Evaluation and Soft-Optimization for R410A Low-GWP Replacement Candidates through Testing and Simulation

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EVALUATION AND SOFT-OPTIMIZATION FOR R410A LOW-GWP REPLACEMENT CANDIDATES THROUGH TESTING AND SIMULATION

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ABSTRACT

The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) announced an industry-wide cooperative research program to evaluate alternative refrigerants that have low global warming potential (GWP). R410A is a common refrigerant for air conditioning and heat pumping application but has a GWP of 2088. Drop-in tests of three R410A low-GWP alternative refrigerants (R32, D2Y60 and L41a) in a 3 ton split heat pump unit are performed according to ASHRAE Standard 116-1995 test matrix. The test matrix is expanded to include extended cooling and heating conditions which represent extreme weather conditions. The results show that R32 and L41a are good replacement candidates for R410A. However, the capacity of D2Y60 is lower than that of R410A by an average of 18% for cooling and 14% for heating. An in-house component based vapor compression system simulation tool is validated against the experimental data. The heat exchanger model used in the system simulation uses an in-house finite volume model. The compressor model uses the 10-coefficient R410A compressor map. A total of 32 experiments are used in the validation. The results of the validation are in good agreement with the experiments. Most of the predicted COP and capacity results lie within 5% of the measured values. The largest error is in the case of low temperature and extended condition tests because the mass flow rates were estimated and not measured. Soft optimization options, by varying the size of the compressor, are carried out to match the capacity to that of R410A. As the compressor size increases, the mass flow rate, power consumption, and capacity increase, while the COP decreases.

Keywords: R410A, R32, D2Y60, L41a, Heat Pumps, AREP, Soft-Optimization, VapCyc, CoilDesigner

1. INTRODUCTION

Two of the main challenges faced by the refrigeration and air conditioning industry are the ozone depletion (ODP) and global warming. Montreal Protocol has achieved a great success in phasing out refrigerants with ozone depletion potential. Kyoto Protocol deals with refrigerants having high global warming potential (GWP). Greenhouse gases, which cause the global warming, absorb the earth's radiation into atmosphere and thus increasing the earth's temperature. The reference value of 1 is used for the GWP of CO_2 based on a 100 years horizon.

R410A was introduced to replace R22 after Montreal Protocol and is a widely used refrigerant for residential applications. Since it has a GWP of 2088 (IPCC, 2007), various alternatives are being investigated. There are several studies in the literature for R410A replacements. For example, Xu *et al.* (2013) compared R410A and R32 in heat pumps with vapor injection cycle and found that the capacity and COP improvements using R32 can reach up to 10% and 9%, respectively. Piao *et al.* (2012) compared R410A to R1234yf, R32 and R32/R1234yf mixture in a unitary system and found that R32 is the best replacement for R410A without any system modifications. Chen *et al.* (2008) compared R410A to R32/R134a mixture and found that the COP can be improved by 8-9% with mixture in a new cycle configuration. The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) announced an industry-wide cooperative research program (AREP) to evaluate alternative refrigerants that have low GWP value. The program aims at testing several refrigerants for major product categories such as air conditioners, heat pumps, chillers and refrigeration equipment. The program consists of three types of testing: compressor calorimeter testing, system drop-in testing and soft-optimized system testing.

Soft optimization tests are carried out to match the capacity of alternative refrigerants to that of R410A. There are several cycle adjustments that can be made to enhance the system performance such as optimizing heat exchangers, optimizing the compressor or adding a suction line heat exchanger (SLHX). Without compressor optimization, using a larger displacement compressor for a given heat pump will increase the capacity as well as the power consumption. The COP, on the other hand, might increase or decrease based on the ratio of capacity-to-power increase. SLHX, which allows lower condensing temperature and thus increases capacity, have been investigated extensively in the literature. For example, Domanski *et al.* (1992) conducted a theoretical study on the performance benefits resulting from the addition of SLHX. They found that the benefit, or loss, of application of SLHX depends on operating conditions and fluid properties, with heat capacity being the most influential property. Klein *et al*. (2000) developed a dimensionless group to evaluate the performance of SLHX for different refrigerants. Mastrullo *et al.* (2007) developed a chart for evaluating the advantage of adding a SLHX.

