Advanced technologies for auto a/c components

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Abstract

Component in the automotive a/c industry have been optimized to minimize size and weight, and for large production volume, but not for energy efficiency. Now facing the potential that regulatory changes that include a/c in auto fuel economy standards, and possible limitations on leakage of greenhouse gases, new forces are driving technological innovation, both at the system and component level. This paper examines the technology changes currently underway, which are being closely watched by the stationary a/c industry as production volumes increase and compactness becomes more important. Special attention is focused on the implications of a shift from a/c-only systems to reversible heat pumps in the automobile sector, as increases in vehicle fuel efficiency limit the amount of waste heat available in the engine coolant.

Introduction

Components are designed to serve the needs of a system, and systems are designed to serve the needs of the user. In the case of automotive air conditioning systems, component designs have changed significantly during the last two decades, as the status of mobile a/c has evolved from optional to standard equipment.

With the exception of the CFC phaseout, the market and regulatory forces driving technological innovation have remained relatively stable in the mobile a/c sector. R134a's thermodynamic and thermophysical properties were quite similar to those of R12, close enough to avoid the need for radical redesign of any component except the lubricant. Consequently the industry was able to improve component and system designs incrementally, a very important factor from the standpoint of maintaining product reliability.

This paper summarizes recent developments in component technology, but focuses mainly on future technology choices. Many of these choices appear risky and threatening to an industry that places such high value on system reliability. Some of the choices involve very substantial changes in component designs.

In the past, design of mobile a/c systems has been driven by the need to minimize system cost while meeting constraints on vehicle fuel efficiency and passenger safety. In turn, these factors created pressures to minimize weight of all components, to make condensers small to permit aerodynamic streamline of the front of the vehicle, and to restrict the volume of evaporators and their ductwork so the dashboard would not protrude too close to the occupants. Refrigerant had to have low toxicity and be present in sufficiently small quantities. Over the same period, economic competition has produced quieter cars with better thermal comfort controls and improved "driveability", stimulating development of air conditioning systems having variable-displacement compressors and sophisticated blower controls.

In the future these same regulatory and market forces are expected to persist. However it is becoming apparent that significant new technology drivers will also emerge, as a result of greenhouse warming and the growing global consensus that it must be controlled. In the past, the auto fuel economy test procedures in the United States did not require that the air conditioner be operating. Naturally, this created incentives to sacrifice efficiency to minimize the weight of all components. Flexible hoses could be used to dampen vibrations, at the cost of allowing diffusion of refrigerant through porous walls. The compressor did not need to operate in extremely cold weather, minimizing the chances of air leakage into the system through the compressor shaft seal. In most cars, engine size and inefficiency guaranteed that enough waste heat was available from the engine coolant to meet heating demands in winter.

The combination of these old and new technology drivers will require that a/c system operating efficiency be improved to minimize greenhouse gas emissions, both carbon dioxide at the tailpipe and refrigerant leakage through the hoses and compressor seal. Simultaneously it will be necessary to improve overall vehicle efficiency, to the point where the amount of waste heat available from the engine coolant will be inadequate to meet heating demands.

The global automobile industry has recognized the need to evaluate alternative technologies at this time. A consortium of manufacturers has undertaken a rigorous experimental comparison involving testing and evaluation of several competing systems, with extensive instrumentation under identical operating conditions (SAE, 2001). The systems are: conventional R134a; advanced R134a; transcritical carbon dioxide; and propane with secondary loop.

New technology choices

Designers of automotive air conditioning systems are being faced with major technology choices, at both the system and component levels. First at the system level, we are considering the possibility of 1) a transition to hermetically-sealed electrically-driven systems; 2) abandoning the subcritical thermodynamic cycle in favor of the transcritical cycle; and 3) obtaining supplemental heat by reversing the cooling cycle. All of these would require substantial redesign of one or several components. Even in the absence of radical technological change at the system level, this new combination of technology drivers may not allow continued reliance on evolutionary, incremental change in compressor and heat exchanger designs.

The following sections explore these technology choices in more detail. They provide some explanation of the factors leading to current designs, but focus mostly on the options for future technological innovation.

Sealed vs open systems

To minimize leakage of fluorocarbon refrigerants in response to increasingly stringent regulations, manufactures have focused on improving the design of compressor shaft seals and hose connections, while reducing diffusion through butyl rubber and nylon hose materials. Additional improvements are likely to be sought in the future, as limits on greenhouse gas emissions are imposed. Hermetically sealed systems offer an alternative, but have not been adopted because of their additional cost, and the weight penalty for an electric motor and larger alternator in today's automobiles. If future cars employ higher-voltage systems the magnitude of the weight and cost penalties may be reduced, but electric and hybrid vehicles will offer the greatest potential because the electric generating capacity will already exist. However the efficiency penalty will not be eliminated completely, as long as shaft power must be converted to electric power and back again.

