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# Development of High Efficiency Carbon Dioxide Commercial Heat Pump Water Heater

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## ABSTRACT

Although heat pump water heaters are today widely accepted in both Japan and Europe, where energy costs are high and government incentives for their use exist, acceptance of such products in the US has been limited. While this trend is slowly changing with the introduction of heat pump water heaters into the residential market, but acceptance remains low in the commercial sector. The objective of the presented work is the development of a high efficiency R744 heat pump water heater for commercial applications with effective utilization of the cooling capability for air conditioning and/or refrigeration. The ultimate goal is to achieve total system COP of up to 8. This unit will be targeted at commercial use where some cooling load is typically needed year round, such as restaurants, hotels, nursing homes, and hospitals. This paper presents the performance results from the development of four R744 commercial heat pump water heater packages of approximately 35 kW and comparison to a commercially available baseline R134a unit of the same capacity and footprint. In addition, the influences of an internal heat exchanger and an enhanced evaporator on the system performance are described and recommendations are made for further improvements of the R744 system.

## 1. INTRODUCTION

Heat pump water heaters (HPWH) are today widely accepted in both Japan and Europe, where energy costs are high and government incentives for their use exist, acceptance of such a product in the US has been slow, with a few thousand units sold per year for the past 10 years (BRG Consulting, 2009). Barriers to HPWH acceptance have historically been performance, reliability, as well as initial and operating costs. The dominant styles of water heaters used today in the U.S. are still electric and gas, split roughly 50/50 in market share. The technology for these systems is quite mature, and all have primary energy efficiencies less than one. Commercial water heaters are rated by a combination of thermal efficiency (essentially heating efficiency) and passive losses to ambient, called stand-by losses. Heat pumps used for water heating are essentially refrigeration machines and have been in use for many years. Thermal efficiency (COP) of electrically driven heat pumps using conventional refrigerants (R22, R134a, R410A) are in the 3 to 5 range, compared to 0.8 to 0.95 seen in electric and gas water heaters.

The use of the natural refrigerant R744 as a refrigerant for heat pumps is relatively new and started first in the automotive industry, but quickly moved to residential HPWHs in the mid 1990's in Japan with sponsorship from the Japanese government. Many Japanese manufacturers have now fully commercialized residential R744 HPWHs (collectively called "Eco-Cute") in Japan, with reported COPs ranging from 4.1 to 4.8. These reliable systems have become fully accepted as the most promising technology for reducing Japan's dependence on oil. They are typically integrated units that take heat from the surrounding low temperature air (source), transferring it to the high temperature water within the tank (sink). However, in these Japanese units, the evaporator is installed outdoors, unlike integrated HPWHs in Europe and North America. Volumes have steadily increased to a few hundred thousand units per year with the help of generous government and utility incentives, essentially reducing the retail cost by 50%, to the same as an electric water heater. Although the dramatic increase in residential HPWH sales in Japan has mainly been due to these fiscal incentives (and the fact that water heating accounts for more than 35% of the home's energy use in Japan), what has undoubtedly helped this increased acceptance has been the use of R744 as the refrigerant of choice, enabling high efficiency operation down to significantly colder ambient temperatures. R744 HPWH residential units are also starting to gain traction in Europe over the past few years, mainly because of huge government incentives. Here, the target market seems to be combined space heating (hydronic) and hot water, not just hot water: this "double-use" will undoubtedly be realized in North America too, further improving the cost-benefit equation for the customer.

In early 2000, United Technologies Research Center started to build up prototype R744 commercial HPWH's in Europe based on their R410A model. In 2004, they started a 2 year DOE sponsored program that looked to minimize the barriers for commercializing this unit for the US market (Radcliff, 2007). Through exhaustive interviews with contractors and consulting engineers, they determined that the top two customer considerations when choosing a commercial water heating system were performance and reliability (the cost of lack of hot water is very high), followed closely by high installation costs not justifying the low hot water expenses and energy cost savings. They also determined that the combined use of cooling and hot water did add to the attraction of the product and that the use of R744 would achieve the required performance (and if adopted, save 5 million metric tons of $CO_2$ emissions annually). If one considers the primary energy used, it can be seen that HPWH's presents a possible 50% or more increase in energy efficiency over a condensing gas or electric storage type water heater, the best conventional water heater available today. The potential national energy use and greenhouse gas emission impact for commercial water heating is large. Approximately 140,000 commercial water heaters are sold in the US every year, split equally between gas and electric. Commercial water heating utilizes 1.1% of the total energy consumption in the US and on average, 11% of the total annual commercial energy consumption (D & R Consulting, 2008). One of the most attractive features of an R744 HPWHs is the ability to achieve relatively high water temperatures without the additional need for electric heaters. Stene (2007) noted that using R744 allowed for the production of hot water in the temperature range from 60 to 85ºC while obtaining the highest possible COP for a HPWH system.

