°F (−1.1°C) and 65°F (18.3°C) in 5°F (2.8°C) increments. Comparative results are presented graphically showing capacity, coefficient of performance (COP), power consumption, and compressor discharge temperature.

In an actual vapor-compression system employing a reciprocating compressor, the isentropic and volumetric efficiencies are functionally dependent on the compressor shaft speed and the pressure ratio. The compressor is not defined in this simulation, as would be the case in a map-based simulation; instead, a range of efficiencies that is reasonably in line with actual compressor efficiencies is specified, creating an approximation of an actual system. To compensate for pressure ratio changes, the isentropic and volumetric efficiencies are made to decrease linearly with the increase in ΔT between ambient air and air entering the indoor coil. The relationships of isentropic and volumetric efficiency to ambient temperature are noted in Table 1.

The behavior of an automotive heat pump differs slightly from that of a comparable unitary heat pump, due mostly to the limited size of, and airflow through, the compartment coil. In particular, the size and airflow are small in relation to the outdoor coil. In a nonautomotive application, the coils are more comparably sized. In this simulation, the indoor coil has a frontal area of 0.58 ft^2 (0.0054 m^2) and an airflow of 225 cfm (6.37 m^3/min), compared to the outdoor coil, which has an area of 3.02 ft^2 (0.28 m^2) and an airflow of 3,000 cfm (85.0 m^3/min). The outdoor coil has more than 5 times the frontal area of the indoor coil and more than 13 times the airflow. The coil differences are a consequence of the system having to fit in an automobile. The outdoor coil, placed alongside the radiator, can be relatively broad and at high speeds has large volumes of air passing over it. The opposite is true with the indoor coil; it is somewhat small and is placed in the dash, with limited airflow due to restrictions imposed by the passenger compartment. Several output parameters, particularly COP and capacity, are decidedly affected by the nature of the coil sizing, as is reflected in their respective output graphs.

Input

Input parameters are the same between cooling and heating simulation, except isentropic and volumetric efficiencies (which were outlined previously), ambient temperature, air temperature entering the indoor coil, and evaporator exit superheat. The temperature of air entering the indoor coil is 78°F (25.6°C) in the cooling mode and 70°F (21.1°C) in the heating mode, while the evaporator exit superheat is 10°F (5.6°C) and 25°F (13.9°C) in the cooling and heating modes,

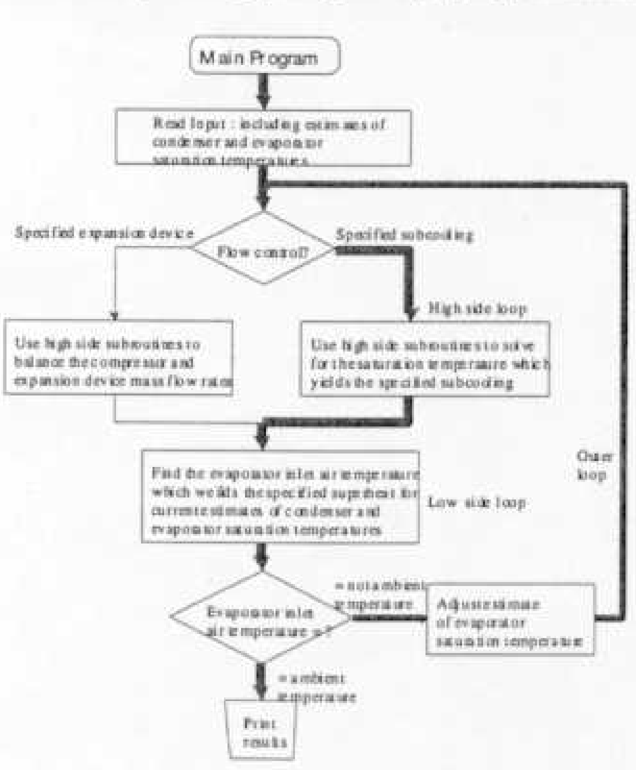


Figure 1 AHPM flow diagram.