The objectives of this paper are to (a) test the performance of an air conditioner and a heat pump using three low GWP refrigerants, R32, D2Y60 and L41a, and compare the results against the baseline refrigerant R410A, (b) validate a simulation model against the experiments and (c) explore soft optimization options using the validated model.

2. REFRIGERANTS COMPARISON

R410A is a binary mixture of R32 and R125 with GWP of 2088 and belongs to A1 safety group. R32 is a pure refrigerant which has a GWP of 675 and belongs to A2 safety group. D2Y60 is a binary mixture of R32 and R1234yf with a GWP of around 300 and belongs to A2L safety group. L41a is a ternary mixture of R32, R1234yf and R1234ze with a GWP of around 500 and belongs to A2L safety group (Johnson *et al.*, 2012). Using REFPROP 9.1 (Lemmon *et al.*, 2013) or property calculation tool from the refrigerant suppliers, there are significant differences in the thermophysical properties between R410A and the alternative refrigerants. For example, the vapor density of R410A is about 40% higher than the alternative refrigerants. D2Y60 and L41a have temperature glides of 5.8 K and 2.5 K in the evaporator, respectively. The condensing vapor pressure for D2Y60 is 20% less than that of R410A. Also, the liquid thermal conductivities of R32 and L41a are both about 30% lower than that of R410A. The latent heats of R32 and L41a are both about 25-50% higher than that of R410A.

3. EXPERIMENTAL TEST FACILITY

3.1 System Description

The heat pump unit that was tested is a 10.55 kW (3 ton) heat pump with rated SEER of 14 Btu/Wh and rated HSPF of 8.7. The unit has 27.5 cm^3 single speed scroll compressor. It has TXV for cooling and orifice for heating. The details of the testing unit and instrumentation can be found in Alabdulkarem *et al.* (2013).

3.2 Test Conditions

The test conditions followed ASHRAE Standard 116 (2010) as shown in Table 1. In addition, two extended conditions of 46.1ºC for cooling and -17.8ºC for heating were added to investigate the system behaviors at severe weather conditions.

Table 1: ASHRAE Standard 116 (2010) Test Matrix.

Test	Indoor		Outdoor		Operation
	db (°C)	wb (°C)	db (°C)	wb (°C)	
Extended Conditions	26.7	19.4	46.1	NA	Steady State Cooling
A			35		Steady State Cooling
B			27.8		Steady State Cooling
C		≤13.9			Steady State Cooling, Dry Coil
D					Cyclic Cooling, Dry Coil
High Temp. 2	21.1	≤15.6	8.3	6.1	Steady State Heating
High Temp. 1			16.7	14.7	Steady State Heating
Low Temp.			-8.3	-9.4	Steady State Heating
Extended Conditions			-17.8	NA	Steady State Heating
High Temp. Cyclic			8.3	6.1	Cyclic Heating
Frost Acc.			1.7	0.6	Steady State Defrost

4. PERFORMANCE EVALUATION

The air-side and refrigerant-side heat transfer capacities can be obtained from the experiment. The air side capacity is calculated by multiplying the air mass flow rate and the inlet and outlet air enthalpy difference in the closed air loop, as described by

$$Q_{air} = \dot{m}_{air} \Delta h_{air} \tag{1}$$

where $\dot{m}_{air}$ is the air mass flow rate in the closed air loop, and Δh_{air} is the inlet and outlet air enthalpy difference of the indoor coil.

Refrigerant side capacity is calculated by multiplying the refrigerant mass flow by the inlet and outlet enthalpy difference, given by

$$Q_{ref} = \dot{m}_{ref} \Delta h_{ref} \tag{2}$$

where $\dot{m}_{ref}$ is the refrigerant mass flow rate and Δh_{ref} is the refrigerant enthalpy difference of the inlet and outlet of the indoor coil. Energy balance was defined as the capacity difference between the refrigerant side and air side divided by refrigerant side capacity, given by

$$\text{Energy Balance} = (Q_{ref} - Q_{air})/Q_{ref} \tag{3}$$

The system cooling and heating coefficient of performance (COP) is defined in Equation (4) as

$$COP = Q/W \tag{4}$$

where W is the total power consumption of the heat pump system.