This paper presents a comparison between a commercially available R134a HPWH with several development stages of a R744 HPWH of the same footprint. The replacement of the working fluid R134a with R744 was done to improve system performance characteristics but at the same time to show the influence of the higher volumetric capacity of R744 compared to R134a. The ultimate goal of this work is to achieve a combined COP of heating and cooling of up to 8 for typical operating condition.

## 2. EXPERIMENTAL FACILITIES

The baseline R134a unit was a commercially available HPWH marketed toward applications in which the unit would be located indoors. This baseline unit is a packaged air-source HPWH with a nominal heating capacity of 35 kW and COP of 3.9. A picture and a simple schematic of the R134a baseline unit are shown in Figure 1. A scroll compressor was used with a condenser of brazed plate design. The evaporator was a round tube plate fin design, and the expansion device was a thermostatic expansion valve.


**Figure 1:** Baseline R134a Heat Pump Water Heater Unit

In addition to possible performance benefits of using R744 as the working fluid, the high volumetric capacity of R744 enables the construction of the same capacity unit with a substantially reduced system volume. Displaying

this potential was a secondary aim of this work beside the COP improvement process. To achieve this, the R744 system was assembled in a frame with the same footprint as the baseline R134a unit, and the volume reduction was demonstrated in the reduced height of the unit. This reduction was primarily achieved through the reduction of heat exchanger size. The reduction of the evaporator volume was achieved by using an evaporator coil originally designed for a 17 kW R134a system. This resulted in a 40 % reduction in face area, primarily in height, and a 55 % reduction in evaporator volume.
For the following development steps of the performance improvement process of the R744 HPWH, a bigger evaporator which basically uses the full height of the R134a HPWH housing was used. This shows the potential of the R744 HPWH when using the full height offered by the R134a housing. For this purpose the baseline R744 evaporator volume was enlarged by 50 % creating an evaporator of almost the same height as the R134a evaporator. A comparison of all three evaporators and the two condensers/ gas coolers is shown in Figure 2.