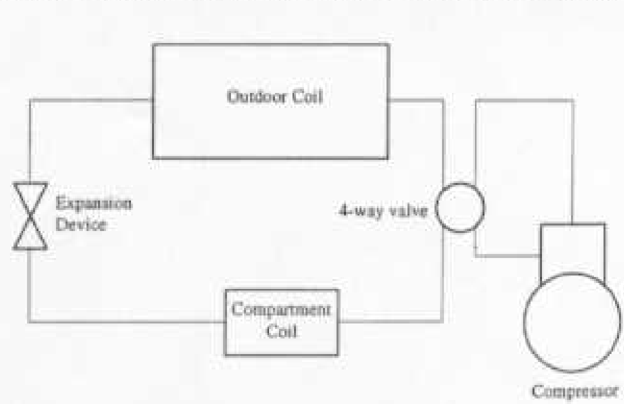


Figure 2 Schematic of simulation.

TABLE 1
Efficiencies vs. Ambient Temperature

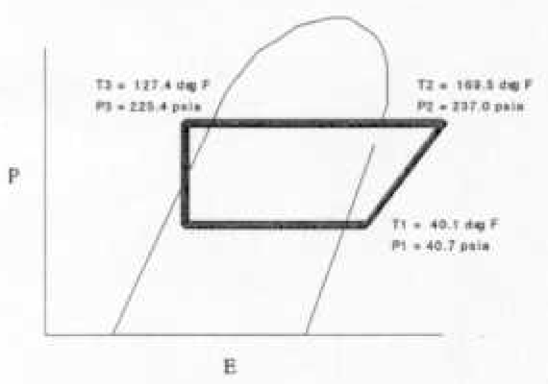
Cooling Mode			Heating Mode		
Ambient Air Temp.	η_{ISEN}	η_{VOL}	Ambient Air Temp.	η_{ISEN}	η_{VOL}
80°F (26.7°C)	0.63	0.73	30°F (−1.1°C)	0.57	0.26
85°F (29.4°C)	0.62	0.72	35°F (1.7°C)	0.58	0.27
90°F (32.2°C)	0.61	0.71	40°F (4.4°C)	0.59	0.28
95°F (35.0°C)	0.60	0.70	45°F (7.2°C)	0.60	0.29
100°F (37.7°C)	0.59	0.69	50°F (10.0°C)	0.61	0.30
105°F (40.6°C)	0.58	0.68	55°F (12.8°C)	0.62	0.31
110°F (43.3°C)	0.57	0.67	60°F (15.6°C)	0.63	0.32
115°F (46.1°C)	0.56	0.66	65°F (18.3°C)	0.64	0.33
120°F (48.9°C)	0.55	0.65	— —	—	—

respectively. Some compressor parameters are specified; the displacement is 9.8 in.^{2} (0.16 L) and the shaft speed is 1,800 rpm.

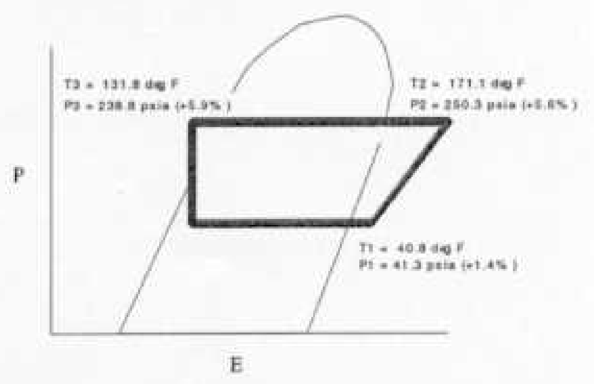
The indoor heat exchanger serves as the evaporator in the cooling mode and as the condenser in the heating mode. It is simulated as a small fin-tube-style heat exchanger, approximately 0.58 ft^2 (0.054 m^2) in frontal area, typical of what is used in automotive air-conditioning systems. Airflow through the indoor coil is held constant at 225 cfm (6.37 m^3/min). The outdoor heat exchanger serves as the condenser in the cooling mode and as the evaporator in the heating mode. It is simulated as a fin-tube heat exchanger approximately 3.02 ft^2 (0.28 m^2) in frontal area, with airflow held constant at 3,000 cfm (85.0 m^3/min). An explicit expansion device is not specified for the simulation; instead, constant condenser exit subcooling is specified at 16°F (8.9°C), from which an appropriately sized expansion device is calculated by the AHPM.