In order to consider the effect of the indoor blower on the system performance, power consumption data from the OEM was used. The average power consumption of the original blower matching the indoor coil is 373 W. The fan volume flow rate is set at 0.57 m^3/s (1200 CFM) according to the manufacturer specifications. The fan power consumption is deducted from the cooling capacity and added to the heating capacity since the fan is located in the psychrometric loop. It influences the net system capacity since the power input to the indoor fan is eventually converted to heat and dissipated into the air.

The compressor isentropic efficiency and volumetric efficiency are given by Equation (5) and Equation (6), respectively, as

$$\text{Isentropic Efficience, } \eta_{Isen} = (h_{out,s} - h_{in})/(h_{out} - h_{in}) \quad (5)$$

$$\text{Volumetric Efficiency, } \eta_{Vol} = \dot{m}_{ref} / (\rho \times V \times RPM) \quad (6)$$

where h is the enthalpy inlet/outlet of the compressor, ρ is the suction density and V is the compressor volume.

5. EXPERIMENTAL RESULTS AND DISCUSSION

5.1 Steady State Cooling Test Results Comparison

The comparisons of the steady state cooling test results are shown in Figure 1 and Figure 2. The results show that R32 has about 4% higher capacity than R410A in all test conditions except the C test condition where the capacity is almost the same. However, R32 resulted in lower COP than R410A due to the degradation in compressor efficiencies. The results of D2Y60 show that it has about 16-19% lower capacity than R410A. Furthermore, D2Y60 has about 2-7% lower COP than R410A. The results of L41aA show that it has about 6-10% lower capacity than R410A. Furthermore, L41a has about 0.8-5% lower COP than R410A.

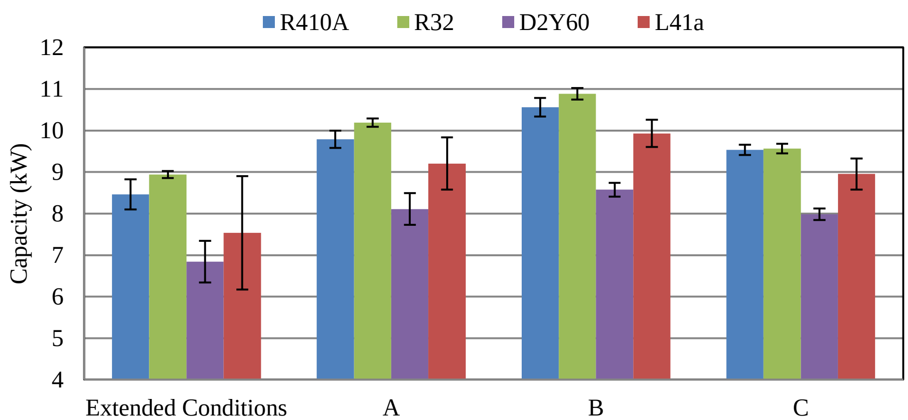

Figure 1: Cooling capacity test results of three alternatives vs. R410A.

