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# Laboratory Evaluation of R407H for Commercial Refrigeration Applications

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# Laboratory Evaluation of R407H for Commercial Refrigeration

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## ABSTRACT

Globally and in the US the commercial refrigeration industry is in a transition to refrigerants below a global warming potential of 1500 to meet regulatory requirements. A Class A1 refrigerant blend in the R407 series was developed, R407H, to meet this requirement for commercial refrigeration applications.

To evaluate the performance of the blend in medium and low temperature conditions a comparison was performed at an independent laboratory (Urbana, IL). The laboratory was instrumented to ASHRAE 72 standard requirements and consists of varying capacity, VFD controlled, tandem scroll compressors, coupled with a controlled condenser and running a set of 5 cases with a combined LT load of 2.98kW and combined medium temperature load of 7.86kW. The equipment selected was standard R22/R404A equipment.

The system was baselined with R22 and a comparison was made to R404A, and R407H. System effects due to glide, such as frosting, condenser and evaporator efficiency were analyzed and potential control parameters proposed and evaluated. It was determined that R407H utilizing traditional R407 refrigerant chemistry without any additional HFO, Hydrocarbon or CO2 components is suitable as a replacement for existing R22 and R404A systems reaching expected efficiency targets at the target GWP.

## 1. INTRODUCTION

In recent years, many refrigerants applications have been to submitted to ASHRAE and over 40 refrigerants have been classified by ASHRAE [5]. Fig.1 shows correlation between GWP and boiling point of refrigerants which have been classified by ASHRAE in the past 5 years. A1 refrigerants ("non-flammable") are shown as "●" and A2L classified refrigerants ("slightly flammable") are shown as "▲". Focusing on the refrigerants having boiling point between -44°C and -48°C similar to R404A, we can categorize those refrigerants into two groups. One group is that consist of non-flammable refrigerant with GWP less than 1500. The other group is that consisted by slightly flammable refrigerant with GWP less than 500. The lower GWP of the refrigerant, the higher flammability of the refrigerant. R407H is one of the newer proposed alternatives for R404A, according to its boiling point, with an A1 designation and a GWP of 1380 per IPCC 5$^{TH}$ assessment report.

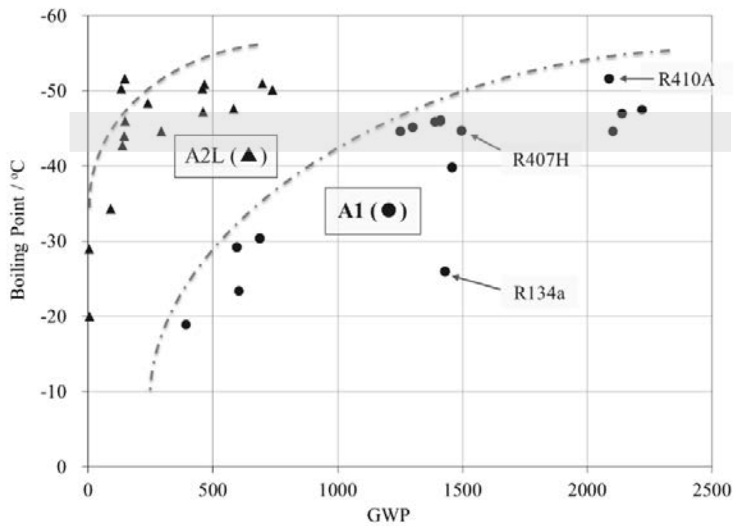


**Figure 1** Interrelation between GWP and Boiling Point of Refrigerants Classified by ASHRAE

R407H theoretical models and compressor Drop-in-Testing has been previously introduced by Shibanuma (2016) at the Kobe symposium. Other refrigerant which fall into the A1 region with GWP < 1500 such as R448A, R449A & R449B have been evaluated by Sjoholm et al. (2014); Yana Motta et al. (2014), Makhnatch and Khodabandeh (2015); Petersen (2016) and others.