VALIDATION

A cooling mode validation comparison was made between a 1993 laboratory test of a mobile air conditioner and its corresponding model using the AHPM. The test rig is a system typical of what is used in light trucks. Simulations were run and comparisons were made at four compressor speeds: 1,210 rpm, 1,395 rpm, 1,585 rpm, and 1,770 rpm. The results for the 1,210-rpm test run are shown in Figure 3. Results are shown with test data on the left and simulation data on the right, with the percentage difference between tested and simulated output shown in parentheses. The top part of the figure is a pressure-enthalpy chart showing the refrigerant state at the low-pressure site exiting the evaporator and at two high-pressure sites—exiting the compressor and exiting the condenser. Agreement of refrigerant states is generally good. The lower part of the figure shows comparisons of four output parameters: mass flow rate, cooling capacity, coefficient of performance, and compressor power consumption. They again are in



Mass Flow = 287.0 lb/h
Capacity = 14,864 Btu/h
COP = 2.39
Compressor
Power = 1,696 Btu/h



Capacity = 274.3 lb/h (−4.4%)
Capacity = 14,476 Btu/h (−2.6%)
COP = 2.24 (−6.2%)
Compressor
Power = 1,770 W (+4.4%)

Figure 3 *Validation at 1,210 rpm.*

reasonable agreement of less than 5% deviation. Results from the other three speeds are given in Domitrovic et al. (1996).

RESULTS OF SIMULATION

Cooling Capacity

Figure 4 graphically shows the relationship between cooling capacity and ambient temperature. Cooling capacities for R-12 and R-134a are nearly the same, diverging slightly at lower ambient temperatures, although differing no more than 1% at any temperature. Cooling capacity ranges from approximately 16,900 Btu/h (4,952 W) at 80°F (26.7°C) to 14,400 Btu/h (4,219 W) at 120°F (48.9°C).

Heating Capacity

Figure 5 shows the heating capacity of R-12 and R-134a as functions of ambient temperature. Again, the capacities are nearly the same across the ambient temperature range, deviating no more than 5%. For R-12, heating capacity ranges from 8,018 Btu/h (2,349 W) at 30°F (−1.1°C) to 14,977 Btu/h (8,320 W) at 65°F (18.3°C). R-134a ranges from 7,663 Btu/h (2,245 W) at 30°F (−1.1°C) to 14,807 Btu/h (4,338 W) at 65°F (18.3°C).

Cooling Mode COP

The coefficients of performance for R-12 and R-134a are nearly the same across the entire cooling mode temperature range from 80°F (26.7°C) to 120°F (48.9°C), as shown in Figure 6. The COP decreases, for both refrigerants, as the ambient temperature increases, ranging from approximately 2.23 at 80°F (26.7°C) down to 1.34 at 120°F (48.9°C). The maximum divergence between COPs of R-12 and R-134a occurs at the extreme temperatures but is never greater than 1%.

Heating Mode COP

In the heating mode, the graphs of the COPs of R-12 and R-134a (shown in Figure 7) are nearly parallel but tend to diverge as the temperature increases. Also notice that the COP of R-12 is always greater than that of R-134a at any given temperature in the range, unlike the behavior of the COP in the cooling mode. For R-12, the COP ranges from 1.88 at 30°F (−1.1°C) up to 2.14 at 65°F (18.3°C), and for R-134a it rises from 1.87 at 30°F (−1.1°C) to 2.09 at 65°F (18.3°C).

