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Review

Review of Residential Air Conditioning Systems Operating under High Ambient Temperatures

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Abstract: This article provides an overview of residential vapor-compression air conditioners operating under high ambient temperatures (HAT). For the purpose of this article, a minimum temperature criterion, 40 °C and above, was developed to evaluate studies that were conducted at HAT. Several HAT organizations and projects were launched with the purpose of assessing the performance of low-GWP (GWP = global warming potential) refrigerants when operating under HAT and accelerating the transition to such refrigerants. Previous studies of air conditioner improvements (i.e., for condensers, evaporators, compressors, and refrigerants) were discussed under HAT conditions. This article also explores the challenges, the possible design modifications, and several limitations of air conditioners operating under HAT. Condenser improvements showed an 18 to 50% higher coefficient of performance (*COP)* and an 8 to 30% higher cooling capacity. Only one study was found for evaporator enhancement under HAT which improved the *COP* by ~7% and cooling capacity by ~10%. Experimental compressor improvements achieved 2 to 17 °C lower discharge temperature and up to 15% higher cooling capacity, whereas the *COP* ranged from −4% to +3% of the baseline values. Under HAT conditions, several A2L refrigerants exhibited an attractive performance compared to R-410A while none outperformed R-22 in terms of both cooling capacity and *COP*. Considering R-22 alternatives, all A1 refrigerants exhibited lower *COP*, A2L refrigerants achieved comparable *COP*, and A3 refrigerants reached higher *COP*.

Keywords: high ambient temperature; hot ambient; hot climate; air conditioning; global warming; climate change; cooling

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1. Introduction

The purpose of this article is to evaluate previous reports related to residential air conditioning systems under high ambient temperatures (HAT) and provide an overview of what has been carried out in this area to date. Most studies investigated residential air conditioners at a wide range of temperatures and hence it can be difficult and time-consuming to evaluate the results only at HAT, especially since there is no common temperature cut-off. Therefore, this article can be helpful for researchers who are interested in this topic for the following reasons: (i) it defines a criterion that segregates the moderate and high ambient temperatures, (ii) it shows the improvements of each technique compared to the baseline under HAT, (iii) it helps in avoiding repeated research work, (iv) it illustrates how A2L and A3 refrigerants operate under HAT compared with baseline refrigerants (e.g., R-410A and R-22), and (v) it recommends promising techniques for further development of AC units with A2L refrigerants under HAT. Lastly, to the best of the authors' knowledge, there is no review article that particularly discusses residential air conditioners operating under HAT. Section 1 reviews the main issues associated with HAT and their impacts on different regions of the world. Section 2 explains the approach followed to find interdisciplinary articles related to this study. Section 3 provides a brief description of different HAT organizations and projects. Section 4 discusses the literature studies of the main components of residential vapor compression systems operating under high ambient temperatures. In Section 5, various design modifications for residential air conditioners under high ambients are explored. Limiting factors of air conditioners operating under HAT conditions corresponding to Kuwait and Phoenix, Arizona, USA are investigated in Section 6. Section 7 summarizes the primary findings of the Section 4 literature studies.

Air conditioners operating under HAT must overcome high summer loads, high discharge temperature, and performance degradation. Consequently, to reduce both direct and indirect emissions, regulatory jurisdictions have added more requirements such as minimum energy efficiency and environmentally friendly refrigerants. The use of air conditioners in hot regions will be increasingly popular as a result of population and economic growth, which may affect the environment adversely, directly and indirectly if no mitigation measures are taken. Therefore, the problem becomes more complex and needs design modifications to alleviate the unit performance degradation, accelerate the use of environmentally friendly refrigerants, and comply with the minimum energy requirements under HAT conditions.

1.1. Climate Change

Since the discovery of modern electric air conditioners by Willis Carrier, there are now ~2 billion air conditioning units that have been installed globally [1]. The use of air conditioning has improved living standards by reducing heat exposure and providing comfortable indoor conditions. Hence, it became a necessity rather than a luxury in hot climates. In 2018, the percentage of households with air conditioners reached ~90% in some countries (i.e., USA and Japan) and less than 10% in other countries [1]. Moreover, ~69% of the installed units worldwide are attributed to China, the USA, Japan, and Korea, respectively, from highest to lowest. However, this is expected to change substantially in the next 30 years. The use of air conditioners will become increasingly popular as a result of population and economic growth, especially in hot regions [1]. By 2038, installed units are expected to double, reaching 4 billion units. India is expected to be the second-largest country in installed household air conditioners. The expansion of installed units is expected to improve human wellbeing in hot countries but can adversely affect the climate. In 2018, the International Energy Agency (IEA) stated that the energy consumption of air conditioners, including electric fans, accounted for 10% of global energy consumption and is expected to triple in the next 30 years [1]. The expected boom of air conditioners will cause a high electricity demand that utilities must be able to provide without harming the environment. “Growing demand for air conditioners is one of the most critical blind spots in today’s energy debate”, according to the IEA Executive Director [2].

Air conditioning systems interact with the environment mainly in three ways: electricity consumption, refrigerant leakage to the atmosphere, and heat rejected to the ambient. Electricity is produced primarily by burning fossil fuels, which represent ~63% of global electricity sources [3]. This process involves the releasing of greenhouse gases (GHG), mainly CO_2, into the atmosphere and eventually contributes to global warming. In fact, 25% of global GHG is due to electricity and heat production, which are the largest global sources of GHG [4]. Hence, the increase in the number of air conditioners indirectly increases the amount of CO_2 in the atmosphere. Moreover, refrigerators and air conditioners have been using several types of working fluids since the last century including: chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), and hydrofluorocarbons (HFCs).

A leakage of those fluids can occur multiple times during the lifecycle of an air conditioner, for several reasons. Unfortunately, the environmental effects of those substances were not always well perceived when they were introduced. Later on, researchers discovered the environmental effect of these refrigerants, which are depleting the ozone layer and acting as potent GHG, up to 1000 times stronger than CO_2. Moreover, the way that air conditioners work is to pump heat from the indoors to the outdoors, which can worsen the heat island effect and increase the cooling demand [5,6]. For example, a study conducted in Phoenix, Arizona, USA found that while the effect of waste heat was negligible during the day it was substantial at night. Up to 1 °C of ambient temperature increase was observed at night, which may lead to a negative feedback loop between energy consumption and the heat island effect [7,8].

In 1984, a catastrophic impact of refrigerants was discovered—an ozone hole over Antarctica, and was attributed to the released chemicals (i.e., CFCs and HCFCs). As a result, the Vienna Convention was held in 1985 and was followed by the adoption of the Montreal Protocol in 1987, which mandated a total phase-out of HCFCs and CFCs [9]. HFCs were used as replacements for the banned refrigerants but as mentioned above they have a high effect on global warming. The world also started to observe that the earth's average temperature has been increasing since the pre-industrial era due to human activities that involved high emissions of GHGs [10], especially in the last 4 decades. Consequently, in 1997, the Kyoto Protocol was adopted to limit and reduce GHG emissions, especially in industrialized countries. There is a continuation of the United Nations Framework Convention on Climate Change (UNFCCC) meetings since the Kyoto Protocol with the goal of preventing harmful human activities from interfering with the climate system, including the Paris Agreement in 2015, which aims to limit global warming to well below 2 °C compared to pre-industrial levels. In fact, the earth's temperature was recorded in 2016 as the highest on record and was estimated to be ~1 °C warmer than the pre-industrial level [11]. In 2016, an important update to the Montreal Protocol was made in the Kigali Amendment and ratified by ~170 countries. This amendment aims to phase down the production and consumption of HFCs by more than 80% by 2047, and eventually limit the earth's temperature increase to below 1.5 °C.

Now that we have better scientific evidence on the environmental effects of human activities and the expected population growth, several measures can be taken to protect the environment. Further reductions in CO_2 emissions can be achieved by using alternative low global warming potential (GWP) refrigerants, energy-efficient systems, and more clean energy sources. Various alternative low-GWP refrigerants were found as suitable replacements but the search is still ongoing for high ambient temperature (HAT) countries. Furthermore, air conditioning manufacturers are improving the systems through several component changes (i.e., fans, compressors, etc.). Higher energy efficiency can also be promoted by governments in several ways, such as by setting minimum efficiency requirements and giving incentives to those who adopt them. For example, the Department of Energy (DOE) in the USA has imposed a minimum energy efficiency standard on appliances while the USA Environmental Protection Agency (EPA) developed the ENERGY STAR program to provide a label on efficient appliances in the market [12,13]. According to the IEA, a reduction of ~40% in energy demand can be achieved, in 2040, by following efficient cooling strategies [1].

1.2. High Ambient Temperature Regions

The high ambient temperature (HAT) exemption of the Kigali Amendment included 34 countries where suitable alternative refrigerants did not appear to exist [14]. Fifteen of those countries are recognized by the United Nations (UN) as least developed countries (LDC), contributing ~2.5% of the world population [15]. According to the UN, LDCs are at high risk of any disaster (i.e., economic, natural, epidemic) and have more than 75% of their population living in poverty [15]. Therefore, they need special consideration when implementing changes related to global warming. Gulf Cooperation Council (GCC) countries are also included in the exemption due to extreme temperatures in the summer.

The ANSI/ASHRAE standard 169-2021 [16] classified thermal climate zones based on Heating or Cooling Degree Days (HDD or CDD). The use of only average temperatures to classify thermal climate zones, however, can be misleading by averaging out an important factor—temperature variations. The CDD Equation (1), shown below, measures how hot the temperature is during a period of days relative to a reference temperature [17]:

$$\text{CDD} = \sum \left[\frac{T_{High} - T_{Low}}{2} - T_{Reference} \right] \quad (1)$$

Figure 1 shows the Cooling Degree Days (CDD) and average high temperature of the hottest month in a number of HAT countries [18,19]. The included HAT countries were selected based on location, average high temperature, and economic condition. Since all exempted HAT countries lie in the Northern Hemisphere, 12 countries from various regions of Africa (i.e., Western, Central, Eastern, and Northern) and 7 countries from Western Asia (i.e., the Arabian Peninsula) were selected. Omitted HAT countries are not expected to show significantly different results given that their average high temperatures are within the range of the included HAT countries. Furthermore, Phoenix, Arizona, USA was included in the analysis since it has considerably high summer temperatures, yet is not included in the exemption of the Kigali Amendment. It must be noted that there is more than one climate zone for large countries, which can give different CDD and temperature values. Moreover, CDD or average high temperature does not show the humidity levels in which some coastal countries may have mildly hot temperatures but extremely high humidity values. However, in this study, we are more concerned with the ambient temperature and its effect on the air conditioning system.

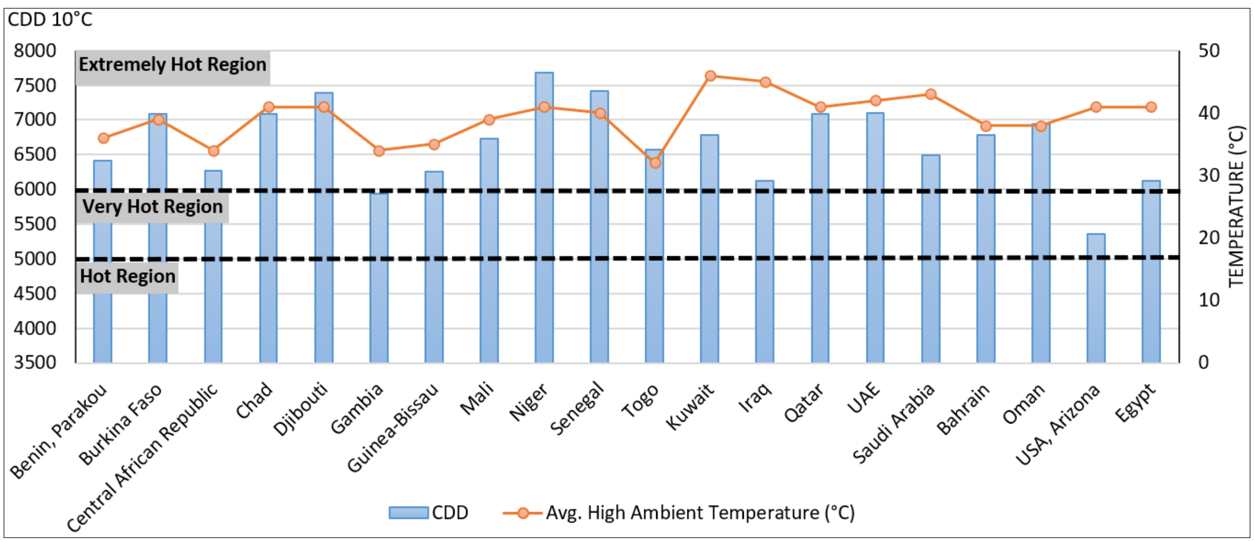

Figure 1. Average high ambient temperature for the hottest month of the year [18] and CDD°10 [19] for a number of high ambient temperature countries and regions.

ASHRAE classification of climate zones is shown in Table 1, where Climate Zone 0, or Extremely Hot, was introduced in 2013 [20]. Despite the fact that approximately 10% of USA counties have shifted to a warmer climate zone, no location falls under the Extremely Hot zone [20,21]. "For the HAT exemption to apply, a country must have an average of at least two months per year over ten consecutive years with a peak monthly average temperature above 35 °C", according to the exemption criteria of the Montreal Protocol [14]. Table 2 shows that almost all HAT countries defined by the Montreal Protocol fall under the Extremely Hot zone, while Phoenix, Arizona, USA lies in the Very Hot zone. The average high temperatures for the hottest month of the year range from 32 °C to 46 °C for Togo and Kuwait, respectively. In comparison, the average high temperature for Phoenix, Arizona, USA is relatively high, but the CDD is the lowest, which implies that the summer season can reach very high temperatures, yet the yearly values are tolerable. On the other hand, Togo has the lowest average high temperature but a relatively high CDD value due to moderate summer temperatures and almost constant ambient temperature throughout the year, ranging approximately from 23 to 32 °C [18].

Table 1. Thermal climate zone definitions from ASHRAE STANDARD 169-2021 [16.]

Thermal Zone	Name	SI Units
0	Extremely hot	$6000 <$ CDD10 °C
1	Very hot	$5000 <$ CDD10 °C ≤ 6000
2	Hot	$3500 <$ CDD10 °C ≤ 5000
3	Warm	CDD10 °C ≤ 3500 and HDD18.3 °C ≤ 2000
4	Mixed	CDD10 °C < 3500 and $2000 <$ HDD18.3 °C ≤ 3000
5	Cool	CDD10 °C ≤ 3500 and $3000 <$ HDD18.3°C ≤ 4000
6	Cold	$4000 <$ HDD18.3 °C ≤ 5000
7	Very cold	$5000 <$ HDD18.3 °C ≤ 7000
8	Subarctic/arctic	$7000 <$ HDD18.3 °C

From the previous examples of Togo and Phoenix, Arizona, USA it was clear that applying either the ASHRAE classification or the HAT definition of the Montreal Protocol may result in regions with mild temperatures, such as Togo, or omit regions with high ambients such as Phoenix, Arizona, USA. Furthermore, since there was no temperature cut-off for HAT countries in the previously mentioned methods, it is essential to define a temperature threshold for the purpose of this study. Based on the mean of average high temperatures of Extremely Hot countries exempted by the Montreal Protocol, in Table 2, high ambient temperature (HAT) conditions are defined here to be 40 °C and above, for the subsequent sections.

Table 2. Average high temperature for the hottest month of the year [18] and CDD°10 [19]

Country	Station [1]	CDD°10 [1]	Average High Ambient Temperature (°C) [2]	Climate Zone
Benin	Parakou, BJ (2.61 E,9.36 N)	6412	36	Extremely Hot
Burkina Faso	Ouagadougou, BF (1.51 W,12.35 N)	7081	39	Extremely Hot
Central African Republic	Bangui, CF (18.52 E,4.40 N)	6270	34	Extremely Hot
Chad	Ndjamena, TD (15.03 E,12.13 N)	7086	41	Extremely Hot
Djibouti	Camp Lemonier, DJ (43.15 E,11.55 N)	7387	41	Extremely Hot
Gambia	Banjul/Yundum, GM (16.63 W,13.20 N)	5933	34	Very Hot
Guinea-Bissau	Bissau Aeroport, GW (15.65 W,11.89 N)	6258	35	Extremely Hot
Mali	Bamako/Senou, ML (7.95 W,12.53 N)	6730	39	Extremely Hot
Niger	Niamey-Aero, NE (2.17 E,13.48 N)	7675	41	Extremely Hot
Senegal	Tambacounda, SN (13.68 W,13.77 N)	7415	40	Extremely Hot
Togo	Lome, TG (1.25 E,6.17 N)	6574	32	Extremely Hot
Kuwait	Kuwait International Airport, KW (47.98 E,29.22 N)	6784	46	Extremely Hot
Iraq	Al Najaf International Airport, IQ (44.40 E,31.99 N)	6124	45	Extremely Hot
Qatar	Doha International Airport, QA (51.57 E,25.26 N)	7087	41	Extremely Hot
UAE	Abu Dhabi International Airport, AE (54.65 E,24.43 N)	7100	42	Extremely Hot
Saudi Arabia	King Khaled International Airport, SA (46.72 E,24.93 N)	6492	43	Extremely Hot
Bahrain	Bahrain International Airport, BH (50.63 E,26.27 N)	6784	38	Extremely Hot
Oman	Seeb International Airport, OM (58.28 E,23.59 N)	6946	38	Extremely Hot
USA, Arizona	Phoenix Sky Harbor International Airport, AZ, US (112.01 W,33.43 N)	5360	41	Very Hot
Egypt	Luxor, EG (32.71 E,25.67 N)	6118	41	Extremely Hot

[1] Obtained from [19] over the period of 1 Decemebr 2020 to 31 November 2021. [2] Obtained from [18] over the period of 2014 to 2021.

