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Review of alternative cooling technologies

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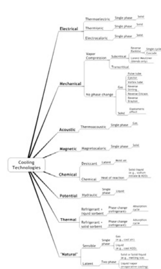
HIGHLIGHTS

- We reviewed the state-of-the-art of alternative cooling technologies.
- Progress in developments has been lower than predicted in 1994.
- Likely to find increased niche market applications in the future.
- Unlikely to widely displace vapor compression technology in the near-term future.

GRAPHICAL ABSTRACT



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ABSTRACT

This paper provides an update on alternative cooling technologies in the context of a report by Fischer et al. [2], which contains an extensive assessment of "not-in-kind" technologies including their state-of-the-art, development issues, and potentials to replace vapor compression equipment. After nearly 20 years, it is now of interest to update the status of alternative technologies considering regulatory actions aimed at refrigerants with high global warming potential. Several technologies are considered with sorption cooling, desiccant cooling, magnetic cooling, thermoacoustic cooling, thermoelectric cooling, and transcritical CO_2 being discussed in some detail. For each technology we present its physical principle, a brief summary of the findings of Fischer et al., the technological advancements since their study leading to the current state-of-the-art, and our assessment as to the potential of each technology to enter the market as a supplement to or replacement of vapor compression equipment in the next 20 year period.

1. Introduction

Vapor compression air-conditioning and refrigeration equipment dominates the market for residential and commercial buildings. For example, greater than 99.9% of all units shipped in the United States in 2005 were based on this electrically-driven technology [1]. This dominant position of vapor compression equipment has been achieved due to its low first cost, superior efficiency (low operating cost), and good personal safety record. However, the most commonly used refrigerants, halogenated alkanes, have been implicated as contributing to destruction of stratospheric ozone and global climate change, which has necessitated an examination of different cooling technology options.

One of the most thorough studies on alternative cooling options was performed by Fischer et al. [2]. They investigated ten alternatives that were emerging or were being developed at the time of their report, and which they believed were potentially able to replace vapor compression technology, thus eliminating the need for chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants. In addition to their study, three other reports [3–5] are not as broad but worth noting. While implementing refrigerants with zero ozone depletion potential (ODP) was the main industry objective in the 1990s, the current primary goal is the identification

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Nomenclature

COP	Coefficient of Performance (first law-based efficiency for thermodynamic refrigerators and heat pumps)
I	irreversibility, kJ
Q	heat transfer, kJ
SEER	Seasonal Energy Efficiency Ratio
T	temperature, °C or K
TEWI	Total Equivalent Warming Impact
W	work, kJ
Z	figure of merit, 1/K
Greek	
ΔT_L	$= T_L - T_{evap}$, °C or K
ΔT_H	$= T_{cond} - T_H$, °C or K
Φ	exergetic efficiency, Eq. (1) (combined First and Second Law-based efficiency)
η	thermal efficiency (a First Law-based efficiency for heat engines)
ζ	$= \eta$ COP, energetic efficiency, Eq. (1) (First Law-based efficiency)
Subscripts	
Carnot	Carnot cycle
cond	condenser
evap	evaporator
H	high-temperature reservoir
HE	heat engine
int	internal to cycle
L	low-temperature reservoir
o	reference
R	refrigerator or heat pump
tot	total

and introduction of high efficiency, low global warming potential (GWP) fluids to minimize both the direct and indirect effects of air-conditioning and refrigeration equipment on the earth's climate. The need for high efficiency systems cannot be overemphasized because the majority of electrical energy is produced from the burning of fossil fuels, and the amount of energy consumed for space cooling and refrigeration is enormous. As an example, space cooling and refrigeration in commercial and residential buildings account for 24.8% of the total electrical energy consumption in the United States [1]. Given increasing primary energy costs, the unequal distribution of primary energy reserves in the world, political instabilities throughout the world, and the increasing awareness by the current generation of its responsibility to use primary energy reserves in a sustainable manner — to mention only a few issues and concerns—it is increasingly incumbent upon the industry to continuously improve the energy efficiencies of its systems.

In light of the above discussion, it is worthwhile periodically to review the status of alternative technologies and readdress the question of whether or not these technologies have been developed to the point of being able to compete with and replace, at least in part, vapor compression technology. In developing our update on alternative technologies, we believe it is important to report not only on the current state-of-the-art but also on the technological progress achieved over a well-defined time frame, as the rate of progress can be an useful indicator of the feasibility of reaching a competitive status for a given technology. For this reason we opted to use the findings of Fischer et al. [2] as the "marker" from which the realized technical advancements are reported. While acceptance of a given technology in the market is dependent on its being able to satisfy a variety of criteria, e.g., cost, physical size, weight, manufacturability, serviceability, reliability, safety, environmental impact, availability of the primary energy source, among many other, this update on the state-of-the-art, similar to [2], focuses on efficiency as the chief parameter indicating the market potential of different technologies during preliminary screening.

This paper considers several technologies. Sorption cooling, desiccant cooling, magnetic cooling, thermoacoustic cooling, thermoelectric cooling, and transcritical CO_2 being discussed in some detail. For each technology the paper presents the physical principle, a brief summary of the findings of Fischer et al. [2], the technological advancements since the 1994 study leading to the current state-of-the-art, and our assessment as to the potential of each of the technologies to reach market viability and compete with vapor compression equipment for space cooling and near-room temperature refrigeration in the next twenty year period. Given the breadth of the topic and editorial prerogatives necessitated by space limitations, the coverage of individual technologies is limited. For the same reason, this paper reports only the key findings and cites just a few publications out of over 100 reviewed. A complete list of references is available from the authors.

2. Considerations for objective performance comparison

2.1. Technical merits of exergetic efficiency

The purpose of space cooling and refrigeration systems is to transfer thermal energy from a low-temperature source to a high-temperature sink while utilizing the least amount of work for a given capacity and source and sink temperatures. The most common performance measure for these systems is a First Law-based efficiency, namely, the Coefficient of Performance (COP). However, the typical definition of COP is less helpful when the primary energy input is not a form of work (mechanical, electrical, ...). To make this point clearer, consider that work requiring and work producing energy systems can be broadly classified into refrigerators/heat pumps and heat engines, and that First Law-based efficiencies can be thought of as measures of "how well energy is used", that is, they can be thought of as ratios of "energy output to energy input". As an example of the first type of energy system mentioned above, the refrigeration system of Fig. 1a typically has a First Law-based efficiency defined as $COP = Q_{L,R}/W_R$. As an example of the second type of energy system mentioned above, the heat engine (produces mechanical work from thermal energy) of Fig. 1b typically has a First Law-based efficiency defined as $\eta = W_{HE}/Q_{H,HE}$. However, neither of these measures is sufficient when the primary energy input for a refrigeration system is thermal energy, such as is shown in Fig. 1c. In this case, the First Law-based efficiency is a combination of η and COP, namely, $\zeta = \eta\ COP = Q_{L,R}/Q_{H,HE}$.

Regardless of the type of refrigeration system, comparing alternatives—or even the same system—solely based on First Law-based efficiencies can be misleading or incomplete. For example, what if the cooling capacity is not fixed and/or the source and sink temperatures are not fixed? In these cases, would a COP or ζ of 5 be better than 4? Not necessarily!