The greatest advantage offered by hermetic systems would appear to be packaging; there would be no need to locate components in the engine compartment. Conceivably most components could be located in the trunk, allowing for cleaner aerodynamic design of the vehicle. Finally, the elimination of shaft seals creates opportunities to use higher-pressure refrigerants such as R410A and R744, as will be discussed below.

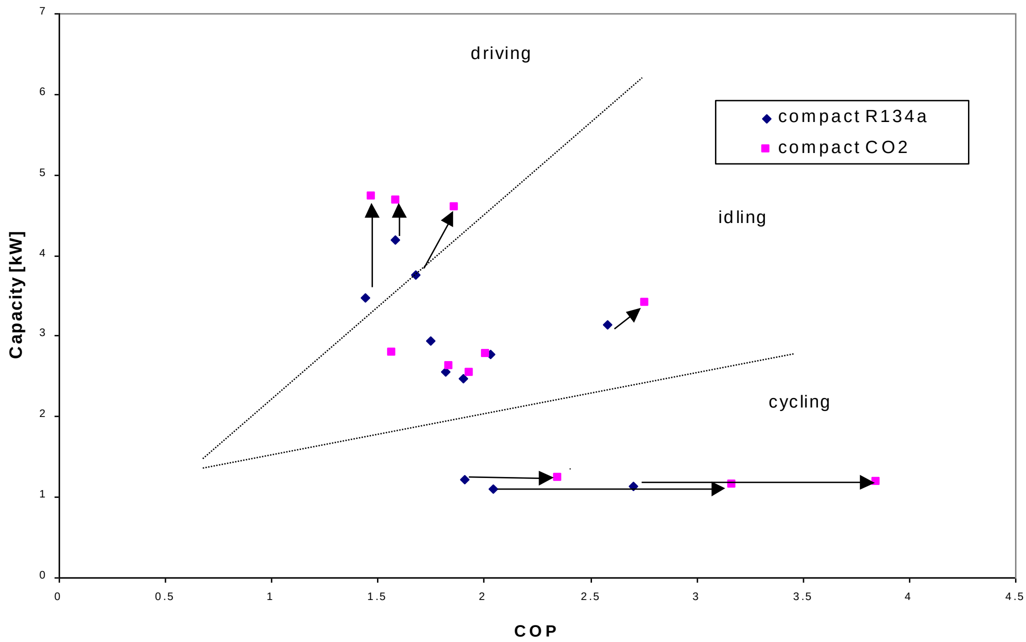

Figure 1. System capacity and efficiency: R134a and R744 mobile a/c data

Subcritical vs transcritical cycle

About a decade ago, the use of carbon dioxide refrigerant (R744) in a transcritical thermodynamic cycle was proposed by Lorentzen and Pettersen (1993), and the concept has been the subject of research investigations conducted by vehicle manufacturers, a/c system suppliers, and component suppliers. Early theoretical analyses focused on the inherently inferior ideal thermodynamic cycle efficiency of R744, and concluded that its total global warming impact would exceed that of R134a, considering both direct (leakage) and indirect (fuel combustion) effects (Bhatti, 1997). Subsequent experimental comparisons of a typical R134a system and early prototype R744 system of comparable weight and package volume revealed that comparable cooling COP's could be obtained at most operating conditions. Some of the results are shown in Figure 1, where R744 is clearly more efficient at the more commonly encountered low-load conditions, and slightly less efficient at the extremely hot ambient condition. Several factors were found to be responsible for this result: 1) higher real compressor efficiency due to lower compression ratio; 2) higher evaporating temperature due to superior thermophysical properties and greater tolerance of pressure drop due to the slope of the vapor pressure curve; 3) closer approach temperature differences at the gas cooler outlet (Boewe et al. 1999b, McEnaney et al. 1998, 1999a, 1999b, 2000).

However the low thermodynamic cycle efficiency of R744 requires an additional component, an internal heat exchanger, to match the overall operating efficiency of R134a. Moreover its high operating pressures require the use of microchannel evaporators instead of the conventional plate-fin designs developed for R134a. The high temperature of heat rejected from the gas cooler provide an opportunity for delivering high-temperature air in heating mode, but only if a counterflow heat exchanger is used. These implications for component design are discussed in more detail below.

Heating vs. cooling-only

As cars become more energy-efficient, less waste heat is available from the engine coolant to heat the passenger compartment. A few of today's cars fall into this category and obtain supplemental heat rather inefficiently from electric heaters (e.g. in seats or steering wheel) or friction heaters which dissipate mechanical energy to heat the engine coolant entering the heater core. A potentially more efficient option is to reverse the air conditioning cycle, pumping heat from an external source into the passenger compartment. Finding an acceptable heat source is problematic; the engine coolant itself is an obvious choice but it has some disadvantages. If too much heat is extracted, especially during startup in cold weather, it will take longer for the engine to reach its proper operating temperature, and tailpipe emissions will increase. The shaft work does place additional load on the engine, causing it to warm up faster, but a greater amount of heat would be extracted from the engine coolant. Other potential sources of heat include the exhaust gases downstream of the catalytic converter, and of course the air passing over the outdoor heat exchanger that would function as an evaporator.