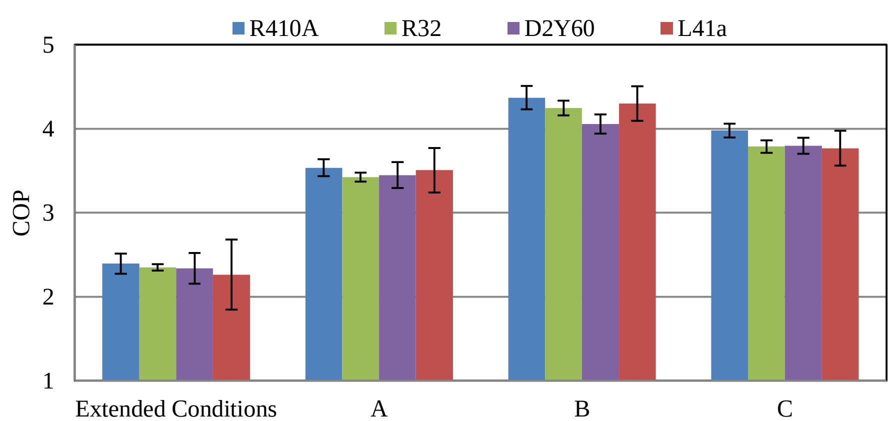

Figure 2: Cooling COP test results of three alternatives vs. R410A.

5.2 Steady State Heating Test Results Comparison

The comparisons of the steady state heating test results are shown in Figure 3 and Figure 4. The results show that R32 has 4% to 7.78% higher heating capacity in all test conditions except the extended test conditions where 4.25% degradation in capacity is observed. However, R32 resulted in lower COP than R410A in case of low temperature test and extended conditions test while the COP remains comparable for the other tests. The results of D2Y60 show that it has about 7-20% lower capacity than R410A. Furthermore, D2Y60 has about 4-8% lower COP than R410A in High Temp. 2 test and Extended test conditions. However, it has about 0.7-3% higher COP than R410A in High Temp. 1 and Low Temp. test conditions. The results of L41a show that it has lower capacity than R410A in two tests whereas it has higher capacity in two tests. Furthermore, L41a has lower COP than R410A in two tests whereas it has higher COP in two tests.

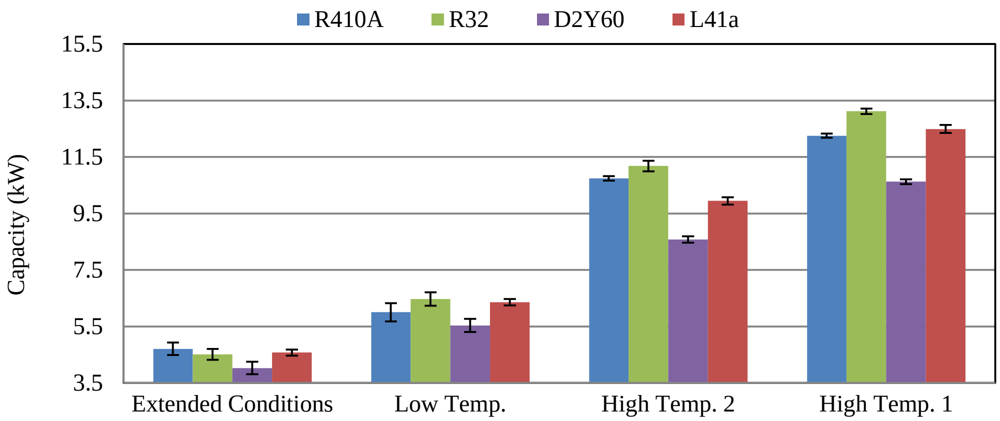

Figure 3: Heating capacity test results of three alternatives vs. R410A.

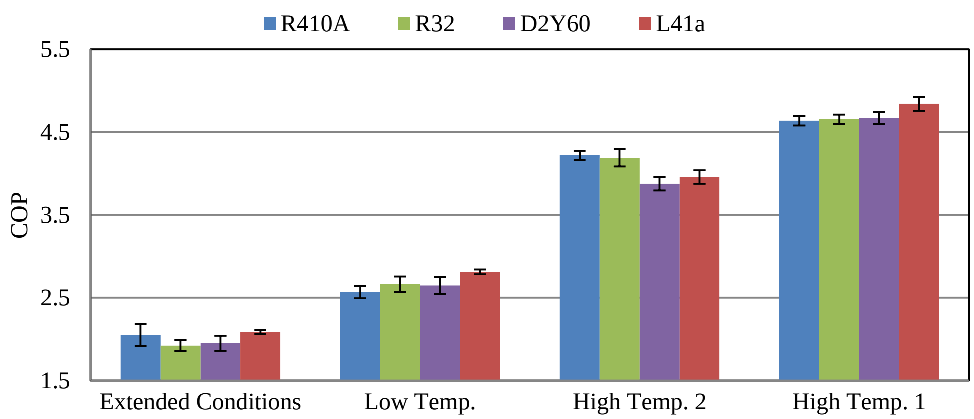

Figure 4: Heating COP test results of three alternatives vs. R410A.