Cooling Mode Power Consumption

Values of compressor power consumption for both R-12 and R-134a follow similar trends, as shown graphically in Figure 8. The respective values for overall power consumption are nearly identical across all ambient temperatures, differing no more than 1%. Of the total power represented, a fraction is a constant draw by the fan motors, equal to 258 watts. With fan power accounted for, the difference in power consumption between R-12 and R-134a across all temperatures remains less than 1%. Power consumption increases from approximately 2,250 watts at 80°F (26.7°C) to 3,125 watts at 120°F (48.9°C).

Heating Mode Power Consumption

Figure 9 shows the heating mode relationship between power consumption and ambient temperature. The curves for R-12 and R-134a are similar, with larger differences at lower temperatures, the greatest being approximately 4% at 30°F (−1.1°C) ambient. For R-12, power consumption increases with ambient temperature from 1,251 watts at 30°F (−1.1°C) to 2,054 watts at 65°F (18.3°C). Similarly, that of R-134a increases from 1,203 watts at 30°F (−1.1°C) to 2,077 watts at 65°F (18.3°C).

Cooling Mode Compressor Discharge Temperature

In the cooling mode, compressor discharge temperatures increase with increasing ambient temperature, as shown in Figure 10. R-12 discharge temperatures are higher than those of R-134a across all input temperatures, ranging from 166°F (74.4°C) at 80°F (26.7°C) ambient to 227°F (108.3°C) at 120°F (48.8°C) ambient. R-134a discharge temperatures increase along a parallel path from 155°F (68.3°C) at 80°F (26.7°C) ambient to 209°F (98.3°C) at 120°F (48.8°C) ambient.

Heating Mode Discharge Temperature

The heating mode graphs of compressor discharge temperature vs. ambient temperature are shown in Figure 11. As in the cooling