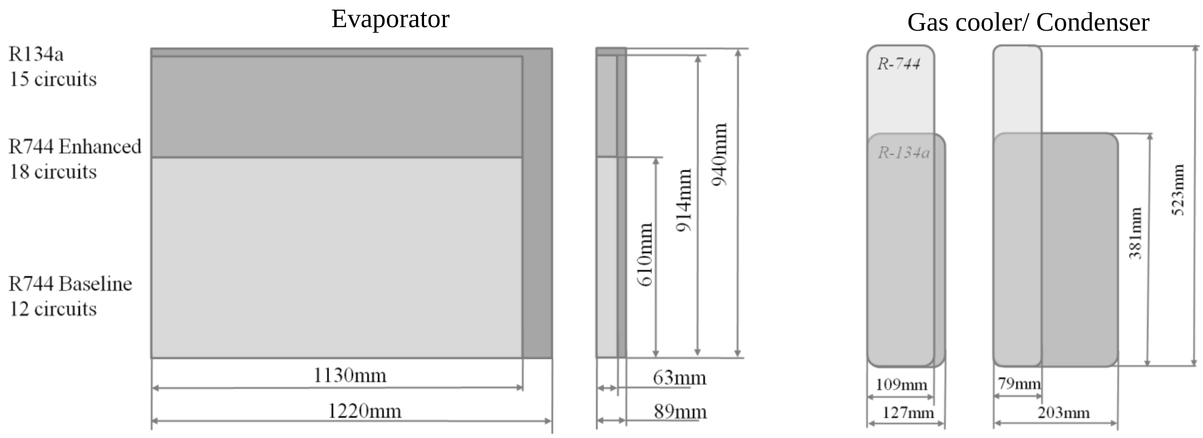


**Figure 2:** Heat exchanger dimensions

The R744 gas cooler was a commercially available model with a much narrower design compared to the R134a condenser. The reduction of the gas cooler volume was approximately 50 %. An R744 compressor that would provide similar capacity at the rating condition for the baseline R134a was chosen. This compressor was of a semi-hermetic reciprocating design. It has been demonstrated several times before (Bullard, 2004 and Elbel and Hrnjak, 2008) that the performance of transcritical R744 system can be optimized using the high side pressure. In order to allow for the ease of this optimization at each test condition, an electronic expansion device was installed that allowed the high side pressure to be varied during testing. The same model of blower was used to move air over the evaporator in both systems, at a constant volumetric flow rate of 1800 l/s. For the R134a system and the R744 with enhanced evaporator, this resulted in an evaporator face velocity of approximately 1.5 m/s. The face velocity in the R744 baseline system with the smaller evaporator was 2.60 m/s.
In addition to the three systems described above, an investigation of the R744 baseline and enhanced evaporator system with internal heat exchange was done. These systems were the same in all respects except for the addition of an internal heat exchanger between the liquid line and the suction line. This internal heat exchanger was designed to have an effectiveness of approximately 70%. The systems that were investigated were as follows:

- R134a HPWH
- R744 HPWH baseline system
- R744 HPWH with IHX
- R744 HPWH with enhanced evaporator
- R744 HPWH with IHX and enhanced evaporator

A schematic of the instrumentation installed the experimental facility of the R744 HPWH with IHX is shown in Figure 3.
The heat pump system was instrumented in such a way as to achieve five energy balances, two on the cooling side of the cycle, and three on the heating side of the cycle. On the cooling side of the cycle, the two balances are achieved on the air stream and the refrigerant stream, respectively. Determination of the cooling capacity on the air stream is determined using a separate wind tunnel directly connected to the evaporator air discharge of the heat pump unit. This wind tunnel was built and instrumented according to ASHRAE Standard 37-2005. While the heat pump unit is equipped “off the shelf” with a blower to provide air flow over the evaporator, this blower is not strong enough to provide the pressure head required to overcome the pressure drop caused by the flow nozzles used to

determine air flow rate. For this reason, an additional "helper" blower has been installed at the exit of the wind tunnel to provide the additional pressure lift required to maintain the flow rate provided by the blower integrated into the heat pump water heater unit. The second determination of the cooling capacity of the system is through measurements obtained on the refrigerant flow stream.

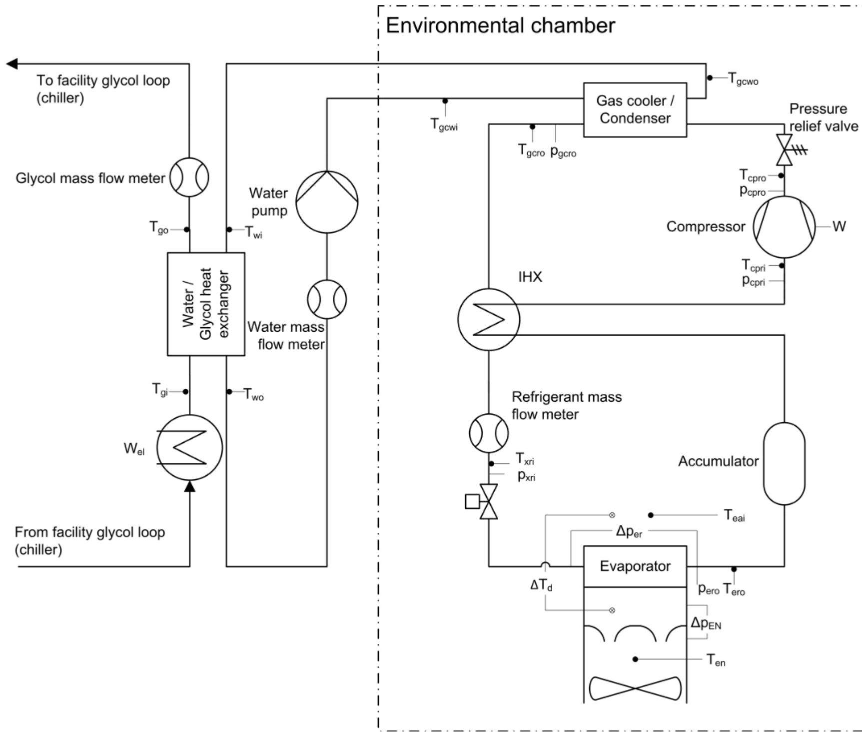


**Figure 3:** Experimental facility

On the heating side of the system, three different energy determination methods were employed. Using the instrumentation on the refrigerant cycle described above, the heating capacity can be determined from the temperatures, pressures, and mass flow values in the condenser/gas cooler. The second heating capacity determination was made using temperature and mass flow measurements on the water stream. This is in accordance with the testing of Type IV heat pump water heaters specified in ASHRAE Standard 118.1. In order to reject the heat input into the water stream by the heat pump, a glycol chiller was used. The third determination of heating capacity was made on this glycol stream. The glycol/pump facility was designed and constructed to transfer heat between the hot water stream and the cold glycol stream through a brazed plate heat exchanger. In addition to the brazed plate heat exchanger, the facility was also equipped with mass flow meters for each fluid stream, trim heaters, and a pump to control the water flow rate.