Figure 2 shows data from experiments on an early prototype air-to-air R744 system in heating mode. Although the cross-counterflow indoor heat exchanger prototype was far from ideal, the data illustrate the essential features of an automotive heat pump: capacity is highest at startup, when it is needed most; the capacity is at least three times higher than what could be obtained from an electric resistance or friction heater, due to the high heat pumping efficiency; capacity and efficiency decline slowly as the car warms up and heat becomes available from the engine coolant. See Giannavola et al. (2000) for a more detailed discussion of heat pump performance.

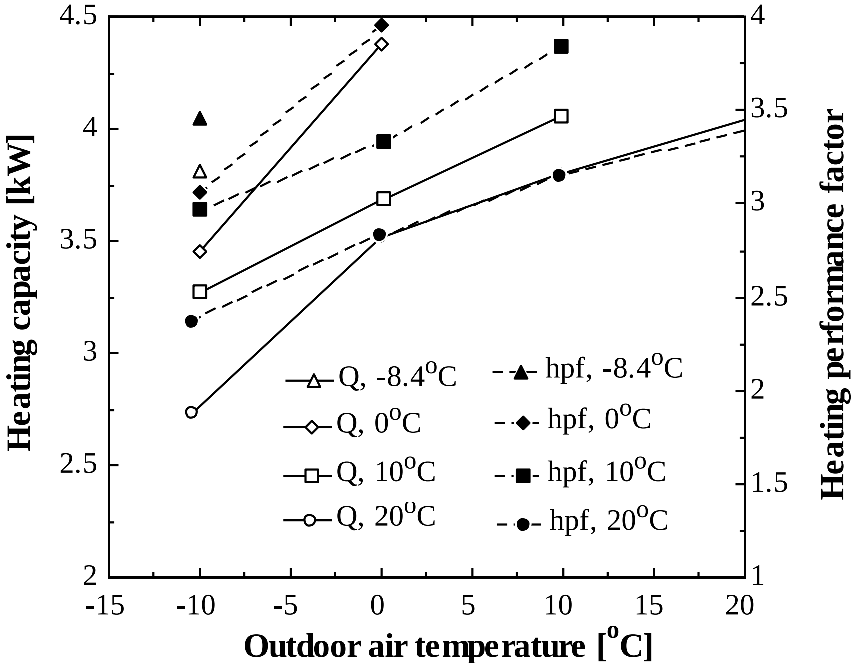

Figure 2 Automotive heat pump performance at different indoor and outdoor conditions

One reason heat pumps are not currently employed in automobiles is that R134a has severe disadvantages as a heat pump fluid. Regardless of the fluid, however, air-to-air heat pumps present substantial technological challenges that have not yet been addressed: the outdoor heat exchanger will accumulate frost, and perhaps ice as water is splashed on it from the road. Very little is known about frosting and condensate drainage from ultra-compact microchannel heat exchangers, and what is known suggests that the difficulties may be substantial (Ito et al. 1996; Kim and Bullard, 2000). Residential heat pumps defrost by switching into air conditioning mode, but that is not a desirable option for automobiles because the there is so little air in the passenger compartment. Connecting to another heat source for defrosting may also be challenging, especially after a series of short trips during which the engine never reached normal operating temperature. On the other hand during normal operation, the heat pump needs only to provide supplemental capacity for a short period after startup, while the engine is warming up. After that, sufficient heat should be available for defrosting; the only problem is how to transfer it.

Refrigerant choices

R134a, which has properties very similar to the R12 it replaced, is currently the fluid of choice for many reasons. Its principal advantage for air conditioning is its low evaporating and condensing pressures, which facilitated the design of lightweight heat exchangers and relatively simple fittings and flexible hoses. In large-diameter tubes and between wide parallel plates, low pressures produce small forces. Today, however, that advantage is not as important, because it is possible to make brazed aluminum heat exchangers of similar weight, using flat multiport tubes that can handle even higher pressures due to their small channel diameters.

One factor leading to re-evaluation of R134a is its greenhouse warming potential. Numerous studies have attempted to quantify the total equivalent warming impact, but the results are quite sensitive to the assumptions made about leakage rates. The lack of verifiable data on the ultimate fate of the R134a currently produced, coupled with the fact that technological innovation is likely to produce better shaft seals and more diffusion-resistant hoses, exacerbate the uncertainty of such estimates. Nevertheless, one trend is apparent: “direct” fugitive emissions of refrigerant constitute a greater fraction of the total warming impact of air conditioners in relatively mild European and Japanese climates, compared to the hotter and more humid driving conditions and mileage typical of North America (Yin et al. 1999, Fischer et al. 1997, Pettersen and Hafner, 1997). Those relative proportions may account for the different emphases in the policy debates around the world. The global warming potential of R410A is even greater, so that alternative us usually discussed in the context of a change to hermetically sealed systems.