1.3. Challenges at High Ambient Temperatures

1.3.1. Energy Consumption and AC Performance

The energy efficiency and cooling capacity of air conditioners degrade as outdoor temperature increases. In most studies mentioned in this article, the degradation becomes substantial at high ambient temperature (HAT) conditions (i.e., 40 °C and above). The energy efficiency of an air conditioner can be described by its coefficient of performance (*COP*), which equals cooling capacity divided by energy consumption. Hence, the decrease in *COP* is observed with not only lower cooling capacity but even higher energy consumption. The ideal *COP*, $COP_{carnot} = T_L/(T_H - T_L)\ - in\ Kelvin$, can be used to explore the effect of outdoor temperatures on a system's performance. For example, when the indoor temperature is held constant at 18 °C, while the outdoor temperature is increased from 35 °C to 55 °C, the degradation of ideal *COP* is ~54%. In addition, the performance of an air conditioner can be greatly affected by refrigerant type, therefore, alternatives not only must have low GWP but show similar performance to the replaced refrigerants. As a result, an exemption of 34 developing countries was given to delay the phase-down of HFCs due to the apparent absence of suitable alternatives under HAT conditions [14].

An increase in energy consumption and demand during elevated ambient temperatures imposes high loads on electricity providers to satisfy the peak load demand. For example, household air conditioners in Kuwait account for 67% of residential electricity consumption and 72% of residential peak demand [22]. The ubiquitous usage of air conditioners poses heavy loads on electricity grids and increases the risk of power outages. This may even require investments in new systems. Consequently, fossil fuel power stations will release even higher amounts of GHGs which, in turn, increase global warming. Lundgren-Kownacki et al. [8] expect the increase in cooling demands to be more focused on fast-growing dense cities in tropical and subtropical regions. Moreover, as the global warming effect is increasing, hot regions are expected to have higher temperatures for longer periods and even more frequent heat waves [23]. The USA EPA stated that if the USA climate warms by 1 °C, an increase in electricity demand for cooling is expected to be 5 to 20% higher. This may strain the electricity grid and lead to power outages [24]. Since there appears to be no viable alternative refrigerant available for HAT countries, they are expected to move from HCFC to HFC refrigerants which have significant Global Warming Potential (GWP) values. As discussed in the previous section, all HAT countries are considered developing countries in Article 5 of the Montreal Protocol, and 15 countries are recognized as least developed countries (LDC) [15,25]. It is clear that the majority of those countries do not have the cutting-edge technology or advanced research and development (R & D) capability to handle this issue. Therefore, the support of developed countries is much needed to overcome this obstacle and transition to efficient air conditioning using low-GWP refrigerants.

1.3.2. Socio-Economic Impacts

The overall cost of an air conditioner includes the initial purchase price, maintenance costs, and energy cost. Hence, all of which must be considered in addition to the system efficiency, when comparing multiple brands. A higher efficiency system is usually more expensive upfront but yields lower energy costs and lower rates of degradation [26]. Furthermore, some low-GWP refrigerants are significantly expensive, such as R-1234yf, which reached prices up to 10 times higher than the baseline R-134a [27]. Owing to the Kigali Amendment, the phase down of HFCs may increase their prices due to lower global production, especially since Non-Article *5* countries (i.e., developed nations) are scheduled to reduce 70% of HFC production and consumption by 2024, whereas Article 5 countries (i.e., developing nations) are scheduled to reduce only 10% until 2035 [28].

The increase in energy consumption and demand during peak hours can lead to extending power plants or even building new ones, to keep up with the demand. As a result, energy prices can be affected and hence consumers may pay part of this cost, especially in the summer [29]. Looking at the total accrued expenses of an air conditioner, richer segments of society are more likely to afford the costs than poorer segments causing thermal inequities and inequalities of heat mortality [8]. This can be even more difficult for less developed countries (LDC) where they have less access to electricity and may end up living in uncomfortable conditions.

1.3.3. Human Health Impact

The human body has a core temperature of 37 °C, on average, and needs a comfortable surrounding environment to avoid health problems related to temperature. Exposure to high ambient temperatures for prolonged periods can cause the body to overheat leading to heat stroke, which can affect organs (i.e., brain, heart, kidney, etc.) and may lead to death [30]. Therefore, the International Organization for Standardization (ISO) introduced the Wet Bulb Globe Temperature (WBGT) index to assess the heat stress on individuals who are working in hot environments [31]. The WBGT index considers four environmental factors that can contribute to human heat stress: air temperature, radiant temperature, air speed, and humidity [32].

Sanderson et al. [33] performed an extensive study of mortality due to high ambient temperatures and found that cases are expected to increase under global warming. Studies have also shown that the human body can acclimatize to hot climates and thus reduce the risk of heat exposure. However, it is not known to what extent [34]. A study conducted in 16 countries has shown the temperature of increased mortality risk varies in different regions and was found to be higher in hot regions [35]. Moreover, Thompson et al. found that higher ambient temperatures increased suicide risk and recommended that mental health problems should be included in health response plans for high ambient temperatures [36].

1.3.4. Possible Mitigations

The challenges of high ambient temperatures are complex and there is no single solution to overcome those concerns. There are several solutions that can help in reducing the impact of high ambient temperatures which should be implemented by individuals and governments. Awareness campaigns to educate people about the current climate change dilemma and how to improve their energy consumption behavior should be a top priority for regulatory authorities. Additionally, average individuals must understand the benefit of transferring to a more efficient system (i.e., lower energy bills, longer life span, better performance, less environmental impacts, etc.). Governments must implement effective strategies for supporting consumers and encouraging them to use efficient systems. Those strategies include subsidizing efficient air conditioners, subsidizing new low-GWP refrigerants, setting minimum thermostat temperatures during peak hours, and imposing minimum energy standards. Successful implementation of such solutions will save energy for individuals and utilities as well as reduce CO_2 emissions.

2. Research Methodology

The purpose of this article is to evaluate previous studies related to residential air conditioning systems under high ambient temperatures and provide an overview of what has been carried out in this area so far. Initially, a minimum temperature criterion was developed, as mentioned in Section 1.2, to evaluate studies that focused only on hot ambient temperatures. A systematic search was developed in a multidisciplinary manner to include all related topics including engineering, industrial design, health, environment, and renewable energy. The database search was conducted mainly using the "ASU Library One Search" engine, which at first assessed any air conditioning system operating under high ambient temperatures. After careful review of each article, only those meeting the 40 °C and above criterion were included. Lastly, the goal was changed to focus only on high ambient temperatures for residential air conditioning systems. It is noted that throughout the review process, new topics were discovered to be related to the purpose of this study, and hence further research was conducted.

3. High Ambient Temperature Organizations and Projects

As mentioned in Section 1.3.1 the performance of an air conditioner degrades substantially at higher ambient temperatures. Hence, a major concern is how low global warming (GWP) alternative refrigerants perform under high ambient temperatures (HAT). To tackle this problem, the US Department of Energy, in cooperation with Oak Ridge National Laboratory, initiated an alternative refrigerant evaluation program specifically under HAT [37,38]. The program goal was to study the performance of low-GWP alternatives and compare them with R-410 and R-22, as baselines. International experts from various areas guided the program to ensure successful implementation. Moreover, the United Nations Environment Programme (UNEP) and United Nations Industrial Development Organization (UNIDO) sponsored two more programs with the same objective. The first was Promoting Low-GWP Alternative Refrigerants in the Air Conditioning Industry for High Ambient Conditions (PRAHA), while the second was the Egyptian Program for Promoting Low-GWP Refrigerants (EGYPRA) [39,40]. Lastly, the Air Conditioning, Heating, and Refrigeration Institute (AHRI) conducted the Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP), through AHRI participants [41]. The main findings of the ORNL and AHRI Low-GWP AREP programs are summarized in Section 7.

3.1. Oak Ridge National Laboratory (ORNL)

The ORNL high ambient temperature program aimed to find low-GWP alternatives for CFCs and HCFCs in mini-split and rooftop air conditioners [37,38]. The program evaluated alternative low-GWP refrigerants in ~136 tests using soft-optimized mini-split air conditioners, provided by Carrier, and drop-in rooftop units, provided by SKM and Petra, under ambient temperatures up to 55 °C. Each test (i.e., drop-in and soft-optimized) included units designed for R-22 and R-410A as baselines. The alternative refrigerant selection was guided by an expert panel that included subject experts from different countries and UNEP and UNIDO personnel. Program results at high ambient temperatures (HAT) are shown in Section 7.

It must be observed that each testing program had different procedures, units, testing conditions, and levels of testing (i.e., drop-in, soft-optimized, purpose-built). Therefore, the results are not meant for direct comparison but, altogether, the findings can be used to understand the overall behavior of each alternative refrigerant. More comprehensive insights can be obtained by including additional factors such as metering device adjustments, charge quantity, and compressor speed to avoid any misinterpretation when comparing different systems [42].

3.2. AHRI Low-GWP Alternative Refrigerants Evaluation Program (AREP)

In response to the global warming effect of HFCs and other refrigerants, AHRI launched an industry-wide program (Low-GWP AREP) to assess new refrigerants and accelerate the transition to low-GWP refrigerants [41]. The evaluation of alternative refrigerants was performed in different applications including air conditioners, ice makers, and chillers. The program included a major category of tests under high ambient temperatures (HAT). Those reports include drop-in and soft-optimized testing with wide capacities of R-410A residential systems. AHRI required participating companies to conduct the tests at their facility using their own equipment, except for measuring the heat transfer coefficient. Nevertheless, AHRI played an important role in coordinating companies to avoid any duplicative work. All test reports were published and can be found on the AHRI website. Program results at high ambient temperatures (HAT) are shown in Section 7.

3.3. Promoting Low-GWP Refrigerants for Air-Conditioning Sectors in High-Ambient Temperature Countries (PRAHA)

PRAHA was a project that aimed to support the assessment of alternative refrigerants for air conditioners in high ambient temperature (HAT) countries [40,43,44]. The project was implemented by UNEP and UNIDO in two phases: PRAHA-I and PRAHA-II. PRAHA had seven local participating manufacturers from Saudi Arabia, Bahrain, Kuwait and the United Arab Emirates, and six international technology providers. PRAHA-I offered collaborative work in the regional air conditioning industry which involved building prototypes and testing alternative refrigerants and identifying minimum energy requirements for new systems. In addition, the project was able to coordinate with various component manufacturers to design compressors that are able to work efficiently with alternative refrigerants under HAT. The main findings of PRAHA-I are: (i) viable alternative refrigerants do exist at HAT but their optimization required some design modifications, and (ii) risk assessment is needed in order to safely use flammable refrigerants in HAT countries.

The second phase, PRAHA-II, focused on three elements: capacity building, design optimization, and risk assessment. The first element was able to create a platform that facilitated cooperation and exchange of knowledge among governments, research institutes, and industry associations. As a result of this awareness, the local industry started testing potential alternative refrigerants to boost the selection process. The second element focused on assessing and optimizing the prototype units. The design assessment involved some modeling work to evaluate the system performance with different components. It was highlighted that a major performance degradation occurred at ambients higher than 46 °C, and hence units were tested at this temperature. The aim of the last element was to provide a risk assessment of using flammable alternative refrigerants in HAT countries. An example of a risk assessment model was provided to test its applicability throughout the life cycle of the equipment. Furthermore, PRAHA recommended that HAT countries expand the assessment by incorporating several actual factors (e.g., local practices, cultural aspects). This can achieve a better model of tailored risk assessment, especially since the criteria for acceptable tolerances may differ among different countries.

3.4. Egyptian Program for Promoting Low-GWP Refriferants (EGYPRA)

EGYPRA is a project implemented by UNEP and UNIDO and initiated by the Arab Republic of Egypt [39]. The program aimed to test 19 purpose-built prototypes with dedicated compressors from several manufacturers, and 16 base units using eight alternative refrigerants under high ambient temperatures (HAT), while R-410A and R-22 are baselines. The systems were split and central units were tested at OEM labs at four different indoor/outdoor temperatures according to the Egyptian Organization for Standardization and Quality (EOS). It must be noted that testing conditions were different than ORNL and AHRI-AREP and hence the refrigerants had different behaviors. Higher cooling capacities were obtained at T_{high} than at T_3 outdoor temperatures (i.e., $T_{high} = 50\ °C\ and\ T_3 = 46\ °C$). This was attributed to the difference between indoor and outdoor temperatures, which affects the system efficiency, as explained in Section 1.3.1.

4. Residential Vapor Compression Systems at High Ambient Temperatures

Table 3 summarizes the studies on air conditioner improvements under HAT conditions. Further details on each study are provided below.

Table 3. Summary of studies related to air conditioner improvements at HAT conditions.

Authors	Objective	Results
Hajidavalloo and Eghtedari [45]	Analyzed the improvement of direct evaporative cooling (DEC) on a 1.5-ton split-unit condenser	Achieved higher *COP*, higher cooling capacity, lower compression ratio, lower electric current consumption, and higher mass flow rate
T. Wang et al. [46]	Investigated the enhancements of direct evaporative cooling (DEC) applied to the condenser inlet air using R-410A	Lower compressor work, higher *COP*, and sub-cooling improvements
Shen and Bansal [47]	Studied the performance improvement for window air conditioners when using submerged sub-cooler and/or slinger	Submerged sub-cooler achieved up to ~5% higher *COP* while using both sub-cooler and slinger achieved up to ~7% higher *COP*
Eidan et al. [48]	Investigated the effect of direct evaporative cooling (DEC) on the condenser inlet air in dry–hot climates	Achieved higher cooling capacity and higher *COP*
Bahman and Groll [49]	Experimentally evaluated an interleaved evaporator circuitry using a 17.6 kW_{th} environmental control unit (ECU) with R-407C	Achieved higher *COP* and higher cooling capacity
Al-Bakri and Ricco [50]	Investigated the heat transfer performance of a horizontal microchannel condenser (i.e., local heat transfer coefficient) using R-410A and a specifically designed test facility	The condensation HTC increases with (i) higher mass flow rate per unit area, (ii) higher vapor quality, (iii) smaller hydraulic diameter, and (iv) lower ambient temperature
López-Belchí [51]	Assessed the performance of R-134a, R-513A, and R-1234yf using a mini-channel condenser at condensing temperatures of 40 °C, 50 °C, and 60 °C	The pressure drop is greatest at higher quality, mass flow rate, and ambient temperature
		The HTCs are highest at high quality and mass flow rate but lower ambient temperatures
Ketwong et al. [52]	Studied three factors affecting air temperature when implementing DEC: FW temperature, mass ratio of water to air, and air wet-bulb temperature (WBT) for both humid and dry climates.	In hot–dry climates lower water-to-air mass ratio is recommended to achieve higher *COP* improvements while in hot–humid climates, a higher water-to-air mass ratio is recommended
Yang et al. [53]	Investigated an atomization cooling element (ACE) that uses condensate water to reduce the ambient air before entering the condenser	Achieved higher cooling capacity, lower power consumption decreased, and higher *COP*
Ding et al. [54]	Introduced a modified air source heat pump (ASHP) system that aims to provide additional sub-cooling and compressor injection, at medium pressure, by the means of an auxiliary circuit	Improved cooling capacity, increased power consumption, and lowered discharge temperatures.
Kang et al. [55]	Investigated the effect of liquid injection technique using an accumulator heat exchanger (AHX)	At constant valve opening, there was a substantial discharge pressure increase beyond the 15% injection ratio.
		At constant mass flow rate, the cooling capacity and *COP* decreased as the injection ratio increased
X. Wang et al. [56]	Investigated two vapor-injection methods in an R-410A 11 kW_{th} residential heat pump	The cooling capacity increased by 14 and 15%, and *COP* increased by 4 and 2%, for IHXC and FTC, respectively
Bahman et al. [57]	Investigated the effect of two technologies on compressors: liquid flooded with regeneration and vapor injection with economizing	The vapor injection method, at HAT, had higher *COP* improvements for all refrigerants except R-1234yf and provided lower discharge temperatures for all refrigerants except R-32, compared to oil flooding
		The vapor injection method showed the best performance when using R-410A while oil flooding had better performance using R-1234yf
Ribeiro [58]	Investigated the feasibility of using a novel refrigerating circuit to cool the outer shell of a compressor in a compact vapor compression unit used at telecommunication stations	The compressor shell temperature was always within 10 °C from the condensing temperature
		The cooling loop allowed the unit to work reliably under a higher temperature range
Bahman et al. [59]	Assessed the improvement of retrofitting economized vapor injection (EVI) in a 17.6 kW_{th} environmental control unit (ECU)	The EVI system improved the cooling capacity and *COP* for both superheated injections by 12.7 and 3.1%, and saturated injections by 11.8 and 1.3%, respectively.