Therefore, in addition to performance indexes based on the First Law of Thermodynamics, it is appropriate to compare different systems, or even the same system operating under different conditions, using performance indexes based on both the First and Second Laws of Thermodynamics, which compare the actual performance to the ideal (Carnot) performance, such as the ones

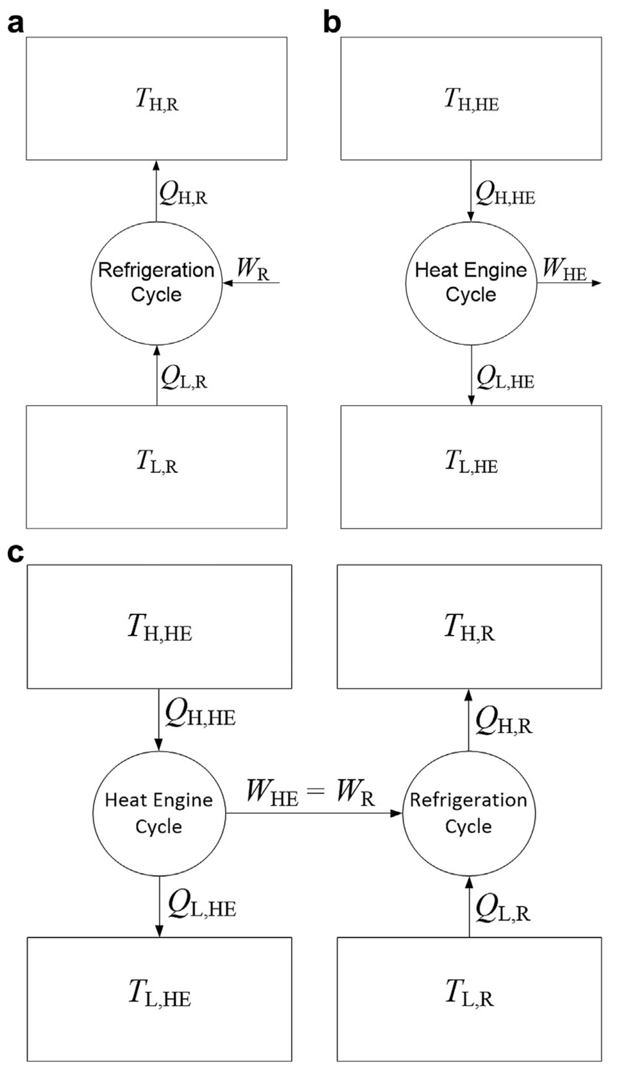

Fig. 1. Basic energy transfers for (a) refrigeration cycle with work as primary energy input, (b) heat engine, and (c) refrigeration cycle with thermal energy as the primary energy input.

provided in Eq. (1), which are often referred to as exergetic efficiencies:

$$\Phi = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{Carnot}}}\ (\text{Fig. 1a});\quad \Phi = \frac{\eta}{\eta_{\mathrm{Carnot}}}\ (\text{Fig. 1b});\quad \Phi = \frac{\zeta}{\zeta_{\mathrm{Carnot}}}\ (\text{Fig. 1c}) \tag{1}$$

where $\Phi < 1$ due to irreversibilities.

To understand why $\Phi < 1$, let us briefly turn our attention to the energy transfer processes that are associated with the basic refrigeration cycle, schematically shown in Fig. 1a. The total system irreversibility consists of those internal to the refrigeration cycle and those associated with the external heat transfer processes, or:

$$I_{\mathrm{tot,R}} = I_{\mathrm{int,R}} + I_{\mathrm{Q,L,R}} + I_{\mathrm{Q,H,R}} \tag{2}$$

where $I_{\mathrm{int,R}}$ is the sum of the internal irreversibilities, which for vapor compression systems result from the throttling process, the non-isentropic compression process, refrigerant pressure drops in the refrigerant lines, refrigerant pressure drops in the evaporator and condenser, etc., and is given by:

$$I_{\mathrm{int,R}} = W_{\mathrm{R}} - Q_{\mathrm{H,R}}\left(1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{cond}}}\right) + Q_{\mathrm{L,R}}\left(1 - \frac{T_{\mathrm{o}}}{T_{\mathrm{evap}}}\right) \tag{3}$$

and where $I_{\mathrm{Q,L,R}}$ and $I_{\mathrm{Q,H,R}}$ are the external heat transfer irreversibilities associated with $Q_{\mathrm{L,R}}$ and $Q_{\mathrm{H,R}}$ across the finite temperature differences, i.e., $T_{\mathrm{L,R}} - T_{\mathrm{evap}}$ and $T_{\mathrm{cond}} - T_{\mathrm{H,R}}$, respectively. The heat transfer irreversibility associated with the low-temperature heat exchanger is:

$$I_{\mathrm{Q,L,R}} = T_{\mathrm{o}} Q_{\mathrm{L,R}}\left(\frac{1}{T_{\mathrm{evap}}} - \frac{1}{T_{\mathrm{L,R}}}\right) \tag{4}$$

and the one associated with the high-temperature heat exchanger is:

$$I_{\mathrm{Q,H,R}} = T_{\mathrm{o}} Q_{\mathrm{H,R}}\left(\frac{1}{T_{\mathrm{H,R}}} - \frac{1}{T_{\mathrm{cond}}}\right) \tag{5}$$

In Eqs. (3)–(5), W_{R} is the work input, T_{o} is an appropriately selected reference temperature, and $T_{\mathrm{L,R}}$, $T_{\mathrm{H,R}}$, T_{evap}, and T_{cond} are temperatures for the low-temperature source, the high-temperature sink, the evaporation process, and the condensation process, respectively.

For systems of Fig. 1a (Carnot and otherwise) for the same cooling capacities ($Q_{\mathrm{L,R}}$), Eq. (1) reduces to $\Phi = W_{\mathrm{R,Carnot}}/W_{\mathrm{R}}$. For the Carnot cycle, $I_{\mathrm{int,R}} = I_{\mathrm{Q,L,R}} = I_{\mathrm{Q,H,R}} = I_{\mathrm{tot,R}} = 0$. Thus, using Eqs. (3)–(5) for the Carnot cycle and assuming that $T_{\mathrm{o}} = T_{\mathrm{H,R}}$, $W_{\mathrm{R,Carnot}} = Q_{\mathrm{L,R}}(T_{\mathrm{H,R}}/T_{\mathrm{L,R}} - 1)$. Therefore, Eq. (1) can be rewritten for Fig. 1a as:

$$\Phi = \frac{\mathrm{COP}}{\mathrm{COP}_{\mathrm{Carnot}}} = \frac{W_{\mathrm{R,Carnot}}}{W_{\mathrm{R}}} = \frac{Q_{\mathrm{L,R}}\left(\frac{T_{\mathrm{H,R}}}{T_{\mathrm{L,R}}} - 1\right)}{W_{\mathrm{R}}} = \mathrm{COP}\left(\frac{T_{\mathrm{H,R}}}{T_{\mathrm{L,R}}} - 1\right) \tag{6}$$