Concerns about global warming alone may not be sufficient to justify changing the refrigerant used for automotive air conditioning, because of the large capital costs for retooling, uncertainties about production cost and reliability, and the relatively small performance benefits obtainable from naturally-occurring refrigerants. In addition, there are concerns about toxicity that would require substantial efforts to minimize charge in carbon dioxide systems, and flammability risks would require use of a secondary loop if hydrocarbon (propane) refrigerant is employed.

If heat pumps enter the picture, however, the range of refrigerant choices widens dramatically. The most significant thermodynamic property may be its vapor pressure, which is subatmospheric for R134a at temperatures below –26^{o}C. Existing a/c systems apparently do not encounter serious problems due to inward leakage of air at such very low ambient temperatures, because the compressor clutch is rarely engaged. However such low evaporating temperatures may be encountered frequently during warmup in heating mode. If the shaft seal is more prone to leakage while the compressor is operating, suction pressure may need to be constrained, and the cooling capacity severely limited at the times it is needed most. Of course the risk would not be present in a hermetically-sealed system.

It is also instructive to compare differences in thermodynamic properties of three alternative refrigerants, measured in terms of their volumetric capacities (quotient of latent heat and suction volume), which provide a rough measure of required compressor displacement rate. R744 (carbon dioxide) has the highest capacity, so it would require the smallest compressor displacement rate. To achieve a given pulldown capacity in cooling mode, compressors must be sized to handle vapor volumes approximately 8 times greater for R134a; 10 times greater for propane; and nearly 3 times as greater for R410A. At the other extreme, when heating loads are greatest and evaporation must occur at $-35^{o}C$ in the outdoor heat exchanger, the capacities of carbon dioxide and propane systems will be about equal, R410A about 10% less, and R134a capacity will be about 25% lower. These thermodynamic property-based figures are only approximate, but they suggest a substantial range of capacity differences. Since compressor isentropic efficiencies tend to be greater when pressure ratios are smallest, efficiency considerations may also favor R410A and R744 for heat pumps.

The final observation about the refrigerant choice is its connection to the transcritical thermodynamic cycle. R744 will rarely operate subcritically, perhaps only when a/c loads are very low and while defogging during cold weather. However R410A will experience supercritical pressures when the outdoor coil encounters high air inlet temperatures. Due to recirculation within the engine compartment, this could happen frequently. To avoid the consequent loss of capacity and efficiency, it may be worthwhile to consider packaging changes that would enable the use of fresh inlet air. Such packaging changes could be accommodated with hermetically sealed systems.

The choice of R744 as a heat pump fluid carries the intrinsic capability to deliver high-temperature (ie $60^{o}C$) air to the passenger compartment, due to the temperature glide in the gas cooler. Stated another way, it can deliver greater heating capacity at a given air flow rate, or it can deliver the same heating capacity at a much lower air flow rate than subcritical systems. However, subcritical systems using HFC's or hydrocarbons can also achieve high discharge air temperatures, but only by increasing compressor discharge pressure and incurring a significant efficiency penalty. See Richter et al. (2001) for a more detailed discussion.

Finally hydrocarbons such as propane are under consideration, because of their low greenhouse warming potential. Because of flammability risks, such systems are being designed with secondary loops. The choices of secondary refrigerant are not far advanced; most preliminary assessments have assumed that glycol would be used. However it may be possible to reduce pumping power considerably by using a two-phase secondary refrigerant, which could operate oil-free in a sealed loop. A clear advantage of any secondary-loop system is the minimization of oil-return problems, which are especially severe in the multi-evaporator systems used in vans and other large vehicles. The most obvious disadvantages are the extra temperature difference between the primary and secondary loops, and the large pumping power required to minimize it if a single-phase fluid is used.

Compressor

Automotive a/c compressors are made mostly of aluminum to minimize weight, and most are of swashplate design. Wobble plate designs are losing in popularity while scrolls are increasing their market share. The tilted swashplate is fixed rigidly to the shaft and located in the crankcase where the pistons ride on shoes that contact the swashplate. There is no sump, so the fraction of lubricant circulating with the refrigerant may be as high as 6%. There is usually an odd number of cylinders (eg 5 on each side of the swashplate, or 7 on one side of a wobble-plate. Another distinguishing feature of automotive a/c compressors is that they are designed for very short lifetimes, compared to those for stationary applications: cars operate for only about 2000 hours during a 50,000-mile warranty period. The a/c usually operates less than half that time, and the clutch is engaged turning the compressor for only a fraction of that period. Therefore it has been possible to make automotive compressors lighter and in some ways less rugged than their stationary counterparts.

Some major changes in compressor technology are currently underway. Scroll compressors are being introduced to achieve further reductions in noise and vibration. And variable-displacement compressors are becoming more common, especially in lighter and quieter vehicles where the sound and feel of the cycling clutch can become an annoyance. In the future, demands for increased system efficiency and passenger comfort will lead to development of compressors whose displacement is externally controlled, ie to set refrigerant flow rate to set cabin dry bulb temperature while blower speed controls humidity.