## 3. RESULTS AND DISCUSSION

The four R744 HPWH development stages were investigated at the condition the R134a unit is rated by the manufacturer. The rating temperature of the water and air inlet temperature of the HPWH is 26.7°C at a water flow rate of 1760 g/s. All four systems were tested under these conditions as well as under a broader range of water inlet temperatures. The combined COP was calculated as the ratio of the useful output (heating and cooling capacity) divided by the HPWH power consumption. The results of the performance tests are shown in Figure 4. Typical water temperature lift values for all systems were approximately 5°C. For the R744 systems, the high side pressure at each condition was optimized to provide the highest heating COP, when operating transcritically. The results in Figure 4 show the decreasing combined COP with increasing water inlet temperature. Even though this high water flow rate is how the manufacturer provides ratings for their systems, this may not represent actual operation of such units, especially if higher water lift conditions are required.

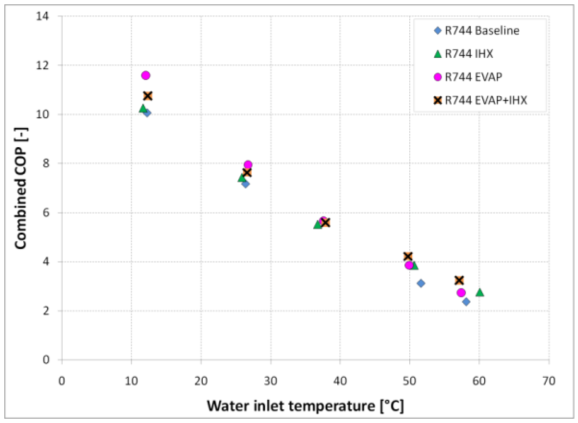


**Figure 4:** Combined COP versus water inlet temperature for different HPWH systems at a water flow rate of 1760 g/s

In order to understand the effect of water temperature lift on system performance and gain insight into a better control strategy, the water flow rate was reduced from the rating value of 1760 g/s down to 1000 g/s and 400 g/s. This was done for water inlet temperatures of 26.7°C while maintaining the air inlet temperature at 26.7°C. This ambient condition was chosen because this model is marketed as an indoor unit. This means, the heat pump will be pumping heat from the building's indoor environment to the water stream. This offers a twofold benefit of providing hot water and air conditioning. Applications in which this would be most beneficial would be somewhere where there is a constant need for hot water and cooling simultaneously. Restaurant kitchens and laundry facilities are both excellent examples of such locations. In addition to a water inlet temperature of 26.7°C, lower (12°C) and higher (50°C) water inlet temperatures were investigated for the R744 HPWH systems. A comparison was done between the investigated R744 HPWH development stages in order to quantify the relative improvement of each step over the R744 baseline system at the water rating mass flow rate of 1760 g/s (Figure 5). It can be seen that the IHX has a higher benefit on the system performance at higher water inlet temperatures than at lower. The highest relative COP improvement of up to 35 % at a water inlet temperature of 50°C is achieved for the R744 EVAP+IHX system. However the trend that is seen at 50°C water inlet temperature cannot be observed at lower temperatures. There the enhanced evaporator system (R744 EVAP) shows the highest combined COPs of the four R744 systems.

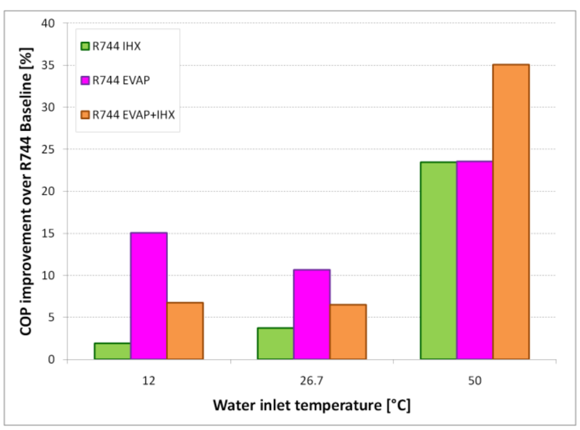


**Figure 5:** COP improvement of R744 HPWH over R744 baseline at different water inlet temperatures

The results of the combined COP for different water inlet temperatures lifts at various water temperature lifts are shown in Figure 6.

