Model validations for low-global warming potential refrigerants in mini-split air-conditioning units

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Introduction

The use of hydrofluorocarbon (HFC) refrigerants as nonozone-depleting fluid alternatives for air conditioning and refrigeration equipment was adopted by the developed countries during the ozone-depleting substances (ODS) phase-out, as described in the Montreal Protocol (Ozone Secretariat 2016). Unfortunately, the commonly used HFCs have higher global warming potential (GWP) compared to the refrigerants that they replaced; for example, R-410A has a GWP of 1924, and R-22 has a GWP of 1760, which are thousands of times higher than natural refrigerants like CO₂. HFCs currently account for only 1% of greenhouse gas emissions, but their use is growing rapidly by as much as 10 to 15% per year, primarily because of their use as replacements for ODS and the increasing use of air conditioners globally, as reported by Ramanathan and Xu (2010) and Xu et al. (2013). Furthermore, according to the Montreal Protocol, Developing Countries, Article 5, countries have started their phase-down schedule for ODS. As such, finding suitable lower GWP refrigerants for HFC and hydrochlorofluorocarbon (HCFC) refrigerants is timely and will avoid a costly two-step transition from HCFC to HFC and then from HFC to lower GWP refrigerants. Therefore, there is potential for significant reduction in greenhouse gas emissions through the substitution of high-GWP HFCs and HCFCs with lower GWP alternatives.

While progress toward widespread application low-GWP refrigerants continues, only limited information regarding the performance of the most commonly proposed low-GWP refrigerants is available. Abdelaziz and Shrestha (2016) conducted extensive experimental tests to assess low GWP alternative refrigerants as drop-in replacements in two mini split air conditioning units designed for high-ambient conditions. One unit was designed for R-22 and the other for R-410A. Table 1 shows the R-22 alternatives; Table 2 shows the R-410A alternatives evaluated by Abdelaziz and Shrestha (2016). In these tables, the temperature glides in the condenser were calculated as the difference between the saturated vapor temperature and liquid temperature at pressure corresponding to 115°F (46.1°C) dew point, and glides in the evaporator were calculated at pressure corresponding to 50°F (18°C) dew point.

The test conditions from Abdelaziz and Shrestha (2016) are listed in Table 3. They reported the measured airside capacities and energy efficiency ratios and compared performances of different refrigerants. Table 4 compares the R-22 alternatives, while Table 5 compares the R-410A alternatives. The experimental results were obtained via a soft-optimized process, as follows:

To identify low global warming potential refrigerants to replace R-22 and R-410A, extensive experimental evaluations were conducted for multiple candidates of refrigerant at the standard test conditions and at high-ambient conditions with outdoor temperature varying from 27.8°C to 55.0°C. In the study, R-22 was compared to propane (R-290), DR-3, ARM-20B, N-20B, and R-444B in a mini split air-conditioning unit originally designed for R-22; R-410A was compared to R-32, DR-55, ARM-71A, L41-2 (R-447A) in a mini split-unit designed for R-410A. To reveal the physics behind the measured performance results, thermodynamic properties of the alternative refrigerants were analysed. In addition, the experimental data were used to calibrate a physics-based equipment model, for example, ORNL heat pump design model. The calibrated model translated the experimental results to key calculated parameters, i.e. compressor efficiencies and refrigerant side two-phase heat transfer coefficients, corresponding to each refrigerant. These calculated values provide scientific insights on the performance of the alternative refrigerants and are useful for other applications beyond mini split air-conditioning units.

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Table 1. Alternative low GWP replacements for R-22.

| Refrigerant | GWP AR4 | GWP AR5 | Safety class | Glide in condenser [K] | Glide in evaporator [K] | Critical temperature [C] |
|-------------|---------|---------|--------------|------------------------|-------------------------|--------------------------|
| R-22        | 1810    | 1760    | A1           | 0.0                    | 0.0                     | 96.16                    |
| Propane (R-290) | 3       | 3       | A3           | 0.0                    | 0.0                     | 96.74                    |
| DR-3        | 148     | 146     | A2L          | 6.5                    | 7.7                     | 88.47                    |
| ARM-20B     | 251     | 251     | A2L          | 5.3                    | 6.0                     | 88.74                    |
| N-20B       | 988     | 904     | A1           | 4.6                    | 5.4                     | 89.62                    |
| R-444B      | 295     | 295     | A2L          | 7.6                    | 8.9                     | 92.11                    |

aDR-3 has mass-based compositions of R-32 (0.215) / R-1234yf (0.785).
bARM-20B has mass-based compositions of R-32 (0.35) / R-1234yf (0.55) / R-152a (0.1).
cN-20B has mass-based compositions of R-32 (0.13) / R-125 (0.13) / R-134a (0.31) / R-1234yf (0.43).
dR-444B has mass-based compositions of R-32 (0.415) / R-1234ze(E) (0.485) / R-152a (0.1).

