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ABSTRACT

The US Army has equipped their HMMWV's (High Mobility Multi-purpose Wheeled Vehicle) with R134a two evaporator A/C systems and is exploring the possibility to implement a two evaporator R744 system. The components of the vehicle A/C systems (R134a and R744) were installed in the laboratory. Different breadboard versions were tested experimentally at identical ambient conditions. This paper presents how the R744 two evaporator breadboard system was optimized to exceed the performance of the R134a two evaporator breadboard system. The subject of different hardware setups (e. g. choice of expansion devices) for the R744 two evaporator system and their implications on performance and stability are addressed. For the case of two controllable expansion devices the iterative process which was used to achieve good performance over a wide range of conditions will be presented. The optimized R744 system showed higher cooling capacity and higher coefficient of performance (COP) compared to the R134a system. The cooling capacity was increased up to 57% and the COP up to 18% compared to the R134a system.

1. INTRODUCTION

The US Army is interested in natural refrigerants for a number of specific reasons. First, the quick pull-down characteristics and high capacity of CO_2 is very attractive for many of the harsh environments in which they operate. Although efficiency is not a high importance criterion, results presented here also indicate an increase in efficiency over the currently used R134a HMMWV system. Second, the use of any HFC requires recycling and recovery, which for a mobile force requires specifically trained personnel and equipment which must be transported around. Finally, using a CO_2 system with all aluminum microchannel heat exchangers provides significant weight and volume reductions for the A/C system, both of which are attractive to a vehicle that needs to be quick and is packaged with lots of equipment (Memory *et al.*, 2004). This paper presents the experimental results for both the R-134a and CO_2 systems which were tested in a breadboard test facility in order to accurately compare capacities and efficiencies. It then details on how a high side pressure correlation for the two evaporator R744 system with two electrical expansion valves (EEV) was derived.

2. EXPERIMENTAL FACILITY FOR HMMWV BREADBOARD SYSTEMS

The experimental facility consists of two environmental chambers (see Figure 1). All refrigerant-to-air heat exchangers are installed in open-loop wind tunnels housed inside the chambers. The compressor is installed between the two chambers. The outdoor chamber contains the condenser/gas cooler, while the evaporators are mounted in the indoor chamber. All refrigerant-to-air heat exchanger measurements were made in triplicate to verify their precision, with a calorimetric chamber balance, air-side balance and a refrigerant balance. There are several welded Type-T thermocouples attached to each chamber surface. The heat transmission losses through the chamber surfaces are determined from temperature differences between the interior and the exterior of the chambers in combination with the UA values determined from chamber calibration experiments. During tests, the ambient conditions are maintained at specified values based on the test conditions. The power consumptions of all electrical devices operated in the chambers are measured with Watt transducers. The condensate formed at the evaporators is collected. The condensate formation rate is determined with two load cells, from which the latent portion of each evaporator load is calculated. By combining the reading of the Watt transducers, the heat loss through the walls and the latent portion of each evaporator load, it is possible to determine the load on the evaporators and condenser/gas cooler. The load on the evaporators determined by this method is called the chamber energy balance. The air-side measurements were designed according to ANSI/ASHRAE Standard 41.2-1987 (RA92). For the air-side energy balance, the latent loads at each evaporator are calculated from the measurements taken by two chilled mirror dew point sensors. This method is independent of the condensate formation rate measurement, which is part of the chamber energy balance. The refrigerant-side energy balance was accomplished from temperature and pressure measurements in combination with two Coriolis-type mass flow meters in each branch of the refrigerant flow. For all data presented in this paper the chamber and air-side energy balance were within ±5%. For R134a the refrigerant side balance could not always be used due to two phase flow in the coriolis mass flow meter.