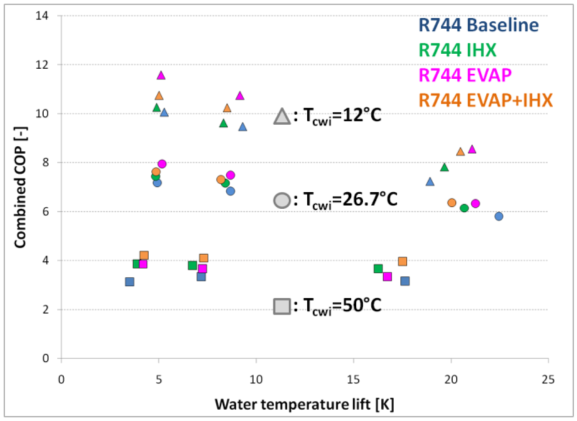


**Figure 6:** Comparison of combined COP versus water temperature lift at different water inlet temperatures

The combined COPs for the variable water flow rate conditions (Figure 6) confirm the behavior of the IHX at different water inlet temperatures for varying water flow rates. The results show the system optimization process going from the R744 baseline system in several development stages to the R744 enhanced evaporator system with IHX. It can be pointed out that the combined COP is more dependent on water temperature lifts at low water inlet temperatures. At 50°C water inlet temperature the results are almost steady whereas at 12°C the COP drops significantly when going to higher temperature lifts. The experimental test data that was measured and analyzed was used to develop a performance map which is shown in Figure 7.

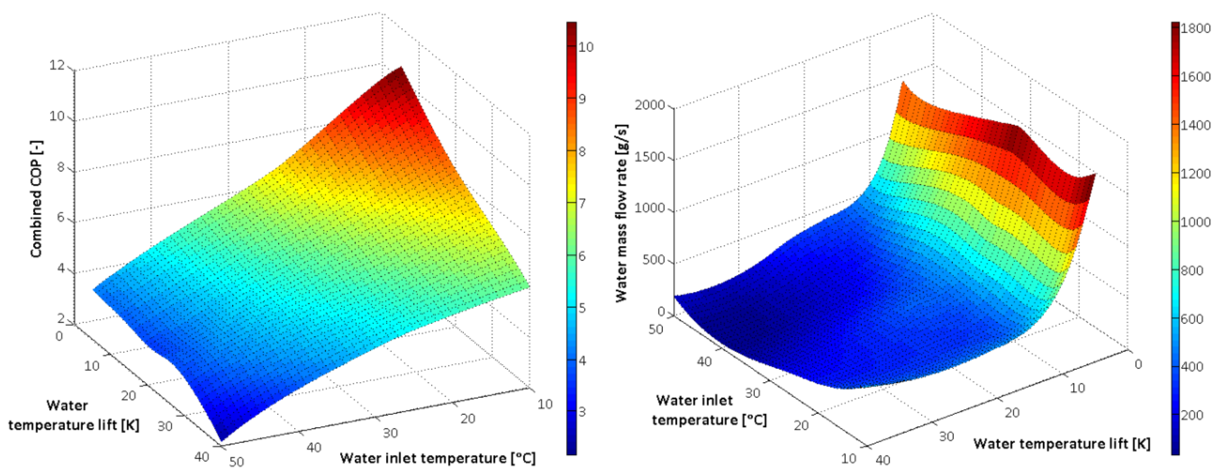


**Figure 7:** Performance map of combined COP and heating capacity for R744 IHX HPWH

The interpolated performance map (Figure 7) shows the main aspects that have to be taken into account when developing a control strategy for HPWH. It includes the water inlet temperature between 10 and 50°C and the desired water temperature lift for a range from 5 K to 40 K. Based on these parameters the combined COP of the HPWH and the resulting water mass flow rate are determined. The developed performance map of combined COP and water mass flow rate summarizes some of the main aspects when developing control strategies for HPWH. The water inlet temperature and the desired water temperature lift have a strong influence on the performance of the HPWH. The performance map is used as a tool to predict the energy consumption of different HPWH using interpolated trends that are based on experimental test results. Predictions about the energy consumption of HPWHs based on test data and extrapolated results over a wide range of water outlet temperatures were presented in Bowers *et al.* (2011). Energy savings compared to a R134a HPWH were as high as 20%. The main reason for this is the inability of R134a to reach the high water temperatures required in sanitary applications, without additional resistance heating, due to a limit of the compressor discharge temperature of 107°C specified by the manufacturer. For this purpose an additional electric resistance heater was used to supplement heating capacity for water outlet temperatures above 82°C. The performance map represents a starting point for tank tests that will be part of the optimization process of the HPWH in subsequent steps of the project. In this way, an ideal control strategy can be developed by predicting the energy consumption according to the operating condition on the water side (inlet temperature and water temperature lift).