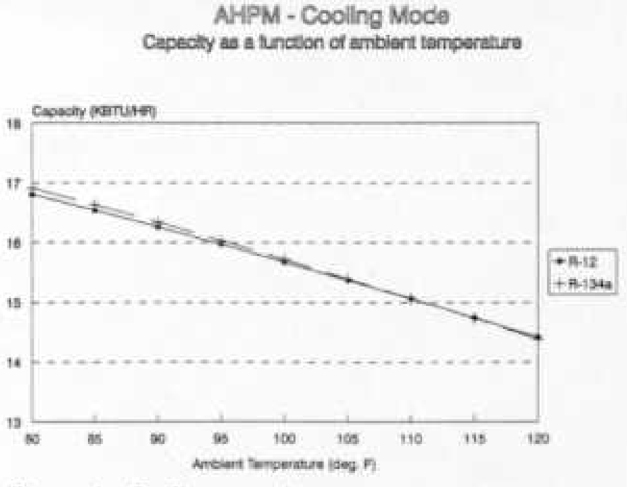


Figure 4 *Cooling capacity.*

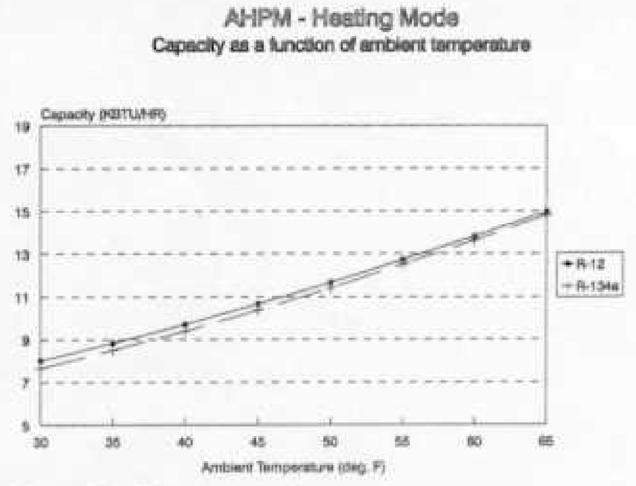


Figure 5 *Heating capacity.*

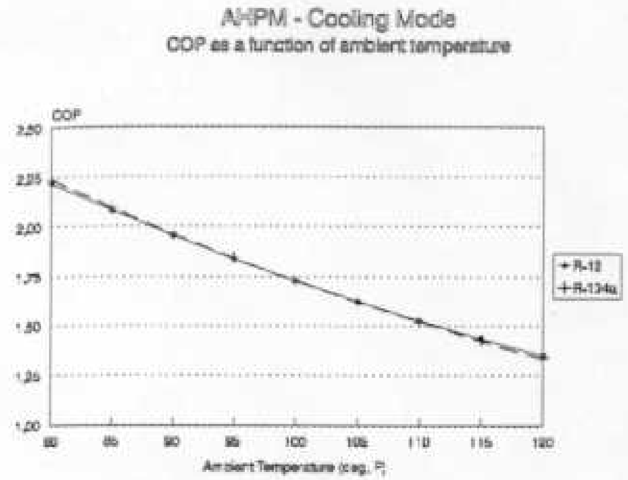


Figure 6 Cooling COP.

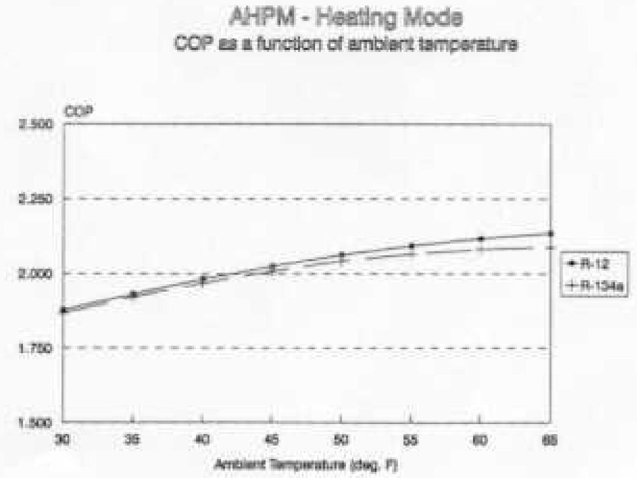


Figure 7 Heating COP.

mode, discharge temperatures for R-12 are higher than those of R-134a at all input temperatures in the ambient range. The graphs are somewhat parallel, tending to converge as ambient temperature increases.

CONCLUSIONS

This simulation was performed to gauge the effectiveness of an automotive heat pump arrangement comparing the results of both R-12 and R-134a in cooling and heating modes. Additionally, the AHPM simulator was compared with test data to provide a level of validity.

There are two basic conclusions to draw from the results of this simulation: first, that the two refrigerants—R-12 and R-134a—produce comparable results, particularly in the more external measured values such as capacity, power consumption, and COP, provided the compressor efficiencies are equal at given ambient temperatures; and second, that under current heat exchanger configurations, an automotive heat pump would operate poorly in the heating mode. The following is a summary of results for heating and cooling mode simulations.

- The cooling capacity for given ambient temperatures is nearly the same across refrigerants, ranging from 16,900 Btu/h at 80°F (26.7°C) to 14,400 Btu/h at 120°F (48.9°C). Heating capacity also shares the cross-refrigerant similarities and ranges from approximately 7,600 Btu/h at 30°F (−1.1°C) to 15,000 Btu/h at 65°F (18.3°C).
- Cooling mode COPs closely match across temperatures for both refrigerants, ranging from 2.23 at 80°F (26.7°C) down to 1.34 at 120°F (48.9°C). In the heating mode, COPs are also nearly the same across refrigerants, although there is a slight divergence at higher ambient temperatures. At 65°F (18.3°C), R-12 has a COP of 2.14, about 2.3% higher than the 2.09 value for R-134a. At 30°F (−1.1°C) ambient, both refrigerants show a COP of about 1.88.
- Power consumption corresponds well across refrigerants in both the heating and the cooling mode. In the cooling mode both refrigerants show a gradual increase from about 2,250 watts at 80°F (26.7°C) to 3,125 watts at 120°F (48.9°C). In the heating mode there is also a gradual increase from approximately 1,250 watts at 30°F (−1.1°C) to 2,050 watts at 65°F (18.3°C) for R-12, with R-134a about 3.95% lower at 30°F (−1.1°C).
- Discharge temperatures are higher for R-12 in both cooling and