The design of future compressors will depend to a great extent on the system and refrigerant choices discussed above. If heat pumps are installed in future vehicles, the design lifetime will probably increase from hundreds to thousands of hours – nowhere near the 80,000 hours required of refrigerator compressors – but radically different than today's automotive a/c compressors. If hydrocarbons become the refrigerant of choice, today's hygroscopic synthetic lubricants can be abandoned in favor of mineral oil, and oil return problems may be diminished because the secondary loop enables the evaporator to be co-located with the compressor. If R744 is selected as the primary refrigerant, leakage may be tolerated and recharging might be done by unskilled workers or even the vehicle owner. Finally if future systems are hermetically sealed to take advantage of packaging flexibility, automotive a/c compressors may look very similar to those used in stationary applications.

Figure 3 shows a 155 cc R134a compressor next to a 21 cc R744 compressor. The relationship between a compressor's mass and its displacement rate is not an obvious one, and will depend on specific design tradeoffs involving piston diameter and stroke and number, rpm, materials, etc. There is no evidence at the present time to suggest that switching to higher-pressure refrigerants for their capacity advantages would necessarily entail substantial weight penalties in the compressor.

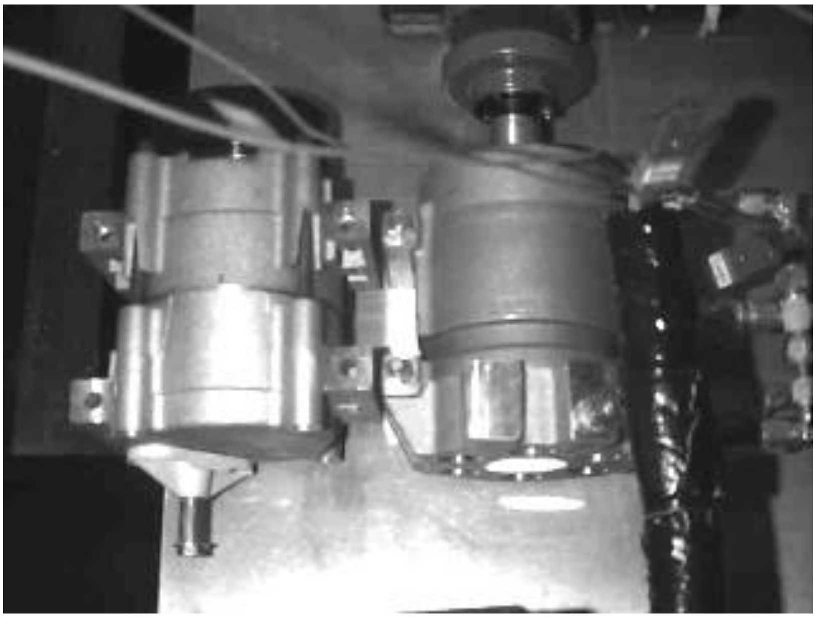

Figure 3. A 155 cm^3 R134a compressor (L) next to a 21 cm^3 R744 compressor (R)

Evaporator

During the 1980's conventional copper tube-aluminum fin evaporators were replaced by lighter, more compact brazed aluminum designs. Refrigerant flows between plates patterned with chevrons or dimples to facilitate refrigerant distribution between the upwind and downwind sides; sets of parallel plates are in crossflow, usually with several passes between headers, with a variable number of plates per pass to control pressure R134a pressure drop.

With plate heat exchangers the auto industry adopted louvered fins, which re-start the boundary every millimeter, increasing air-side heat transfer coefficient by 50-100% over that of plain fins. Condensate retention and blowoff remain a problem, which companies address in a variety of ways to minimize performance degradation and mold growth: eg tilting the heat exchanger about 10 degrees off the vertical, optimizing louver angle and spacing; and applying hydrophilic coatings and biocides.   See Jacobi (2001) for a thorough review of air-side issues in heat exchangers with noncircular tubes.

Since most heat transfer resistance is on the air side, and higher face velocities are possible with noncircular tubes due to lower pressure drop, the introduction of plate evaporators resulted in smaller evaporators yielding greater capacity. Figure 4 shows such an evaporator, sized for a sport utility vehicle. Next to it is a prototype R744 evaporator that has much smaller face area, is only slightly deeper in the air flow direction, yet it produces greater capacity due to its use of microchannel tubes. The fins on each heat exchanger are of the louvered folded type. The enhanced performance is attributable to the fact that microchannel tubes are thinner than the brazed plates, allowing the same air volume to pass with greater face velocity through a deeper heat exchanger, without a pressure drop penalty. In both plate evaporators and microchannel evaporators, the challenge is to distribute the two-phase flow uniformly through the parallel circuits.