The work done an independent laboratory in Urbana Illinois, seeks to add another dimension in characterizing R407H and other R404A/R22 alternatives by running a simulated commercial refrigeration environment. Applications to a physical system demonstrate more accurate pressure drop, heat transfer effects, frosting effects, glide and other properties and as such provide a more accurate assessment of overall system performance

## 2. PROPERTIES AND MODELING

### 2.1 Basic Properties

The properties for the refrigerants evaluated are shown in Table 1. We can see that the saturated pressure of the refrigerants is comparable to R404A

**Table 1:** Basic Refrigerant Properties

| | R22 | R404A | R407H | R448A |
|---|---|---|---|---|
| Global Warming Potential (GWP AR5) | 1760 | 3943 | 1380 | 1273 |
| Ozone Depletion Potential (ODP 1) | 0.055 | 0 | 0 | 0 |
| Boiling Point at 1atm (°C) | -40.8 | -46.2 | -44.6 | -46.1 |
| Molecular Weight | 86.5 | 97.6 | 79.1 | 86.3 |
| Saturated Liquid Pressure at 23.9°Cw (Mpa) | 1.01 | 1.22 | 1.20 | 1.19 |
| Critical Pressure (Mpa) | 4.99 | 3.73 | 4.85 | 4.66 |
| Critical Temperature (°C) | 96.1 | 72.1 | 86.6 | 83.7 |
| Liquid Density at 25°C (kg/m$^3$) | 1190 | 1044 | 1112 | 1100 |
| Vapor Density at 25°C (kg/m$^3$) | 44.2 | 65.7 | 41.7 | 46.1 |
| ASHRAE Classification | A1 | A1 | A1 | A1 |
| Lubricant | MO | POE, PVE | POE, PVE | POE, PVE |
| Composition by wt. | R22(100%) | R125(44%)<br>R134a(52%)<br>R143a(4%) | R32(32.5%)<br>R125(15%)<br>R134a(52.5%) | R32(26%)<br>R125(26%)<br>R134a(21%)<br>R1234yf(20%)<br>R1234ze(E)(7%) |

### 2.2 Modeling Evaluation Conditions

In this work the refrigerants are compared for application in commercial refrigeration equipment, or more specifically, in supermarket rack type systems. To that end a basic simulation of performance was first conducted to obtain a baseline or predicted result for comparison. Evaluation conditions are listed in table 2.

**Table 2:** Modeling Evaluation Conditions

| | |
|---|---|
| Condensing Temperatures | 25°C / 35°C |
| Evaporating Temperature | -4°C (MT) / -30°C (LT) |
| Evaporator Superheat | 8 K |
| Suction Line Superheat | 10 K |
| Subcooling | 1.5 K |
| Compressor Efficiency | 70%(MT) / 65% (LT) |
| Compressor Model | Simple Cycle (MT) / Intermediate Liquid Injection (LT) |

A simple cycle one stage model was not appropriate for the LT condition as discharge temperatures at the compressor would exceed typical design conditions thus a liquid injection model was used.

### 2.3 Modeling Evaluation Results

Modeling was performed utilizing Daikin Ref 10.0 and NIST Refprop 9.1

**Table 3:** Modeling Results

| Medium Temperature Modeled COP | R22 | R404A | R407H | R448A |
|---|---|---|---|---|
| 35°C Condensing | 3.96 | 3.66 | 3.88 | 3.83 |
| 35°C Condensing (as % of R22) | 100% | 92.4% | 98% | 96.7% |
| 25°C Condensing | 5.61 | 5.36 | 5.52 | 5.5 |
| 25°C Condensing (as % of R22) | 100% | 95.5% | 98.4% | 98% |

| Low Temperature Modeled COP | R22 | R404A | R407H | R448A |
|---|---|---|---|---|
| 35°C Condensing | 2.16 | 1.71 | 1.90 | 1.88 |
| 35°C Condensing (as % of R22) | 100% | 79.1% | 88.0% | 86.9% |
| 25°C Condensing | 2.52 | 2.27 | 2.42 | 2.41 |
| 25°C Condensing (as % of R22) | 100% | 90.1% | 96.0% | 95.6% |

The modeled results show that R407H has a 4-12% reduction in efficiency when compared to R22 at the low temperature set point and a 2% reduction compared to R22 at the medium temperature condition.