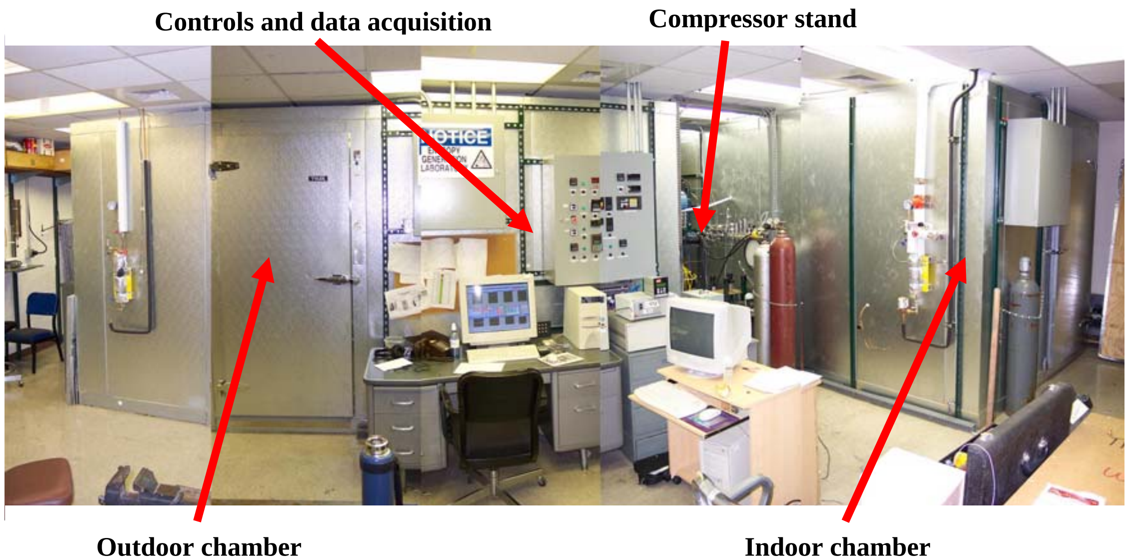


Figure 1: Laboratory for HMMWV breadboard systems

3. HMMWV R134A BREADBOARD TESTS

The HMMWV R134a breadboard System #1 was comprised of all the original components from the vehicle including the A/C hoses. Additional short piping was used around the heat exchangers. Mass flow meters, sight glasses, pressure and temperature sensors were also added. Two TXVs (thermostatic expansion valve) were used as expansion devices (see Table 3). The only major difference between vehicle and breadboard was the piping distance between condenser and the evaporators. This was because the wind tunnels in the indoor chamber are physically installed on top of each other. As a result, the distance between the condenser and the evaporators was the same. In the vehicle, the distance between the condenser and the rear evaporator was shorter than the distance between the condenser and the front evaporator. A refrigerant charge determination test was carried out according to the conditions specified in Table 1. All charge tests mentioned in this paper were conducted under those conditions.

Table 1: Condition for charge determination test

Condenser/Gas Cooler			Evaporator (Front & Rear)		
T [^{0}C]	V [m^3/s]	RH [%]	T [^{0}C]	V [m^3/s]	RH [%]
43	0.897	30	43	0.100	30

Based on the test results, the optimum system charge was found to be 3500g for the R134a baseline system which was higher than the recommended charge (Memory, *et al.*, 2005). This higher charge ensured satisfactory cooling capacities for test conditions exceeding the charge test specifications. The test matrix for the HMMWV R134a system in breadboard version comprises 15 test conditions (see table 2). According to the charge determination test all test conditions listed in Table 2 were carried out with a charge of 3500g.

Table 2: Test matrix for HMMWV breadboard tests

Name	Compressor speed [RPM]			Gas cooler		Evaporator (front and rear)			
	R134a System #1	R744 System #2 & #3	R744 Sytem #4	T [^{o}C]	V [m^3/s]	T [^{o}C]	RH [%]	Tdpei [^{o}C]	V [m^3/s]
I64-49-0.1	1150	1500	810	64	0.802	49	20	20.0	0.100
L64-49-0.1	2200	2880	1550	64	0.897	49	20	20.0	0.100
H64-49-0.1	3400	4480	2410	64	0.991	49	20	20.0	0.100
I49-49-0.1	1150	1500	810	49	0.802	49	20	20.0	0.100
L49-49-0.1	2200	2880	1550	49	0.897	49	20	20.0	0.100
H49-49-0.1	3400	4480	2410	49	0.991	49	20	20.0	0.100
I58-43-0.1	1150	1500	810	58	0.802	43	30	21.7	0.100
L58-43-0.1	2200	2880	1550	58	0.897	43	30	21.7	0.100
H58-43-0.1	3400	4480	2410	58	0.991	43	30	21.7	0.100
I43-43-0.1	1150	1500	810	43	0.802	43	30	21.7	0.100
L43-43-0.1	2200	2880	1550	43	0.897	43	30	21.7	0.100
H43-43-0.1	3400	4480	2410	43	0.991	43	30	21.7	0.100
I35-35-0.1	1150	1500	810	35	0.802	35	40	19.4	0.100
L35-35-0.1	2200	2880	1550	35	0.897	35	40	19.4	0.100
H35-35-0.1	3400	4480	2410	35	0.991	35	40	19.4	0.100

The overall cooling capacity ranged from 5.5kW to 8.1kW and the COP ranged from 1.2 to 3.6. The results for cooling capacity and COP are shown in Figure 5 and Figure 6, respectively where they are compared as a baseline against the R744 results.