While details are not provided here, similar equations to Eqs. (2)–(5) could be written for the systems shown in Fig. 1b and c. If this is done, Eq. (1) can be rewritten for refrigeration systems such as shown in Fig. 1c where thermal energy is the primary energy input as:

$$\Phi = \frac{\zeta}{\zeta_{\mathrm{Carnot}}} = \frac{Q_{\mathrm{H,HE,Carnot}}}{Q_{\mathrm{H,HE}}} = \frac{\zeta}{\left(\frac{T_{\mathrm{H,HE}} - T_{\mathrm{L,HE}}}{T_{\mathrm{H,HE}}}\right)\left(\frac{T_{\mathrm{L,R}}}{T_{\mathrm{H,R}} - T_{\mathrm{L,R}}}\right)} = \frac{Q_{\mathrm{L,R}}/Q_{\mathrm{H,HE}}}{\left(\frac{T_{\mathrm{H,HE}} - T_{\mathrm{L,HE}}}{T_{\mathrm{H,HE}}}\right)\left(\frac{T_{\mathrm{L,R}}}{T_{\mathrm{H,R}} - T_{\mathrm{L,R}}}\right)} \tag{7}$$

The advantage of equations such as Eqs. (1), (6), and (7) is that they allow for refrigeration systems to be compared thermodynamically regardless of the primary energy input, type of cycle, and other details (e.g., types of hardware, controls, etc.). Consequently, the exergetic efficiency is a suitable parameter for comparing the performance of different cooling technologies.

2.2. Application considerations

When evaluating the thermodynamic performance of different technologies, one must consider what the comparison benchmark is in order to be certain that an equal comparison is being made between different technologies and systems and further recognize that the benchmarks reported in the literature are most often First Law-based efficiencies, e.g., COP, and not combined First and Second Law-based efficiencies. Thus, one should ask questions such as: Is the COP of the alternative technology based on an analysis of only the cycle [internal irreversibilities as given by Eq. (3)] or does it also include external heat transfer irreversibilities [Eqs. (4) and (5)]? One reason for this is that the later will almost certainly be included in an experimentally determined COP of a baseline vapor compression cycle but may not be included in analytical/theoretical analyses or experimental evaluations of emerging technologies due to their early developmental status.

Since external heat transfer irreversibilities are not always included in efficiency calculations for alternative technologies but can represent a significant fraction of the total system irreversibility (often well more than half), it is worthwhile to explicitly demonstrate the effects that external heat transfer irreversibilities have on the Second Law-based efficiency defined in Eq. (1), which can be rewritten for a system where T_{evap} and T_{cond} are constant as:

$$\Phi = \frac{T_{\text{H,R}}/T_{\text{L,R}} - 1}{T_{\text{cond}}/T_{\text{evap}} - 1} \tag{8}$$

where all temperatures are in Kelvin.

Fig. 2 plots Eq. (8) for $T_{\text{L,R}} = 0\ ^\circ\text{C}$ and $T_{\text{H,R}} = 40\ ^\circ\text{C}$ as a function of increasing driving temperature differences in the evaporator and condenser. Fig. 2 shows, for example, that driving temperature differences of 10 °C in both the evaporator and condenser reduce the Second Law-based efficiency by 35.8%. Note: ΔT refers to the driving temperature difference between the sink or source and the condenser or evaporator, respectively.

A second important question to ask is: Do the reported performance measures account for auxiliary powers? For vapor compression systems these are, for example, controls and fan powers, which may contribute on the order of 20% to the required total energy input to the system, thus reducing the Second Law-based efficiency by a similar amount. Each alternative cooling technology has its unique auxiliary subsystems, and power requirements for these subsystems should be considered.

Finally, when making comparisons among different technologies, one also must take into account the continuing improvements made in baseline vapor compression equipment over time. For example, according to AHRI statistical profiles [6], at the time of the Fischer et al. [2] report, the average seasonal cooling efficiency, referred to as SEER, of unitary equipment shipped in the U.S. was 10.61 (equivalent to a COP of 2.80); whereas, in 2007 the average SEER value had climbed to 13.66 (equivalent to a COP of 3.60), a 28.7% increase in average efficiency over a more or less fifteen year period. Today it is possible to purchase ultra-high efficient units

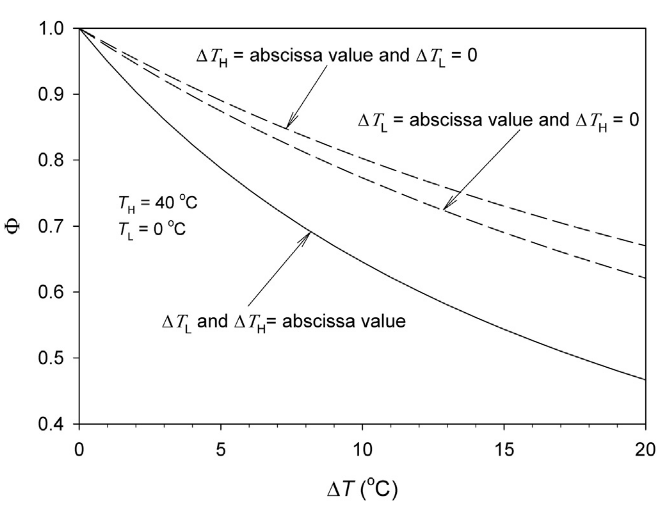

Fig. 2. Impact of external heat transfer irreversibilities on the Second Law efficiency.

with SEER values on the order of 22 (equivalent to a COP of 5.80). As noted by Calm and Domanski [7], the "A" condition of standard ratings for unitary air-conditioning equipment corresponds to ideal cycle conditions of $T_{\text{evap}} = 10\ ^\circ\text{C}$ and $T_{\text{cond}} = 35\ ^\circ\text{C}$, which yields a $\text{COP}_{\text{Carnot}} = 11.3$ (equivalent to a SEER value of 42.9). Therefore, the Second Law efficiencies of average unitary equipment shipped in 1994 was 0.248, in 2007 was 0.318, and today, ultra-high efficiency equipment can have values over 0.50.

3. Analysis

3.1. Classification of cooling technologies

Fig. 3 presents a taxonomy of cooling technologies based on the primary energy input (i.e., electrical, mechanical, etc.). This is, of course, not the only taxonomy that could be developed. For example, one could be developed based on operating temperature range (e.g., cryogenic, low-temperature, medium-temperature, high-temperature), one could be developed based on the phase of the working fluid (e.g., solid, liquid, gas, multiphase), one could be based on whether the primary energy input is in the form of mechanical work or thermal energy (heat), etc. Regardless, the divisions shown in Fig. 3 are not exhaustive, and, in some cases, could