6. MODEL DEVELOPMENT AND VALIDATION

6.1 Model Development

An in-house simulation tool for modeling various types of heat exchangers is used to validate the indoor and outdoor units of the experimental setup using 32 tests (Jiang *et al.* 2006). The validated units are then used by another in-house simulation tool which models split heat pump systems (Richardson *et al.* 2002, Winkler et al. 2008). In order to simulate the low GWP refrigerants, a user-defined mixture capability in the models is used. The condenser side air mass flow rate is not measured during the experiments and thus an energy balance is applied on the condenser to obtain the missing parameters. The evaporator inlet state is also not measured in the experiments. In the extended conditions and low temperature heating experiments, the refrigerant mass flow rate is not measured. However, the split heat pump model does not require the mass flow rate as an input. In the experiments, a flash tank is used to ensure a saturated vapor suction condition. In the model, an assumption of 0.01 K superheat suction is assumed for these tests. The compressor model used is a ten-coefficient database compressor model. The coefficients are supplied by the manufacturer for R410A while accurate coefficients for the other refrigerants are not readily available. In order to use the compressor model for the other refrigerants some adjustment factors are applied to the mass flow rate and power consumption shown in Table 2. The adjustment factors are calculated based on a component tester model as compared to the experiments data for each refrigerant.

Table 2: Compressor adjustment factors.

Refrigerant	Mass flow rate	Power
R410A	1	1.12
R32	1.22	0.69
D2Y60	0.86	0.72
L41a	1.08	0.71

6.2 Model Validation

The validation results of the cooling experiments are shown in Figure 5. Most of the results of R410A, R32, and L41a are within 5% of the experimental tests data. However, the mass flow rate of D2Y60 in the extended cooling test is within 9% of the experimental value. However, the other results of D2Y60 agree within 5% as well.