		The discharge temperature was lower by 5 and 1.7 °C for saturated and superheated injections, respectively.
J. Wu et al. [60]	Investigated the reliability of an R-290 rotary compressor under various conditions using mineral oil and synthetic oil (i.e., PAG)	Higher ambient temperatures decreased viscosities but did not have a major effect on solubility
		Higher suction temperatures increased viscosities and decreased solubility
		Higher ambients decreased bearing film thickness
C. Wang et al. [61]	Investigated the variations of dynamic pressure and oil viscosity of an R-290 rotary compressor using a 2.82 kW_{th} room air conditioner	At surface roughnesses of 0.2 and 0.6 μm, the peak contact forces were 3 and 174 N, and the minimum oil film thicknesses were 0.44 and 0.6 μm, respectively
Motta et al. [62]	Simulated the performance of a vapor compression system with R-22 and four alternatives: R-134a, R-290, R-410A and R-407C	Fluids with low critical temperature exhibited a large reduction in cooling capacity, while the compressor power increase was unaffected
		Adding a liquid-line/suction-line heat exchanger improved the *COP* for all refrigerants
Payne et al. [63]	Compared the performance of R-22 and R-410A in a split air conditioning system	The performance degradation was higher for R-410A as outdoor temperature increased
		At 54.4 °C outdoor temperature, R-410A exhibited lower cooling capacity and *COP* compared to R-22
Devotta et al. [64]	Assessed the performance of R-407C in a 5.28 kW_{th} window air conditioning unit designed for R-22	R-407C had lower cooling capacity, lower *COP*, and higher power consumption, compared to R-22
		Discharge pressure of R-407C was higher than R-22
Devotta et al. [65]	Assessed the performance of R-290 as a drop-in using a 5.13 kW_{th} window air conditioner designed for R-22	R-290 had lower cooling capacity, higher *COP*, and lower power consumption, compared to R-22
		The discharge pressure of R-290 was lower than R-22
Westphalen [66]	Investigated the possibility of using alternative drop-in refrigerants instead of R-407C in an environmental control unit (ECU)	R-1270 had higher cooling capacity, and higher *COP* compared to R-407C
		The ECU size could be reduced by 50 mm (i.e., condenser height) and still maintain the baseline refrigerant performance
Y. Wu et al. [67]	Investigated the feasibility of R-161 in a residential air conditioner	R-161 exhibited lower cooling capacity, higher COP, and lower discharge temperature compared to R-22
Barve and Cremaschi [68]	Compared the drop-in performance of R-32 and R-1234yf using a 17.6 kW_{th} split heat pump with R-410A in residential applications	R-32 had higher cooling capacity, similar *COP*, and 20 to 30 °C higher discharge temperature compared to R-410A
		R-1234yf had 50% lower cooling capacity, higher COP, and lower discharge temperature compared to R-410A
Biswas and Cremaschi [69]	Assessed the characteristics of new low-GWP refrigerants DR-4 and DR-5 as a drop-in, using a 17.6 kW_{th} split heat pump with R-410A in residential applications	DR-5 achieved higher cooling capacity and higher *COP* compared to R-410A, while DR-4 exhibited lower cooling capacity and higher *COP* compared to R-410A
		Optimization increased the cooling capacity of DR-4 by 5 to 8% and *COP* by 2 to 6%, with respect to values from the drop-in test
J.H. Wu et al. [70]	Investigated the performance of R-290 and R-1270 in a 2.4 kW_{th} wall room air conditioner designed for R-22	R-290 had lower cooling capacity and higher *COP* compared to R-22, while R-1270 achieved higher cooling capacity and higher *COP*, compared to R-22
		After retrofitting the larger displacement compressor, the cooling capacity increased by approximately 9 and 15% while *COP* decreased by about 9 and 2%, for R-290 and R-1270, respectively
Joudi and Al-Amir [71]	Compared the performance of R-22 and three alternatives: R-410A, R-407C, and R-290, in 3.52 and 7.03 kW_{th} residential split air conditioners	R-290 system had the smallest optimum charge, power consumption, condensing temperature, pressure ratio, and highest *COP*
		Results showed that R-290 had the smallest value of TEWI

Sethi et al. [72]	Evaluated alternative low-GWP refrigerant of R-22 in a 6.2 kW_{th} mini-split air conditioner	The cooling capacity and COP for R-444B were within 2% of R-22, while R-407C had 2 to 3% lower cooling capacity and 4 to 7% lower *COP* compared to R-22
		R-444B had the lowest direct environmental impact due to its low GWP, low charge, and energy efficiency
Abdelaziz et al. [37]	Evaluated alternative low-GWP refrigerants of both R-410A and R-22, using soft optimized 5.25 kW_{th} mini-split air conditioners	R-22 alternatives: A2L had slightly higher discharge temperatures, cooling capacity within 5%, and *COP* within ~10%, while A3 had lower discharge temperatures, ~8% higher *COP* but within 10% lower cooling capacity
		All R-410A alternatives were A2L and showed promising results as alternatives at high ambients
Abdelaziz et al. [38]	Evaluated drop-in alternative low-GWP refrigerants for R-410A and R-22, using 27.2 and 38.7 kW_{th} roof top units	L41z(R-447B) and ARM-71a achieved higher *COP* and higher cooling capacity, compared to R-410A
		ARM-20a achieved about 1% higher *COP* and L-20A(R-444B) achieved about 2% higher cooling capacity, compared to R-22
Taira et al. [73]	Investigated the performance of low-GWP HFO-mix refrigerant R-32/R-125/R-1234yf (67/7/26) and R-32 using a 7.1 kW mini-split air conditioner	At the same cooling capacity, the HFO mix had higher compressor speed, higher power input, and lower *COP* compared to R-32.
		The discharge temperature of R-32 was higher by 3.5 to 5.5 °C than the HFO mix
Oruç et al. [74]	Evaluated R-22 drop-in alternatives, R-422A, R-422D, R-417A, and R-424A using a 2.05 kW_{th} split air conditioner with a rotary compressor and capillary tube expansion device	R-22 had the highest cooling capacity, lowest compression ratio, and highest *COP*, while R-424A had comparable performance
		R-424A discharge temperature, cooling capacity, and *COP* were lower than the baseline by 19 °C, 20%, and 2.5%, respectively

4.1. Evaporators and Condensers

4.1.1. Introduction

Condensers are heat exchangers responsible for rejecting heat from the vapor compression system and are placed downstream from the compressor (i.e., operating at high pressures). In residential air conditioning systems, condensers are usually air-cooled and have aluminum fins for heat transfer enhancements (i.e., due to larger surface area). In a typical condenser, the refrigerant enters as a superheated vapor and leaves as a sub-cooled liquid, which can be factored into three stages: de-superheating, condensation, and sub-cooling. During the two-phase condensation stage, refrigerant velocity decreases due to the density increase when the refrigerant state changes from vapor into liquid [75]. The sub-cooled liquid at the exit sustains the metering device performance and helps increase the cooling capacity by allowing larger amounts of liquid to enter the evaporator (i.e., if there is no sub-cooling, there is a chance of flashing due to liquid-line pressure drop). For the condenser to reject heat, its temperature must be sufficiently higher than ambient air and any increase in ambient temperature requires higher condensing temperatures. With that being said, high ambient temperatures can bear higher loads on the compressors, lower the *COP* significantly, and increase the risk of compressor shutdown [46,48].

Microchannel heat exchangers are increasingly being used due to their lighter weight, smaller size, higher contact surface-area-to-volume ratio, reduced refrigerant charge, and providing similar heat transfer compared to conventional designs [50,51,76]. Their effectiveness is increased through internal and external fins and they can be used as condensers and evaporators [76]. The channels in micro and mini channel heat exchangers have hydraulic diameters less than 3 mm [77]. Fluid flow characteristics and heat transfer can be different when using microchannels instead of conventional size channels [78]. Al-Bakri and Ricco [50] found that condensation at near-critical pressure resulted in unique heat transfer behavior and could not be predicted by the literature studies. López-Belchí [51] attributed the increase in the heat transfer coefficient (HTC) of mini-channels to the flow pattern that reduces the liquid film leading to lower liquid resistance between the core gas and tube wall. In addition, he also mentioned the small hydraulic diameter effect on internal shear stress which can lead to high frictional pressure drop.

Many researchers evaluated the improvement resulting from pre-cooling air before entering the condenser using direct evaporative cooling (DEC). This cost-effective method showed promising enhancements in *COP* and cooling capacity, especially in hot–dry climates [45,46,52]. A common configuration is with a cooling pad placed at the inlet of the condenser, causing additional pressure drop, and injecting feedwater (FW) at the top of the media pad. Another study used a disk type atomization cooling element (ACE) to spray water droplets radially into air entering the condenser (i.e., this system required modifications that are explained in the literature section) with the use of condensate water from the evaporator, which can be questioned in desert climates (i.e., very low humidity).

On the other hand, evaporators are heat exchangers responsible for absorbing heat into the vapor compression system and placed upstream of the compressor (i.e., operating at low pressures). When the low-temperature two-phase refrigerant enters the evaporator, heat transfers from the return air and boils off the refrigerant into a vapor state. The refrigerant is usually superheated before entering the compressor to avoid liquid compression. Unlike condensers, evaporators deal with external sensible and latent heat transfer (i.e., from the air side) and they must be designed properly based on the climate and building demand. The evaporator coil must be sufficiently lower than the dew-point temperature (DPT) of the return air to make sure condensation is taking place, and evaporator temperature fluctuations can adversely affect the supply air condition. During refrigerant boiling, the kinetic energy is increased leading to higher pressure drops along the coil which reduce the evaporator saturation temperature. Uniform fluid distributions (i.e., refrigerant side and air side) are critical to achieving the desired cooling capacity without degrading system performance. Refrigerant maldistribution in parallel flow channels can be caused by (i) coils of different lengths or diameters, and (ii) uneven distribution of liquid–vapor at channel inlets, while air maldistribution can be affected by (i) evaporator geometry, (ii) fans, and (iii) dirty coils [79].

Several studies were conducted to assess the effect of air maldistribution and found a reduction in heat exchanger performance by up to 30% [80]. Another study by Choi et al. [81] found maximum cooling capacity degradations of 8.7% and 30% for air and refrigerant maldistribution, respectively. It must be noted that the effects of non-uniform distributions may lead to different conclusions based on the combinations of different factors (i.e., either exacerbate or outbalance the effects) [79,80,82]. Bahman and Groll [49] discussed passive and active controls (i.e., active control regulates the mass flow rate to control the exit superheat while passive control controls the exit superheat through design modification) to improve the evaporator performance under HAT. The passive control (i.e., interleaved circuitry) was preferred due to its reliability and lower implementation cost, and attractive improvements for both *COP* and cooling capacity at HAT. Studies related to condenser or evaporator improvements at HAT are discussed in Section 4.1.2 and summarized in Table 3.

4.1.2. The Literature

Hajidavalloo and Eghtedari [45] experimentally analyzed the improvement of direct evaporative cooling (DEC) on a 5.28 kW_{th} (1.5 RT) split-unit condenser under high ambient temperatures of 35 °C, 44 °C, and 49 °C. Experiments were conducted in two runs, with and without DEC, at each ambient temperature. Results showed significant improvements at ambient temperatures of 35 to 49 °C with 31.7 to 50.6% higher *COP*, 16.4 to 20.1% higher cooling capacity, 13 to 17% lower compression ratio, 11.6 to 20.3% lower electric current consumption, and 9.7 to 6% higher mass flow rate. It must be noted that greater improvements were observed at higher ambient temperatures.

T. Wang et al. [46] experimentally investigated the enhancements of direct evaporative cooling (DEC) applied to the condenser inlet air of a purpose-built system (i.e., similar to residential AC) using R-410A. The experiment was conducted in two runs with and without DEC under various outdoor temperatures including 44.5 °C. The evaporator and condenser air velocities were set to 1.6 m/s and 1.14 m/s, respectively, while the evaporative cooling pad was made from porous cellulosic paper. Experimental results showed improvements in sub-cooling, reduced compressor work, and increased *COP* by 18%, at 44.5 °C. Furthermore, the authors conducted a cost analysis based on USA water and electricity prices and showed that DEC is economically viable at high ambients.

Shen and Bansal [47] experimentally analyzed the performance of a window air conditioner (WAC) using a modified heat pump design model (HPDM) and evaluated the effectiveness of both a submerged sub-cooler and slinger (i.e., a ring placed around the condenser fan blades to collect and spray water in the ambient air entering the condenser). The unit had a cooling capacity of 2.93 kW_{th} (0.8 RT) with a single-speed rotary compressor and fin-tube evaporator and condenser. Moreover, the fin-tube sub-cooler was submerged in a water container, which used condensed water from the evaporator coil, placed downstream to provide additional sub-cooling. Results showed *COP* improvements when using a submerged sub-cooler of up to ~5% while using both sub-cooler and slinger achieved up to ~7%, over the range of ambient temperatures ~32.2 to 43.3 °C.

Eidan et al. [48] experimentally investigated the effect of direct evaporative cooling (DEC) on the condenser inlet air of a purpose-built system with a capacity of 7.0 kW_{th} (2.0 RT) under dry–hot climates (i.e., 10% RH = relative humidity, and 45 °C, 50 °C, 55 °C DBT = dry-bulb temperature). Three air velocities were imposed across the evaporative pads (1 m/s, 2 m/s, and 3 m/s) in which the highest velocity showed the highest pressure drop yielding the lowest evaporative cooling efficiency. Results showed significant improvements when applying evaporative cooling for all runs: lower compression ratio, higher refrigerant mass flow rate, higher cooling capacity by ~27 to 33%, and higher *COP* by ~17 to 33%.

Bahman and Groll [49] experimentally evaluated an interleaved evaporator circuitry using a 17.6 kW_{th} (5.0 RT) environmental control unit (ECU) with R-407C under outdoor temperatures up to 51.7 °C. Local air velocities were measured at the face of the evaporator to determine the maldistribution of the airflow. Based on the percentage of airflow at different locations, the arrangement of the interleaved circuitry was determined (i.e., refrigerant exiting a circuit with high airflow was redirected to a circuit with low airflow and so on) which yielded an additional refrigerant pressure drop of ~20 kPa compared to an unmodified evaporator. Results found that refrigerant superheat distribution was uniform across the interleaved evaporator circuitry (i.e., differences ranged from 1 °C to 5 °C). Moreover, the *COP* improved by 5.9 to 7.7% while cooling capacity improved by 8.4 to 10.6%, at outdoor temperatures of 40.6 to 51.7 °C.

Al-Bakri and Ricco [50] experimentally investigated the heat transfer performance of a horizontal microchannel condenser (i.e., local heat transfer coefficient) using R-410A and a specifically designed test facility. Ambient temperatures were 35 and 45 °C while condensing pressures were 70 and 80% of refrigerant critical pressure (i.e., near-critical). It was found that the condensation HTC increases with (i) higher mass flow rate per unit area, (ii) higher vapor quality, (iii) smaller hydraulic diameter, and (iv) lower ambient temperature. Results were validated using literature correlations and found unsatisfactory discrepancies which are attributed to the high operating conditions (i.e., ambient temperature and condensing pressure). The authors recommended further experimental analysis using R-410A at critical conditions to have a better understanding of heat transfer in microchannel condensers.

López-Belchí [51] experimentally and numerically assessed the performance of R-134a, R-513A, and R-1234yf using a mini-channel condenser at condensing temperatures of 40 °C, 50 °C, and 60 °C. The effect of variable flow rate, saturation pressure, vapor quality, and tube geometry was evaluated by studying both (i) local HTC, and (ii) frictional pressure drop, for the three refrigerants. Experimental results showed that the pressure drop is greatest at higher quality, mass flow rate, and ambient temperature. The HTCs are highest at high quality and mass flow rate but lower ambient temperatures. After validating the experimental data with model predictions from the literature, it was found that R-134a had the best thermal performance. Total equivalent warming impact (TEWI) analysis found R-513 applicable for very limited conditions while R-134a is better in most cases.

Ketwong et al. [52] theoretically studied the performance of direct evaporative cooling (DEC) for dry and humid conditions under ambient temperatures up to 40 °C. The feedwater (FW) temperature was varied in three scenarios: (i) less than air inlet wet-bulb temperature (WBT), (ii) equals air inlet WBT, and (iii) higher than air inlet WBT. Moreover, the mass ratio (MR: water mass flow rate to air mass flow rate) varied from 0.2 to 2.0. Numerical results showed that DEC was more effective in lowering ambient air temperatures, in hot–dry climates than in hot–humid climates. At 40 °C ambient temperature and 30 °C FW temperature, hot–dry conditions (30% RH) required lower MR to reduce ambient air temperature since the FW temperature was higher than air inlet WBT, while hot–humid conditions (70% RH) required higher MR to reduce the ambient air temperature since the FW temperature was lower than the air inlet WBT. Furthermore, simulation has shown that the lower the FW temperature the higher the improvement that could be reached for both climates.

Yang et al. [53] experimentally investigated a method that uses condensate water to reduce the ambient air before entering the condenser, under ambient temperatures, up to 43 °C. A 2.65 kW_{th} (0.75 RT) split unit, with R-22, was modified by changing the fan orientation (i.e., blowing towards the condenser instead of sucking air) and installing an atomization cooling element (ACE) disk on the fan. The effect of installing the ACE and changing the fan orientation was explored. Results at 43 °C showed that when using condensed water with an ACE disk, cooling capacity increased by 8.1%, power consumption decreased by 9.5%, and *COP* increased by 20%. The only concern with this method is when the condensate water is insufficient (i.e., in dry climates), the performance is expected to degrade and become lower than the baseline unit.

4.2. Compressors

4.2.1. Introduction

Compressors (i.e., positive displacement) are mechanical devices responsible for raising the pressure of the refrigerant by decreasing its volume and circulating the refrigerant in the cycle from high to low pressure. The low pressure-temperature refrigerant undergoes a compression process exiting as superheated with a high pressure-temperature state. It is imperative that the inlet refrigerant state is vapor otherwise the compressor (sometimes called a vapor pump) can be damaged since liquids are incompressible. There are several methods in real air conditioning systems that can provide proper superheating upstream of the compressor including an electronic expansion valve (EEV), thermostatic expansion valve (TXV), and an accumulator. The compression ratio (CR), absolute discharge pressure divided by absolute suction pressure, is key in identifying the compressor load. When the load on a compressor increases from higher ambient temperature or lower indoor temperature, the CR increases accordingly, indicating higher power consumption and lower performance [54,56,83]. Compressors in residential air conditioners are usually cooled by the refrigerant itself, hence proper mass flow rate and suction temperature must be maintained to provide sufficient cooling. Another important variable is the lubricant viscosity, which changes according to different operational parameters. The oil type must be compatible with the refrigerant and should maintain acceptable viscosities under various conditions. Yokozeki [84] discussed the behavior of viscosity and solubility of refrigerant-oil mixtures and the importance of choosing suitable oil viscosity and degree of solubility to avoid lowering system performance. When the ambient temperature increases, the discharge temperature and pressure will also increase so that the condenser is at a sufficiently higher temperature than the ambient, and hence heat can be rejected from the system. As a result, the CR will increase and the unit performance will decrease (i.e., lower cooling capacity, higher energy consumption, and lower *COP*). Therefore, compressors play a critical role in determining the cycle behavior and must be given special attention when developing improvements needed for HAT. Residential air conditioning compressors currently used are positive displacement types (i.e., reciprocating, rotary, and scroll).