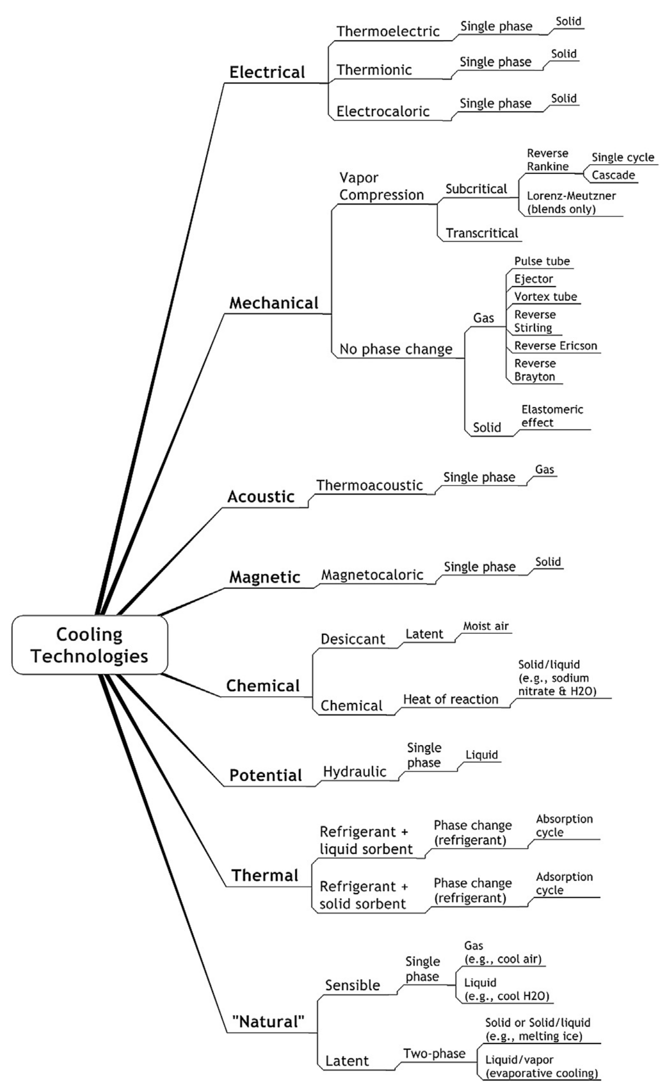

Fig. 3. Taxonomy of cooling technologies.

be combined. The usefulness of the taxonomy shown in Fig. 3 is that it categorizes systems first by the primary energy input and then subdivides them by the basic working principle and the "refrigerant".

3.2. Discussion of alternative cooling technologies

Following is a discussion of selected cooling technologies. Section 3.2.1 groups ones that are judged to have a low performance potential. Sections 3.2.2–3.2.7 discuss six cooling technologies which are deemed to have higher performance potentials and thus higher market promise. Throughout, only a few select papers are referenced; an exhaustive bibliography is not claimed.

3.2.1. Low-promise cooling options

While the report of Fischer et al. [2] discussed hydraulic refrigeration (experimental $\Phi \approx 0.25$ [8]), evaporative cooling (experimental $\Phi \approx 0.19$ [9]), Stirling cycle refrigeration (experimental $\Phi \approx 0.2$ to 0.3 [2]), Malone cycle refrigeration (experimental $\Phi \approx 0.2$ [2]), and thermoelastic heat pump (theoretical $\Phi \approx 0.35$ [2]), their assessments at that time were that these technologies likely would not be able to compete with vapor compression technology for the foreseeable future for comfort cooling and near-room temperature refrigeration. Our literature search revealed that little progress has occurred in any of these areas over the last 15+ years to fundamentally alter their assessments. Therefore, we affirm the assessments of Fischer et al. [2] regarding these technologies and thus will not discuss them any further in this paper.

In addition to the technologies discussed by Fischer et al. [2] our literature review showed some research activity on a few other cooling options. They are: (a) Ranque–Hilsh tube or vortex tube (see, e.g., Simões-Moreira [10] who achieved a $\Phi \approx 0.05$ when testing an air-standard cycle Ranque–Hilsh tube); (b) various gas cycles, e.g., Brayton cycle or Ericsson cycle (see, e.g., Hugenroth et al. [11] who achieved a $\Phi \approx 0.03$ when testing a liquid-flooded Ericsson cycle cooler) or; (c) pulse tube refrigerator (used in cryogenics but is not competitive for comfort cooling); (d) Einstein absorption cycle (see, e.g., Mejbri et al. [12] who discuss the feasibility of the Einstein refrigeration cycle and conclude that air-cooled chillers based on the cycle are not feasible and report a theoretical $\Phi \approx 0.05$); and (e) ejector cycle refrigeration (see, e.g., Pridasawas and Lundqvist [13] or Angelino and Invernizzi [14] who both point to the low cycle efficiency of ejector cycle refrigeration, with [14] reporting maximum theoretical values for $\Phi \approx 0.16$–0.28. The cycle continues to be investigated for heat driven systems, e.g., Dennis [15]). Consequently, our general assessment of these technologies is that they are, and for the foreseeable future will be, unable to compete with vapor compression systems.

A recent report by Brown et al. [4] lists thermionic and thermotunneling cooling among their five most promising alternative cooling technologies. However, our literature review did not provide us with sufficient evidence to support this opinion. Perhaps among the two, the easiest for us to dismiss is thermionic cooling since Brown et al. [4] assess it as having the lowest potential (theoretical $\Phi \approx 0.2$–0.3 and experimental $\Phi \approx 0.1$) of the five alternative cooling technologies they discussed in detail in their report. On the other hand, they rank thermotunneling cooling as third, ahead of both thermoelectric and thermionic, pointing out the large number of patents held by a single private company, which may serve as an indicator of a rapid R&D effort, and citing its theoretical performance at room temperature ($\Phi \approx 0.5$–0.8) as being similar to that of vapor compression technology. One should keep in mind that no details (cooling capacities, temperature lifts, experimental verification) are provided for this theoretical performance estimate and that it does not account for the irreversibilities associated with coupling to the source and sink. Thus, we disagree with the conclusion that these technologies hold promise to compete with vapor compression technology for comfort cooling and near-room temperature refrigeration applications. Even if all technical barriers associated with thermionic and thermotunneling cooling could be overcome, these technologies are more likely to be applied for small cooling capacity and high heat flux applications, e.g., electronics cooling, because of expected difficulties in scaling them to capacities required in comfort cooling and near-room temperature refrigeration applications. Moreover, our review showed little research and development activity is occurring for these two technologies other than for spot cooling of electronics.

3.2.2. Sorption cooling

3.2.2.1. Physical principle. Absorption and adsorption cooling systems operate much like the conventional vapor compression refrigeration cycle where perhaps the main difference is that thermal energy is the principle driver of the cycle instead of mechanical work. Therefore, in lieu of a mechanical compressor a generator/absorber pair or generator/adsorber pair and mechanical pump drive the working fluid. Typical working fluids include ammonia/water and lithium bromide/water for absorption systems, and activated carbon/ammonia, activated carbon/methanol, silica gel/water, zeolite/water, metal chlorides/ammonia, or composite adsorbents for adsorption systems [16]. The main difference between absorption and adsorption systems is that in the former a substance in one phase absorbs into a second phase to form a solution, and in the later a substance in one phase is adsorbed/desorbed onto a solid (e.g., silica gel, zeolite, activated carbon, etc.) surface.

3.2.2.2. Assessment of Fischer et al. [2]. Because of the particular energy market (cost of electricity, cost of gas, and subsidies), absorption chillers were widely used (e.g., Japan) or represented a minor share (e.g., United States). A second use was to incorporate them into a hybrid system together with a conventional electric-driven chiller to manage peak electricity loads; however, their efficiencies were lower than conventional electric-driven chillers. Triple-effect absorption chillers were under development that promised comparable efficiency to conventional electric-driven chillers. Smaller absorption systems were under development for residential and light commercial applications.

