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# Cooling performance of R510A in domestic water purifiers †

Ki-Jung Park[1], Yohan Lee[2] and Dongsoo Jung[1,*]
[1]Department of Mechanical Engineering, Inha University, Incheon 402-751, Korea
[2]Graduate School, Inha University, Incheon 402-751, Korea

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**Abstract**

Cooling performance of R510A is examined both numerically and experimentally in an effort to replace HFC134a in the refrigeration system of domestic water purifiers. Although the use of HFC134a is currently dominant, it is being phased out in Europe and most developed countries due to its high potential contribution to global warming. To solve this problem, cycle simulation and experimental measurements are conducted with a new refrigerant mixture of 88%RE170/12%R600a using actual domestic water purifiers. This mixture has been recently numbered and listed as R510A by ASHRAE. Test results show that, due to the small internal volume of the refrigeration system of the domestic water purifiers, system performance with R510A is greatly influenced by the amount of charge. With the optimum charge amount of 20 to 21 g, approximately 50% that of HFC134a, the energy consumption of R510A is 22.3% lower than that of HFC134a. The compressor discharge temperature of R510A is 3.7℃ lower than that of HFC134a at the optimum charge. Overall, R510A, a new, long term, and environmentally safe refrigerant, is a good alternative for HFC134a. Furthermore, it requires only minor changes in the refrigeration system of the domestic water purifiers.

*Keywords*: Alternative refrigerants; Domestic water purifiers; Alternative refrigerants; R510A; HFC134a; Ozone depletion; Global warming

## 1. Introduction

For the past few decades, chlorofluorocarbons (CFCs) have been widely used in various refrigeration and air-conditioning applications. However, CFCs have been found to be responsible for the destruction of the stratospheric ozone layer. In 1987, the Montreal Protocol proposed to phase out these ozone-depleting substances [1]. CFCs were completely phased out in developed countries in 1996, while complete removal in developing countries is expected to be achieved in 2010.

In order to fill the gap caused by the phase-out of CFCs, the refrigeration and air-conditioning industry has carried out extensive research and development activities to find alternative refrigerants with zero ozone depletion potential (ODP). As a result of these efforts, for the past decade, HFC134a has been successfully developed and adopted in new domestic refrigerators and water purifiers. HFC134a demonstrates similar vapor pressure and performance as CFC12.

Recently, global warming has become one of the most important issues facing mankind. In 1997, the Kyoto Protocol was proposed to control greenhouse gases (GHG), including HFCs [2]. Consequently, HFC134a was identified as one of the controlled GHG. The global warming potential (GWP) of HFC134a is 1430, as compared to that of $CO_2$, which has a GWP of 1. Therefore, it needs to be replaced by more environmentally safe, long-term refrigerants in the near future. In fact, the European Union F-Gases Regulation and the Mobile Air-Conditioning Directive have banned the use of HFC134a in automobile air-conditioners of newly manufactured vehicles starting in 2011 [3]. The Mobile Air-Conditioning Directive specifically prohibits the use of refrigerants with GWP higher than 150.

Due to the rapidly changing environmental and energy situation, more energy efficient domestic refrigerators and water purifiers charged with a refrigerant with GWP of less than 150 are essential. In order to maintain the system change to a minimum level for cost effective manufacturing of such products, conventional vapor compression refrigeration technology with similar compressor size and technology is necessary. One way of achieving this goal with the present technology is to use refrigerant mixtures of similar vapor pressure with low GWP.

Owing to this trend of banning the use of HFC134a in refrigeration and air-conditioning equipment, many countries have adopted flammable hydrocarbon refrigerants. Traditionally, flammable refrigerants have not been accepted in normal refrigeration and air-conditioning applications due to safety concerns. This trend, however, has been recently relaxed due

† This paper was recommended for publication in revised form by Associate Editor Ji Hwan Jeong
*Corresponding author. Tel.: 82 32 860 7320, Fax.: 82 32 868 1716
E-mail address: dsjung@inha.ac.kr

to an environmental mandate. Some flammable refrigerants have been applied to specific applications as pure working fluids, or as one of the components of mixed working fluids [4, 5]. For instance, R600a (isobutane) has dominated the European refrigerator/freezer sector for the past decade [6]. In fact, hydrocarbons are known to offer such advantages as no ODP, low GWP (typically less than 3), low cost, availability, and compatibility with conventional mineral oil [4, 5].