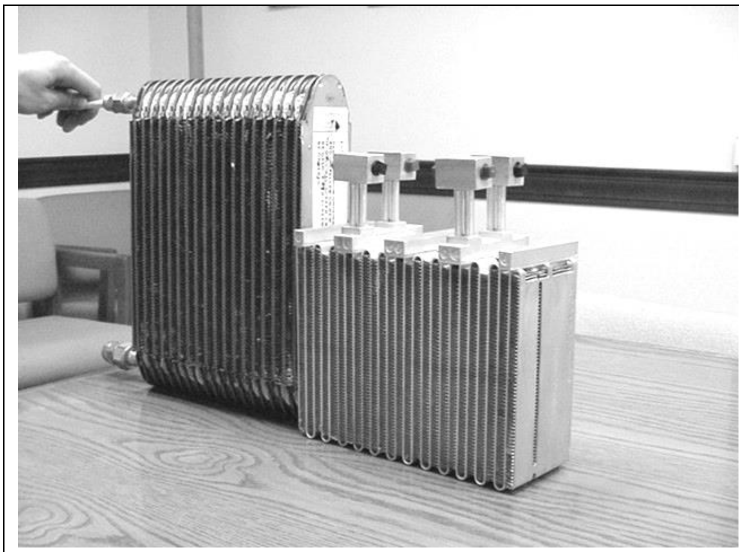

Figure 4. Plate evaporator (L) and microchannel (R)

Because energy efficiency has not been a strong force driving automotive a/c technology, there is probably room for improvement of plate evaporators in the future. Higher pressure refrigerants may require microchannel evaporators, for which the distribution problem is just now being addressed. If future systems are to be accommodate reversed flow in heating mode, intermediate headers may need to be eliminated in order to ensure uniform refrigerant distribution. If future heat pumps use R744, the indoor heat exchanger should be counterflow in order to maximize the supply air temperature.

Also emerging is yet another approach to design of evaporators for automotive applications. That is to use many parallel circuits of small-diameter circular aluminum tubes. The small diameters enable use of higher pressure refrigerants, R744 included. Most importantly these heat exchangers use flat fins, enhanced with waves or offset strips, and are much less susceptible to degradation by condensate retention, and are more easily defrosted. In that sense they are taking advantage of all that has been learned through the design of stationary air conditioners and heat pumps for stationary applications. Unlike stationary systems, however, these heat exchangers have much greater primary surface area. At this time there is relatively little published data that would allow detailed comparisons of these heat exchangers with the microchannel type, because their geometry lies outside the range of many correlations developed for stationary applications.

Condensers and gas coolers

The outdoor heat exchanger is located ahead of the radiator and designed to be as compact as possible in order to allow for streamlining the front of the vehicle. The air inlet temperature can exceed that of the ambient air by 10^{o}C, and even more when the vehicle is idling on hot pavement and when wind conditions exacerbate air recirculation within the engine compartment. To achieve compactness, flat tubes have almost completely replaced round tube designs. To allow for even more aerodynamic streamlining to meet the fuel efficiency standards many vehicles have smaller openings for ram air, and use controlled electric fans instead of belt drives to achieve high performance at idling conditions.

As condensing pressures rise, the compressor is able to pump less refrigerant, making condenser performance the key to maximizing capacity by minimizing such degradation at extreme conditions. Since heat transfer area is highly constrained by limitations on weight and volume, high performance demands enhancements of heat transfer coefficients on both the refrigerant and air sides; both are provided by microchannel tubes and louvered, folded fins. The flat tubes enable face velocities to be increased because air side pressure drop is lower than with round tubes. High face velocities are important for two reasons. First, they increase air side heat transfer coefficient. Second, they reduce the air temperature "glide" which ultimately places a lower bound on the condensing temperature. Since R134a systems always operate subcritically, low condensing temperature is the key to energy efficiency and system capacity, as seen for the subcritical condenser in Figure 5. For comparison, curves are also shown for the supercritical gas cooler, illustrating how finite air flow rates (large temperature glides) can be achieved without raising the high-side pressure.

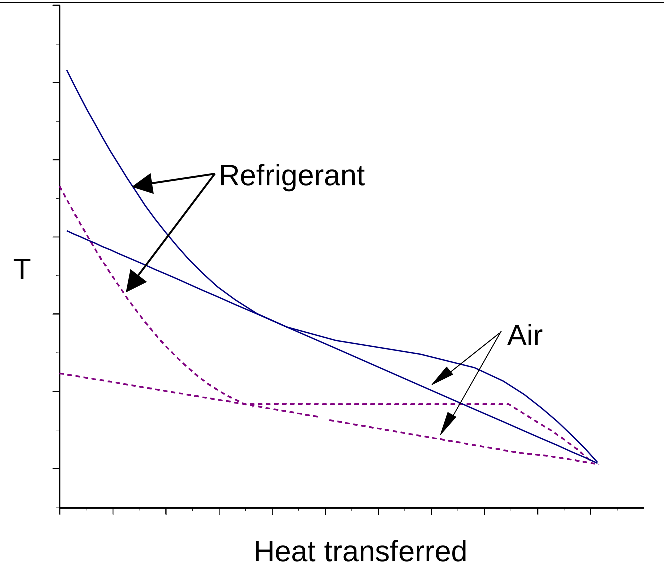

Figure 5. High-side air temperature glides; supercritical and subcritical

Condensers for automotive R410A systems have not yet been designed, but experience with residential a/c and heat pump systems suggests that the same design principles apply, even as the cycle becomes transcritical at the higher outdoor air temperatures encountered in automotive applications. As the critical pressure is exceeded, capacity and efficiency continue to decline, but nothing dramatic happens because the refrigerant exit is at subcritical temperature.