3.2.2.3. Advancements and state of technology. Absorption chillers continue to represent a large fraction of chillers sold in some markets (e.g., Japan and China) and a more minor share in the other large markets, with Asian markets having approximately 85 % of the stock of absorption chillers with capacities greater than 350 kW [17]. This continues to be driven primarily by the cost of electricity, the cost of gas, and subsidies in the particular market. Since the report of Fischer et al. [2], perhaps the two most important innovations in absorption cooling have been the introduction of triple-effect chillers and the introduction of generator–absorber heat exchanger (GAX) technology.

Triple-effect chillers were commercially introduced in 2005 [18]. Deng et al. [18] report that the efficiencies of triple-effect lithium bromide/water absorption machines can be 40–70% greater than double-effect machines and up to as much as 300% greater than single-effect machines. One of the major disadvantages for triple-effect chillers is that they require heat source temperatures greater than about 200 °C, while double-effect machines require source temperatures in the range from 120 °C to

170 °C, and single-effect machines require heat source temperatures of about 80 °C to 120 °C [18].

Smaller, single-effect, gas-fired ammonia/water absorption systems for residential and light commercial applications were commercially introduced, which was made possible by the commercial development of GAX technology. GAX technology can improve the COP by 10–40% [19] compared to conventional ammonia/water absorption systems.

With increased societal emphasis on energy recovery and alternative energy sources, in addition to direct-fired absorption chillers, both absorption and adsorption systems have seen greater commercial application and increased research and development effort for applications involving waste heat recovery (e.g., industrial processes, fuel cells, and gas turbines), low-temperature sources (e.g., solar and geothermal), and tri-generation (cooling, heating, and power). In addition to technical advancements, regulations and incentives have begun to be implemented that support and promote the use of sorption cooling. For example, the Renewable Heat Incentive Scheme [20] provides incentive payments to non-domestic installations of sorption chillers that were commissioned on or after July 15, 2009.

Deng et al. [18] report that depending on configuration, heat source, and application, thermal COPs (cooling capacity divided by required thermal input) of sorption systems range from approximately 0.1 to 1.7. (Using Eq. (6) for a temperature lift of 20 °C at room temperature for a heat source temperature of 600 K would result in $\Phi \approx 0.01$ to 0.09 for adsorption, $\Phi \approx 0.03$ to 0.09 for single-effect absorption, $\Phi \approx 0.11$ to 0.16 for double-effect absorption , and $\Phi \approx 0.19$ to 0.23 for triple-effect absorption.)

3.2.2.4. Overall assessment. For the next twenty year period, our judgment is that research and development will continue; however, absorption and adsorption cooling will retain a small share in most markets unless the cost mix of primary energy significantly changes. These systems will continue to be used and will expand into applications where management of peak electricity demand is important, where sources of waste heat are readily available, and in solar applications.

3.2.3. Desiccant cooling

3.2.3.1. Physical principle. In desiccant cooling systems, the desiccant removes water from an incoming airstream, which is subsequently cooled by an evaporative cooler or air conditioner. Desiccant systems are particularly attractive for spaces with low sensible heat ratios (SHRs), that is, for spaces with high latent loads. When coupled with a conventional vapor compression refrigeration cycle, the latent load can be managed by the desiccant subsystem, and the sensible load can be managed by the vapor compression refrigeration cycle. Desiccant materials are either liquid or solid sorbents, both of which must be regenerated (e.g., using waste heat, solar heating, etc.). Desiccant cooling can reduce the size of conventional vapor compression equipment and can lead to increased evaporation temperatures, and hence increased system efficiencies.

In liquid desiccant-based systems, the airstream is brought into direct contact with the liquid desiccant which absorbs some of the airstream's water vapor. In solid desiccant-based systems, the airstream is brought into direct contact with the solid desiccant, which is typically contained in a rotating wheel; one half of the wheel is exposed to the airstream while the other half is exposed to a heat source for regeneration.

3.2.3.2. Assessment of Fischer et al. [2]. Desiccant cooling systems were commercially available. The authors noted, however, that capital costs had to be lowered for desiccant systems to gain wider market acceptance. They also noted that better performing desiccants were needed, and that heat and mass losses in systems had to be reduced to improve system efficiencies. Finally, they noted that more research and development was needed to address cyclic performance of desiccants and the effects of dust buildup on desiccant wheel-based systems.

3.2.3.3. Advancements and state of technology. Since desiccant systems must be coupled with conventional systems to manage sensible cooling loads, they have not achieved widespread market penetration. Furthermore, as Dieckmann et al. [21] point out, liquid desiccant units have not achieved widespread market penetration, selling only several thousand units per year, due to low efficiencies. Dieckmann et al. [21] point to three reasons that the efficiencies of these units are limited: (1) irreversibilities in the absorber, (2) large liquid and air pressure drops, and (3) the COPs of single-effect systems are inherently limited to values less than one. In fact, according to them, thermal COPs (cooling capacity divided by required thermal input) of existing units are approximately 0.5–0.6. (Using Eq. (6) for a temperature lift of 20 °C at room temperature for a heat source temperature of 600 K would result in $\Phi \approx 0.07$.) Much research and development activity has been directed toward desiccant materials. Halide salts are the most commonly used liquid desiccant material because of their non volatility and their excellent absorbing characteristics; however, they are corrosive and expensive [22]. Typical materials for solid desiccant wheels include silica gels, activated alumina, natural and synthetic zeolites, titanium silicate, lithium chloride, and synthetic polymers [23]. Solid desiccant wheels suffer from low thermal COPs, maintenance issues, pressure drop, and the need for a heat source to regenerate the desiccant.

3.2.3.4. Overall assessment. For the next twenty year period, our judgment is that it is likely that the overall market penetration of desiccant cooling systems will remain low; however, research and development will continue in the areas of solid and liquid desiccant materials, improving cycle performance, reducing maintenance issues, and reducing costs. The most likely continued use of desiccant-based systems will be to manage latent loads in high humidity areas while being coupled with smaller-sized, conventional vapor compression technology to manage sensible loads.

3.2.4. Magnetic cooling

3.2.4.1. Physical principle. Magnetic cooling is based on the magnetocaloric effect (MCE). For normal (most) magnetocaloric materials, magnetization will lead to heating of the material, and demagnetization will lead to cooling of the material. An important parameter for characterizing magnetic cooling is the Curie Temperature, which is the point where a ferromagnetic material loses its permanent magnetism and becomes paramagnetic. The magnetocaloric effect is most pronounced near the Curie Temperature. Increasing the magnetic field intensifies the effect.

According to Kitanovski and Egolf [24], most heat pump prototypes are based either on moving magnetocaloric materials through a static magnetic field or on moving a magnet mechanism relative to a static magnetocaloric material bed. In either case, as the magnetocaloric material heats or cools, it can be coupled to an external heat transfer fluid to realize a heat pumping effect. For comfort cooling and refrigeration, the most suitable materials are ones which have a Curie Point near room temperature: one such material is the rare earth metal gadolinium.

3.2.4.2. Assessment of Fischer et al. [2]. Design studies had been performed for rotary magnetic-based machines and their efficiencies and performances predicted for various operating conditions; however, only one working prototype at Los Alamos Laboratory was noted. One of the largest hurdles noted to developing magnetic heat pumps was the unavailability of low-cost, room temperature superconducting materials to enable strong electromagnets. The report noted one study from the early 1990s by the Electric Power Research Institute that indicated it would take 15+ years for the materials to become available.