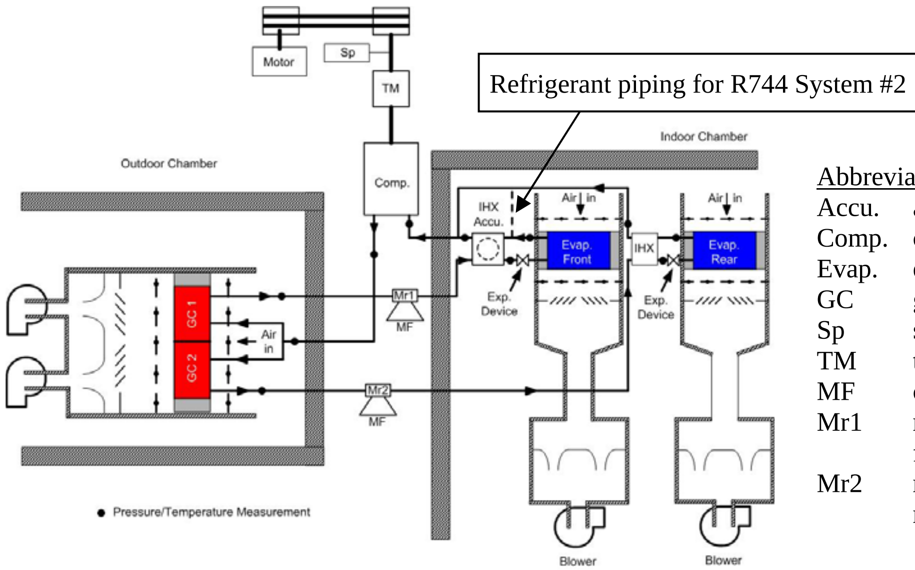
4. HMMWV R744 BREADBOARD TESTS

For the R744 HMMWV breadboard system three different set-ups were evaluated. Figure 2 shows a incorporated schematic of those three set-ups. The only differences between the set-ups were the use of different expansion devices and the mixing point of the refrigerant streams downstream of the evaporators. For System #2 the refrigerant stream leaving the rear IHX (internal heat exchanger) was mixed with exit stream of the front evaporator. For System #3 and #4 the refrigerant streams were mixed downstream of both IHXs just before the compressor inlet. Table 3 gives an overview of which expansion devices were used in which set-up.

Table 3: Overview of expansion devices by system

System	Set-up	Expansion Device
System #1	R134a baseline	2 TXV's
System #2	R744 Set-up 1	2 orifice tubes
System #3	R744 Set-up 2	1 orifice tube; 1 EEV
System #4	R744 Set-up 3	2 EEV

An independent charge determination test where the charge was stepwise increased was carried out for all three systems according to the conditions specified in Table 1. Based on this test, the optimum charge for R744 System #2 was found to be 1150g. As explained below, for System #2, the test also indicated that the refrigerant exit qualities for the two evaporators were significantly different for this optimum charge, as was also found on the vehicle (Memory *et al.*, 2005). For Systems #3 & #4, the optimum charge was found to be 1100g. Furthermore, it was possible to achieve equal exit qualities at the evaporators for both systems as explained below. However, for System #3, high side pressure could not be controlled. With the set-up of System #4 the high side pressure could be controlled and coincided with maxima for both COP and capacity were observed for a charge of 1100g.



Abbreviations:

Accu.	accumulator
Comp.	compressor
Evap.	evaporator
GC	gas cooler
Sp	speed meter
TM	torque transducer
MF	coriolis mass flow meter
Mr1	refrigerant mass flow rate at front evaporator
Mr2	refrigerant mass flow rate at rear evaporator

Figure 2: Schematic for R744 breadboard systems

4.1 Comparison between R744 set-up 1 and R134a baseline

For R744 System #2 two orifice tubes from the vehicle were used in place of the TXVs from the R134a baseline system (thereby requiring no control). When compared to the R134a baseline the R744 System #2 showed an increase in cooling capacity between 5% and 35% at the conditions specified in the test matrix. However, the COP was significantly lower by 54% to 64%. To improve the COP some tests were done to match the cooling capacity of the R134a system. One could achieve this by reducing the displacement of the R744 compressor. The COP did increase about 15% for equal cooling capacity, but was still lower (by 23%) than for the R134a system.

The reasons for these lower COPs were identified as follows. First, the exit qualities of the evaporators were too low and unequal (e.g. 0.7 and 0.9 at the charge condition with optimized charge). Second, the discharge pressures were too low because of the high inlet densities to the compressor due to the mixing point. Third, the total refrigerant mass flow rate of up to 100 g/s had to flow through the IHX/Accumulator resulting in pressure drop of over 200 kPa. In addition to the low performance, System #2 revealed the possibility of instabilities occurring. When the system was run at steady state and then the load at the evaporators was disturbed, mass flow rates and exit qualities were sometimes transposed after a new steady state was attained. The sum of evaporator capacities and the COP were essentially the same as for the previous steady state. Furthermore, when starting the system, one could not predict which of the two stable points would be reached at steady state.