The transcritical carbon dioxide cycle is fundamentally different, because the outdoor air and refrigerant exit are above the critical temperature. A small change in refrigerant exit temperature can therefore produce a large change in enthalpy because specific heat becomes infinite at the critical point. This means system performance is very sensitive to gas cooler design, and there is an extra degree of freedom to select the high-side pressure that maximizes system performance. Figure 6 shows how COP can be increased 11% and discharge pressure reduced 5 bar by a gas cooler that cools the refrigerant exit 2 ^{o}C more.

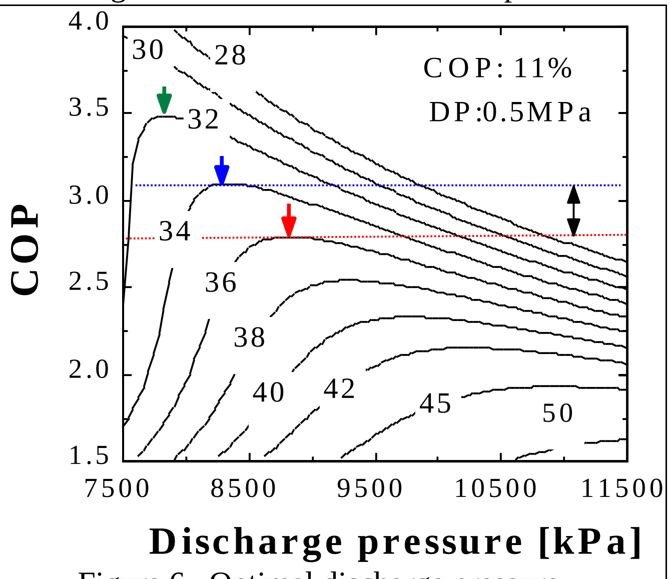

Figure 6. Optimal discharge pressure

Moreover the steep refrigerant temperature glide, apparent in Figure 5, allows for ideal cycle efficiency to be achieved at finite air flow rate, in contrast to the infinite air flow required to achieve ideal efficiency in the subcritical cycle.

Figure 7 shows the round-tube condenser technology (top) that has now been mostly replaced by the flat-tube microchannel condensers of the type used on most cars today (bottom). The difference in free flow area is obvious, owing to the fact that the flat tubes are usually less than 2 mm thick. What they have in common is their single slab crossflow configuration that exposes all tubes to the inlet air. The same crossflow configuration would probably be used for systems using other refrigerants that operate subcritically (eg hydrocarbons).

Figure 7. Round- and flat-tube condensers

Figures 8 and 9 show the design of a prototype R744 gas cooler, which differs in several major respects. First, its multi-slab overall counterflow configuration concentrates the cool air stream on the exiting refrigerant, because the transcritical cycle is so sensitive to this exit condition. It achieves approach temperature differences <2^{o}C at most operating conditions because air flowing over the first slab undergoes only a small temperature change, and that DT is what places an upper bound on the approach temperature difference. Second, the flat tubes are vertical in this prototype, to facilitate condensate drainage and defrosting in heating mode. Finally the refrigerant flows in a single pass from the inlet to outlet, with no intermediate headers, to accommodate reversibility and facilitate refrigerant distribution in heating mode.