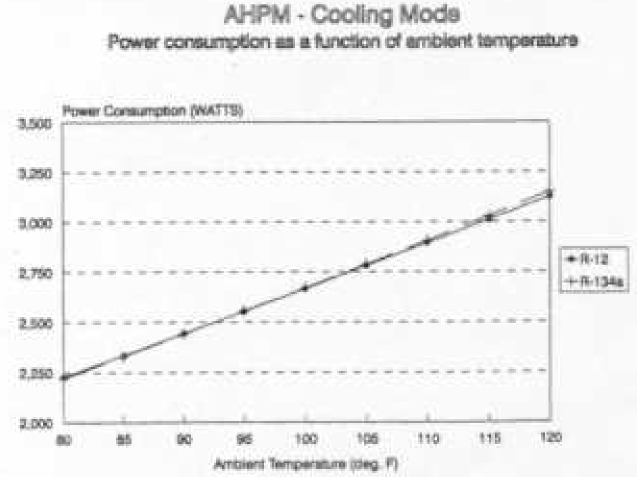


Figure 8 Heating power consumption.

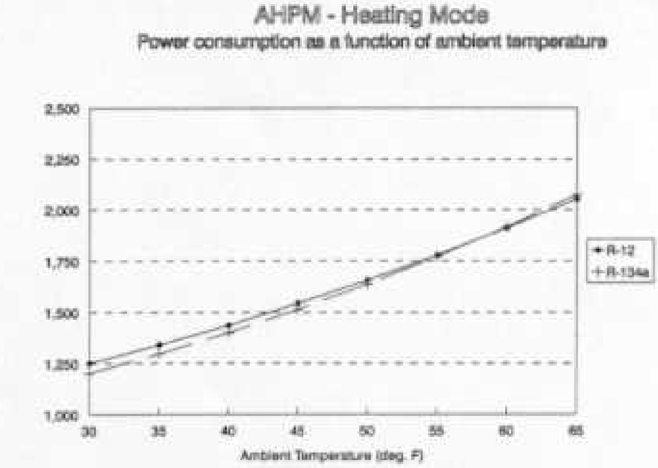


Figure 9 Cooling power consumption.

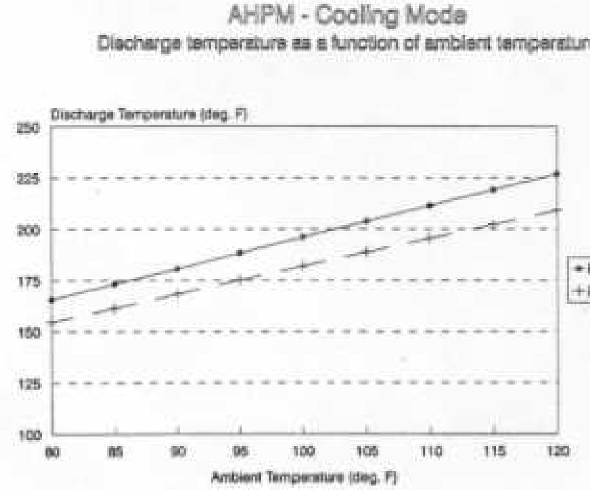


Figure 10 Cooling discharge temperatures.