## 3. EXPERIMENT

### 3.1 Experimental Set Up

An environmental chamber with dimensions 36’ x 18’ x 18.5’ was set up to according to ASHRAE 72 specifications. The chamber was populated with five 8ft long display cases. With an estimated combined medium temperature load of 7.86kW and a combined low temperature load of 2.98kW.

**Table 4:** Set Up Details

| | Estimated Load | # of TC / Simulators | Total Load (kg) |
|---|---|---|---|
| Case 1 (MT Case) | 1.12kW | 12 / 308 | 118 |
| Case 3 (MT Case) | 2.18kW | 18 / 462 | 177 |
| Case 4 (MT Case) | 3.38kW | 30 / 506 | 193 |
| Case 5 (MT/LT Case) | 0.59kW/1.49kW | 48 / 1092 | 417 |
| Case 6 (MT/LT Case) | 0.59kW/1.49kW | 48 / 1092 | 417 |

For control each case was equipped with electronic expansion valve (EEV) and an Electronic Evaporation Pressure Regulator (EEPR) managed by an a traditional system controller, model number E2 controller RX300. Each case uses the EEPR to adjust its own evaporating pressure according to the temperature control set-point, the EEPR of the leading case was at 100% open. The leading case, determine by the lowest evaporating temperature, was Case 4 when running at MT conditions, and Case 6 when running at LT conditions. The superheat of each case is controlled by the EEV using PID loops built into the controller. For the medium temperature applications defrost was completed by system OFF cycle, with evaporation temperature set at 5.5°C for case 1,2,3 and 8.9°C for case 5 and 6. Each case was populated with product simulators and instrumented with thermal couples to obtain product temperatures over the course of the experiment.

Additionally, the chamber was conditioned to a constant temperature and humidity with a dry bulb temperature of 24°C with a dew point of 18°C. The heating was provided via radiant heat from the white gloss floor. Air currents in the chamber were maintained at less than 0.25 m/s from the suspended ceiling with returns at floor level. The chamber was lit with fluorescent lighting resulting in a minimum luminescence of 800 lux. Case 5 and Case 6 were equipped with automated door openers

The condenser was enclosed in temperature-controlled environment with the ability to control inlet temperature to 25°C or 35°C. The condensing unit itself consisted of two tandem scroll compressors, specifically model numbers ZF11 and ZF06. Each having a MT rated capacity of 6.5kW and 3.7kW respectively, and a LT rated capacity of 2.4kW and 1.4kW. Both compressors are equipped with liquid injection which was turned off for the MT testing.

Compressors and the system was operated utilizing POE oil, with an oil separator and compressor oil return level monitored and controlled.