Some studies have shown the difficulty of finding pure alternative fluids to replace existing refrigerants unless major changes, including a compressor redesign, are undertaken [7, 8]. One of the ways of avoiding a major system change is the use of refrigerant mixtures that can tune thermodynamic properties, such as vapor pressure, to provide similar capacity. Refrigerant mixtures such as R407C and R404A are good examples, and are currently popularly used for R22 and R502 applications. For these mixtures, similar size compressors can be used in the same system. In particular, refrigerant mixtures of small temperature glide during the phase change, often called near-azeotropes, and with similar vapor pressure would be the best choice due to a decreased chance of fractionation caused by a leakage in the system.

In applying alternative refrigerants, determination of the optimum amount of charge is very important. Houcek and Thedford [9] found that there is an optimum amount of charge for highest coefficient of performance (COP) and capacity. Frazad and O'Neal [10] made measurements with a capillary tube in an air-conditioner and observed that the charge has a great influence on the overall performance of the system. Choi and Kim [11] also observed that the change in mass flow rate resulting from a change in charge affects the system performance.

The objective of this paper is to measure the performance of a new refrigerant mixture of R510A in the refrigeration system of domestic water purifiers in order to gauge its suitability to substitute for HFC134a. R510A is a binary mixture, which is an azeotrope composed of 88% RE170 and 12% R600a by mass, without a temperature glide, and has recently been listed by ASHRAE [12]. It has no ODP and no GWP; hence, it can be used as a long-term alternative refrigerant in domestic water purifiers. In this study, reference baseline pull-down tests are performed in the original water purifier charged with HFC134a. Pull-down tests are then performed on the same unit charged with R510A for various amounts of refrigerant charges. The results are compared with the baseline test results.

## 2. Thermodynamic cycle analysis

Before the experiments were conducted, thermodynamic performance of R510A was compared to that of HFC134a by thermodynamic cycle analysis. Fig. 1 shows a simplified two-dimensional sketch of the water purifier. A water container is placed at the top of the purifier and evaporator coil is located inside the water container. As the refrigeration operation begins, water gives out heat to the refrigerant in direct contact

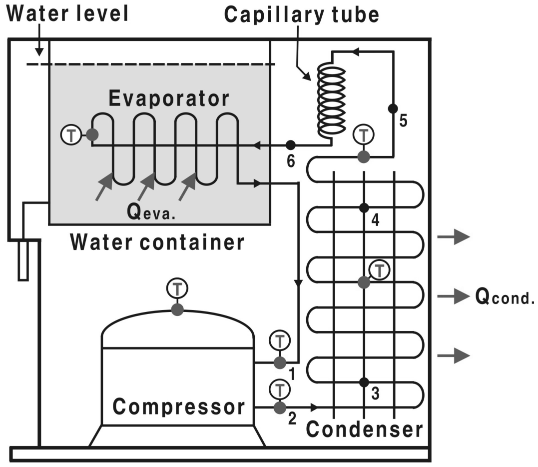


Fig. 1. Refrigeration system for domestic water purifier.

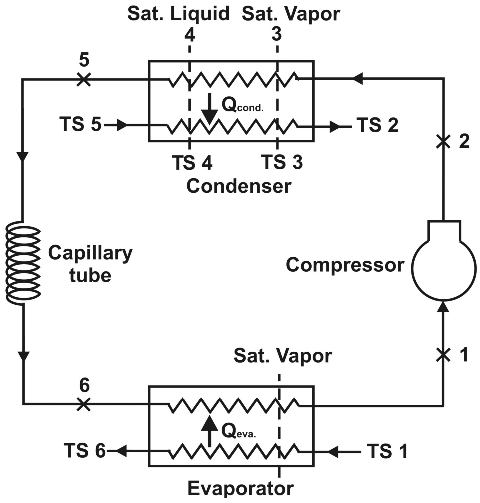


Fig. 2. Schematic of a single evaporator refrigeration system for water purifiers.

with the evaporator. Ice is formed on the outside the evaporator coil. A hermetic compressor is used for cooling and a capillary tube is used as an expansion device. A natural draft condenser is placed at the back of the purifier.