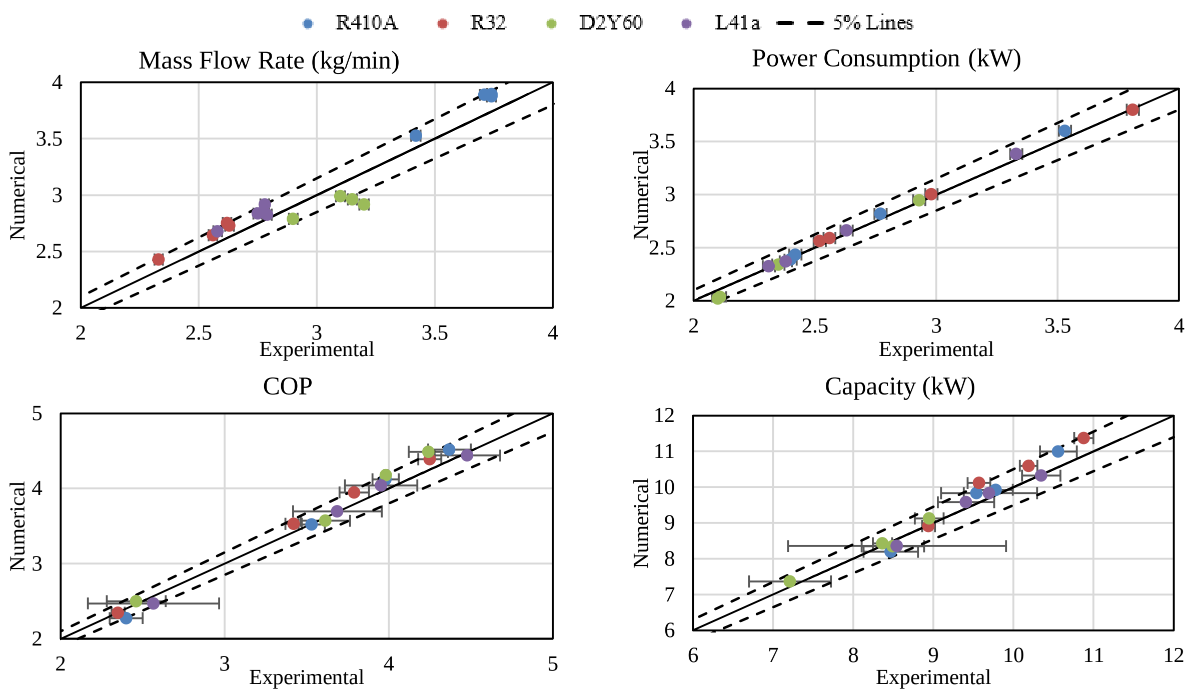

Figure 5: Cooling experiments validation results.

The validation results of the heating experiments are shown in Figure 6. As previously stated, the refrigerant mass flow rate is not measured in two of the four heating experiments. The mass flow rate of D2Y60 tests agree within 17%, while one of the L41a tests agree within 22%. This results show that a more accurate compressor performance prediction is required in order to be able to accurately model the low GWP refrigerant mixtures. However, most of the other results agree within 5% from the experimental data.

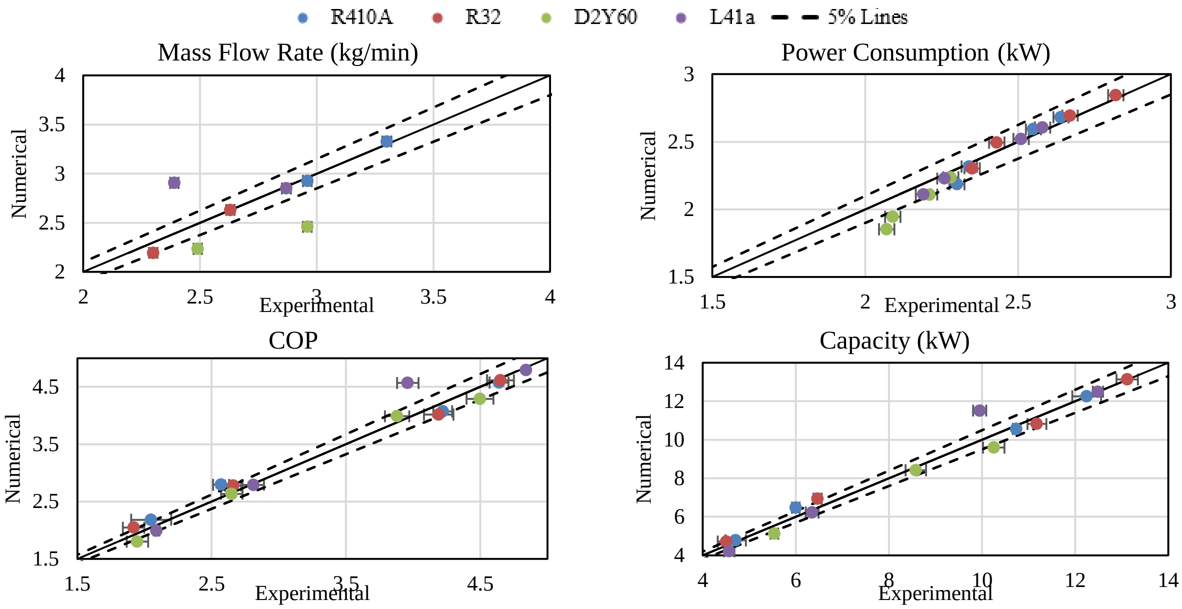

Figure 6: Heating experiments validation results.

7. SOFT OPTIMIZATION OPTIONS

7.1 Alternative Compressor Size

Since the capacity of D2Y60 is lower than that of R410A, the use of larger compressor sizes is investigated. Three compressors are investigated having 7.1%, 19%, and 29.7% larger capacity than the baseline compressor used in the experiments. Figure 7 shows the percentage difference in COP and capacity when using larger compressors than the baseline compressor for R410A and the alternative refrigerants in the cooling A test. All the other tests show similar results. The results show that as the compressor size increases, the capacity increases for all the refrigerants. However, the compressor power consumption increases at a higher rate than the capacity increases resulting in an overall deterioration of the COP.