The diagram for the control scheme of system can be seen in the figure below

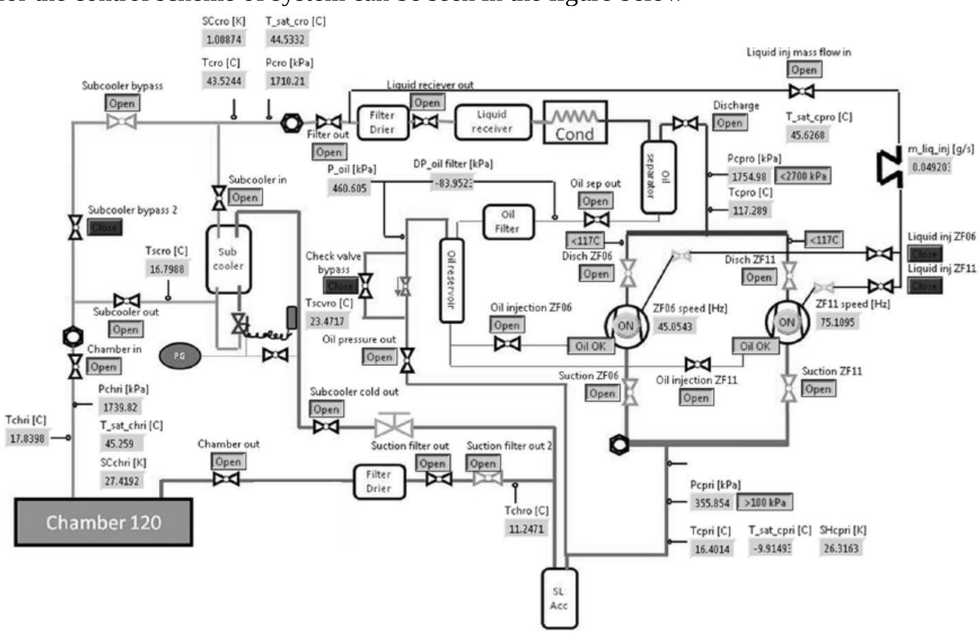


**Figure 2.** Experiment Control Diagram

### 3.2 Experimental Results

After reaching a steady state condition, typically 3-4 days, each test was conducted over 24 hours period. An example of a test can be seen in figure 3.

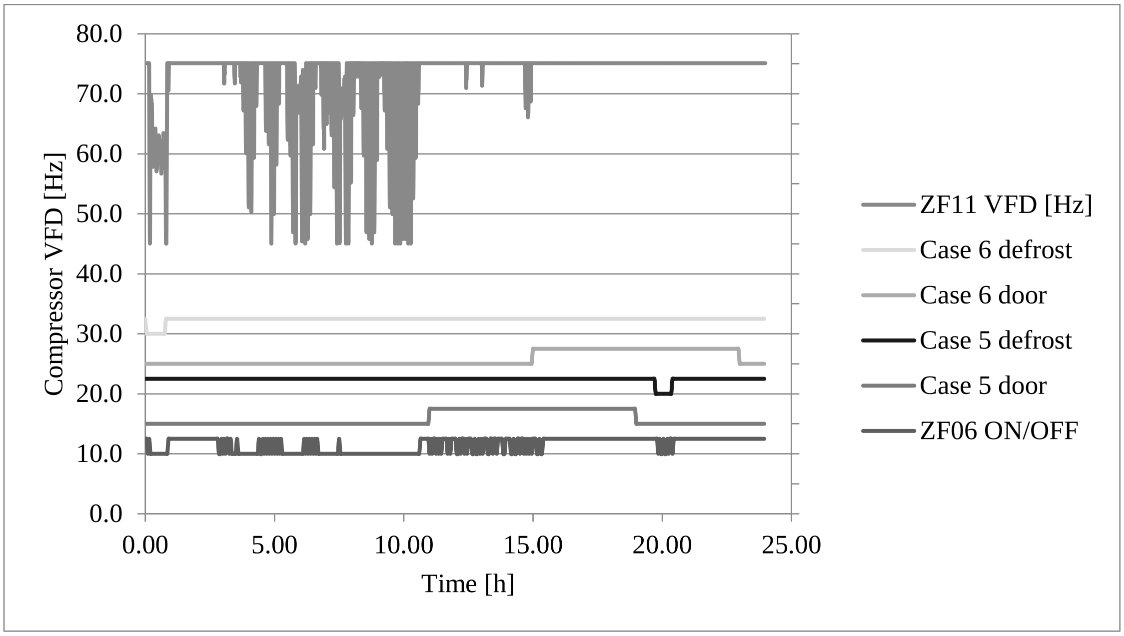


**Figure 3.** Example Test R407H LT Condition

We can see from figure 3 that over the course of the 8 hours where the door openers are simulating customer behavior the load on the system is increased. This can be observed by the increased run time of the ZF06 compressor as well as the higher frequency of the VFD driving the ZF11 compressor.