Table 2. Alternative low GWP replacements for R-410A.

| Refrigerant | GWP AR4 | GWP AR5 | Safety class | Glide in condenser [K] | Glide in evaporator [K] | Critical temperature [C] |
|-------------|---------|---------|--------------|------------------------|-------------------------|--------------------------|
| R-410A      | 2088    | 1924    | A1           | 0.1                    | 0.1                     | 71.34                    |
| R-32        | 675     | 677     | A2L          | 0.0                    | 0.0                     | 78.12                    |
| DR-55       | 698     | 676     | A2L          | 1.2                    | 1.3                     | 79.68                    |
| ARM-71A     | 460     | 461     | A2L          | 1.8                    | 2.1                     | 81.52                    |
| L41-2 (R-447A) | 583   | 572     | A2L          | 3.8                    | 4.6                     | 82.63                    |

aR-410A has mass-based compositions of R-32 (0.5) / R-125 (0.5).
bDR-55 has mass-based compositions of R-32 (0.67) / R-125 (0.07) / R-1234yf (0.26).
cARM-71A has mass-based compositions of R-32 (0.68) / R-1234yf (0.26) / R-1234ze(E) (0.06).
dL41-2 has mass-based compositions of R-32 (0.68) / R-125 (0.035) / R-1234ze(E) (0.285).

1. Select capillary tube length using appropriate correlations (ASHRAE 2002 and the method used by Yana Motta 1999) and fabricate.
2. Run charge optimization procedure at the AHRI A condition to maximize coefficient of performance (COP) and decide the optimized charge of M_{opt,ref#}; collect steady-state data for 10 min at each condition.
3. Run the unit with M_{opt,ref#} and the selected capillary tube at T3 conditions to ensure adequate subcooling and superheating; if not, adjust the charge accordingly (approximately 2 oz at a time with 10 min of steady-state data collected).
4. Evaluate the system performance for all test conditions listed in Table 3.

As indicated in Table 4, all R-22 alternative refrigerants have smaller cooling capacities, and only propane provides better COPs than R-22. Table 5 reveals that R-32 leads to larger capacities and COPs than R-410A, and DR-55 lead to similar capacities and COPs as R-410A. Other refrigerant drop-ins of the R-410A alternative refrigerants result in smaller capacities and lower COPs at conditions of B and A. At high ambient temperatures, for example, conditions of T3, T3*, hot and extreme, all R-410A alternatives show better

Table 3. Test conditions.

| Test condition | Outdoor Dry-bulb temperature | Indoor Dry-bulb temperature | Wet-bulb temperature | Dew point temperature | Relative humidity |
|---------------|-----------------------------|-----------------------------|----------------------|----------------------|-------------------|
| AHRI B        | 27.8 (82)                   | 26.7 (80.0)                 | 19.4 (67)            | 15.8 (60.4)          | 50.9              |
| AHRI A        | 35.0 (95)                   | 26.7 (80.0)                 | 19.4 (67)            | 15.8 (60.4)          | 50.9              |
| T3            | 46 (114.8)                  | 26.7 (80.0)                 | 19 (66.2)            | 15.0 (59.0)          | 48.8              |
| Hot           | 52 (125.6)                  | 29 (84.2)                   | 19 (66.2)            | 13.7 (56.6)          | 39                |
| Extreme       | 55 (131)                    | 29 (84.2)                   | 19 (66.2)            | 13.7 (56.6)          | 39                |

aThere is no specification for the outdoor relative humidity as it has no impact on the performance.
bDew-point temperature and relative humidity evaluated at 0.973 atm (14.3 psi).
cPer AHRI Standard 210/240.
dT3* is a modified T3 condition in which the indoor settings are similar to the AHRI conditions.
Table 4. Performances of low GWP alternatives for R-22.