3.2.4.3. Advancements and state of technology. Since the report of Fischer et al. [2], there has been considerable increase in research activity indicated by the large number of papers and patents from many different research groups published during the 2000's. For example, Gschneidner and Pecharsky [25] report that prior to the discovery of the giant MCE (explained below) in 1997, approximately 10–20 papers were published annually with the word "magnetocaloric" in the title, abstract, or among the keywords, and that after 1997 the number of papers increased rapidly reaching well over 250 per year by the time of the writing of their paper in 2007. Another example provided by Yu et al. [26] is the number of European patents issued for magnetic refrigerators and heat pumps. In particular, during the period 1997–2009, a total of 135 patents were issued, with 68% of them having been issued during 2002–2009.

Several significant developments have been reported over the last decade or so, a few of which will be highlighted here. Related to materials development, Pecharsky and Gschneidner [27] discussed the discovery of the so-called giant MCE in $Gd_5(Si_2Ge_2)$ with approximately 50% greater MCE than that of pure gadolinium. However, subsequent direct adiabatic temperature change measurements by Giuere et al. [28] showed a significantly smaller MCE, and work by Provenzano et al. [29] pointed to large hysteretic losses of this material. Second, as noted by Yu et al. [26], Astronautics Corporation of America successfully developed in 2001 the world's first room temperature magnetic refrigerator based on permanent magnets. Third, as noted by Engelbrecht et al. [30], layered regenerator beds consisting of layers of several MCE materials were developed with the promise to increase system performance. Fourth, as noted by Yu et al. [26], there are now 41 working prototypes that have been built and are (or were) operating since the first room temperature machine was introduced by Brown [31].

The research activity noted above, as well as other activity, continues. For example, extensive materials research is ongoing, where state-of-the-art materials are the $La(Fe,Si)_{13}H_y$ family, NiMnGa shape-memory alloys, $MnFeP_xGe_{1-x}$ alloys, and $Gd_5(SiGe)_4$ intermetallic compounds [32]. In addition to materials research, prototypes continue to be designed, developed, and tested, although all of the prototypes designed and developed to date have been small capacity machines. For example, of the 41 working prototypes, the maximum cooling capacity is 600 W, and of the 19 machines with reported cooling capacity, the average cooling capacity is approximately 125 W [26]. Most of the working prototypes use gadolinium.

Despite the presence of these 41 working prototypes and the ongoing work, the literature lacks reliable experimental data for comparing magnetic refrigeration and vapor compression technology; only three of the machines report some efficiency measurements. In particular, a permanent magnet refrigerator produced a maximum cooling power of 540 W with a COP of 1.8 at a temperature lift of 0.2 °C [33], and a permanent magnet refrigerator produced a COP of 1.6 with a cooling capacity of 50 W and a temperature lift of 2 °C [34]. Both of these machines are inefficient with very low values of Φ. Gschneidner and Pecharsky [32] report that a superconducting magnet refrigerator [35] produced a cooling capacity of 600 W with a COP of nearly 10 ($\Phi \approx 0.6$) with a temperature lift of 10 °C; however, at a temperature lift of 22 °C, the cooling capacity dropped to 150 W and a COP of 2 ($\Phi \approx 0.2$).

3.2.4.4. Overall assessment. Considerable research and development activity needs to continue in MCE materials, magnets, regenerators, system modeling, and systems engineering. Considerable steps need to be made to scale-up the cooling capacities and temperature lifts so that they are both appropriate for comfort cooling, and the irreversibilities, such as, hysteresis, coupling to an external heat transfer fluid, internal conduction, and large pressure drops through the regenerator bed (single-phase flow), if present, need to be addressed. Considerable activity needs to be undertaken to provide system performance test results so that reliable comparisons can be made with other technologies. Finally, R&D efforts must address materials and system costs, the processing of large quantities of MCE materials, and the larger size required per cooling capacity as compared to conventional vapor compression technology [30]. Hirano et al. [36] highlight a large Japanese research effort in which they hope to move from the COP value of 1.8 of Okamura et al. [33] to a COP above 10; however, no timeline or research details are provided.

Therefore, for the next twenty year period, our judgment is that it is likely that intense research and development activity will continue in the area of room temperature magnetic cooling; however, significant breakthroughs in magnetocaloric materials and design concepts are needed to bring magnetic cooling to the marketplace and make it competitive with vapor compression technology.

3.2.5. Thermoacoustic cooling

3.2.5.1. Physical principle. Thermoacoustic cooling is based on the conversion of acoustic energy to thermal energy. The presence of an acoustic wave (standing or traveling) expands and contracts a working fluid (gas). As the gas expands, its pressure and temperature are reduced; likewise, as the gas contracts, its pressure and temperature are increased. In addition, the pressure differences lead to movement of the working gas. To achieve cooling and heating, the working gas must be coupled to an external heat transfer fluid through heat exchangers.

Thermoacoustic devices are contained in a sealed pressure vessel and consist of an acoustic driver (e.g., loudspeaker), which radiates acoustic energy into a resonator containing a porous "regenerator" or "stack". The "regenerator" or "stack" serves to allow the working fluid (gas) to displace within it and to transfer thermal energy to and from the gas. The "cold" and "hot" heat exchangers are used to transfer thermal energy to and from an external working fluid. The acoustic waves can be either standing-wave or traveling-wave, which Garrett [37] uses to classify thermoacoustic refrigerators as either: (1) standing-wave stack-based devices or (2) traveling-wave (acoustic-Stirling or pulse tube) regenerator-based devices. The distinction he makes between "regenerator" or "stack" is beyond the scope of the present discussion; however, the terms are used to distinguish the physical geometry, where a non-dimensional number (Lautrec number), which compares the hydraulic radius to the thermal penetration depth, is used to make the distinction. In any case, a "regenerator" or a "stack" serves the same purposes; therefore, hereafter, we simply refer to them as a regenerator. The technical differences between standing-wave and traveling-wave devices will not be discussed further here, other than to note that the acoustic network of a traveling-wave device is more complex than that of a standing-wave device [37] and that since a traveling-wave device is based on the Stirling cycle it can theoretically achieve the Carnot efficiency.

3.2.5.2. Assessment of Fischer et al. [2]. Prototype thermoacoustic refrigerators had been built and tested with low levels of efficiency measured relative to conventional vapor compression technology. The report noted there were no fundamental reasons why the efficiency levels could not be improved with sufficient development; however, the overall assessment was that thermoacoustic refrigerators held little promise for replacing vapor compression equipment, with commercialization prior to 2000 being unlikely. The report further noted that extensive effort was needed: (1) to address the constraint imposed by the resonator on the size of the external heat exchangers, (2) to address the likelihood of shocks waves being created, and (3) to develop acoustic compressors.

3.2.5.3. Advancements and state of technology. While Minner et al. [38] predicted a COP = 1.7 for a thermoacoustic refrigerator, which was comparable to conventional household refrigerators of that time, the measured Second Law efficiency for prototypes has typically ranged from about 0.03 to 0.21 [39], which is significantly below contemporary vapor compression values, which range from about 0.3 to 0.5.