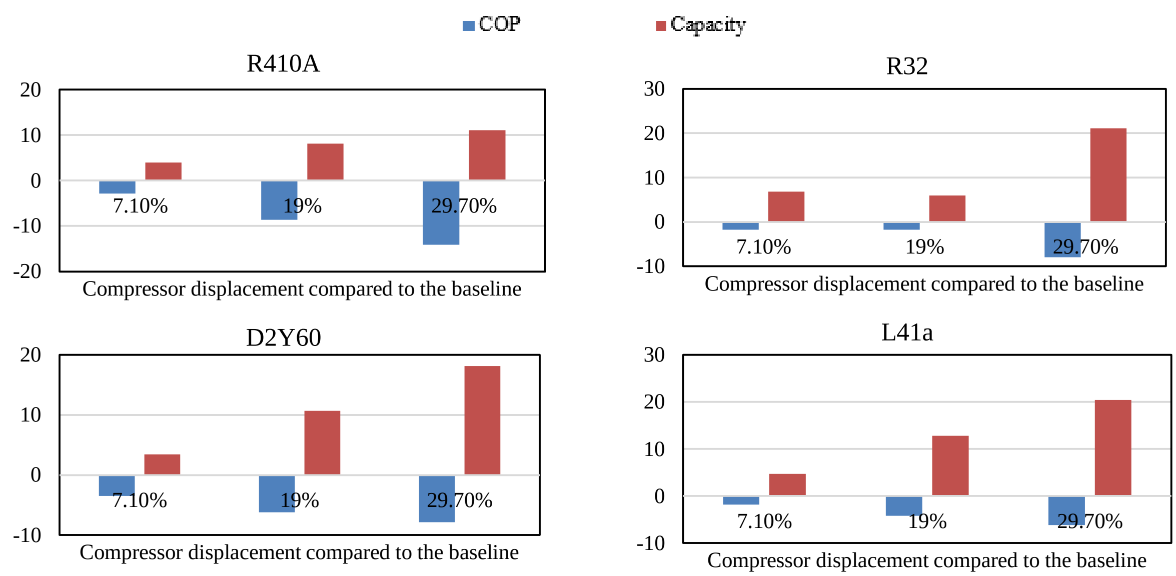

Figure 7: Larger compressors simulation results for R410A and alternative refrigerants in cooling A test.

8. CONCLUSIONS

R410A is compared against three low GWP refrigerants using a 10.55 kW heat pump. The test results show that R32 is superior compared to R410A in regards to cooling and heating capacities, while L41a is superior compared to R410A in regards to COP, and D2Y60 is superior compared to R410A in regards to compressor discharge temperature. Generally, L41a is superior in terms of SEER and HSPF values compared to other alternative refrigerants. It is also observed that there is a significant refrigerant charge reduction for all refrigerants compared to R410A.

The cycle model is developed and validated against the experimental data. Most of the model predictions are within 5% of the measured values. Larger compressor sizes are simulated showing an increase in capacity and power consumption, while the COP deteriorates. Further soft-optimization options will be investigated using the developed model. The prediction of the compressor performance for different refrigerants is necessary to better study the performance of alternative refrigerants and thus, must be further studied.

NOMENCLATURE

Abbreviations

P	Pressure
$\dot{m}$	Mass flow rate
h	Enthalpy
s	Entropy
$\dot{W}$	Power consumption
T	Temperature
ρ	Density
SLHX	Suction line heat exchanger
VCC	Vapor compression cycle
Ref	Refrigerant
RPM	Compressor rotational speed per minute
c	Specific heat
r	Latent heat of vaporization
COP	Coefficient of performance
Q	Cooling capacity

Greek symbols

η	Efficiency

Subscripts

in	Inlet
Out	Outlet

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