Product simulator temperatures were measured throughout the experiment and can be seen in Figures 3 and 4. The average temperature is indicated by the marker and the average of the warmest and coldest product simulators over the course of the test are represented by the error bars.

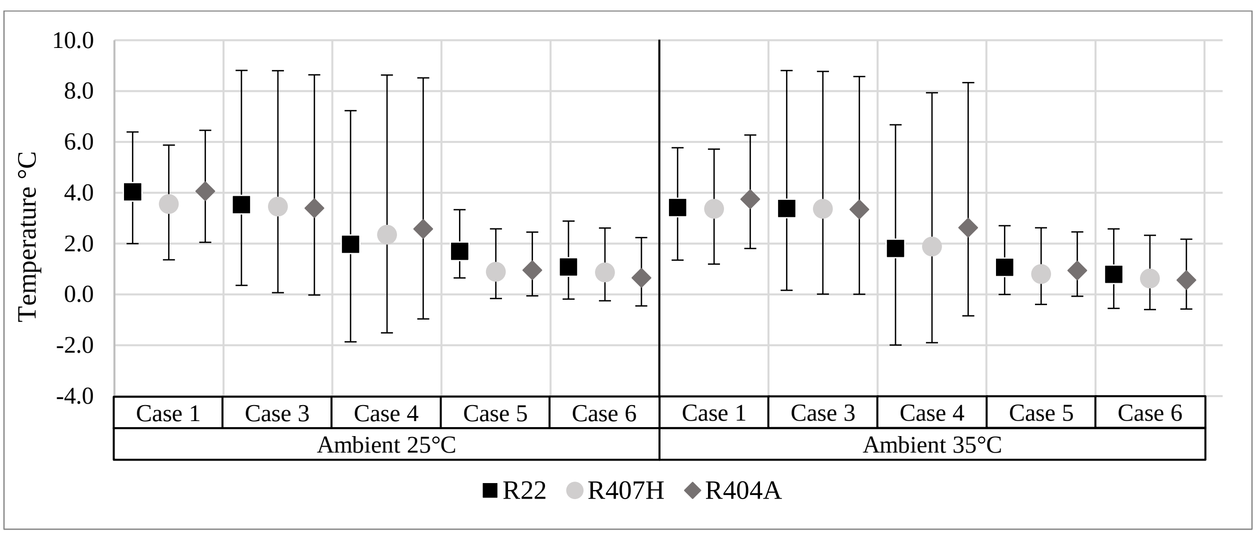


**Figure 4.** Product Simulator Temperatures at MT conditions

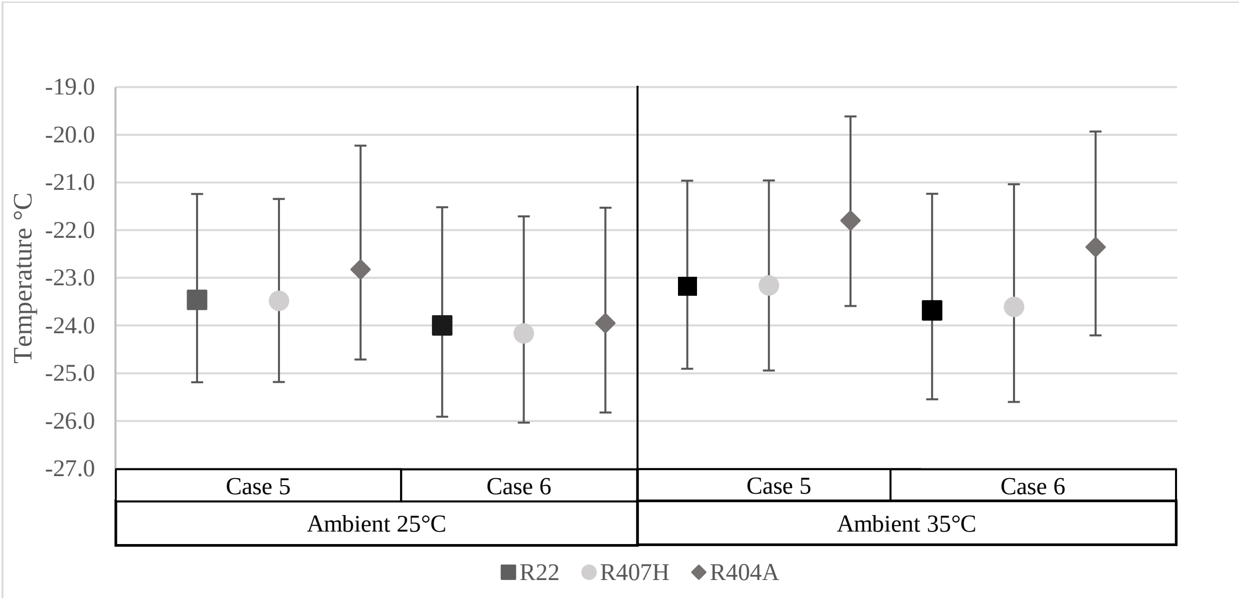


**Figure 5.** Product Simulator Temperatures at LT conditions

All three refrigerants maintained a very similar product simulator temperature, and product temperature spread in the medium temperature condition at both condensing temperatures. At low temperature conditions we see that R404A showed slightly elevated temperatures as compared to R22 and R407H, and all three