Most early development work was based at Los Alamos Laboratory, with much follow on work being performed at Purdue University and Penn State University. In 2004, Professor Garrett of Penn State University created a consulting company, ThermoAcoustics Corporation, dedicated to thermoacoustic, which developed thermoacoustic cycle-based prototypes for NASA, the U.S. Navy, and Ben & Jerry's (ice cream producer), with some commercial promise (see, for example, Smith [40]). However, based on our review, we are led to believe that none of these projects currently is being actively pursued. Continuing issues with prototypes include low cooling capacities, large physical size, heat exchanger inefficiencies, and the parasitic heat conduction from the hot heat exchanger to the cold heat exchanger.

3.2.5.4. Overall assessment. For the next twenty year period, our judgment is that it is likely that thermoacoustic cooling will remain uncompetitive with vapor compression technology, but will likely continue to be a topic of research interest. For example, while not commercially viable, Zink et al. [41] considered a thermoacoustic engine for automotive applications which could be powered by engine exhaust and used to drive a thermoacoustic air conditioner. Finally, while outside the scope of this paper, some interest is being shown in thermoacoustically-driven pulse tube cryocoolers because of their reliability and overall size.

3.2.6. Thermoelectric cooling

3.2.6.1. Physical principle. Thermoelectric cooling is based on the Peltier effect: when an electrical current is applied to two conductors of dissimilar metals, a temperature difference will develop across the two junctions, that is, one junction will become colder and the other one hotter. Thermoelectric materials are characterized by a so-called figure of merit Z, which is a function of material properties and the absolute temperature. However, rather than reporting solely Z, thermoelectric materials are typically characterized by the dimensionless figure of merit ZT, which, if increased, can be shown to lead to increased COP [42]. ZT values on the order of unity result in Second Law efficiency values from 0.10 to 0.15 [42].

3.2.6.2. Assessment of Fischer et al. [2]. State-of-the-art materials had ZT values of approximately 1, with a maximum value on the order of 2 having been suggested based on theoretical analysis. Citing Mathiprikasam [43], the report noted that ZT values greater than 3 would be needed for thermoelectric refrigeration to become energy efficient on a cycle basis, but if coupling with external heat transfer fluids were to be considered in the analysis, even greater values of ZT would be needed to make thermoelectric cooling competitive with vapor compression technology. The report concluded that, with the efficiency much below that of vapor compression technology, thermoelectric cooling may only be attractive in niche applications, e.g., where compactness or portability are needed.

3.2.6.3. Advancements and state of technology. Modern thermoelectric cooling devices are based on N-type or P-type semiconductor materials, e.g., Bi_2Te_3 with ZT values of approximately unity. Considerable materials research has been conducted with breakthroughs claiming up to $ZT = 2.4$ at 300 K [44]; however, none of these materials have been commercialized. As noted by Bell [42], the commercialization of these materials faces considerable hurdles, and in practical devices, as of yet, have not been able to surpass the performance of state-of-the-art materials such as Bi_2Te_3 (bismuth telluride) and Sb_2Te_3 (antimony telluride). The search for improved materials continues with particular focus on nanostructures, e.g., quantum dot superlattices [45]. However, one should keep in mind that Z increases with decreasing electric resistivity and decreasing thermal conductivity. Thus one inherent problem in increasing Z is that materials that are good conductors of electricity (needed for thermoelectric devices) are also typically good conductors of heat.

The maximum theoretical efficiency achievable [2] is:

$$\Phi = \frac{(1+ZT)^{0.5} - T_H/T_L}{(1+ZT)^{0.5} + 1} \tag{9}$$

Fig. 4 plots Eq. (9) for conditions defined by $T_H = 40\,°C$ and $T_L = 0\,°C$. The figure shows that (1) state-of-the-art materials have a maximum theoretical $\Phi \approx 0.18$, (2) the maximum reported ZT value of 2.4 results in theoretical $\Phi \approx 0.37$, and (3) that a ZT value of approximately 4.4 would be needed to achieve a theoretical $\Phi \approx 0.50$, which is achievable by today's vapor compression technology. It is to be noted that Eq. (9) does not include auxiliary powers and external heat transfer irreversibilities that occur when coupling the thermoelectric device to an external heat transfer fluid.

3.2.6.4. Overall assessment. For the next twenty year period, our judgment is that it is likely that thermoelectric cooling will remain uncompetitive with vapor compression technology other than in certain niche applications with low cooling requirements and

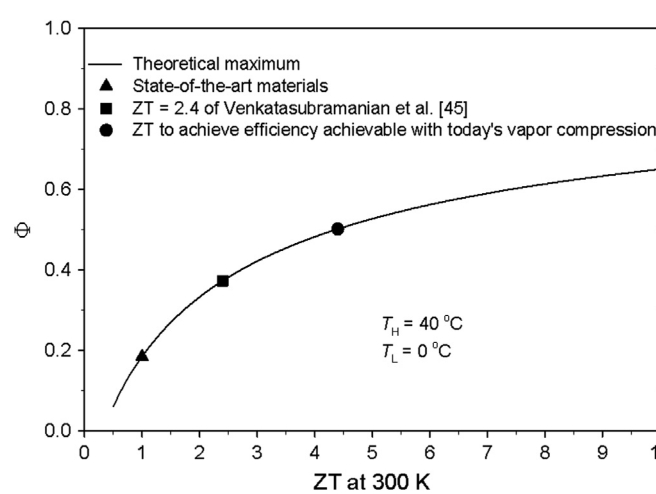

Fig. 4. Second Law efficiency as a function of non-dimensional figure of merit.

modest temperature lifts, e.g., certain electronic cooling applications, certain military and space applications, certain medical applications, recreational cooling, etc.

3.2.7. Transcritical CO_2

3.2.7.1. Physical principle. The transcritical refrigeration cycle is a vapor compression refrigeration cycle whose heat rejection occurs above the critical temperature of the working fluid. Thus, three of the four basic processes of the transcritical cycle (heat addition, compression, and expansion) are the same as for the subcritical vapor compression cycle; whereas, the heat rejection process for a transcritical cycle does not result in a phase change for the working fluid. Instead, the temperature, pressure, and density of the working fluid change continuously during the heat rejection process.

Any working fluid whose critical temperature approaches or is above the sink temperature would yield a transcritical cycle. CO_2, because of its low critical temperature (approximately 31 °C), results in a transcritical cycle for many comfort cooling and near room temperature refrigeration applications. Most common working fluids are selected so they operate in subcritical cycles to achieve better theoretical efficiency.

Generally speaking for vapor compression cycles, the volumetric cooling capacity increases and COP decreases with decreasing critical point temperature. When compared with medium and high-temperature working fluids, low critical temperature working fluids result in higher working pressures, have better transport properties (particularly in the region of the critical point), have larger throttling irreversibilities, have larger heat rejection irreversibilities for heat exchangers without glide matching, and result in better compressor efficiencies because of their lower pressure ratios.

3.2.7.2. Assessment of Fischer et al. [2]. Transcritical CO_2 was under development for automotive applications, where early prototypes demonstrated feasibility of the technology. They noted that there were no independent evaluations of performance, safety, durability, manufacturing cost, and maintenance problems, all which needed to be resolved before the technology could be viable in consumer products. Based on reported prototype results, they projected the CO_2 Total Equivalent Warming Impact (TEWI) could be as low as approximately 60% that of R134a. They further noted that the technology differs significantly from conventional air conditioners and thus did not foresee significant market penetration before 2003 to 2010.