In order to better understand the influence of the heat exchanger modifications (IHX, evaporator) on the system performance, the cycles are compared in a R744 pressure-specific enthalpy diagram. A comparison of the high-side pressure optimized four cycles at the rating water mass flow rate and an air and water inlet temperature of 26.7°C is shown in Figure 8.

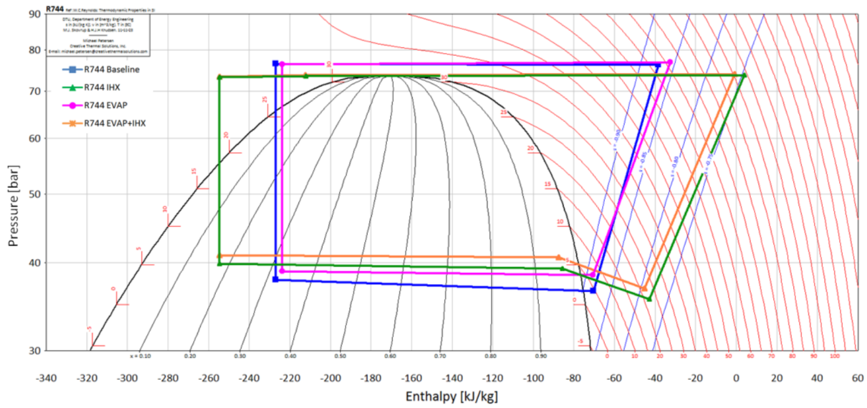


**Figure 8:** Comparison of R744 HPWH cycles in logarithmic pressure versus specific enthalpy diagram

It can be seen that the systems without IHX (R744 Baseline, R744 EVAP) operate at a higher high side pressure than the systems with IHX (R744 IHX, R744 EVAP+IHX). For the systems without IHX a constant superheat at the evaporator outlet of 5 K was used. This was done to prevent liquid refrigerant from entering the compressor which could lead to damage. The IHX systems operate at a lower optimized high side pressure resulting from more subcooling and therefore lower evaporator inlet qualities. The evaporator outlet condition was kept constant at a quality of 0.95.

The IHX consists of two parallel microchannel heat exchangers. These compact heat exchangers provide good performance characteristics regarding capacity and effectiveness. However the full potential of cycle improvements is not realized due to a relatively high pressure drop on the low pressure side of the heat exchanger. The pressure drop on the suction side was determined for all four systems in order to compare the influence of the IHX. For the systems without IHX, the low side pressure drop is measured across the inlet and outlet of the evaporator. For the cycle with IHX, the low side pressure drop is measured between evaporator and compressor inlet. The low side pressure drop of the R744 HPWHs at different water flow rates at an air and water inlet temperature of 26.7C are shown in Figure 9.

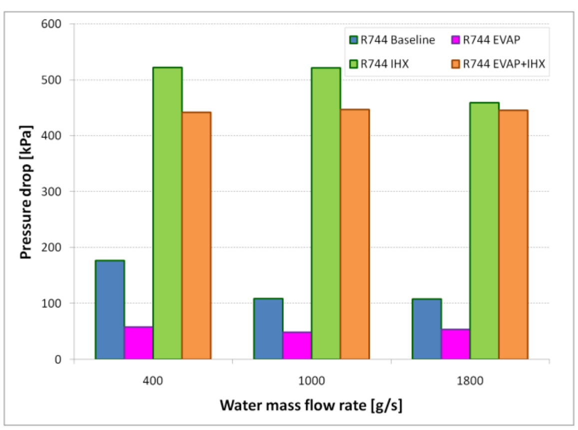


**Figure 9:** Low side pressure drop at different water mass flow rates

Even though the R744 HPWH with IHX improves performance characteristics over the baseline, it can be seen in Figure 9 that the IHX causes a large pressure drop on the suction side of the compressor. This pressure drop has to

be compensated during the compression process increasing the compressor power consumption. Consequently the overall combined COP increases are offset. A minimization of these pressure losses would further improve the COP values significantly.
Another characterizing parameter which can be used to analyze the influence of the IHX is the pressure ratio of the system which is calculated by the ratio of compressor discharge and suction pressure. The pressure ratio and combined COP at different water temperature lifts at an air and water inlet temperature of 26.7°C are shown in Figure 10.