| Test condition | R-22 | N-20B | DR-3 | ARM-20B | R-444B | Propane (R-290) |
|----------------|------|-------|------|---------|--------|-----------------|
| COP            |      |       |      |         |        |                 |
| B              | 3.48 | 3.04  | 2.88 | 3.06    | 3.02   | 3.85            |
|                | −13% | −17%  | −12% | −13%    | −13%   | 11%             |
| A              | 3.07 | 2.68  | 2.57 | 2.71    | 2.72   | 3.30            |
|                | −13% | −16%  | −12% | −11%    | −11%   | 7%              |
| T3*            | 2.34 | 2.05  | 1.99 | 2.09    | 2.15   | 2.49            |
|                | −12% | −15%  | −11% | −8%     | −7%    | 6%              |
| T3             | 2.34 | 2.06  | 2.01 | 2.07    | 2.17   | 2.49            |
|                | −12% | −14%  | −11% | −7%     | −7%    | 7%              |
| Hot            | 1.98 | 1.77  | 1.70 | 1.76    | 1.85   | 2.12            |
|                | −11% | −14%  | −11% | −7%     | −7%    | 7%              |
|                |      |       |      |         |        |                 |
| Capacity, kW   |      |       |      |         |        |                 |
| B              | 6.26 | 5.42  | 5.52 | 6.05    | 5.53   | 5.93            |
|                | −13% | −12%  | −3%  | −12%    | −5%    |                 |
| A              | 6.10 | 5.25  | 5.40 | 5.91    | 5.58   | 5.62            |
|                | −14% | −11%  | −3%  | −9%     | −8%    |                 |
| T3*            | 5.41 | 4.56  | 4.81 | 5.28    | 5.17   | 4.90            |
|                | −16% | −11%  | −2%  | −4%     | −9%    |                 |
| T3             | 5.42 | 4.59  | 4.83 | 5.24    | 5.19   | 4.91            |
|                | −15% | −11%  | −3%  | −4%     | −9%    |                 |
| Hot            | 5.00 | 4.26  | 4.41 | 4.84    | 4.79   | 4.50            |
|                | −15% | −12%  | −3%  | −4%     | −10%   |                 |
|                |      |       |      |         |        |                 |
| Extreme        | 4.76 | 4.10  | 4.21 | 4.62    | 4.59   | 4.33            |
|                | −14% | −12%  | −3%  | −4%     | −9%    |                 |

Table 5. Performances of low GWP alternatives for R-410A.

| Test condition | R-410A | R-32 | DR-55 | L41-2 (R-447A) | ARM-71A |
|----------------|--------|------|-------|----------------|---------|
| COP            |        |      |       |                |         |
| B              | 3.95   | 3.99 | 4.03  | 3.62           | 3.94    |
|                | 1%     | 2%   | 3%    | −8%            | 0%      |
| A              | 3.40   | 3.55 | 3.50  | 3.22           | 3.38    |
|                | 4%     | 3%   | 3%    | −5%            | −1%     |
| T3*            | 2.47   | 2.57 | 2.63  | 2.48           | 2.52    |
|                | 4%     | 6%   | 0%    | 2%             |         |
| T3             | 2.49   | 2.59 | 2.52  | 2.49           | 2.48    |
|                | 4%     | 1%   | 0%    | 0%             |         |
| Hot            | 2.07   | 2.17 | 2.14  | 2.13           | 2.11    |
|                | 5%     | 3%   | 3%    | 2%             |         |
|                |        |      |       |                |         |
| Extreme        | 1.87   | 1.98 | 1.93  | 1.96           | 1.90    |
|                | 6%     | 3%   | 5%    | 2%             |         |
| Capacity, kW   |        |      |       |                |         |
| B              | 5.35   | 5.46 | 5.15  | 4.49           | 4.97    |
|                | 2%     | −4%  | −16%  | −16%           | −7%     |
| A              | 5.14   | 5.42 | 5.01  | 4.44           | 4.75    |
|                | 5%     | −2%  | −14%  | −8%            |         |
| T3*            | 4.39   | 4.76 | 4.42  | 4.01           | 4.17    |
|                | 8%     | 1%   | −9%   | −5%            |         |
| T3             | 4.41   | 4.79 | 4.27  | 4.03           | 4.12    |
|                | 9%     | −3%  | −9%   | −7%            |         |
| Hot            | 3.98   | 4.43 | 3.99  | 3.77           | 3.83    |
|                | 11%    | 0%   | −5%   | −4%            |         |
|                |        |      |       |                |         |
| Extreme        | 3.75   | 4.23 | 3.76  | 3.63           | 3.62    |
|                | 13%    | 0%   | −3%   | −3%            |         |
COPs because the R-410A performance degrades drastically when the ambient temperature approaches its critical temperature, i.e., 71.34°C.