3.2.7.3. Advancements and state of technology. Transcritical CO_2 air conditioning and refrigeration has been a very active research area in alternative technology refrigeration over the past 15 years. Since the report of Fischer et al. [2], there have been many hundreds of articles published on topics such as compressors, heat exchangers, ejectors, expanders, controls, lubricants, properties, heat transfer, materials, safety, environmental impact, applications, etc. The literature is indeed vast and thus no attempt will be made here to reference the body of work. Instead, for brevity, of the many articles that could be referenced, we refer the interested reader to Kim et al. [46], Hrnjak [47], or Groll and Kim [48] for more information.

There have been a large number of application areas investigated for transcritical CO_2, some of the most promising of which

Table 1
Exergetic efficiencies of cooling technologies.

Cooling/refrigeration technology	Φ Parasitic losses not included	Φ Parasitic losses included	Comments
Vapor compression cycle	0.63 (e)	0.50 (e)	Mature technology; large range of cooling capacities and application temperatures; mainly driven by electric energy
Absorption cycle		0.03–0.09 (e) 0.11–0.16 (e) 0.19–0.23 (e)	Single-effect Double-effect Triple-effect Heat driven cycle; solar applications; uncompetitive in many electricity markets
Adsorption cycle		0.01–0.09 (e)	Heat driven cycle; solar applications; uncompetitive in many electricity markets
Desiccant cooling		0.07 (e)	Coupled with a conventional system; not viable in all operating conditions; high cost
Einstein refrigeration cycle	0.05 (t)		Not feasible for many operating conditions; the cycle is not easy to realize
Ejector refrigeration cycle	0.16–0.28 (t)		Large irreversibilities related to normal shock and fluid mixing
Ericsson cycle		0.03 (e)	
Evaporative cooling	0.19 (e)		Coupled with a conventional system; not viable in all operating conditions; high cost
Hydraulic refrigeration	0.25 (e)		Large size and weight; limited progress since report of Fischer et al. [2]
Magnetic refrigeration/cooling	0.2–0.6 (e)		Strong Φ degradation with increasing temperature lift; limited number of and availability of materials; large internal irreversibilities; high costs; large size; limited reliable experimental results available for comparison purposes
Malone cycle	0.2 (e)		Impractical for most working fluids since it operates near the critical point; limited progress since report of Fischer et al. [2]
Pulse tube refrigeration			Used in cryogenics
Ranque–Hilsh tube (vortex tube)	0.05 (e)		
Stirling cycle	0.2–0.3 (e)		Used in low-temperature applications; low cooling capacities; high costs; large internal irreversibilities; limited progress since report of Fischer et al. [2]
Thermionic cooling	0.2–0.3 (t) $<$0.1 (e)		Large number of technical barriers need to be overcome; immature technology; small cooling capacities
Thermoacoustic cooling	0.5 (t) 0.03–0.21 (e)		Large size; high costs
Thermoelastic heat pump	0.35 (t)		Limited progress since report of Fischer et al. (1994)
Thermoelectric cooling	0.18–0.37 (t)		Used in niche applications; limited availability of high performance materials; low weight; strong Φ degradation with increasing temperature lift
Thermotunneling cooling	0.5–0.8 (t)		Large number of technical barriers need to be overcome; immature technology; small cooling capacities
Transcritical CO_2 cycle			Similar Φ to conventional vapor compression cycle at lower ambient temperatures with significant Φ degradation as ambient temperature increases; high refrigerant pressure; high costs

(t) = theoretical; (e) = experimental.

are: water heaters, vending machines, residential air conditioners and heat pumps, and automotive air conditioners [48]. Moreover, because of the increased research activity related to the transcritical cycle, CO_2 has received increased interest as a refrigerant in conventional subcritical cycles, e.g., the low-temperature refrigerant in secondary loop systems for supermarket applications. The application which has received the most interest is automotive air conditioning, including the possibility of its near-term commercial use.

The CO_2 transcritical cycle was proposed in the 1990s, in part, to reduce the global warming impact from HFC-based air conditioning and refrigeration systems. However, there has been considerable debate whether the use of the CO_2 transcritical cycle instead of conventional HFC-based vapor compression equipment reduces the impact on the climate for individual applications. This debate centers around the fact that the impact on the climate strongly depends on the system efficiency in addition to the refrigerant's GWP and its leak rate. For example, Kikuchi and Ikegami [49] concluded that there is no clear evidence that the environmental impact of CO_2 automotive air-conditioning systems is smaller than R134a systems. A second example is a study referenced by SAE [50], which shows that the Life-Cycle Climate Performance (LCCP) of a CO_2 automotive air conditioner is approximately 25% worse than R134a for Phoenix, Arizona and approximately 5% worse than R134a for Frankfurt, Germany.

3.2.7.4. Overall assessment. Imminent regulations aiming to phase down the use of high-GWP refrigerants will further increase interest in applying transcritical CO_2 in comfort cooling and refrigeration applications. For the next twenty year period, our judgment is that it is likely that the transcritical CO_2 cycle will continue to be an area of research interest with possible larger scale commercial introduction, which would help to reduce the system costs to a more competitive level. If large scale commercialization is realized, it is likely to be for small size, small capacity applications benefiting from the large volumetric cooling capacity of CO_2. Moreover, the transcritical CO_2 cycle will likely expand in niche applications, e.g., heat pump water heaters. Considering the thermodynamic disadvantage compared to the subcritical vapor compression cycle, for the transcritical CO_2 cycle to be more widely applied, it is likely that outside factors will need to come into play, e.g., taxes, regulations, legislation, or public perception. Some issues that still need to be addressed are: efficiency at high ambient temperatures, high operating pressures, cost, safety, and maintenance.

4. Conclusions

We reviewed the state-of-the-art of alternative cooling and near-room temperature refrigeration technologies using as a starting point the assessment of these technologies by Fischer et al. [2]. Table 1 lists the considered technologies and provides general comments regarding their merits and demerits along with demonstrated values of their exergetic efficiencies. Among the new technologies studied, only the transcritical CO_2 cycle has entered the commercial market to date since 1994, primarily for water heating and commercial refrigeration, though its market share remains small. We expect additional transcritical CO_2-based products to be introduced as the technology develops and regulations limiting the use of HFC refrigerants are implemented. The attractiveness of absorption, a "conventional" alternative technology, will also increase due its capability to utilize waste heat and solar energy, which improves the life-cycle economics of this equipment compared to vapor compression machines.

The progress in the technological development of the remaining emerging technologies since 1994 has been much slower than the predictions made at that time by experts working in the respective fields. Among them, magnetic refrigeration has the highest level of current research activity, and is considered to hold some, but not imminent, promise for implementation as better magnetocaloric materials and other technical breakthroughs are needed to realize this promise. In the coming years, we do expect some of the different cooling technologies to find increased market application, particularly niche ones. However, we do not foresee any of these technologies achieving widespread displacement of vapor compression technology in the immediate future because of their lower energy efficiencies or higher costs, or both. Significant technical breakthroughs are needed to advance the market prospect of novel cooling concepts. One goal of a recent $30 million funding package [51] is to accelerate commercial implementation of these systems.

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