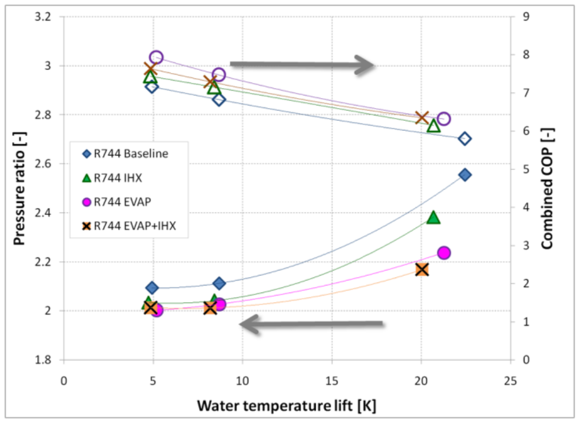


**Figure 10:** Pressure ratio and combined COP versus water temperature lift

It can be noted that the pressure ratios of the systems with IHX are less dependent on the water temperature lift. This has a positive effect on the combined COP. The increased low-side pressure for the systems with enhanced evaporator (EVAP and EVAP+IHX) keeps the pressure ratio on a lower level which is beneficial for the compressor power consumption. Consequently they show the best combined COPs.
The logarithmic mean temperature difference (LMTD), as defined in Equation 1, describes the average temperature difference between the two fluid streams in the heat exchanger.

$$LMTD = \frac{\left(T_{en} - T_{ero,sat}\right) - \left(T_{eai} - T_{eri,sat}\right)}{\ln \frac{\left(T_{en} - T_{ero,sat}\right)}{\left(T_{eai} - T_{eri,sat}\right)}} \tag{1}$$

A lower LMTD indicates a better performance of the heat exchanger. The temperature difference is calculated using the air side inlet and outlet temperatures as well as the evaporation saturation temperatures at the in- and outlet of the evaporator. Another characteristic factor that has to be considered when modifying the evaporator is the overall heat transfer coefficient U. The results for the LMTD and U are shown in Figure 11 and Figure 12 respectively.