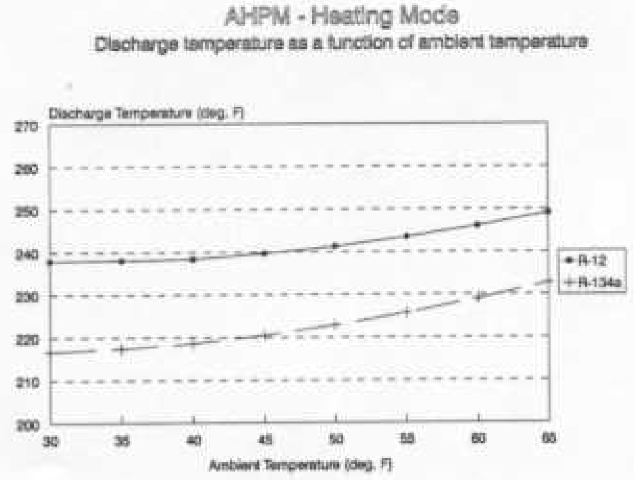


Figure 11 Heating discharge temperatures.

heating modes across all ambient temperatures. In the cooling mode, temperatures follow parallel paths, increasing approximately 60°F (15.6°C) across the input range, with R-12 temperatures averaging slightly higher. Heating mode discharge temperatures also increase across the input range but only by approximately 15°F (8.3°C), with R-12 temperatures averaging about 20°F (11°C) higher.

The first conclusion, that the two refrigerants produce comparable results, is rather straightforward, as indicated by the graphs of capacity, power consumption, and COP. Although the behavior of each refrigerant is different when observed via internal measures such as discharge temperature, the net effect of each system is nearly the same, with R-12 exhibiting a slight edge. Recall, however, that the compressor efficiencies were specified and do not represent a single compressor across the refrigerants. If the simulation was based on the performance of a single compressor, there would be differences in efficiencies between R-12 and R-134a at corresponding ambient temperatures. For example, if a compressor was designed for use with R-12 and it was switched for duty with R-134a, its efficiencies and, hence, the overall system performance would be reduced.

The second conclusion, like the first, is also drawn in a straightforward manner from the presented data, though its causes and consequences are more compelling. In a typical heat pump, perhaps a residential unit of several tons, one normally sees a ratio of heating capacity to cooling capacity on the order of 1.0, when capacities are measured at standard rating ambient temperatures (95°F [35°C] for cooling and 47°F [8.3°C] for heating). Given comparable conditions, the heating mode COP of a heat pump can be slightly greater than the cooling mode COP because the heat added to the refrigerant by the compressor is beneficial in the heating mode, whereas it must be rejected in the cooling mode. In the simulated automotive system, little is comparable from the cooling mode to the heating mode because of the disparity between indoor (compartment) and outdoor coil sizes. Indeed, the ratio of heating capacity to cooling capacity at temperatures similar to those above (cooling mode at 95°F [35°C] and heating mode at 47°F [8.3°C]) is on the order of 0.66, well off the usual mark of 1.0. Approached from the standpoint of energy balance, it is apparent that the ratio of evaporator heat gain to compressor heat input is low in the heating mode simulation, meaning that although the evaporator (outdoor coil) is reasonably large, it is not permitted to work at full capacity due to the limits of the heat-rejecting abilities of the condenser (compartment coil).

For a heat pump to operate in the heating mode in an automobile with an appreciable effect, some adjustments must be made to the size of the heat exchanger and the airflow through the heat exchanger in the passenger compartment. Such adjustments could also have beneficial effects on the cooling mode.

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DISCUSSION

John Fitzgerald, Professor, Massasoit Community College, Canton, Mass.: Have any of the auto manufacturers considered using heat pumps for defrosting windshields or first stage heating until the engine comes up to temperature?

Vince C. Mei: The authors appreciate this very interesting question. Heat pump for automobile application so far is limited to electric vehicles only, because they do not have waste heat available for heating purpose. The authors are not aware of using heat pump for the first stage heating by any of the auto manufacturers. This concept, while workable, will increase the cost of the vehicles, such as the addition of a four-way valve and extra piping, etc., that makes this concept not practical. However, it is possible that when the future engines' energy efficiency become so high that not enough waste heat is available for heating, this concept will probably be revisited.