demonstrated a similar temperature range. Additionally, during low temperature application we see that, on average, temperatures increased 0.5°C when going from the lower condensing temperature to the higher.

From the data gathered we calculate the COP of the refrigeration system. The COP is taken as the ratio of the cooling energy for all of the display cases, calculated from refrigerant side measurements, and the total electric energy consumed by the condensing unit and all of the display cases over the 24-hour period.

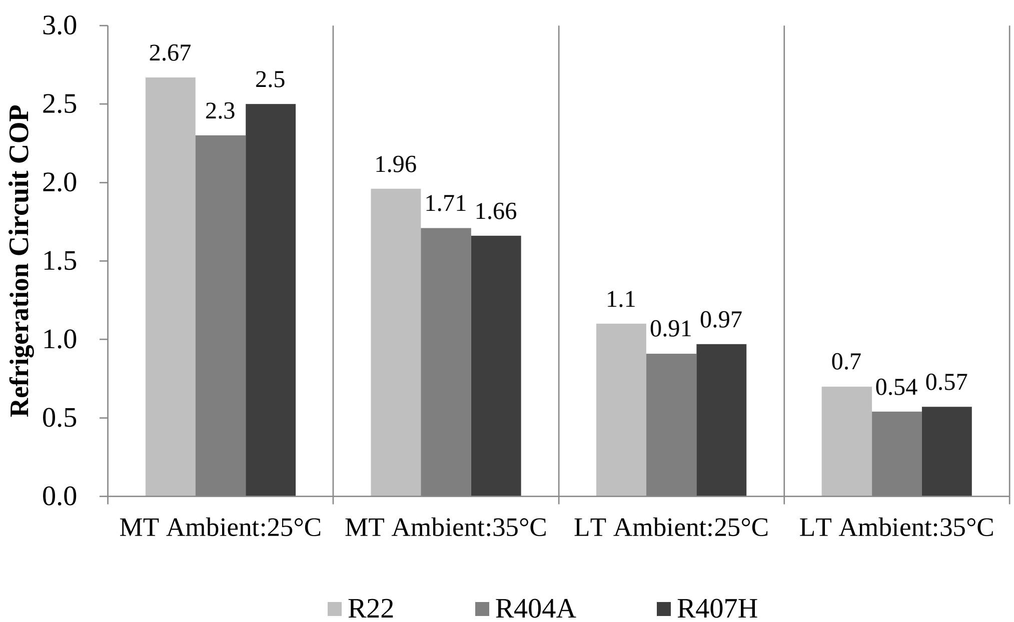

**Figure 6.** Commercial refrigeration system COP

Figure 6 demonstrates the relative nominal COPs for the three tested refrigerants. Except for the medium temperature test condition at 35°C Ambient R407H falls in between R22 and R404A.