Fig. 2 shows the schematic of a typical single-evaporator refrigeration system for water purifiers. Due to the heat exchange in the water container, evaporation occurs. Superheated vapor usually leaves the evaporator at State 1. During evaporation, the refrigerant temperature rises for mixtures (gliding temperature effect), while the temperature remains constant for pure components if pressure drop is not considered. The vapor is compressed with work addition, and becomes superheated vapor at high pressure and temperature (State 2). This vapor is desuperheated and condensed in the condenser. Generally, subcooled liquid or slightly saturated two-phase fluid leaves the condenser at State 5. Finally, expansion occurs through a capillary tube to complete the cycle. The two-phase refrigerant then enters the evaporator at State 6.

In thermodynamic simulation, a simple UA model was em-

Table 1. Simulation input conditions.

| | |
|---|---|
| Evaporator temperature | -5℃ |
| Condenser temperature | 50℃ |
| Evaporator superheating | 5℃ |
| Condenser subcooling | 5℃ |
| UA (Evaporator) | 12.9 W/C |
| UA (Condenser) | 7.7 W/C |
| Compressor efficiency | 0.7 |
| Cooling load | 150 W |

Table 2. Simulation results.

| Refrigerant | COP | Difference in COP (%) | VC (kJ $m^{-3}$) | Difference in VC (%) | $T_{dis}$ (℃) |
|---|---|---|---|---|---|
| HFC134a | 1.84 | | 1348 | | 93.2 |
| R510A | 2.15 | 16.9 | 1415 | 5.0 | 101.6 |

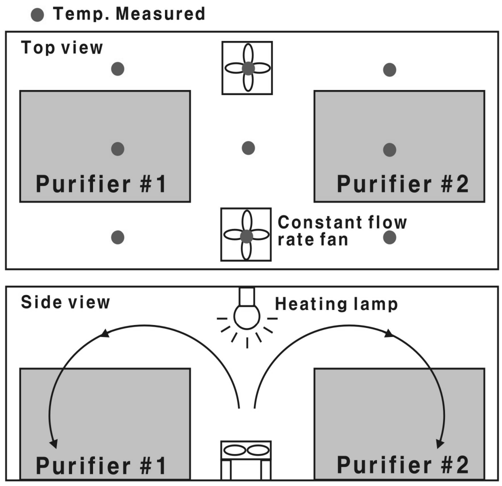


Fig. 3. Schematic of the interior of the constant temperature chamber.

ployed with a constant cooling load for both HFC134a and R510A. This is similar to the experiment by Jung and Radermacher [13]. In the model, the evaporator and condenser were specified by the product of an overall heat transfer coefficient and an area (UA). Typically, the evaporator and condenser saturation temperatures for HFC134a are -5℃ and 50℃, respectively and UA values are adjusted to yield similar temperatures for HFC134a. For R510A, the same UA values were imposed in the condenser.

The compressor is modeled by simply specifying an isentropic efficiency. The unknowns are determined by solving a set of nonlinear equations by the Newton-Rahpson method. This method requires initial guesses for the variables and iteratively finds the solution by forcing the residual equations to become zero. Since Jung and Radermacher [13] have presented all the details of the simulation calculations, details will not be repeated here. For reference, Table 1 lists the simulation input variables.

Cycle analysis was performed for both HFC134a and R510A. All thermodynamic properties needed for the simulation were computed by REFPROP program [14]. Table 2 summarizes the simulation results. Cycle simulation indicates that R510A offers significant increases in COP and capacities of 16.9% and 5.0% respectively while the compressor discharge temperature is 8.4℃ higher than that of HFC134a.

## 3. Experiments

In this study, a constant temperature chamber was built to carry out pull-down tests with HFC134a and R510A in domestic water purifiers. The heat needed to maintain the constant temperature inside the chamber was supplied by two 100 W incandescent lamps installed on the upper portion of the chamber. Two small fans were likewise installed for air circulation. Fig. 3 shows the schematic of the chamber for the pull-down tests. Temperatures were measured at nine locations inside the chamber. During the tests, water purifiers were placed such that the natural draft condensers of the purifiers faced against the wall with 50 mm space as in actual typical installation.