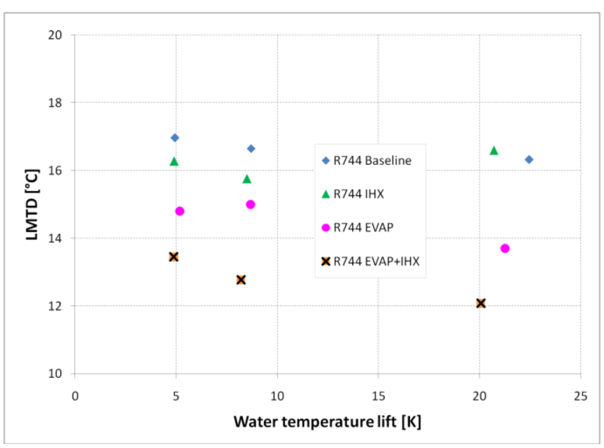


**Figure 11:** Evaporator LMTD versus water temperature lift for different R744 HPWH

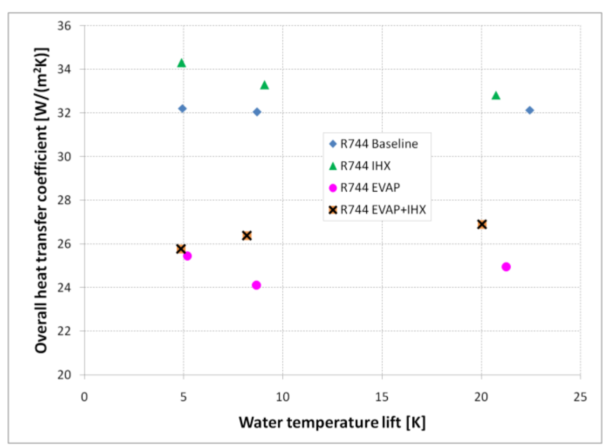


**Figure 12:** Overall heat transfer coefficient versus water temperature lift for different R744 HPWH

The evaporator was modified to utilize the volume that is provided by the R134a HPWH housing. For this the evaporator volume was enlarged by 50%. This larger heat exchanger volume leads to an increase in heat transfer area which decreases the approach temperature difference between air and refrigerant side. The low side pressure increases, reducing the compressor work necessary to provide the high side pressure. The trends for evaporator LMTD and overall heat transfer coefficient confirm the behavior that was expected: The increase of the heat transfer area reduces the LMTD. At the same time the overall heat transfer coefficient is reduced. The combination of evaporator enhancement (lower LMTD) and internal heat exchange (higher low side pressure) provides the best evaporator performance results.
The improvement of the R744 HPWH was done in order to achieve a combined COP of 8 at rating condition. At the same time the baseline R134a HPWH performance was supposed to be matched or even exceeded. A comparison of the combined COP of the R134a HPWH and the four development stages of the R744 system at a water inlet temperature of 12°C and an air inlet temperature of 26.7°C at the rating water mass flow rate of 1760 g/s is shown in Figure 13.

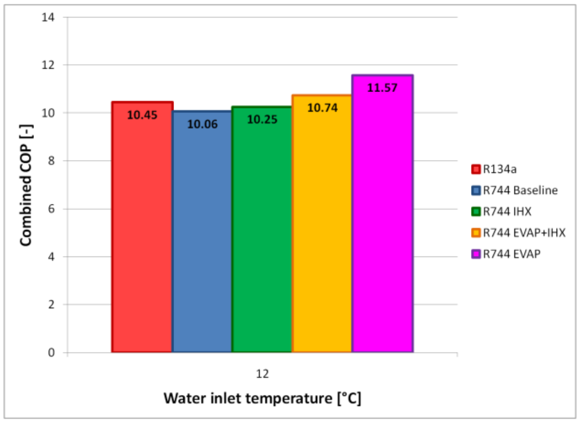


**Figure 13:** Comparison of combined COP for R134a and R744 HPWHs

The first two development stages of the R744 HPWH show a lower combined COP compared to the R134a system. The R744 HPWH was optimized and the system with enhanced evaporator and internal heat exchanger as well as the system with enhanced evaporator beat the R134a performance. The maximum performance has the enhanced evaporator system which has a 10.7% higher combined COP compared to R134a.

## 4. CONCLUSIONS AND FUTURE WORK

The development and optimization process of a R744 HPWH was described aiming for a combined COP of 8. The R744 HPWH was modified with an internal heat exchanger and an enhanced evaporator creating a total of four R744 HPWH systems. It was shown that the combined COP of the system increased for each development step compared to the R744 baseline system. At low (12°C) and medium (26.7°C) water inlet temperatures the system with enhanced evaporator showed the highest relative performance increase with approximately 15% and 11% respectively. At a water inlet temperature of 50°C the system with enhanced evaporator and internal heat exchange showed the best performance with a relative improvement of 35 % compared to the R744 baseline system. The influence of the IHX and the enhanced evaporator on the system performance was described. It was pointed out that the downside of the IHX is its relatively high pressure drop. A component with similar performance but with lower pressure drop could combine the beneficial characteristics of the internal heat exchange without the disadvantage of an increase in compressor power consumption.
In the next steps of the project further efforts in optimizing the HPWH system performance will be attempted. The utilization of an ejector in order to recover expansion throttling losses will be part of the project. The minimization of conduction losses in the gas cooler of the HPWH will be investigated by using multiple heat exchangers. Furthermore the use of high efficiency interior permanent magnet compressor motors is anticipated to deliver better compression efficiencies which will positively affect the combined COP of the HPWH. Another aspect of the investigation of the HPWH is the connection of the system to a tank to investigate stratification inside the tank. Also the interaction between HPWH and tank as well as the development of control strategies are aspects that influence the optimization of the system performance. As a starting point the developed performance map will be used to predict energy consumption of the different systems.

## NOMENCLATURE

| | |
|---|---|
| COP | Coefficient of performance |
| EVAP | Enhanced evaporator (50% volume increase) |
| IHX | Internal heat exchanger |
| LMTD | Logarithmic mean temperature difference (°C) |
| p | Pressure (kPa) |
| T | Temperature (°C) |
| U | Overall heat transfer coefficient (W/($m^2$K) |
| W | Power (kW) |

**Subscripts**

| | |
|---|---|
| a | Air |
| comb | combined |
| cp | Compressor |
| d | Dew point |
| e | Evaporator |
| el | Electric |
| g | Glycol |
| gc | Gas cooler |
| i | Inlet |
| n | Nozzle |
| o | Outlet |
| r | Refrigerant |
| ratio | Ratio |
| sat | Saturation |
| w | Water |
| x | Expansion valve |

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