This article presents a follow-up study to reveal the physics behind the comparisons in Tables 4 and 5. The experimental data were used to calibrate a physics-based equipment model, i.e., ORNL heat pump design model developed by Shen and Rice (2014) and Rice (1981). The calibrated model is able to translate the experimental results to key parameters, i.e., compressor efficiencies and refrigerant side two-phase heat transfer coefficients, corresponding to each refrigerant.

**Thermodynamic properties of low-GWP alternative refrigerants**

The temperature-enthalpy diagram of a refrigerant illustrates two critical properties: its span between the saturated liquid line and saturated vapor line (i.e., latent heat of vaporization) and critical temperature (working range). Volumetric vaporization heat, i.e., latent heat × vapor density at similar mid-point temperatures, indicates the evaporating capacity per volumetric flow rate. Refrigerants with smaller volumetric vaporization heat have reduced cooling capacities at a constant mid-point temperature and compressor displacement volume, as with the case with drop-in performance evaluation of replacements.

The evaporation mid-point temperature is impacted by the evaporation heat transfer performance of an alternative refrigerant and the heat exchanger surface area normalized to the evaporating capacity. The heat transfer performance is dictated by two factors, i.e., refrigerant heat transfer coefficient and glide. Glide degrades heat transfer performance in a conventional heat exchanger which is designed for R-22 or R-410A due to reduction in average temperature, driving potential between the air and the refrigerant. If the heat transfer performance of an alternative refrigerant is worse than the baseline refrigerant, it decreases the suction saturation temperature and the refrigerant flow rate due to the decreased UA in the same heat exchanger. Consequently, the cooling capacity of an alternative refrigerant depends on the volumetric vaporization heat and heat transfer performance.

**R-22 Alternatives**

For the R-22 alternatives, Figure 1 illustrates the temperature-enthalpy diagram. It can be seen that propane has the widest span between the saturated liquid and vapor lines. Critical temperatures of the refrigerant other than propane are lower than that of R-22. However, these critical temperatures are all higher than 80°C and, therefore, impose no limits on the air conditioning application which tends to have condensing temperature below 70°C. Figure 2 illustrates the volumetric vaporization heat as function of the average saturation temperature. It indicates that R-22, ARM-20B, and R-444B have a similar capacity, and other refrigerants have smaller capacities when compared at similar mid-point temperature.

For the R-410A alternatives, Figure 3 illustrates the temperature-enthalpy diagram of each individual refrigerant. The R-410A alternatives all have wider domes and higher critical temperatures than R-410A, indicating they are better refrigerants for high ambient operations. Figure 4 illustrates the volumetric vaporization heat as function of the average saturation temperature.
saturation temperature. It indicates that R-410A, R-32, and DR-55 have similar volumetric vaporization heat, and R-32 is the largest. Except R-32, the other refrigerants are likely to have smaller cooling capacities if keeping the same heat transfer performance as R-410A.

Besides the thermodynamic properties and heat transfer performance, whether an alternative refrigerant will lead to better cooling capacity and efficiency is also affected by the compressor volumetric efficiency and isentropic efficiency. These will be revealed in the following sections.

Unit information

Table 6 describes parameters of the two mini split air-conditioning units. They are single-speed units, having constant indoor and outdoor airflow rates and compressor speed.

During the laboratory investigations, the two mini split units were instrumented the same way as illustrated in Figure 5. In the figure, the symbol “P” means refrigerant pressure transducers which were placed before the expansion device and at the evaporator exit. “M” means a refrigerant mass flow meter which was placed in the liquid line. “W” means watt transducers used to measure power consumptions of the compressor, indoor blower, and outdoor fan. “T” means temperature probes inserted to refrigerant flow placed at the compressor suction and discharge, evaporator exit, and the liquid line. “Ta” means a temperature sensor to measure the condenser inlet air temperature. “Ta, RH” means temperature and relative humidity sensors to measure the inlet and outlet states of the indoor airflow.

Model description

ORNL Heat Pump Design Model (HPDM) is a well-recognized public domain HVAC equipment modelling and design tool. It has a web interface to support public use, which has been accessed over 300,000 times by U.S. and worldwide engineers. Some features of the HPDM related to this study are introduced below.