Pull-down tests were performed using two purifier units. One purifier unit was used as a reference unit for all tests to check the reliability of the chamber and test methods. It was placed without any modification inside the chamber during the entirety of the tests. Another unit was used for actual pull-down tests with HFC134a and R510A.

Before the tests, the interior chamber temperature and water temperature inside the water container were set to 35℃ and 30℃ respectively. The power compressor was then placed into the units. The tests were completed when the water temperature dropped from 30℃ to 5℃. For all tests, the volume of the water inside the container was maintained at 8.2 L.

As shown in Fig. 1, six temperatures were measured in the refrigeration cycle during the tests, as follows:

(1) Evaporator inside the water container
(2) Compressor suction
(3) Compressor discharge
(4) Condenser exit
(5) Condenser center
(6) Compressor dome

For all tests, T-type thermocouples of ±0.1℃ accuracy were used to measure all temperatures. The instantaneous power input to the compressor was measured by a Watt meter of ±0.2% accuracy, and the total power consumption was obtained by integrating the instantaneous power. During the tests, all temperatures and power input values were acquired by a data logger (HP34970) every 10 s and stored in a computer.

For R510A, the optimum charge was not known before the experiments. Hence, an initial charge was determined approximately by comparing the liquid densities of HFC134a and R510A. According to the REFPROP refrigerant property routine [14], liquid densities of HFC134a and R510A at 50℃ were 1102 and 599 kg/$m^3$, respectively. Based on these values,

Table 3. Summary of test results.

| Refrig-erant | Charge (g) | Starting tempera-ture (℃) | Ending tempera-ture (℃) | Operating time (min) | Energy consump-tion (W·h) | Differ-ence in energy consump-tion (%) | $T_{dis}$ (℃) |
|---|---|---|---|---|---|---|---|
| HFC134a | 42 | 30.2 | 5.6 | 109.7 | 228.2 | | 93.5 |
| R510A | 18 | 30.2 | 12.8 | 149.5 | 190.4 | -16.6 | 81.4 |
| | 19 | 30.2 | 5.7 | 130.5 | 197.8 | -13.3 | 89.6 |
| | 20 | 30.2 | 5.7 | 111.3 | 177.3 | -22.3 | 89.8 |
| | 21 | 30.1 | 5.6 | 110.7 | 183.6 | -19.5 | 91.1 |

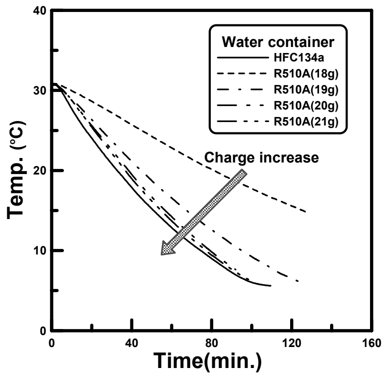


Fig. 4. Water temperature during pull-down tests.

the initial charge of R510A was determined to be approximately 23 g. During the tests, a variety of measurements was taken for various amounts of charge with an interval of 1 g for R510A.

Since the objective of this study was to examine the drop-in feature of R510A, no change was made to the system during the study, including compressor, oil, and capillary tube. As for the tests of reference fluid of HFC134a, commercial mass production units currently available were chosen without any changes.

## 4. Results and discussion

Table 3 lists the test results for HFC134a and R510A. For these refrigerants, tests were repeated at least three times to confirm the data. All data in Table 3 have good repeatability, showing less than 1% scatter in energy consumption. At the beginning, an initial amount of 23 g of R510A was charged into the purifier. Test results, however, showed that the system was overcharged severely when the charges of 22-23 g were used. Therefore, in this section, tests results with 18-21 g of R510A with 1 g interval are presented.

Fig. 4 shows the water temperature inside the container during the pull-down tests. As seen in the figure, the pull-down was greatly influenced by the amount of charge. Since the internal volume of the refrigeration system was very small, even 1 g made a significant difference in pull-down test results.