Several methods of improving the compressor performance were initially developed for extreme low ambient temperatures and were found to be applicable for high ambient temperature applications in which cooling capacity can be greatly enhanced [57,85]. Two techniques were investigated for improving compressor performance: (i) liquid flooded compression (i.e., using oil), and (ii) refrigerant vapor injection. The latter was found to be effective in enhancing cooling capacity and increasing *COP* in certain conditions. It can be implemented by injection at intermediate pressure using an economizer: flash tank (i.e., injecting saturated vapor or liquid–vapor mixture), or an internal heat exchanger (i.e., injecting superheated vapor) [56,57,59]. On the other hand, (i) oil-flooded compression, and (ii) liquid refrigerant injection techniques, at the accumulator inlet, were evaluated to identify possible improvements in preventing high discharge temperature and compressor overheating [55,57]. In addition, exploiting a heat exchanger between the compressor inlet and condenser outlet (i.e., regenerator) prevents wet-compression, increases the sub-cooling, and hence improves the cooling capacity as well [55,57]. Many improvements were evaluated in the literature for compressors in residential air conditioners but for the purpose of this study, only feasible studies at HAT are discussed in Section 4.2.2 and summarized in Table 3.

4.2.2. The Literature

Ding et al. [54] introduced a modified air source heat pump (ASHP) system that aims to provide additional sub-cooling and compressor injection, at medium pressure, by the means of an auxiliary circuit (i.e., economizer heat exchanger, thermostatic expansion valve TXV, and solenoid valve). The auxiliary circuit starts from the condenser outlet to the compressor inlet, which can be used by activating the solenoid valve based on ambient temperatures. The condensing temperature reached 60 °C while the evaporating temperature was maintained at 2 °C. Results showed that using the auxiliary circuit improved cooling capacity, increased power consumption, and lowered the discharge temperatures. It must be noted that the compressor power increase was more significant at higher ambients (i.e., higher condensing temperatures), while the increase in cooling capacity was at the expense of a lower *COP*. Despite the lower *COP*, the supplementary circuit increased the operating range and the compressor reliability at HAT.

Kang et al. [55] investigated the effect of the liquid injection technique using an accumulator heat exchanger (AHX) in a 9 kW_{th} (2.6 RT) vapor compression cycle at 43 °C ambient temperature. The first test, without liquid injection, was conducted by varying the mass flow rate and measuring the superheat and sub-cooling effects. It was observed that the sub-cooling and superheat effects decreased with higher mass flow rate and higher evaporating pressure. The second test, with liquid injection, was conducted by varying the liquid injection ratio at two operating conditions: (i) constant valve opening, and (ii) constant flow rate. At condition (i) there was a substantial increase in the discharge pressure beyond the 15% injection ratio, while at condition (ii) the cooling capacity and *COP* decreased as the injection ratio increased. Therefore, the authors recommended optimizing the system performance by using higher flow rates at lower injections while limiting the flow rates at high injection ratios to ensure discharge pressure values are within an acceptable range.

X. Wang et al. [56] experimentally investigated two vapor injection methods in an R-410A 11 kW_{th} (3.13 RT) residential heat pump at 46.1 °C outdoor DBT where the baseline compressor was replaced with a vapor-injected scroll compressor. The two methods were the flash tank cycle (FTC) and the internal heat exchanger cycle (IHXC). The injection ratio for both methods varied to reach optimum performance, in which IHXC allowed for a wider range of ratios than FTC due to the use of a thermostatic expansion device. Moreover, the second-stage expansion valve used in the FTC was replaced with a larger one, rated for 18 kW_{th} (5.1 RT), and showed an improved performance. It was found that the IHXC and FTC had comparable performance improvement at 46.1 °C compared to the baseline, in which cooling capacity increased by up to ~14% and *COP* increased by up to ~4%.

Bahman et al. [57] numerically investigated the effect of two technologies on compressors: (i) liquid flooded with regeneration using a polyolester (POE) oil as a flooding agent, and (ii) saturated vapor injection with a flash tank economizer. A parametric study was conducted for the four refrigerants, R-410A (baseline), R-290 (propane), R-32, and R-1234yf, using the two technologies at different ambient temperatures of 25 to 55 °C. At HAT (i.e., 40 °C and above), the vapor injection method achieved higher *COP* improvements for all refrigerants except R-1234yf, compared to oil flooding. Moreover, vapor injection provided lower discharge temperatures for all refrigerants except R-32, compared to oil flooding. It must be noted that R-32 had extreme discharge temperatures of ~110 °C and hence, special attention must be paid when used at high ambients. Finally, the vapor injection method showed the best performance when using R-410A, while oil flooding had better performance using R-1234yf.

Ribeiro [58] experimentally investigated the feasibility of using a novel refrigerating circuit to cool the outer shell of a compressor in a compact vapor compression unit used at telecommunication stations, at ambient DBT up to 55 °C. The experiment was conducted using a linear compressor with a maximum operating temperature of 85 °C due to glued parts. It was found that the compressor shell temperature was always within 10 °C from the condensing temperature, in which at 55 °C ambient the condensing temperature was ~70 °C and the shell was ~79 °C. The cooling capacity was observed to increase at higher ambient temperatures due to the greater effect of increased evaporator temperature than increased condenser temperature.

Bahman et al. [59] experimentally assessed the improvement of retrofitting economized vapor injection (EVI) in a 17.6 kW_{th} (5.0 RT) environmental control unit (ECU) designed for military applications in extreme weather. The unit used R-407C as a working fluid and was tested under ambient temperatures up to 51.7 °C. A plate heat exchanger (PHX) economizer with EEV was used instead of a flash tank, due to the easier control of injected mass vapor flow rate. Experimental results showed that superheated and saturated vapor had better performance than the baseline case. Additionally, at HAT, the EVI system improved cooling capacity and *COP* for superheated injection by up to 12.7 and 3.1%, while for saturated injection, improvements were up to ~11.8 and 1.3%, respectively. On the other hand, the discharge temperature was lower by 5 and 1.7 °C for saturated and superheated injections, respectively.

J. Wu et al. [60] investigated the reliability of an R-290 rotary compressor under various conditions including ambients of 35 °C, 46 °C, and 55 °C, using mineral oil (MO) and synthetic oil (i.e., PAG = polyalkylene glycol). The performance of the R-290 rotary compressor, using PAG oil, was compared with different refrigerants—R-32, R-22, and R-410A—at similar cooling capacities (i.e., using different stroke volume compressors). Under various testing conditions, the following was observed: (i) higher ambient temperatures decreased viscosities but did not have a major effect on solubility, (ii) higher suction temperatures increased viscosities and decreased solubility, and (iii) higher ambients decreased bearing film thickness. Therefore, from tests (ii) and (iii), higher suction superheat can be used to compensate for the lower viscosity values (i.e., mineral oil but not synthetic oil) and hence avoid a too low oil film thickness that may lead to metallic contact.

C. Wang et al. [61] experimentally investigated the variations of dynamic pressure and oil viscosity of an R-290 rotary compressor using a 2.82 kW_{th} (0.8 RT) room air conditioner under ambient temperatures up to 50 °C. Results found that as outdoor temperature increased, the compression process took longer to reach higher discharge pressures.

Moreover, the load on the crank part at 50 °C was nearly double the load at 30 °C. As the ambient temperature increased, there was a decrease in cooling capacity, *COP*, and mineral oil viscosity. Numerical simulations, at a 50 °C outdoor temperature, showed that at a surface roughnesses of 0.2 and 0.6 μm, the peak contact forces were 3 and 174 N, and the minimum oil film thicknesses were 0.44 and 0.6 μm, respectively. Hence, controlling the oil sump viscosity or using higher viscosity oil should be considered to maintain acceptable oil film thickness at high ambients.

4.3. Refrigerants

4.3.1. Introduction

Refrigerants play a substantial role in determining the performance of vapor compression cycles. They are used as working fluids to transfer heat between indoor and outdoor environments essentially by exploiting their latent heat characteristic. The most common refrigerants groups discussed in this section include chlorofluorocarbons (CFCs), hydrochlorofluorocarbons (HCFCs), hydrofluorocarbons (HFCs), hydrofluoroolefins (HFOs), and hydrocarbons (HCs). The history of refrigerants has evolved in four generations, which is well explained by Calm [86]. In 1985, the Vienna Convention was held with the objective of protecting the environment from harmful effects of ozone layer depletion [87], while in 1987 the Montreal Protocol was held with the aim of controlling and ultimately eliminating ozone-depleting substances (ODS) which created an ozone hole over Antarctica [86,88]. The Montreal Protocol is considered the most successful environment protection agreement, ratified by 198 countries, which provided a different ODS phase-out timetable for developed and developing countries [89]. In developed countries, CFCs and HCFCs were totally phased out by 1996 and 2020, respectively [90]. In developing countries, CFCs were phased out in 2010 while HCFCs are delayed until 2030 [90]. As a result, HFCs were introduced as a third-generation refrigerant due to their zero ozone depletion potential (ODP), relatively low flammability, and suitable thermophysical properties [91].

In 1997, the Kyoto Protocol was adopted with the goal to limit and reduce greenhouse gases (GHG), in which HFCs were designated as one of them [92]. In fact, HFCs are considered strong GHG and their impact on global warming can be 100 to 1000 times larger than CO_2 per unit mass [93]. This raised a concern about their impacts as they become increasingly used as alternatives to ODS in air conditioning and refrigeration applications [94]. Researchers suggested that HFC emissions are projected to reach 9 to 19% of global CO_2 emissions by 2050 if no mitigation measures are taken [95]. Therefore, the Kigali Amendment (2016) was introduced with the goal of protecting the ozone layer and the climate and was agreed upon by more than 170 countries [96]. The amendment, which came into force in 2019, aims to phase down the consumption of HFCs by more than 80% over the next 30 years [96]. Consequently, fourth-generation refrigerants are meant to replace the high-GWP HFCs with low-GWP alternatives (i.e., HFO/HFC blends, HFOs, HCs) [9,97]. Concerns were expressed about whether new alternatives are suitable under high ambient temperature (HAT) conditions, and an exemption was given to HAT countries which allowed for a delay in HFC reduction [14,98]. Since developed countries, such as the USA, have already phased out HCFCs, they are transitioning from high to low-GWP refrigerants [99]. On the other hand, HAT organizations are putting efforts into helping HAT countries bypass the high-GWP refrigerants and instead transition from HCFCs directly to low-GWP refrigerants [97].

Total equivalent warming impact (TEWI) is a measure of a refrigerant's impact on global warming, which takes into account both direct, due to leakages, and indirect emissions, from electricity consumption. It is worth mentioning that direct emissions represent about 2.9% of global GHG while indirect emissions represent about 4.9% of global GHG [100]. Hence, it is imperative for alternative refrigerants to also demonstrate adequate energy efficiency and cooling capacity, especially under HAT conditions. Furthermore, the performance of air conditioners (i.e., *COP* and cooling capacity) degrades substantially at high ambient temperatures. This adds complexity for HAT countries where low GWP might not be sufficient to reduce TEWI if the energy performance of the system is significantly penalized [68].

There are several organizations that have evaluated air conditioning systems (i.e., ORNL, PRAHA, AHRI, and EGYPRA) at various testing conditions under HAT, as described in Section 3. Depending on the system type and alternative refrigerant, different testing levels can be conducted, as defined in detail by AHRI low-GWP AREP. A drop-in test is the simplest type that only allows for minor changes: (i) charge optimization, (ii) expansion valve adjustment, and (iii) compressor speed adjustment [101]. A soft-optimized test allows additional modifications including (i) compressor displacement and/or motor size, (ii) use of variable speed compressor motor, (iii) flow control, (iv) lubricant, (v) size of tubing, and (vi) ratio of heat transfer area of condenser and evaporator, at constant total area [101]. Fully optimized types of equipment are those built specifically for certain refrigerants which is a more complex and time-consuming process. Despite the potential of higher efficiency levels for the fully optimized method, most tests conducted by organizations and companies are either drop-in or soft-optimized. Abdelaziz et al. [37] suggested that depending on measurement uncertainties, the performance is expected to improve slightly with further soft-optimization and substantially with additional engineering work. The engineering work can be any change to the unit beyond soft-optimization and drop-in modifications such as compressor cooling technologies and increasing the total area of heat exchangers (i.e., evaporator and condenser).

Refrigerants are classified in the ANSI/ASHRAE Standard 34-2019 according to two hazards involved, toxicity and flammability [102]. Lower toxicity is denoted by class A while higher toxicity is denoted by class B. Flammability is divided into four groups: high flammability denoted by class 3, low flammability denoted by class 2, mildly flammable denoted by subclass 2L, and non-flammable denoted by class 1. For example, an A2L refrigerant is non-toxic and mildly flammable (i.e., low burning velocity). An alternative refrigerant selection must consider several factors including environmental impact, thermo-physical properties, chemical properties, and safety. Despite there being no upper limit for GWP when nominating refrigerants, the candidate must show a significant reduction in GWP relative to the baseline [101]. Moreover, the performance of the alternative low-GWP refrigerant must be evaluated to limit indirect emissions. Desirable thermophysical properties include high enthalpy of vaporization, high thermal conductivity, high critical temperature, and low vapor and liquid viscosities [71]. Low-GWP refrigerants include pure HCs, inorganics, HFOs, and HFO/HFC blends; blends can be either zeotropic or azeotropic. Many alternative refrigerants that have attractive (i.e., low) GWP and ODP values are categorized as flammable refrigerants, hence adding another challenge related to personal safety and risks associated with using flammable refrigerants [103]. Mildly flammable refrigerants (A2L) have been used in Japan for about a decade in residential air conditioners [104]. Furthermore, the EPA has approved the use of low-GWP hydrocarbon refrigerants, subject to use conditions, in various air conditioning and refrigeration applications (e.g., vending machines, room air conditioning units) [99]. Studies carried out for alternative refrigerants in residential air conditioning systems under high ambient temperature conditions are discussed in Section 4.3.2 and summarized in Table 3.

4.3.2. The Literature

Motta et al. [62] simulated the performance of a vapor compression system with R-22 (HCFC-type) and four alternatives: R-134a (HFC-type), R-290 (HC-type), R-410A (HFC-type), and R-407C (HFC-type), at an outdoor DBT of 25 to 55 °C. The results showed that R-410A had a higher degradation than other refrigerants, as the outdoor temperature increased. It was concluded that fluids with low critical temperature exhibited a large reduction in cooling capacity, while the compressor power increase was unaffected. The author performed a similar analysis after adding a liquid-line/suction-line heat exchanger to the cycle. This modification improved the *COP* for all refrigerants which varied depending on the different refrigerants' molar heat capacity.

Payne et al. [63] experimentally compared the performance of R-22 and R-410A in a split air conditioning system. The outdoor DBT ranged from 27.8 to 54.4 °C, while the indoor condition remained unchanged. Refrigerants had a comparable performance at low outdoor temperatures but the performance degradation was higher for R-410A as the outdoor temperature increased. The cooling capacity was similar at a 35 °C outdoor temperature but the R-410A *COP* was lower than R-22 by 4%. When the outdoor temperature increased to 54.4 °C, the cooling capacity and *COP* of R-410A were lower than R-22 by approximately 9 and 15%, respectively. The authors used identical evaporators and condensers, similar design compressors, and different lubricants. Furthermore, the authors conducted a higher outdoor temperature test at 68.3 °C using a customized compressor. R-410A reached a supercritical condition at the condenser inlet without noticeable changes in noise level or system operation.

Devotta et al. [64] experimentally assessed the performance of R-407C in a 5.28 kW_{th} (1.5 RT) window air conditioning unit designed for R-22. Outdoor DBT/WBT varied from 35 °C/30 °C to 46 °C/24 °C, while indoor DBT/WBT ranged from 27 °C/19 °C to 29 °C/19 °C. R-407C had approximately 2 to 8% lower cooling capacity, 8 to 13% lower *COP*, and 6 to 7% higher power consumption, compared to R-22. The discharge pressure of R-407C was higher by 11 to 13% than R-22. In this study they also simulated the finned-tube heat exchangers, using the EVAP-COND model developed by the National Institute of Standards and Technology (NIST), USA [105]. The model accuracy was within $\pm$3% of experimentally measured cooling capacities. It was used to further investigate the performance of heat exchangers for each fluid. Simulation results showed that both refrigerants had the lowest evaporator and condenser capacity at the highest outdoor temperatures. Simulated pressure drops of R-407C in the evaporator and condenser were lower than R-22 by up to ~16 and 41%, respectively. Despite the lower performance of R-407C, using it as a retrofit can extend the R-22 unit's life. It should be noted that retrofitting R-407C involved procedures for changing the unit's oil.

Another similar study was conducted by the same authors [65] to experimentally assess the performance of R-290 (the HC propane) as a drop-in using a 5.13 kW_{th} (1.5 RT) window air conditioner designed for R-22. Identical test conditions were applied. R-290 had an approximately 6 to 10% lower cooling capacity, 3 to 8% higher *COP*, and 12 to 13% lower power consumption, compared to R-22. The discharge pressure of R-290 was lower by ~13 to 18% than R-22. The authors used the same model, which was within $\pm$4% of the experimentally measured cooling capacities, to simulate the performance of heat exchangers. The worst performance of the heat exchangers was observed at the highest outdoor temperature for both fluids. Simulated pressure drops of R-290 in the evaporator and condenser were lower than R-22 by up to ~48%. Moreover, R-290 had up to 22% lower condensing pressure and up to 3% lower evaporating pressure compared to R-22.

Westphalen [66] investigated the possibility of using alternative drop-in refrigerants instead of R-407C in a 17.6 kW_{th} (5.0 RT) environmental control unit (ECU). Different HC refrigerants were investigated based on their characteristics and R-1270 (propylene) was selected for the experimental test. Baseline and alternative refrigerants had similar characteristics except that R-1270 did not exhibit temperature glide. This was a concern for the microchannel heat exchangers design used in the ECU, which did not easily allow optimization for high-glide refrigerants. Experimental findings at a 51.7 °C outdoor DBT showed that R-1270 had a 12% higher cooling capacity and a 10% higher *COP* compared to R-407C. This improvement was partially attributed to the elimination of temperature glide when using R-1270. The author concluded that the condenser height could be reduced by 50 mm and still maintain the baseline refrigerant performance.