Compressor model

In order to model the alternative refrigerants in the same compressor, basic efficiencies were used to model the compressor, i.e., volumetric efficiency shown in Equation 1 and isentropic efficiency shown in Equation 2

\[
m_r = \frac{\text{Volume}_{\text{displacement}} \times \text{Speed}_{\text{rotation}}}{\text{Density}_{\text{suction}} \times \eta_{\text{vol}}} \quad (1)
\]
Fig. 5. Experimental instrumentations.

\[
\text{Power} = m_r \times (H_{\text{discharge},s} - H_{\text{suction}})/\eta_{\text{isentropic}} \tag{2}
\]

where \(m_r\) is compressor mass flow rate, \(\text{power}\) is compressor power, \(\eta_{\text{vol}}\) is compressor volumetric efficiency, \(\eta_{\text{isentropic}}\) is compressor isentropic efficiency, \(H_{\text{suction}}\) is compressor suction enthalpy, and \(H_{\text{discharge},s}\) is an enthalpy obtained at the compressor discharge pressure and suction entropy.

**Heat exchanger models**

**Segment-to-segment fin-and-tube condenser.** This uses a segment-to-segment modeling approach which divides a single tube to numerous mini segments. Each tube segment has individual airside and refrigerant-side entering states and considers possible phase transition. An \(\varepsilon\)-NTU approach is used for heat transfer calculations within each segment. The airside fin is simplified as an equivalent annular fin. Both refrigerant and airside heat transfer and pressure drop are considered. The coil model can simulate arbitrary tube and fin geometries and circuitries, any refrigerant-side entering and exit states, misdistribution, and accept 2D airside temperature, humidity, and velocity local inputs. The tube circuitry and 2D boundary conditions are provided by an input file. The flow-pattern-dependent heat transfer correlation published by Thome (2003a, 2003b) is used to calculate the condenser two-phase transfer coefficient. The pressure drop correlation published by Kedzierski and Choi (1999) is used to model the two-phase pressure drop.

**Segment-to-segment microchannel condenser.** The model uses a segment-to-segment modeling approach. Each microchannel port segment has individual airside and refrigerant-side entering states and considers possible phase transition; the coil model can simulate arbitrary port shapes (round, triangle, etc.), fin geometries and circuitries (serpentine, slab, etc.). The heat transfer correlation published by Dobson and Chato (1998) is used to calculate the condenser two-phase heat transfer coefficient. The Kedzierski (2000) correlation is used to calculate the two-phase pressure drop.

**Expansion devices**

The compressor suction superheat degree and condenser subcooling degree is explicitly specified. As such, the expansion device is not solved here, and a simple assumption of constant enthalpy process is assumed.

**Fans and blowers**

Single-speed fan: The airflow rate and power consumption were direct inputs from the laboratory measurements.

**Refrigerant lines**

The heat transfer in a refrigerant line is ignored, and the pressure drop is calculated using a turbulent flow model as a function of the refrigerant mass flux.

**Refrigerant properties**

Interface to REFPROP 9.1 (Lemmon and Huber 2010): The authors programmed interface functions to call REFPROP 9.1 directly; the current models accept all the refrigerant types in the REFPROP 9.1 database. Also, a new refrigerant can be simulated by making the refrigerant definition file according to the REFPROP 9.1 format.

REFPROP 9.1 can run fairly slow. To speed up the calculation, the authors have an option to generate property look-up tables, based on REFPROP 9.1: the current program uses 1D and 2D cubic spline interpolation algorithms to calculate refrigerant properties via reading the look-up tables; this would greatly boost the calculation speed, given the same accuracy.

**Model calibration**

HPDM has a flexible solver so that any variable can be given or solved. That means users can switch between knowns and unknowns in the system solving. For the model calibrations,
the authors input the measured refrigerant mass flow rate, compressor power, and the pressure and temperature measurements in Figure 5 as known variables and then solve the compressor efficiencies and the two-phase heat transfer coefficients in the condenser and evaporator as unknowns. This two-step solving procedure was applied to decouple the refrigerant-side heat transfer and the airside heat transfer. As the HPDM was validated tremendously against R-22 and R-410A units in previous work (Rice 1997; Shen 2006), its original refrigerant-side heat transfer calculations were considered accurate for the R-22 and R-410A data. For the first step, the airside heat transfer coefficient was reduced for the R-22 and R-410A baseline units using the original refrigerant-side heat transfer correlations. As the mini split units have constant indoor and outdoor airflow rates, the airside heat transfer did not change when running the other alternative refrigerants. For the second step, the calculated airside heat transfer coefficients were treated as knowns and calculated the refrigerant side two-phase heat transfer coefficients specific to each alternative refrigerant, and the phase allocation ratios, i.e., single-phase versus two-phase, are predicted by the HPDM model.