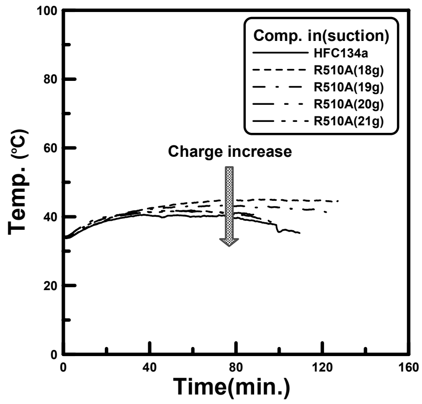


Fig. 5. Evaporator exit temperature during pull-down tests.

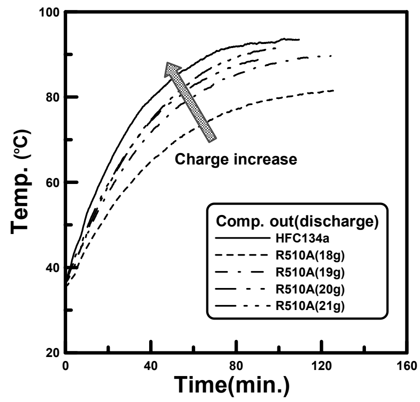


Fig. 6. Compressor discharge temperature during pull-down tests.

Tests results indicate that the optimum charge for R510A is approximately 20 to 21 g. For these charges, the pull-down time was similar to that of HFC134a, and the energy consumption (EC) was 22.3% lower than that of HFC134a. The cycle simulation prediction showed that the coefficient of performance (COP) of R510A would be 16.9% higher than that of HFC134a. Actual performance data, however, showed 22.3% increase in energy efficiency. The difference may be explained by the good oil miscibility and heat transfer with R510A.

Figs. 5 and 6 show the temperatures at the evaporator exit and compressor discharge. In Fig. 5, the system was undercharged with 18 and 19 g of R510A, which was indicated by large evaporator superheat. For the charge of 20 g of R510A, the average compressor discharge temperature was 3.7℃ lower than that of HFC134a. This is shown in Fig. 6 and Table 3, and indicates that R510A would be more reliable than HFC134a in the compressor. This is a good feature for long-term reliability of the system.

Figs. 7 and 8 show the temperatures at the exit and center of the condenser. When the amount of charge increased, these two temperatures likewise increased. For the charges of 20 and 21 g, the temperatures were similar to those of HFC134a.

Finally, Fig. 9 shows the compressor dome temperatures. With the increase in the amount of charge, the compressor

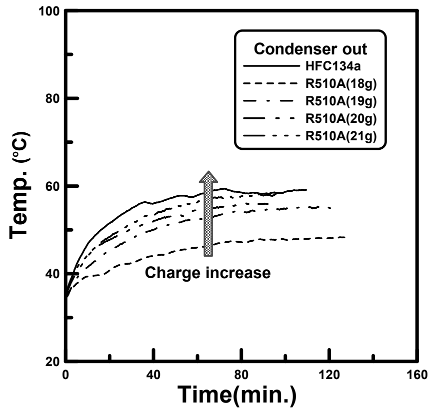


Fig. 7. Condenser exit temperature during pull-down tests.

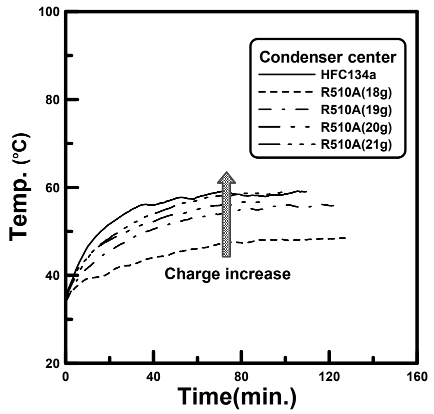


Fig. 8. Condenser center temperature during pull-down tests.

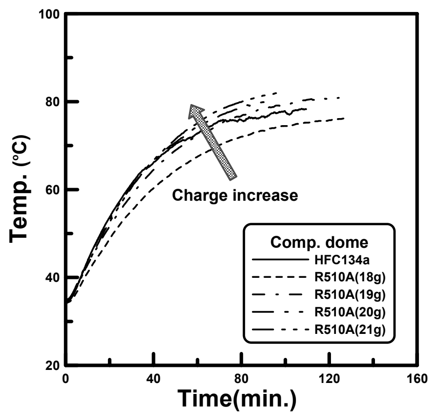


Fig. 9. Compressor dome temperature during pull-down tests.

dome temperature increased due to increased power input to the compressor.