Y. Wu et al. [67] investigated the feasibility of R-161 (HFC-type) in a residential air conditioner. Theoretical simulations were conducted to investigate the performance of three refrigerants: R-161, R-22, and R-290 under various conditions. Theoretical results showed that R-161 had a better thermodynamic performance than R-290, a lower cooling capacity than R-22, and the highest *COP*. R-290 had the lowest discharge temperature. Furthermore, experimental analyses were conducted to verify the findings for R-161 and R-22 using a 3.5 kW_{th} (1.0 RT) residential air conditioning system where outdoor DBT ranged from 27 to 48 °C. At 48 °C, R-161 had a lower cooling capacity, higher *COP*, and lower discharge temperature than R-22 by 5.1%, 10.0%, and 3 °C, respectively.

Barve and Cremaschi [68] experimentally compared the drop-in performance of R-32 (HFC-type) and R-1234yf (HFO-type) using an R-410A 17.6 kW_{th} (5.0 RT) split heat pump in residential applications. The unit was tested at various outdoor DBTs including 43 °C and 46 °C. Experimental findings showed that R-32 had about a 10% higher cooling capacity, similar *COP*, and a 20 to 30 °C higher discharge temperature than R-410A. On the other hand, R-1234yf had a 50% lower cooling capacity, higher *COP*, and a lower discharge temperature. The unsatisfactory cooling capacity of R-1234yf, even after optimizing the thermostatic expansion valve (TXV), eliminated the refrigerant as a possible drop-in replacement.

Biswas and Cremaschi [69] experimentally assessed the characteristics of low-GWP refrigerants DR-4 and DR-5 as drop-in replacements, using an R-410A 17.6 kW_{th} (5.0 RT) split heat pump in residential applications. The unit was tested at various outdoor DBTs including 43 °C and 46 °C. The experimental assessment showed that DR-5 had up to 4% higher cooling capacity, up to 7% higher *COP*, and a higher discharge temperature than R-410A. At the same testing conditions, DR-4 showed up to 18% lower cooling capacity, up to 6% higher *COP*, and a lower discharge temperature than R-410A. Optimization of refrigerant charge and TXV increased the cooling capacity of DR-4 by 5 to 8% and *COP* by 2 to 6% with respect to values from the drop-in test, however, there was no significant improvement for DR-5.

J. H. Wu et al. [70] experimentally investigated the performance of R-290 and R-1270 in a 2.4 kW_{th} (0.7 RT) wall room air conditioner designed for R-22 under ambient temperatures up to 40 °C. At 35 °C, the experimental results showed that R-290 had a 5% lower cooling capacity and about a 10% higher *COP* compared to R-22. On the other hand, R-1270 had about a 2% higher cooling capacity and about a 1% higher *COP* compared to R-22. It was observed that, as the outdoor temperature increased, the cooling capacity of R-290 and R-1270 showed higher degradation rates than R-22. Moreover, the study also carried out refrigerant charge distribution tests since the practical charge was always higher than the allowable charge of flammable HC refrigerants. It was found that for both refrigerants, up to 63% of the charge was within the condenser and about 18% in the compressor. Hence, the authors suggested further investigations are needed on how to reduce the charges within the condenser, compressors, and liquid lines.

Joudi and Al-Amir [71] experimentally compared the performance of R-22 and three alternatives: R-410A, R-407C, and R-290, in 3.52 and 7.03 kW_{th} (1.0 and 2.0 RT) residential split air conditioners. Outdoor DBT varied from 35 to 55 °C to replicate the hot arid climate of Iraq. Experimental results showed that the R-290 system had the smallest optimum charge, power consumption, condensing temperature, pressure ratio, and highest *COP*. TEWI analysis was conducted for all refrigerants using the two air conditioning systems at variable outdoor temperatures. Results showed that R-290 had the smallest value of TEWI. Hence, R-290 is a very attractive candidate for R-22 replacement under high ambient temperatures.

Sethi et al. [72] evaluated alternative low-GWP replacement refrigerants for R-22 in a 6.2 kW_{th} (1.8 RT) mini-split air conditioner under high ambient temperatures. Experimental results under ambients up to 52 °C found that R-444B (HFC-type) was 5% more efficient than R-407C across most of the operating range. Life cycle climate performance (LCCP) was carried out to determine the environmental impacts of each refrigerant under summer ambient conditions in Kuwait. The LCCP analysis showed that the indirect impacts of each refrigerant showed significantly higher contributions to global warming than the direct impacts (i.e., ranging from 98.4 to 99.8% of the total contributions). Nevertheless, future electricity generation is likely to be less carbon-intensive, hence, the indirect impacts should be less important and the benefit of using low-GWP refrigerants would be greater.

Abdelaziz et al. [37] conducted an extensive evaluation of alternative low-GWP replacements for both R-410A and R-22, using soft optimized 5.25 kW_{th} (1.5 RT) mini-split air conditioners provided by Carrier. Outdoor DBT varied from 27.8 to 55 °C while indoor DBT/WBT ranged from 26.7 °C/19.4 °C to 29 °C/19 °C. The alternative refrigerant selection was guided by an expert panel, consisting of members of various nations, UNEP, and UNIDO personnel. R-22 alternative refrigerants were N-20B, DR-3, ARM-20B, L-20B(R-444B), R-290, and DR-93, while R-410A alternatives were R-32, DR-55, L41(R-447A), ARM-71A, and HPR-2A. R-22 alternatives showed promising results at high ambients where two of the A2L refrigerants had slightly higher discharge temperatures, cooling capacity within 5%, and efficiency approximately within 10%; A3 refrigerants had lower discharge temperatures, about 8% higher efficiency but within 10% lower cooling capacity. On the other hand, all R-410A alternatives were A2L and showed significant potential as alternatives at high ambients. R-32 had consistently better efficiency and capacity but 12 to 21 °C higher discharge temperatures.

Another similar study was conducted by ORNL [38] to evaluate drop-in alternative low-GWP refrigerants for R-410A and R-22, but this time using two roof top units where the first was 27.2 kW_{th} (7.7 RT) provided by SKM and the second was 38.7 kW_{th} (11.0 RT) provided by Petra. R-22 alternatives were L-20A (R-444B), ARM-20b, DR-7 (R-454A), and ARM-20a, while R-410A alternatives were DR-55, L41z (R-447B), ARM-71a, and R-32. Outdoor and indoor testing conditions were similar to the mini-split experiment mentioned previously. The experimental results found that at high ambient, L41z (R-447B) and ARM-71a had more than 7% higher *COP* and 3% higher cooling capacity compared to R-410A. However, all R-410A alternatives exhibited higher discharge temperatures. For the R-22 alternatives, at high ambients, ARM-20a had 0.8% higher *COP* and L-20A (R-444B) had 1.8% higher cooling capacity compared to R-22. It was noted that all refrigerants including the baseline showed a substantial efficiency degradation at higher ambient temperatures. Moreover, the testing units were designed for R-22 or R-410A, hence alternative refrigerants should not be expected to outperform the baseline refrigerants. Nevertheless, their performance could be expected to improve when manufacturers implement design modifications.

Taira et al. [73] explained the refrigerant market status for residential AC in Japan. In fact, R-410A had been phased out and R-32 was selected as a better alternative. In this study, the authors investigated the performance of low-GWP HFO-mix refrigerant R-32/R-125/R-1234yf (67/7/26) and R-32 using a 7.1 kW_{th} (2.0 RT) mini-split air conditioner under temperatures up to 52 °C. A variable frequency drive compressor was used to evaluate the system performance at different speeds, and electrical expansion was also used to regulate the mass flow rate as needed. It was found that at the same cooling capacity, the HFO mix had a higher compressor speed, lower discharge temperature, 72 W higher power input, and 6.3% lower *COP* compared to R-32. The authors explored the possible reasons for the HFO mix behavior and concluded that R-32 is superior to the HFO mix because of its latent heat characteristic, especially under high ambient conditions.

Oruç et al. [74] experimentally evaluated R-22 drop-in alternatives, R-422A (HFC-type), R-422D (HFC-type), R-417A (HFC-type), and R-424A (HFC-type) using a 2.05 kW_{th} (0.6 RT) split air conditioner with a rotary compressor and capillary tube expansion device. Ambient DBT ranged from 35 to 41 °C, while indoor DBT was set to 18 °C. Results showed that R-22 had the highest cooling capacity, lowest compression ratio, and highest *COP*, while R-424A had comparable performance. Discharge temperatures were highest for R-22 whereas R-424A had the lowest discharge temperatures. Finally, at an ambient of 41 °C, R-424A discharge temperature, cooling capacity, and *COP* were lower than the baseline by 19 °C, 20%, and 2.5%, respectively. Therefore, R-424A was the best alternative refrigerant, however, its GWP is higher than R-22 which makes it unfavorable due to its higher GWP and cost.

5. Practical Design Modifications for High Ambient Temperatures

Air conditioner manufacturers are usually obligated to certify their products before selling them in the markets and the certification requirements vary based on the minimum energy efficiency of local regulations. Policymakers are following this win-win strategy to curtail electrical demand, mitigate environmental impacts, and reduce utility bills for consumers. Davis et al. [106] conducted a study in Mexico City to evaluate the effect of replacing house appliances with efficient models and found higher energy consumption for replaced air conditioners. This was attributed to the consumer behavior that must be considered carefully in the evaluation of energy-efficiency programs. The performance of air conditioners is usually rated in two methods: (i) Energy Efficiency Ratio (*EER*), "the ratio of cooling capacity in Btu/h to the total power in Watts at 95°F/75°F outdoor and 80°F/67°F indoor DBT/WBT", and (ii) Seasonal Energy Efficiency Ratio (*SEER*), "the total heat removed from the conditioned solace during the annual cooling season in Btu divided by the total electrical energy in Watt*hours consumed by the air-conditioner during the same season" [107]. Both *COP* and *EER* represent the ratio of the cooling capacity to the input power, where *EER* has units of Btu h^{-1} W^{-1} and *COP* is unitless. Equation (2) can be used to convert between the two quantities:

$$EER = COP * 3.41 \tag{2}$$

In the USA, there are different minimum efficiency requirements for residential central AC depending on the region (i.e., North, Southeast, and Southwest) that must be met when selling an AC unit [108]. Nowadays, the minimum cooling efficiency of the Southern and Northern regions are 14 *SEER* and 13 *SEER*, and both are going to rise in 2023 to ~15 *SEER* and 14 *SEER*, respectively [109]. It must be noted that an additional minimum *EER* is required for the Southwest region only (i.e., Arizona, California, New Mexico, and Nevada) [110]. On the other hand, Kuwait has a minimum of 9.6 *EER* for direct-expansion (DX) units with and without inverters in 2022, and 10 *EER* starting in 2024 while phasing out DX units without inverters [111]. It must be noted that the previously mentioned *EER* value for Kuwait is measured at 46 °C, not 35 °C. The Air Conditioning, Heating, and Refrigeration Institute (AHRI) provide certification programs for air conditioners to ensure the products perform according to manufacturers' published claims, in which tests are conducted by an approved independent third-party laboratory and administered by AHRI. Those globally recognized certification programs help (i) manufacturers to distinguish their products in the market, sell more products and comply with local regulations, and (ii) customers to confidently compare different units' efficiencies (i.e., that are tested under similar conditions) [112]. In addition to the standard test condition T1 at 35 °C outdoor DBT, AHRI provides a wider range for high ambient countries, such as GCC countries with T3 being at 46 °C and T4-Kuwait being at 48 °C [113,114].

A motivation to know the practical modifications followed by AC manufacturers led to comparing units that are currently available in different markets. Four USA air conditioning companies were compared with three GCC companies to determine the main differences among nineteen air conditioners (i.e., 10 GCC units and 9 US units) working under high and mild ambient temperatures. The units include split, mini-split, packaged, and wall-mount types with a cooling capacity range from 10.55 to 26.38 kW_{th} (3.0 to 7.5 RT). The selection was random but the availability of detailed unit specifications was very limited and hence reduced the number of companies and units. The following observations were made:

1. The average energy efficiency ratio (*EER*) for the USA market is ~6% higher than in the GCC. This can be attributed to the more stringent regulations in the USA.

2. Almost all companies provided the *EER* and cooling capacity at the AHRI T1 condition.
3. A very limited number of AC companies provided face areas of evaporators and condensers.
4. The average condenser face area per ton and indoor air cubic feet per minute (CFM) per ton were inconclusive. This could be due to the use of mini-channel heat exchangers and the implementation of other design modifications.
5. The average condenser CFM per ton values were in the range from ~800 to 1100 CFM/ton.
6. Only GCC companies provided performance tests at 52 °C. This could be due to local regulations. For example, Kuwait requires the unit to work at 52 °C for at least two hours without tripping or overheating [111].
7. Both USA and GCC companies provided allowable operating temperatures of 52 °C while two ducted mini-split units of a GCC company had a 55 °C allowable temperature. In general, USA companies are providing a wide temperature range to meet the Southwest ambient temperatures, however, units are not allowed for installation unless they meet the minimum criteria for both *SEER* and *EER* ratings [115].
8. No GCC company provided the *SEER* value of any unit. This explains the higher importance of using *EER* when rating units operating under high ambient temperatures for long periods [108].

Due to the limited results found from the previous survey, further investigations were conducted to find common manufacturers' design modifications for high ambient temperatures (HAT). The USA regulations have set a minimum *EER* and *SEER* for the hot climate region (the Southwest) and alerted manufacturers to adhere to specific installation requirements. Some of the requirements can be critical in hot ambients such as (i) an outdoor and indoor unit combination must match the minimum certification ratings in the region, (ii) when the outdoor unit is not matching the indoor unit, the details of the indoor coil must be mentioned including face area, fin density, fin and tube materials, (iii) the outdoor temperature that locks out the low capacity operation when using a two-capacity compressor must be mentioned, and (iv) only a 5% variation is allowed in the face area and total fin surface area of the outdoor coil [116–118]. The two-capacity (or two-stage) compressor is a single or group of compressors operating with only two stages of capacity (i.e., a full compressor stage and a low compressor stage), according to AHRI 210/240 [107]. These specific requirements can be very helpful in assuring only efficient units are installed and can also help in future research and development purposes. A study conducted by the Proctor Engineering Group and AMAD explored the possible modifications of air conditioner components at a 46 °C ambient temperature, which are illustrated in Table 4 [119].

Table 4. Possible design modifications of residential air conditioners for high ambient conditions (46 °C) [119].

Component	Modification	Note
Evaporators/Condensers	Increase face area	Space constraint Uniform airflow distribution is critical
	Increase number of tube rows	Higher pressure drop in air-side and refrigerant-side
	Improved refrigerant circuit configuration	Can be predicted via simulations and optimized based on airflow distribution Can raise manufacturing challenges
	Improved fin design	Geometry and higher fin density Higher air-side pressure drop and susceptible to trapping more dust
	Flat tube micro-channel	Higher ratio of surface area to volume Lower air-side pressure drop Issues with moisture drainage if used as evaporator
	Desuperheater	Heat exchanged with ambient air or suction-line refrigerant

Compressor	Customized models	Increased durability under high ambient temperatures Can increase the unit cost significantly
Fans and Motors	Indoor/outdoor fans	Indoor: centrifugal type blowers with a molded Styrofoam housing, cross flow fans Outdoor: propeller type fans In dry climates, running indoor fan after compressor shut-off can provide further air cooling by evaporating coil moisture for limited periods Higher airflow can improve the heat transfer through larger diameter or higher RPM Higher flow rates can increase fan power draw, increase noise, and raise concerns related to condensate drainage
	Electronically Commutated Motor (ECM)	Higher efficiency even at reduced speeds, compared to conventional permanent split capacity (PSC) motors
Controls	Fan cycling	Ability to circulate air using the fan when the compressor is off
	Variable speed controls	Inverter-driven variable speed compressor can lower consumption at lower demands The benefit can be eliminated during periods of high demand

Design modifications are expected to achieve better annual performance when optimized based on the dominant climate conditions. Thus, when a unit is designed for the highest ambient temperatures that rarely occur, the system can be oversized for long periods of the year leading to unnecessary penalties [120]. For example, oversized condensers can reduce the head pressure and compression ratio to some extent, yet the pressure should not be too low to ensure sufficient refrigerant mass flow rate is circulated and hence components work properly. Therefore, the optimized design depends on several factors and there is no specific change followed by manufacturers. Table 5 shows different design features found in units sold in the GCC market.

Table 5. Air conditioner design features found in HAT units sold in the GCC market.

Component	Feature
Compressor	Internal protection from high discharge temperature Overcurrent protection Low pressure and high pressure protection Voltage protection Cooled by refrigerant to limit discharge temperature Two-stage compressor and variable speed Short cycling protection Limiting liquid refrigerant in the compressor for improved durability
Evaporator/Condenser	Copper tubes with copper or aluminum fins Tubes: internally grooved, rifled, and ripple finned Fin: corrugated, lanced, and louvered Integral sub-cooler for condensers
Motor/Fan	Electronically Commutated Motor (ECM) Motor thermal protection Motor protection from water, particles, and solid objects
Housing	Thermal insulation for evaporator section

Under HAT, the condensing temperature can reach high values so that enough heat rejection is taking place (i.e., it must be sufficiently higher than the ambient temperature), which can lead to excessive discharge temperatures and compressor tripping. The condenser size must be suitable to lower the temperature difference (i.e., between the refrigerant and the ambient air) and hence reduce the discharge temperatures. Design modification toward a high efficiency can be achieved through the use of a larger heat exchanger surface area (i.e., condenser and evaporator), which helps in reducing the temperature difference, discharge temperature, and compression ratio, leading to improved compressor performance. In certain situations where both extreme and mild temperatures can occur, the condenser can have variable speed fans which can be regulated based on the ambient temperature. Moreover, at high humidity levels, the evaporator coil should be sufficiently low for dehumidification to take place, but it can be slightly higher in dry climates. The idea is to trade the latent cooling capacity for sensible cooling capacity and hence increase the sensible cooling efficiency [121]. Bhatia [122] explored the heat rejection of air-cooled condenser coils and the factors affecting their performance, including the condenser fan designs. The condenser fans were designed for either (i) draw-through airflow which has uniform air distribution across the coil but passes the hot discharge air over the fan and drive motor, or (ii) blow-through airflow which passes ambient air over the fan and drive motor but has less uniform air distribution across the coil. Noting that different refrigerants yield different behaviors depending on their characteristics, as mentioned in Section 4.3, the modification must also consider the type of refrigerant used. For example, in the AHRI-AREP test [123], evaluating R-32 as an alternative refrigerant for R-410A required a lower compressor speed to match the baseline cooling capacity and when the ambient temperature increased to ~49 °C, its discharge temperature increased significantly to its maximum allowable limit (i.e., 121 °C) and the unit stopped working.