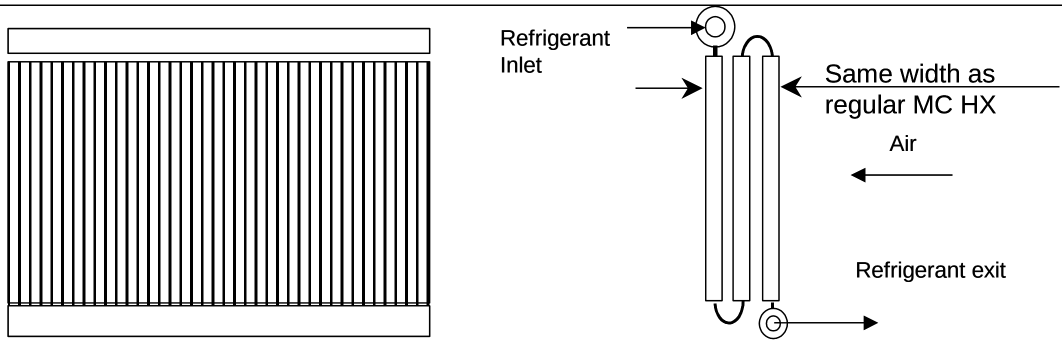

Figure 7. Multi-slab counterflow configuration with vertical tubes

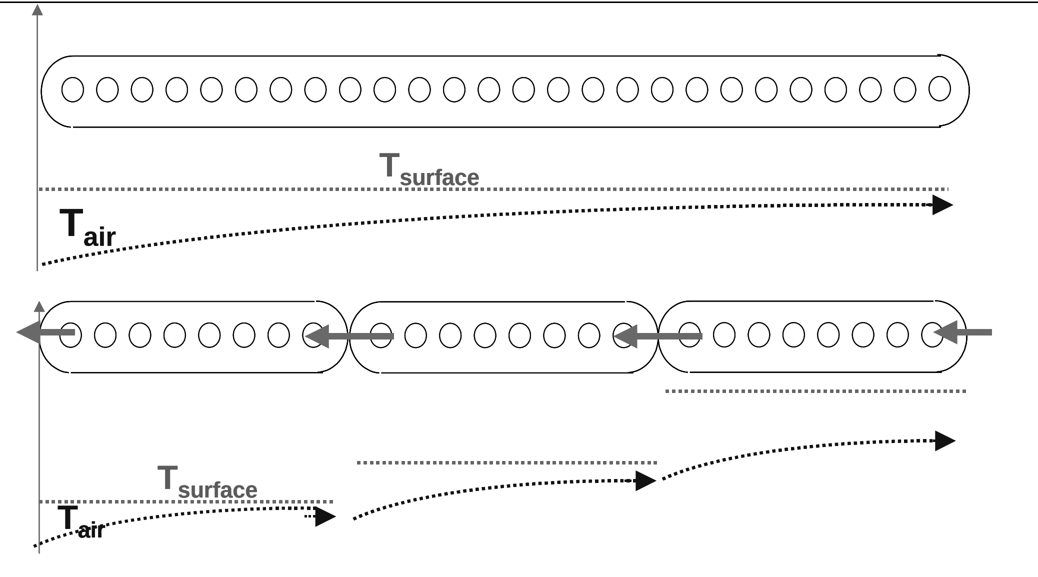

Fig. 10 Principle of multislab, counterflow arrangement

It is clear that flat tubes must be oriented vertically for any air-source heat pump for reasons of defrost and condensate drainage, regardless of the refrigerant choice. However the counterflow configuration offers no inherent advantage for subcritical systems that use R134a or hydrocarbons, because the condenser is nearly isothermal. For R410A at near-critical or transcritical operating conditions, the counterflow configuration may offer significant advantages for the same reasons cited for R744.

Internal heat exchanger

Suction-liquid heat exchange has been shown by Domanski et al. (1994) to increase ideal cycle efficiency of some refrigerants (eg R134a), and to have decrease COP for others (eg R22). The effect for R410A cycle efficiency is neutral, and for R744 it is very positive. Boewe et al. (1999a, 2001) showed how effective internal heat exchange increased cycle efficiency up to 25%. In automotive a/c systems, internal heat exchange provides the greatest enhancement when it is needed most, while idling at high ambient temperatures. In systems without internal heat exchangers, the capacity- and efficiency-maximizing discharge pressures are far apart. Internal heat exchange reduces both, and brings them closer together, creating opportunities for using less precise or simpler control systems and strategies. Their analysis showed how three microchannel tubes could be stacked to provide many parallel ports to control pressure drop in the cold suction gas, while forcing the supercritical fluid through smaller ports to maximize heat transfer coefficients and areas upstream of the expansion device where larger pressure drop can be tolerated. Compared to conventional concentric tube designs,

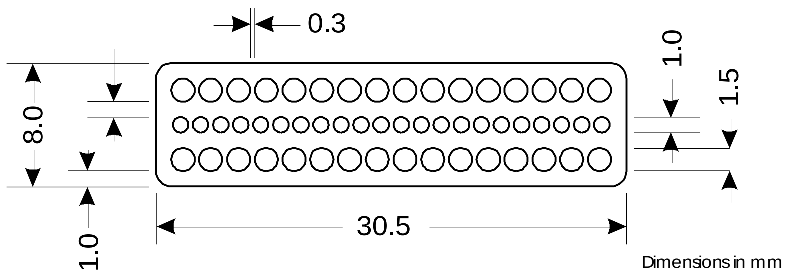

Figure 11. Internal heat exchanger made of three microchannel tubes

the microchannel shown in Figure 11 reduced material requirements by 50% while eliminating the need for long suction and liquid lines, and increasing effectiveness by 10%.

Conclusions

As energy efficiency and global warming become more important factors driving technological change in the automotive industry, substantial changes can be expected in automotive a/c component technology. Because of the need for supplemental heating sources, and the need to reduce leakage of high-GWP refrigerants, major technology changes are being considered at the system level as well. As a result, many commitments to new component technology are being deferred until decisions are made about whether to move to heat pumps, hermetic systems, or new refrigerants. Research and product development continue, however, and are revealing many exciting new options.

At the same time, the stationary a/c industry is placing increasing emphasis on heat pumps and is starting to feel pressures to make heat exchangers more compact because the conventional approach of adding surface area has reached a costly stage of diminishing returns. This confluence of interests is creating a unique opportunity for the mobile and stationary a/c companies to exchange and share technology, especially component technology.

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