Figure 6 shows calculated volumetric efficiencies as a function of the compressor pressure ratio for the R-22 alternative refrigerants. Figure 7 shows the isentropic efficiencies. Propane appears to have noticeably better volumetric and isentropic efficiencies than the other refrigerants. One reason is that propane has smaller pressure ratios in the ambient temperature range from 27.8°C to 55°C. The other refrigerants lead to similar efficiencies as R-22.

Figure 8 shows calculated volumetric efficiencies as a function of the compressor pressure ratio for the R-410A alternative refrigerants. Figure 9 shows the isentropic efficiencies. The alternative refrigerants result in no apparent deviations from the R-410A efficiencies. R32 has two outliers at pressure ratios of 2.43 and 2.92, respectively. These correspond to performance evaluation using the originally capillary tube and optimizing the refrigerant charge at AHRI A and ISO T3 conditions, respectively. At these test conditions, the liquid subcooling measured at the mass flow meter inlet was less than 0.6°C (1°F), which suggests that possible flashing in the mass flow meter might have resulted in a false reading.

Figures 10 and 11 show calculated two-phase heat transfer coefficients in the fin-and-tube condenser and evaporator of the R-22 alternative refrigerants versus the ambient temperature, respectively. It should be noted that the heat transfer coefficient at one ambient temperature may vary with the indoor wet bulb temperature and the corresponding refrigerant mass flow rate. Furthermore, at 46°C (115°F) outdoor conditions, there were two indoor conditions investigated (T3 and T3*), and there were three tests for R-22 using baseline oil, using POE oil, and a re-run using mineral oil at the end. There were also two tests for propane: one with POE oil and one with mineral oil. Figure 10 indicates that R-22 and propane have similar evaporation two-phase heat transfer coefficients which are noticeably better than the other alternatives. Figure 11 shows that R-22, propane, and ARM-20B have comparable condensation two-phase heat transfer coefficients.
Fig. 9. Calculated isentropic efficiencies of R-410A alternative refrigerants.

Fig. 10. Calculated evaporator two-phase heat transfer coefficients for R-22 alternatives.

Fig. 11. Calculated condenser two-phase heat transfer coefficients for R-22 alternatives.

Fig. 12. Calculated evaporator two-phase heat transfer coefficients for R-410A alternatives.

Figures 9, 10, and 11 show the calculated efficiencies and heat transfer coefficients for R-410A, R-22, and R-22 alternatives, respectively. The graphs illustrate the performance of different refrigerants under various conditions.

Summary

Following the experimental study of Abdelaziz and Shrestha (2016) for low GWP alternative refrigerants of R-22 and R-410A, we used the HPDM to model the baseline mini split units and calibrated the models against the experimental data. The calibrated equipment models were used to estimate the compressor efficiencies and two-phase heat transfer coefficients from this drop-in study. These predicted values provide further insights on the performance of the alternative refrigerants and are useful for other applications beyond mini split air conditioning units.

By comparing the R-22 alternatives, one can see that R-22 has the highest heat transfer coefficients, no temperature glide, and the largest volumetric vaporization heat. Consequently, it has larger cooling capacities than the other refrigerants. Propane has similar heat transfer performance as R-22 but smaller vaporization heat. As a result, propane reaches smaller cooling capacities at various ambient temperatures. On the other hand, propane leads to higher COPs than R-22 due to its reduced capacity with the same heat exchangers and increased compressor efficiencies. Except propane, all other R-22 alternatives have lower capacities and efficiencies than R-22 because of their smaller volumetric vaporization heat, degraded heat transfer coefficients, and temperature glides.

Figures 9 to 12 illustrate the calculated efficiencies and heat transfer coefficients for different refrigerants under various conditions. The graphs show the performance differences and provide insights into the suitability of alternative refrigerants for mini split air conditioning units.
By comparing the R-410A alternatives, one can see R-410A has the best heat transfer performance. R-32 and DR-55 have slightly lower heat transfer performance. R-410A and DR-55 have similar volumetric vaporization heat rate—smaller than the other alternatives. R-32 has the largest volumetric vaporization heat. Therefore, R-32 leads to the highest cooling capacity and COP at various ambient temperatures. DR-55 has similar capacities and COPs of R-410A. ARM-71a and L41-2 have slightly lower heat transfer performance. R-410A and DR-55 have the best heat transfer performance. R-32 and DR-55 have similar volumetric vaporization heat—smaller than the other alternatives. R-32 has the largest volumetric vaporization heat rate.

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