6. Air Conditioner Limitations at High Ambient Temperatures

6.1. Climate

Scorching summer ambient temperatures can reach above 50 °C in the shade in the GCC region [49,72], leading to a limited unit performance in conjunction with higher cooling needs. A comparison between Kuwait and Phoenix, Arizona, USA climates was conducted to investigate the potential unit degradation resulting from high ambient temperatures. The hottest period of the day, for both locations, was found to be from approximately 9 a.m. to 5 p.m. (Figure 2) [124]. Subsequently, annual high ambient temperature (HAT) hours were evaluated for Kuwait and Phoenix (i.e., ASHRAE climate zones 0 and 1, respectively) to analyze the necessity for more efficient air conditioners in such climates. In 2017, Phoenix and Kuwait's annual hours, where ambient temperatures are 40 °C and above, were found to be ~423 and ~1430 h, respectively [124]. They are distributed over 2 months in Phoenix (~7.1 h/day) while over 5 months in Kuwait (~9.5 h/day). Figure 3 shows the average annual temperatures, from 9 a.m. to 5 p.m., for Kuwait and Phoenix. Therefore, it is essential to use efficient air conditioners (i.e., rated and certified for high temperatures) in Kuwait, but for Phoenix, a meticulous evaluation must be conducted to avoid any penalties for oversizing the unit when operating under mild temperatures. Moreover, an evaluation of average temperature increase in both locations, from 2010 to 2021, was conducted to assess the climate change effect on HAT seasons (i.e., June and July for Phoenix, Arizona, USA; May to September for Kuwait) [125]. Figure 4 shows the increased values to always be higher for the Kuwait HAT season, except in 2013 and 2016. The temperature increase in the Kuwait HAT season has always been higher than 1.2 °C over the past 6 years, raising another concern of global warming that is aggravating even the HAT levels of Kuwait.

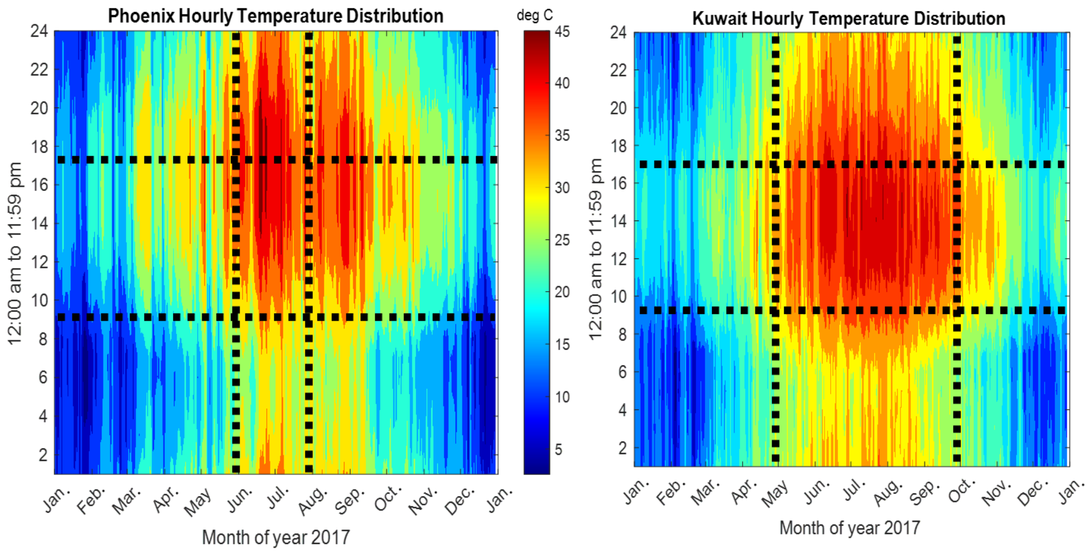

Figure 2. Hourly temperature distribution for Phoenix, Arizona, USA and Kuwait in 2017 [124].

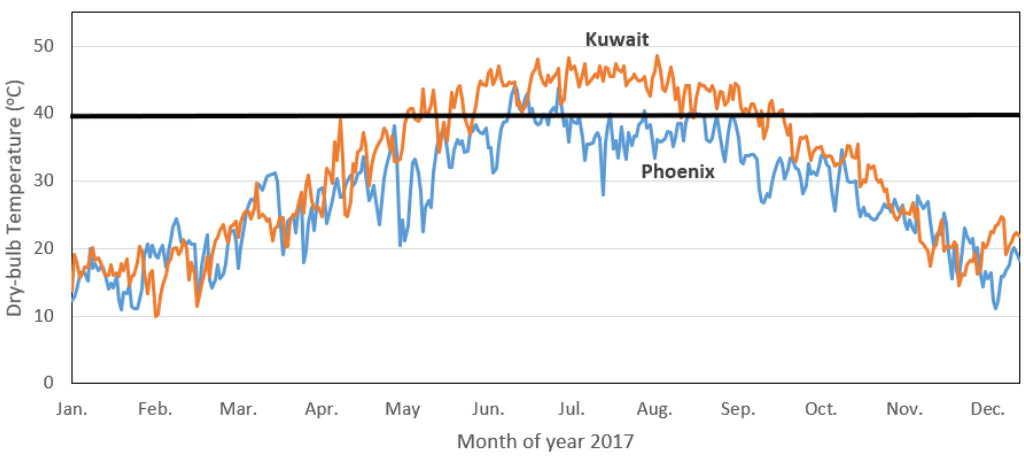

Figure 3. Annual high ambient temperature (HAT), averaged from 9 a.m. to 5 p.m., for Kuwait and Phoenix, Arizona, USA [124].

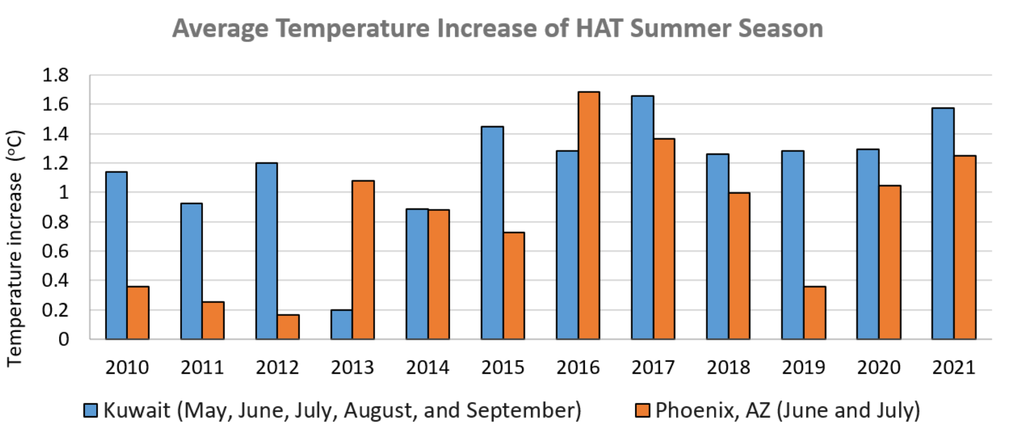

Figure 4. Average temperature increase for Kuwait and Phoenix, Arizona, USA with respect to the average temperatures for the period from 1981 to 2010 [125].

The air conditioning process of residential air conditioners (i.e., split, mini-split, window and packaged) requires both cooling and dehumidification, which makes the indoor coil temperature a very critical factor in providing the proper thermal comfort. The dehumidification process takes place when the indoor return air passes over the evaporator cooling coil at a temperature lower than the air dew-point temperature (DPT), yet it must not be set very low (i.e., below 0 °C) to avoid coil freezing. In hot–humid climates, the dehumidification needs can be as critical as sensible cooling, especially in regions with very high relative humidities. The total air heat content can be attributed to sensible and latent heat, in which latent heat represents the moisture content. Hence, hot–humid climates bear additional cooling loads on the evaporator, which are attributed to increased moisture removal, leading to higher energy consumption. In addition, evaporators impose a very large pressure drop in the vapor compression cycle, and increased load fluctuations can adversely affect the cycle performance. As a result, the evaporating temperature is particularly limited in hot–humid climates than in hot–dry climates.

6.2. Components

Before discussing the performance limitations of an air conditioning unit under HAT, it is essential to understand how the unit efficiency is determined. As we mentioned in Section 5, there are some design modifications followed by manufacturers to improve the unit performance under HAT conditions, and there is no specific single strategy. Engineering judgment is the primary determinant of what modifications are needed and if the unit will reach the desired efficiency considering those changes [126]. The modifications are carefully optimized based on several factors including space, application type, ambient conditions, availability, economic feasibility, reliability, and mandatory regulations. A good design philosophy is to consider the annual peak temperatures but mainly design the system based on the ambient temperatures that occur for the longest periods throughout the year. If a scroll compressor is operating with a higher compression ratio than optimal, the gas is over-compressed leading to higher energy consumption than needed. While if it is operating with a lower compression ratio than optimal, the gas will not reach the desired discharge pressure until it floods back to the scroll pockets and some of it is then compressed twice, leading to higher energy consumption.

An air conditioning system consists of main and auxiliary components in series, where each component has its own limitations (e.g., compressor, condenser, and evaporator). The compressor is the heart of the unit and it is critical to know how it performs based on the selection of the remaining main parts. Proper condenser design at high ambient temperatures is crucial in determining how the compressor will function, where higher rates of heat rejection lead to lower compression ratio and hence lower work. Evaporators usually have high-pressure drops due to the increase in refrigerant velocity as evaporation is taking place leading to unwanted losses and contributing to a higher compression ratio. On the other hand, compressors have several limitations that must not be exceeded to avoid performance degradation or compressor damage. Suction superheat at the compressor suction or return gas temperature is usually specified by compressor manufacturers to be maintained at a minimum value to prevent liquid refrigerant flood back that can cause serious lubrication issues (e.g., oil dilution). Since many HAT compressors are refrigerant cooled, the superheat should not be too high to avoid the compressor overheating. Moreover, compressors also must have a limited discharge temperature to avoid premature compressor failures resulting from excessive wear and oil breakdown.

Favorable refrigerants are to be used especially under high temperatures, as discussed in the refrigerants Section 4.3.1. Another limitation is the use of A2L (mildly flammable) or A3 (highly flammable) refrigerants under high ambients, which can improve the system performance but are subject to additional design requirements to minimize the chance of system failure leading to a refrigerant leak and hence creating a fire hazard. Environmental impacts also limit the options of refrigerants since the new trend is towards zero ODP and extremely low (or zero) GWP.

6.3. System Performance

The compressor's operating envelope represents the allowable simultaneous condensing and evaporating temperatures that are determined based on refrigerant and oil mixtures. For high ambient temperatures it is vital to use compressors with an extended operating range to ensure smooth operation at normal evaporator temperatures, but high condensing temperatures. If the unit is not guaranteed to operate within those limits, compressor protective measures must be added, which are often seen in high ambient temperature AC units (Section 5). To assess the effect of changing evaporating or condensing temperatures on the pressure ratio, the evaporating temperature was varied by 7 °C whereas the condensing temperature was held constant, and vice versa. The saturation temperatures in the evaporator and condenser were assumed to be 10 °C different from the indoor and outdoor temperatures, respectively. Three refrigerants were compared, as shown in Figure 5, and they display the same behavior. The increase in compression ratio from changing the indoor temperature, 26 °C to 19 °C, was higher than from changing the outdoor temperature from 39 °C to 46 °C for all refrigerants (Table 6) [127–129]. Therefore, it is crucial to maintain the evaporator temperature at the design condition during high ambient temperatures to avoid an unwanted rise in the compression ratio leading to higher compressor work [71].

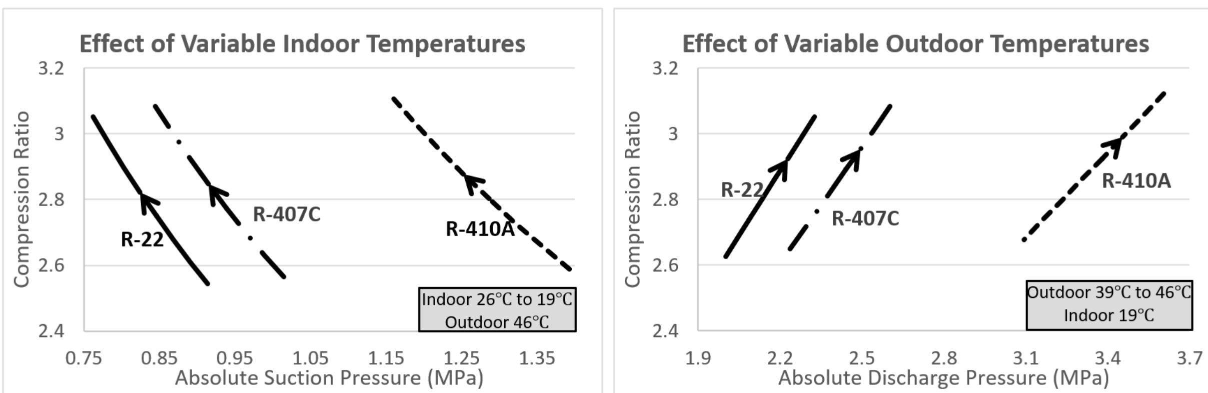

Figure 5. Effects of variable outdoor and indoor temperatures on compression ratio.

Table 6. Effects of variable outdoor and indoor temperatures on the compression ratio (CR).

		R-22	**R-410A**	**R-407C**
Variable Indoor	CR increase %	19.9%	20.5%	20.2%
	Magnitude	0.51	0.53	0.52
Variable Outdoor	CR increase %	16.2%	16.5%	16.3%
	Magnitude	0.43	0.44	0.43

6.4. Irreversibilities

At high ambient temperatures, the need for cooling increases since more heat enters the houses. However, unfortunately, the *COP* and cooling capacity of the unit drops significantly at high temperatures, presenting a limitation in the operating range of the unit, and a need for using more efficient units that can operate under HAT. The Carnot *COP*, mentioned in Section 1.3.1, represents the system's maximum theoretical efficiency according to the first law of thermodynamics and depends only on indoor and outdoor temperatures. Actual *COP*s are always less than ideal and most of the experimental studies showed drastic declines at higher ambients. The reason can be attributed to the irreversibilities of the air conditioner components, especially when operating at high ambient temperatures. Yumruta et al. [130] numerically analyzed how different condensing and evaporating temperatures would affect the exergy losses (i.e., irreversibilities) in a vapor compression system. It was found that at higher temperature differences between the evaporator and indoor air, exergy losses are increased in the evaporator. Similar behavior was observed for the condenser, where higher temperature differences led to higher exergy losses in the condenser. Kalaiselvam and Saravanan [131] experimentally tested scroll compressors and recommended some operating conditions to minimize the compressor irreversibilities: 4 °C evaporating temperature, 35 to 40 °C condensing temperature, within 65 °C discharge temperature, and 14 °C suction temperature. Ahamed et al. [132] reviewed different studies of exergy analysis for vapor compression systems and found that decreasing the condenser temperature or increasing the evaporating temperature improved the system *COP*, whereas the total system irreversibility was decreased. Bahman and Groll [133] experimentally investigated the components' irreversibilities of Environmental Control Units (ECUs), at capacities of 5.28, 10.55, and 17.58 kW_{th} (1.5, 3.0, and 5.0 RT), at a 51.7 °C outdoor DBT. The analysis showed that irreversibility contributions were significant for three system components: the compressor up to 42.5%, the evaporator up to 32.9%, and the condenser up to 22.4%. The 5-ton ECU's evaporator showed the highest irreversibility due to high refrigerant pressure drop (i.e., at the evaporator distributor) and air maldistribution.

6.5. Operation and Maintenance

People's behavior can also limit the performance of air conditioners from operation and maintenance aspects. A study conducted in Florida found that unmaintained air conditioners operating for long periods (i.e., more than ~1500 h/year) are susceptible to increased degradation rates, especially for larger capacity systems. The air conditioning unit degradation is another major AC-limiting factor leading to higher energy consumption with less ability for cooling, where the reasons for continuous degradation can be attributed, but not limited to, coil fouling, filter clogging, and refrigerant charge problems [26].

6.6. Possible Improvements

1. Proper control of indoor DBT during high ambient temperatures can compensate for the performance degradation due to high outdoor DBT. For example, setting an indoor thermostat at moderate temperatures can provide higher cooling capacity and higher *COP* of the unit, which leads to lower energy consumption and cost.
2. Applying optimized design modifications, Section 5, to maintain the overall system performance and achieve stable operation under harsh conditions.
3. Performing routine maintenance as recommended by local manufacturers to sustain the system performance and avoid higher operational costs.
4. Positioning the outdoor unit to face North or East can reduce the amount of direct sunlight as opposed to West or South facing.
5. Proper unit sizing, installation procedures, air distribution, and duct designs also have significant effects but are not included in this study.