## 5. Conclusions

In this study, the performance of new refrigerant R510A, as well as HFC134a, was measured in an environmental chamber during the pull-down tests. Based on the test results, the following conclusions can be drawn:

(1) Due to the small internal volume of the refrigeration system in water purifiers, the system performance was greatly influenced by the amount of charge.
(2) With the optimum charge amount of 20 to 21 g, approximately 50% that of HFC134a, the energy consumption of R510A was 22.3% lower than that of HFC134a.
(3) For the charge of 20 g of R510A, the compressor discharge temperature of R510A was 3.7℃ lower than that of HFC134a.
(4) Overall, R510A is a good alternative for HFC134a in domestic water purifiers. R510A is a new, ASHRAE-listed, long-term, and environmentally safe refrigerant that requires little change in the system. Capillary tube optimization and the use of proper lubricant may further increase the system performance.

## Acknowledgement

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## References

[1] United Nations Environmental Programme, Montreal protocol on substances that deplete the ozone layer, Final act, United Nations, New York, USA, (1987).
[2] Global Environmental Change Report, A brief analysis of the Kyoto protocol, IX (24), (1997).
[3] Official Journal of the European Union, Directive 2006/40/EC of the European parliament and of the council, (2006).
[4] H. Kruse, The state of the art of the hydrocarbon technology in household refrigeration, Proceedings of the International Conferences on Ozone Protection Technologies, Washington, DC, USA, (1996) 179-188.
[5] D. Jung, C. Kim, K. Song and B. Park, Testing of propane/isobutane mixture in domestic refrigerators, *Int. J. Refrigeration,* 23 (2000) 517-527.
[6] Int. Energy Agency's Heat Pump Center, Informative fact sheet: Hydrocarbons as refrigerants in residential heat pumps and air-conditioners, (2002).
[7] J. J. Jetter, N. D. Smith, K. N. Ratanaphruks, M. W. Tufts and F. R. Delafield, Evaluation of alternatives for HFC-134a refrigerant in motor vehicle air conditioning, Proceedings of the International Conferences on Ozone Protection Technologies, Washington, DC, USA, (1996).
[8] J. J. Jetter and F. R. Delafield, Alternatives for CFC12 refrigerant in automotive air conditioning, Proceedings of the International Conferences on Ozone Protection Technologies, Baltimore, USA, (1997).
[9] J. Houcek and M. Thedford, A research into a new method of refrigeration charging and the effects of improper charging, Proceedings of the First Annual Symposium on Efficient Utilization of Energy in Residential and Commercial

Buildings, Texas A&M Univ., TX., USA, (1984).
[10] M. Frazad and D. L. O'Neal, System performance characteristics of air conditioner over a range of charging conditions, *Int. J. Refrigeration,* 14 (1991) 321-328.
[11] E. Choi and J. Kim, The effects of the refrigerant charge on the performance of an air conditioner with capillary tube expansions, *Korean Journal of Air-Conditioning and Refrigeration Engineering,* 14 (5) (2002) 359-364.
[12] ANSI/ASHRAE Addenda a, b, c, d, e, f, g, and h to ANSI/ASHRAE Standard 34-2007, Designation and safety classification of refrigerants. ASHRAE, (June) (2007).
[13] D. Jung and R. Radermacher, Performance simulation of single-evaporator domestic refrigerators charged with pure and mixed refrigerant. *Int. J. Refrigeration,* 14 (1991) 223-232.
[14] E. W. Lemmon, M. L. Huber and M. O. McLinden, NIST Reference Fluid Thermodynamics and Transport Properties, REFPROP version 8.0, (2007).

**Ki-Jung Park** received Ph.D. degree in mechanical engineering from Inha University in 2009. He was a postdoctoral fellow at Inha University. He is currently a guest researcher at National Institute of Standards and Technology (NIST). His primary research interests are refrigeration and heat pumping system design and experimental two phase heat transfer.

**Dongsoo Jung** is currently a professor in mechanical engineering at Inha University. His primary research interests are refrigeration and heat pumping system design and experimental two phase heat transfer.