## 4. DISCUSSION

### 4.1 Glide Behavior and Defrost

As a non-azeotropic refrigerant R407H is expected to demonstrate glide. This behavior was most observed during medium temperature operation, specifically in in the leading case 4. The entering and leaving temperature at the evaporator for the R22 and R404A was at -4.4°C and -3.7°C respectively. For the R407H the entering temperature was -7.7°C and the leaving temperature was -2.2°C. This affected the speed and location of the ice formation. The length of time for defrosts went from 2.5 hours for R22 and R404A to 3.5 hours for R407H with additional frosting at the inlet.

To achieve good system performance control modifications were done for the R407H system. In the experimental results presented the defrost was allowed to reach a higher termination temperature. Which was raised from 5.6°C to 10°C for the evaporator air outlet temperature. Alternatively, shortening the periods between the defrost cycles also achieved appropriate system behavior, as did adjusting the evaporator superheat setpoint. The change in defrost termination temperature did not affect the product simulator temperatures.

### 4.2 Modeled vs Experimental results

When comparing the predicted modeled results shown in table 3 and the experimental outcomes shown in figure 6 we can see a significant divergence in nominal COP. This can be explained by the losses associated with realistic systems as well as with the parameters of the test itself.

- The experiment relied heavily on data acquisition probes and mass flow meters at each case and at the condenser which adds additional pressure drop.
- Modeling results do not account for energy consumption associated with: condenser fans, evaporator fans, refrigerated case lighting, electric defrost
- Modeling results did not account for realistic compressor efficiency envelopes nor any potential VFD losses.

When comparing relative results however we can summarize in the table below

**Table 5:** Results Comparison as % of R22

| | MT Modeled COP | | | MT Experimental COP | | |
|---|---|---|---|---|---|---|
| | R22 | R404A | R407H | R22 | R404A | R407H |
| 35°C Condensing | 100.0% | 92.4% | 98.0% | 100.0% | 87.2% | 84.7% |
| 25°C Condensing | 100.0% | 95.5% | 98.4% | 100.0% | 86.1% | 93.6% |

| | LT Modeled COP | | | LT Experimental COP | | |
|---|---|---|---|---|---|---|
| | R22 | R404A | R407H | R22 | R404A | R407H |
| 35°C Condensing | 100.0% | 79.1% | 88.0% | 100.0% | 77.1% | 81.4% |
| 25°C Condensing | 100.0% | 90.1% | 96.0% | 100.0% | 82.7% | 88.2% |

The experimental results fit the initial expectations. R407H shows a slight improvement in COP over R404A at the medium temperature conditions and a significant improvement in the COP at the low temperature conditions. The exception to the trend are the results for the medium temperature at high ambient condition where R404A showed a better relative performance than predicted. This condition accounts for the highest refrigeration load in the experiment which we postulate can lead to some instability or capacity thresholds at the evaporator, especially when considering the effects of glide at the condenser most recently discussed by Makhnatch et. Al (2017)

## 5. CONCLUSIONS

R407H an A1 refrigerant with a GWP 1380 consisting of R134a, R32 and R125 was evaluated by independently in Urbana, IL for both medium temperature and low temperature refrigeration applications per the ASHRAE 72 standard. Based on modeling results and verified through experiment R407H exhibits a significant COP improvement over R404A in low temperature applications and a match or moderate improvement in medium temperature applications. Based on system performance and final product temperature R407H appears to be a suitable replacement for both R22 and R404A refrigerants in commercial refrigeration applications.

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