7. Results and Discussion

Air conditioners operating under high ambient temperatures (HAT) must overcome high summer loads, high discharge temperatures, and performance degradation (i.e., lower cooling capacity and *COP*). Regulatory jurisdictions have added more requirements such as minimum energy efficiency and environmentally friendly refrigerants. Considering all previous factors, the problem becomes more complex and requires design modifications to comply with them. The climate change motivated the transition to low-GWP (A2L) refrigerants and ongoing efforts are underway to ensure the workability under high ambient conditions. In addition to the research completed by different entities, four main organizations conducted extensive work to investigate different A2L refrigerants under

HAT (Section 3). The following results include two of those organizations (i.e., ORNL and AHRI Low-GWP AREP) and all related research articles found in the literature, where all of them comply with our definition of high ambient temperature, 40 °C and above, as mentioned in Section 3.1. It was found that the definition of the Montreal Protocol for hot countries, Section 3.1, was in compliance with the hottest ASHRAE climate zone. Nevertheless, a more stringent temperature was defined for the purpose of investigating the air conditioner performance and degradation, especially since most studies detected higher performance degradation at temperatures above 40 °C. Furthermore, due to the limited comprehensive resources in such areas, it was decided to focus on electric residential systems to investigate the drawbacks of each individual component.

7.1. Condensers and Evaporators

There were several studies conducted for the improvement of heat exchangers (i.e., evaporators and condensers) under HAT (Figure 6 and Table 7). Condenser improvements in the range from ~18 to 50% for *COP* and 8 to 30% for cooling capacity were achieved through direct evaporative cooling (DEC). Moreover, two studies tested mini-channel condensers at HAT where Al-Bakri and Ricco [50] found unique condensation heat transfer coefficients (HTC) at near-critical pressure and recommended further testing at critical pressures. Nevertheless, both studies [50,51] agreed that local HTC increases at higher quality and mass flow rate, but decreases at lower ambient temperature. On the other hand, only one study was found describing evaporator improvements at HAT [49], which showed improvements of ~7% in *COP* and ~10% in cooling capacity. This method implemented passive controls to improve the refrigerant circuitry based on the external air flow.

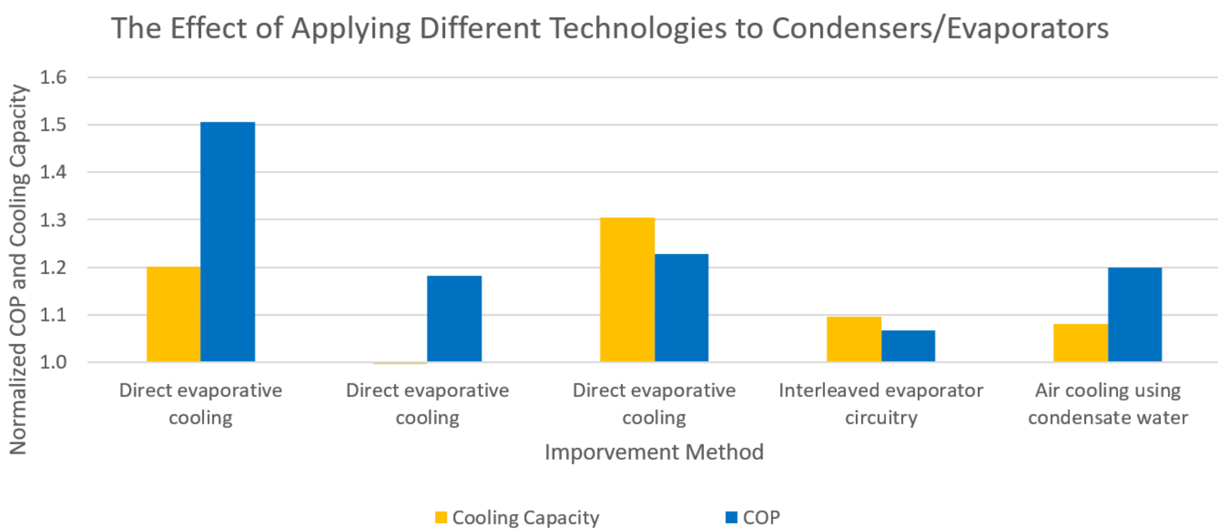

Figure 6. Improvements of different technologies applied to either evaporator or condenser [45,46,48,49,53].

Table 7. Details of condenser or evaporator improvements for each method.

Method	Refrigerant	Size	Test Type	Outdoor Condition	Cooling Capacity	COP	Note	Reference
Direct evaporative cooling	N/A	1.5 ton [5.3 kW_{th}]	Experimental	49.0 °C	+20.1%	+50.6%	-	[45]
Direct evaporative cooling	R-410A	N/A	Experimental	44.5 °C	N/A	+18.2%	-	[46]
Direct evaporative cooling	R-22	2.0 ton [7.0 kW_{th}]	Experimental	50.0 °C	+30.3%	+22.9%	-	[48]

Interleaved evaporator circuitry	R-407C	5.0 ton [17.6 kW_{th}]	Experimental	40.6 to 51.7 °C	+9.5%	+6.8%	At average values for tests 1,2, and 3	[49]
Air cooling using condensade water (ACE)	R-22	0.8 ton [2.8 kW_{th}]	Experimental	43.0 °C	+8.1%	+20.0%	-	[53]

A mini-channel heat exchanger shows attractive characteristics in increasing the surface area and decreasing refrigerant charge, which are essential considerations for HAT air conditioners. The mini-channel heat exchangers could also be tested under HAT with A2L refrigerants (i.e., mildly flammable) since charge limitations are applied for flammable refrigerants. Direct evaporator cooling has shown significant improvements, especially under hot–dry conditions, but water availability is a major concern. Few studies explored the possibility of utilizing condensate water, recognizing that the amount can be insufficient in dry climates, and hence it is not recommended to depend solely upon it. The literature also revealed only a very limited number of studies conducted on evaporator improvements. Bahman and Groll [133] discussed the irreversibility of an environmental control unit (ECU) and showed that evaporator losses can be very significant under HAT. Hence, an active control method, where the refrigerant can be regulated to control the exit superheat, could be investigated under HAT and compared with passive control improvements [49].

7.2. Compressors

Many compressor improvements found at high ambient temperatures were essentially induced from studies related to low ambients. Experimental and numerical improvement techniques found in the literature are (i) vapor injection, saturated or superheated, at the intermediate stage, (ii) liquid flooding, (iii) accumulator heat exchanger (AHX), and (iv) external shell cooling. Moreover, some studies analyzed the compressor reliability and the behavior of lubricants under HAT conditions.

Table 8 shows the experimental vapor injection methods [54–56] achieved up to 3% higher *COP*, 15% higher cooling capacity, and 4.5 °C lower discharge temperature. This method requires installing an additional economizer, which is either an internal heat exchanger or flash tank, where both types showed comparable performance, but the former provides wider mass flow rate control [56,59]. The cooling capacity improvement is attributed to the higher sub-cooling resulting from using an economizer. Additionally, the optimized numerical analysis showed potential for significant improvements using a new compressor (i.e., larger envelope and designed for higher condensing temperature) and properly sized economizer, as shown in Figure 7 [59]. Liquid flooding, using POE oil, was also numerically investigated and demonstrated that only R-1234yf had high improvements in *COP*, yet higher discharge temperatures, and hence needs further experimental testing [57]. Suction-line liquid injection through AHX, Figure 7, showed significant discharge temperature reductions of up to ~17 °C at the cost of lowering the *COP* by ~4.2% when using R-22 [55]. An external shell cooling method, using refrigerant, was conducted experimentally for a compact size compressor and led to attractive improvements [58]. This method is not included in Figure 7 and Table 8 since it was applied to a compact vapor-compression unit and was compared to thermoelectric coolers which yielded about 75% *COP* improvement under HAT. Moreover, it needs further investigation using residential size compressors, since the heat generated inside the compressor can be far from the surface and hence make it more difficult to apply such external cooling.

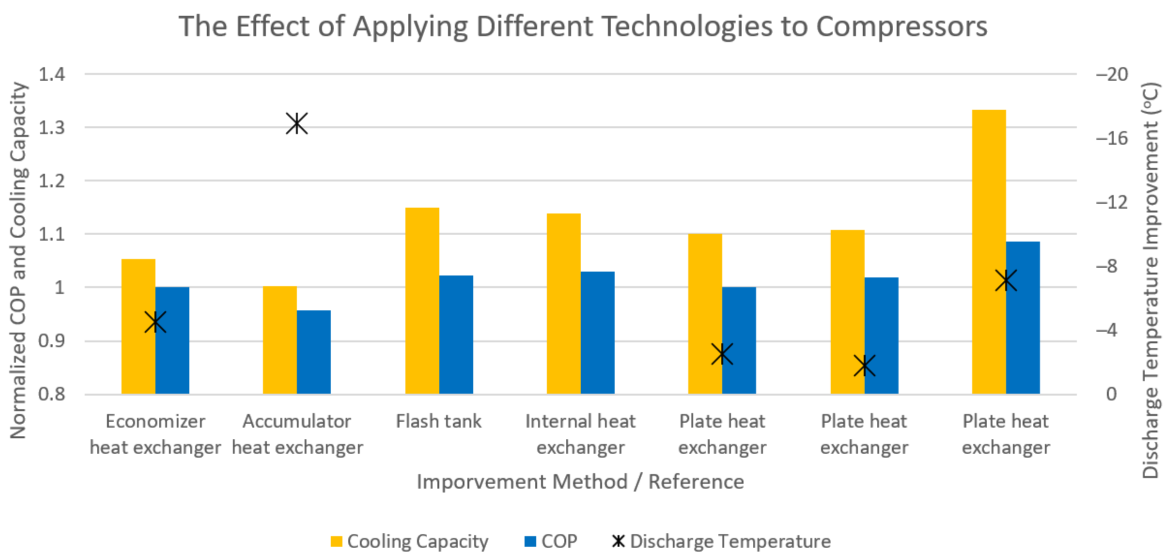

Figure 7. Improvements of different technologies applied to compressors under HAT conditions [54–56,59].

Table 8. Details of compressor improvements for each method.

Method	Injection	Refrigerant	Size	Test Type [1]	Outdoor Condition	Cooling Capacity	COP	Discharge Temperature	Note	Reference
Economizer heat exchanger	Vapor injection	R-22	NA	Experimental	N/A	+5.3%	0	−4.5 °C	60 °C condensing temperature	[54]
Accumulator heat exchanger	Suction-line refrigerant injection	R-22	2.6 ton [9.1 kW_{th}]	Experimental	43.0 °C	+0.2%	−4.2%	−16.9 °C	Injection ratio 15% using open EEV 36%	[55]
Flash tank	Saturated vapor injection	R-410A	3.1 ton [10.9 kW_{th}]	Experimental	46.1 °C	+15.0%	+2.2%	N/A	At 26% injection ratio	[56]
Internal heat exchanger	Superheated vapor injection	R-410A	3.1 ton [10.9 kW_{th}]	Experimental	46.1 °C	+13.9%	+3.0%	N/A	At 20% injection ratio	
Plate heat exchanger	Saturated vapor injection	R-407C	5.0 ton [17.6 kW_{th}]	Experimental	46.1 °C	+10.0%	+0.1%	−2.5 °C	[2]	[59]
	Superheated vapor injection	R-407C	5.0 ton [17.6 kW_{th}]	Experimental	46.1 °C	+10.8%	+1.9%	−1.8 °C	[2]	
	Superheated vapor injection	R-407C	5.0 ton [17.6 kW_{th}]	Numerical	46.1 °C	+33.3%	+8.7%	−7.1 °C	Optimized model [2]	

[1] All compressors used are scroll types. [2] Shown data are at average values for tests 1, 2, and 3.

The compressor improvement techniques mentioned above can be extended to experimentally assessing mildly flammable refrigerants (A2L) under HAT, especially for the high discharge temperature challenge of R-32. Since numerical analysis showed R-32 could reach high discharge temperatures when using a vapor injection technique, two-phase injection within allowable compressor limits could be tested. Of note, this technique has higher costs, is difficult to control, and provides small improvements when compared to vapor injection [57]. In addition, increasing the injection ports at the compressor may lead to only slight improvements and hence it is not always feasible [57]. Utilizing an accumulator heat exchanger showed significant discharge temperature reductions for R-22 and could also be used for assessing R-32 under HAT.

Other studies explored the effects of high ambient temperature (HAT) on the compressor internals and oil viscosity. It was found that HAT decreased oil viscosity and the film thickness of bearings and increased the risk of metallic contact [61]. Moreover, at 50 °C ambient temperatures, the load on the crank of a rotary compressor was found to be double the load at 30 °C, while the compression process took longer periods to reach discharge pressure [61]. Therefore, the following techniques can be used to lower the risk of damaging the compressor internals: (i) maintain relatively higher suction temperatures, (ii) use relatively higher viscosity oils, (iii) limit the compression ratio, (iv) use bearings with lower surface roughness, and (v) shut-off controls at excessive temperatures.

7.3. Refrigerants

Many studies were conducted to test different refrigerant performances at high ambient temperature (HAT) for the purpose of replacing R-22 and R-410A. The test levels are mainly drop-in and soft-optimized, as explained in Section 4.3.1. The data presented here include: (i) all related literature studies including ORNL results for both R-22 and R-410A and (ii) the AHRI low-GWP AREP program. A comparison among different refrigerants is shown in the following data based on: (i) cooling capacity, (ii) *COP*, (iii) discharge temperature, (iv) refrigerant classification, (v) indoor and outdoor temperatures, and (iv) unit size. The refrigerants are not meant to be compared directly since each test was subject to different test levels and conditions, but the data can help in determining the general behavior of each refrigerant under certain test conditions. The selection of new refrigerants is essentially dictated by the climate protocols that aim to lower the environmental impacts. Hence, using low GWP and zero ODP will most likely lead to higher flammability, which adds complexity to the system and more safety requirements. With that being said, the mildly flammable refrigerants (A2L) show attractive performance and are being tested by several researchers.

Table 9 shows all related studies found in the literature with the aim of replacing R-22 with different refrigerants (i.e., A1, A2L, and A3 types). It is obvious that there is no A1 refrigerant that outperformed R-22, and they all had lower *COP* with a wide range of cooling capacities (Figure 8). R-424A showed a similar *COP* but at the cost of a significantly lower cooling capacity. Moreover, A2L R-444B refrigerant reached a similar *COP* and slightly higher cooling capacity than R-22, while ARM-20a had a slightly higher *COP* but lower cooling capacity. ARM-20a showed a substantially lower discharge temperature by 22 °C. In general, most A2Ls are within 4% of the R-22 cooling capacity and within *COP* reductions of up to 15% (Table 9). Most A3 refrigerants showed higher *COP* than R-22 but almost all of them had a lower cooling capacity. The interesting behavior of R-1270 was observed when a larger displacement compressor was used (i.e., higher mass flow rate), which significantly improved the cooling capacity at comparable *COP* and had a slightly lower optimum refrigerant charge [70]. For the same study, R-290 showed cooling capacity improvements but much lower *COP* values, which explains the importance of using an optimum-sized compressor for each refrigerant. It is noted that discharge temperatures of both R-290 and R-1270 were not given in their respective studies (Table 9).

Table 9. Experimental results for R-22 alternative refrigerants at HAT conditions.

System Type	Size	Baseline Refrigerant	Alternative	Test Level	Baseline [1] Cooling Capacity/COP	Indoor Condition	Cooling Capacity [1]	COP [1]	Discharge Temperature [1]	Outdoor Condition [2]	Legend [3]	Reference
Residential heat pump AC	1.0 ton [3.5 kW_{th}]	R-22	R-161	N/A	2.9 kW_{th}/0.68	DBT/WBT 27 °C/19 °C	−5.1%	+10.0%	−3 °C	DBT/WBT 48 °C/30 °C	●	[67]
Residential split AC	1.0 ton [3.5 kW_{th}] 2.0 ton [7.0 kW_{th}]	R-22	R-290 (2.0 ton) R-407C (2.0 ton) R-410A (2.0 ton)	N/A	3.0 kW_{th}/2.32 (6.7 kW_{th}/2.60)	DBT/WBT 25 °C/19 °C	−1.8% (−4.0%) −1.2% (−1.0%) +3.2% (+2.3%)	+19.4% (+3.1%) −6.0% (−6.1%) −5.4% (−9.3%)	N/A	DBT 40 °C to 55 °C	□	[71]
Split AC system	N/A	R-22	R-410A	N/A	9.3 kW_{th}/2.28	DBT/WBT 26.7 °C/19.4 °C	−9.0%	−15.0%	N/A	DBT 54.4 °C	O	[63]

Window AC	1.5 ton [5.3 kW$_{th}$]	R-22	R-407C R-290	N/A Drop-in	4.2 kW$_{th}$/1.84 4.1 kW$_{th}$/1.76	DBT/WBT 29 °C/19 °C	−7.9% −9.7%	−13.5% +2.8%	N/A	DBT/WBT 46 °C/24 °C	◇	[64,65]
Wall room AC	0.7 ton [2.5 kW$_{th}$]	R-22	R-290 (20% larger disp. compressor) R-1270 (20% larger disp. compressor)	N/A	2.4 kW$_{th}$/0.87	DBT/WBT 27 °C/19 °C	−11.3% (−2.9%) −2.9% (+13.0%)	+3.7% (−7.1%) −5.7% (−7.1%)	N/A	DBT/WBT 40 °C/24 °C	△	[70]
Mini split AC	1.5 ton [5.3 kW$_{th}$]	R-22	N-20B DR-3 ARM-20B L-20A (R-444B) R-290 DR-93	Soft optimized	4.9 kW$_{th}$/1.9	DBT/WBT 29 °C/19 °C	−14.5% −12.0% −3.0% −4.0% −9.5% −7.5%	−10.5% −14.5% −11.0% −7.0% +7.5% −14.5%	−7 °C −4 °C +3 °C +6 °C −13 °C −2 °C	DBT 52 °C, 55 °C	X	[37]
Rooftop AC	7.7 ton [27.1 kW$_{th}$]	R-22	L-20A (R-444B) ARM-20b DR-7 (R-454A) ARM-20a	Drop-in	19.8 kW$_{th}$/1.84	DBT/WBT 29 °C/19 °C	+1.8% +1.2% +0.6% −6.8%	−5.3% −10.5% −14.0% +0.8%	−2 °C −8 °C −11 °C −22 °C	DBT 52 °C	+	[38]
Mini split AC	1.8 ton [6.3 kW$_{th}$]	R-22	R-444B R-407C	N/A	5.2 kW$_{th}$/1.80	DBT/WBT 32 °C/13 °C 29 °C/19 °C	+1.0% −2.5%	0 −5.0%	N/A	DBT/WBT 53 °C/32 °C 46 °C/29 °C	✱	[72]
Split AC	0.6 ton [2.1 kW$_{th}$]	R-22	R-424A	Drop-in	1.7 kW$_{th}$/2.41	DBT 18 °C	−20%	−2.5%	−19 °C	DBT 41 °C	—	[74]

[1] Values are approximated from graphs at the mentioned average outdoor conditions unless they are numerically presented by author(s). [2] Only high ambient temperatures were selected (i.e., 40 °C and above). [3] Symbols are illustrated in Figure 8.

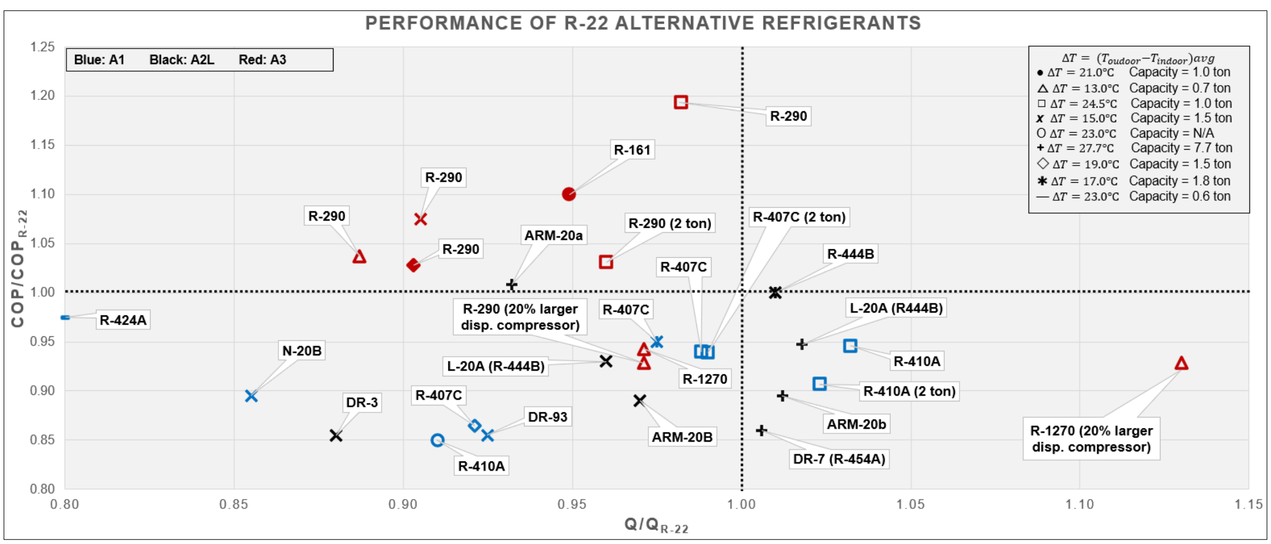

Figure 8. Experimental results for R-22 alternative refrigerants at HAT conditions, where Q is cooling capacity.

All the literature studies evaluated only A2L refrigerants as R-410A alternatives. In Table 10, R-1234yf showed extremely low cooling capacity which was attributed to the need for additional system modifications, beyond charge and TXV optimization, which led to lower flow rates and pressure ratios [68]. Most of the remaining alternatives showed improvements in both cooling capacity and *COP*, however, all of them had higher discharge temperatures than R-410A, except for refrigerant DR-4 (i.e., had lower discharge temperatures at the cost of considerably lower cooling capacity). Therefore, the behavior of any nominated refrigerant must be carefully observed under high ambient temperatures. The discharge temperatures of R-32 are higher than R-410A, ranging from ~16 °C to 30 °C, which makes it imperative to utilize compressor cooling technologies and large condensers. Moreover, after soft optimizing the split system using drop-in DR-5 refrigerant, there was a negligible change in performance but higher discharge temperatures which makes it a better drop-in alternative. All R-410A replacement refrigerants in Table 10 are plotted in Figure 9 excluding R-1234yf [37,38,68,69].

Table 10. Experimental results (except for AHRI Low-GWP AREP) for R-410A alternative refrigerants at HAT conditions.

System Type	Size	Baseline Refrigerant	Alternative	Test Level	Baseline [1] Cooling Capacity/COP	Indoor Condition	Cooling Capacity [1]	COP [1]	Discharge Temperature [1]	Outdoor Condition [2]	Legend [3]	Reference
Residential split heat pump	5.0 ton [17.6 kW_{th}]	R-410A	R-32 R-1234yf	Drop-in	N/A	N/A	+10.0% −49.0%	−1.0% +7.0%	+30 °C −29 °C	DBT 43 °C, 46 °C	□	[68]
			DR-4 (Soft optimization) DR-5 (Soft optimization)	Drop-in	N/A	DBT 21.1 °C	−16.5% (−9.0%) +3.5% (+3.0%)	+5.0% (+10.0%) +6.0% (+7.0%)	−9 °C (−2 °C) +3 °C (+8 °C)	DBT 43 °C, 46 °C	O	[69]
Mini split AC	1.5 ton [5.3 kW_{th}]	R-410A	R-32 DR-55 L41 (R-447A) ARM-71A HPR-2A	Soft optimized	3.4 kW_{th}/1.97	DBT/WBT 29 °C/19 °C	+12.0% 0 −4.5% −3.5% 0	+5.5% +3.0% +4.0% +2.0% +5.5%	+21 °C +8 °C +14 °C +8 °C +14 °C	DBT 52 °C, 55 °C	X	[37]
Rooftop AC	11 ton [38.7 kW_{th}]	R-410A	DR-55 L41z (R-447B) ARM-71a R-32	Drop-in	30.7 kW_{th}/1.80	DBT/WBT 29 °C/19 °C	+1.9% +1.3% +2.9% +5.9%	+3.5% +8.0% +7.6% 0	+7 °C +10 °C +7 °C +19 °C	DBT 52 °C, 55 °C	+	[38]

[1] Values are approximated from graphs at the mentioned average outdoor conditions unless they are numerically presented by the author(s). [2] Only high ambient temperatures were selected (i.e., 40 °C and above). [3] Symbols are illustrated in Figure 9.

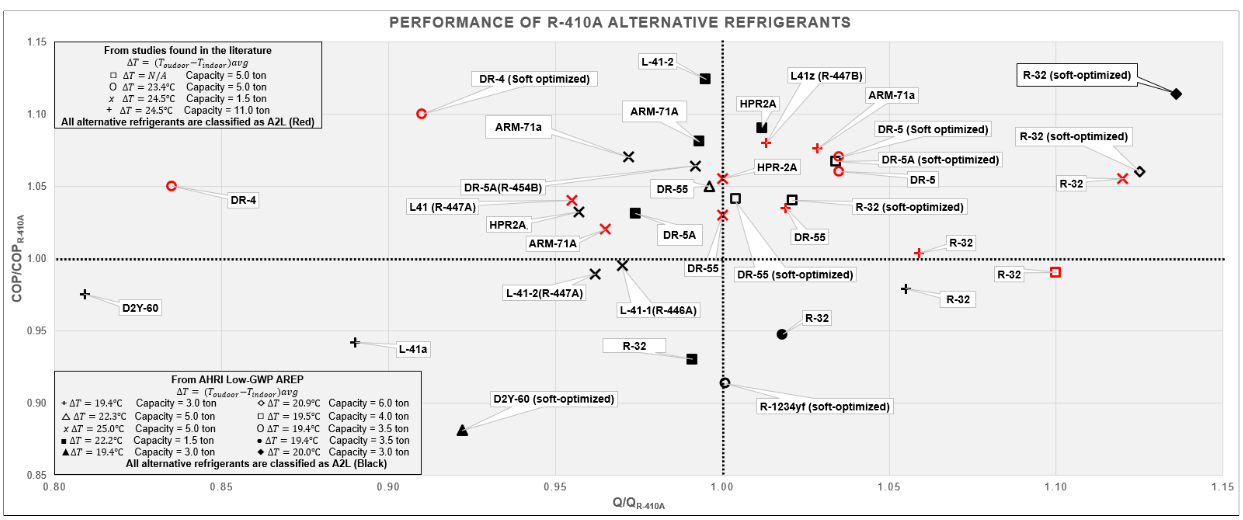

Figure 9. Experimental results for R-410A alternative refrigerants at HAT conditions, where Q is cooling capacity.

The AHRI low-GWP program [123,134–142], discussed in Section 3.2, conducted several tests under high ambient temperatures, Figure 9 and Table 11. All R-410A alternatives are A2L types, and many of these refrigerants showed higher *COP* and cooling capacity than R-410A. R-32 with soft-optimization achieved the highest performance among all refrigerants, however, the discharge temperature was higher by ~16 °C. R-32 discharge temperatures for different experiments ranged from 16 to 26 °C. Various refrigerants had higher *COP* and similar cooling capacity to R-410A with only a small increase in discharge temperature of less than 10 °C. Hence, A2L refrigerants are capable of replacing R-410A but their flammability requires further risk assessment under HAT to decide which level of replacement is recommended (i.e., soft optimized or new equipment).

Table 11. AHRI Low-GWP AREP results for R-410A alternative refrigerants at HAT conditions.

System Type	Size	Baseline Refrigerant	Alternative	Test Level	Baseline [1] Cooling Capacity/COP	Indoor Condition	Cooling Capacity [1]	COP [1]	Discharge Temperature [1]	Outdoor Temperature [2]	Legend [3]	Reference
Rooftop AC	4.0 ton [14.1 kW_{th}]	R-410A	DR-55	Soft-optimized	12.1 kW_{th}/2.25	DBT/WBT 26.6 °C/19.4 °C	+0.4%	+4.1%	+4.6 °C	DBT 40.5 °C to 51.7 °C	□	[123]
			R-32				+2.1%	+4.0%	+20.2 °C			
			DR-5A				+3.4%	+6.7%	+6.2 °C			
Rooftop AC	6.0 ton [21.1 kW_{th}]	R-410A	R-32	Soft-optimized	18.1 kW_{th}/2.48	DBT/WBT 26.6 °C/19.4 °C	+12.5%	+6.0%	+20.7 °C	DBT 40.5 °C to 54.4 °C	◇	[134]
Rooftop AC	5.0 ton [17.6 kW_{th}]	R-410A	DR-55	Drop-in	14.5 kW_{th}/2.29	DBT/WBT 26.6 °C/19.4 °C	−0.4%	+5.0%	+4.6 °C	DBT 46.1 °C, 51.7 °C	△	[135]
Split system heat pump	3.0 ton [10.6 kW_{th}]	R-410A	ARM-71a	Drop-in	7.4 kW_{th}/1.87	DBT/WBT 26.7 °C/15.6 °C	−2.8%	+7.0%	+7.9 °C	DBT 51.7 °C	x	[136]
			DR-5A(R-454B)				−0.8%	+6.4%	+9.0 °C			
			HPR2A				−4.3%	+3.2%	+1.6 °C			
			L-41-1(R-446A)				−3.0%	−0.5%	+14.7 °C			
			L-41-2(R-447A)				−3.8%	−1.1%	+8.2 °C			
Rooftop AC	5.0 ton [17.6 kW_{th}]	R-410A	L-41-2	Drop-in	14.5 kW_{th}/2.29	DBT/WBT 26.7 °C/19.4 °C	−0.5%	+12.4 %	+6.4 °C	DBT 46.1 °C, 51.7 °C	■	[137]
			ARM-71A				−0.7%	+8.1%	+6.9 °C			
			HPR2A				+1.2%	+9.0%	+13.5 °C			
			DR-5A				−2.6%	+3.1%	+9.2 °C			
			R-32				−0.9%	−7.0%	+26 °C			
Split AC system	3.0 ton [10.6 kW_{th}]	R-410A	R-32	Soft-optimized	8.4 kW_{th}/2.02	DBT/WBT 28.9 °C/18.9 °C	+13.6%	+11.4 %	+15.9 °C	DBT/WBT 46.1 °C/25.7 °C 51.7 °C/27.7 °C	◆	[138]
Air source heat pump	3.0 ton [10.6 kW_{th}]	R-410A	D2Y-60	Soft-optimized	8.5 kW_{th}/2.4	DBT/WBT 26.7 °C/19.4 °C	−7.8%	−11.9 %	−1.6 °C	DBT 46.1 °C	▲	[139]
Air source heat pump	3.0 ton [10.6 kW_{th}]	R-410A	R-32	Drop-in	8.5 kW_{th}/2.4	DBT/WBT 26.7 °C/19.4 °C	+5.5%	−2.1%	+18.1 °C	DBT 46.1 °C	+	[140]
			D2Y-60				−19.1%	−2.5%	−11.5 °C			
			L-41a				−11.0%	−5.8%	+6.7 °C			
Split system heat pump	3.5 ton [12.3 kW_{th}]	R-410A	R-1234yf	Soft-optimized	10.9 kW_{th}/2.64	DBT/WBT 26.7 °C/19.4 °C	+0.1%	−8.7%	−17.8 °C	DBT 46.1 °C	O	[141]
Split system heat pump	3.5 ton [12.3 kW_{th}]	R-410A	R-32	Drop-in	10.9 kW_{th}/2.64	DBT/WBT 26.7 °C/19.4 °C	+1.8%	−5.3%	+23.2 °C	DBT 46.1 °C	●	[142]

[1] Values are approximated from graphs at the mentioned average outdoor conditions unless they are numerically presented by author(s). [2] Only high ambient temperatures were selected (i.e., 40 °C and above). [3] Symbols are illustrated in Figure 9.

8. Conclusions

This article provides an overview of residential vapor-compression air conditioners operating under high ambient temperatures (HAT). For the purpose of this article, a minimum temperature criterion, 40 °C and above, was developed to evaluate studies that were conducted at HAT. Several HAT organizations and projects (i.e., ORNL, AHRI AREP, EGPYRA, and PRAHA) were launched with the purpose of assessing the performance of low-GWP refrigerants when operating under HAT and accelerating the transition to such refrigerants. Previous studies of air conditioner improvements (i.e., for condensers, evaporators, compressors, and refrigerants) were discussed under HAT conditions. This article also discussed the challenges, the possible design modifications, and several limitations of air conditioners operating under HAT. The main challenges at high ambient temperature are (i) the reduced efficiency, (ii) the higher energy consumption, (iii) the high discharge temperature associated with promising A2L refrigerants, (iv) the higher minimum energy efficiency requirements, and (v) the ambient temperature increase due to global warming.

Climate limitations were explored for Kuwait and Phoenix, Arizona, USA, and the following was observed:

- The yearly HAT season of Kuwait was found to be ~5 months and therefore designing efficient air conditioners for such a climate is necessary.
- The yearly HAT season in Phoenix, Arizona, USA was found to be ~2 months, and hence a careful evaluation must be conducted to avoid any penalties for oversizing the unit when operating under mild temperatures.
- Over the past 6 years in both Kuwait and Phoenix, Arizona, USA, the average temperatures during the HAT seasons increased by up to ~1.7 °C over the period 1981 to 2010, indicating already high ambient temperatures are increasing even further.
- After surveying the units sold in the USA and GCC regions, it was found that both provided allowable operating temperatures of up to 52 °C.
- In Kuwait, the units are certified by meeting the minimum value of *EER*, while in the Southwest region of the USA, the unit must meet the minimum values of both *EER* and *SEER*.

Main findings and suggested improvements:

- Refrigerant R-32 showed the highest improvements in terms of cooling capacity and *COP*, however, the discharge temperature was too high for a conventional unit to run continuously without tripping.
- Refrigerants DR-5, DR-5a, and ARM-71a showed better performance than R-410A but a slightly higher discharge temperature of +2 to +9 °C, which is expected to be mitigated using compressor and condenser enhancements techniques.
- The compressor cooling techniques that yielded the highest improvements (i.e., lowest discharge temperatures) are saturated vapor injection and accumulator heat exchanger at the suction line.
- Condenser improvements were mainly related to evaporative cooling techniques which showed significant improvements in cooling capacity and *COP*.
- The irreversibilities of evaporators were found to be significant at HAT and only one study of evaporator improvement was found under such conditions. The evaporator circuitry was interleaved based on the air maldistribution, which showed considerable improvements in cooling capacity and *COP* but needs to be implemented at the design stage.
- The TEWI and LCCP analyses showed that the indirect impact poses a significantly higher effect on the environment than the direct impact for an air conditioner operating under HAT conditions and hence, the energy efficiency remains the dominant obstacle that needs further improvement. Therefore, to meet the purpose of lowering global warming and CO_2 emissions, HAT countries should only allow the transition to similar or better energy efficiency A2L units.

Recommended research paths:

- Several studies assessed microchannel condensers under HAT using a bespoke test facility where one of them, at near-critical pressure, found a unique heat transfer behavior and recommended further testing at critical pressure to have a better understanding of the heat transfer behavior.

- Evaporators can be assessed using active control methods which control the refrigerant flow to obtain a uniform exit superheat.
- The literature studies with the purpose of evaluating the performance of different A2L and A3 refrigerants compared to the baseline R-410A provided sufficient data and hence further assessments are not recommended.
- Risk assessments represent a major concern for the next transition to mildly flammable refrigerants and hence, any further development in this area can significantly accelerate the transition to A2L or even A3 refrigerants.

9. Future Work

This article was primarily focused on residential air conditioners only, while the following ideas could be further investigated at HAT:

1. Applying active controls of evaporator refrigerant flow to regulate the exiting superheat.
2. Investigating the condensation behavior of a microchannel condenser at critical pressure since it showed unique findings at near-critical pressure.
3. Applying external cooling techniques to the compressor shell using an additional heat exchanger (e.g., coolant can be refrigerant or phase change material).
4. Investigating the energy savings when using a two-stage compressor compared to a single-stage under real HAT conditions.

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