4.2 Comparison between R744 set-up 1, R744 set-up 2 and R134a baseline

To address the problem of the unequal refrigerant exit qualities at the evaporators, the fixed orifice tube upstream the front evaporator was replaced with an electrical stepper motor expansion valve in the R744 System #3 which made it possible to keep the exit qualities equal at both evaporators for all tests. In addition to that, the mixing point of the two refrigerant branches was changed. For System #3, the two branches were combined downstream the IHX/Accumulator to lower the pressure drop across this component. Figure 3 shows a comparison between Systems #1, #2 and #3.

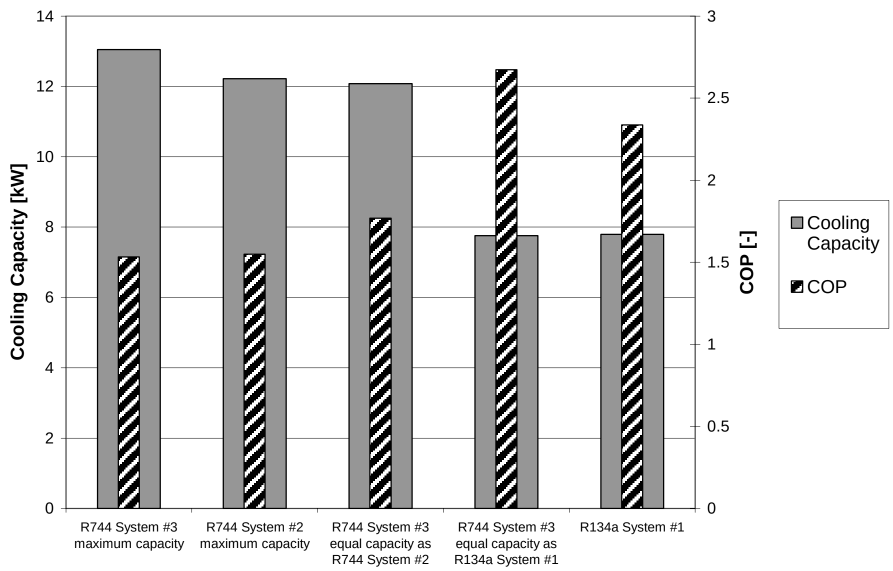


Figure 3: Comparison of Systems #1, #2 and #3 at condition L43-43-0.1

In conclusion, Figure 3 shows that R744 System #3 provided 7% more cooling capacity than R744 System #2. In addition, the COP was increased by 4%. By reducing the displacement of the compressor to achieve equal cooling capacities between Systems #2 & #3, System #3 then provided a 14% higher COP than System #2. Comparing R744 System #3 with R134a System #1 at equal cooling capacities, the R744 System #3 provided a 14% higher COP than the R134a System #1. This result is significant and shows that proper adjustment of the system provides higher efficiency.

Another way to improve COP whilst maintaining capacity is to reduce compressor speed while increasing or maintaining displacement. To study the effect of reducing compressor speed (rather than displacement) on COP, two tests were compared: R744 System #3 (one EEV) was run at the charge condition (Table 1). The displacement of the compressor was then reduced until the cooling capacity matched the 7.8 kW cooling capacity of the R134a baseline system. The second test was performed the same way except that the displacement was kept at the maximum and the compressor speed was reduced until the capacity reached 7.8 kW. This resulted in a 9% higher isentropic efficiency for the case of reduced speed when compared with reduced displacement, resulting in an increase of system COP of 4.5%. Therefore, as expected, it is better to reduce compressor speed rather than displacement to improve COP. The main conclusion from the R744 System #3 experiments is that the exit qualities of the evaporators could be kept constant due to controlling the electrical expansion valve, and therefore the R744 System #3 had a better performance compared to the R744 System #2.

4.3 Optimization of R744 set-up 3

The major difference between R744 System #4 and R744 System #3 is that EEVs were used for both evaporators. With one expansion valve and one orifice tube (R744 System #3), it is possible to achieve equal evaporator exit qualities, but it is not possible to control the high side pressure at the same time. By exchanging the orifice tube with a second electrical expansion valve, thereby controlling the high side pressure, further COP improvement was expected. The difficulty is to determine the optimum high side pressure for each condition which maximizes the COP. The optimum high side pressure for a CO_2 system is typically given as a function of the air inlet temperature to the gas cooler or as a function of the refrigerant exit temperature of the gas cooler. For a typical R744 two evaporator system one usually has the following degrees of freedom which influence the system performance (for a fixed condition with given compressor speed): charge, exit qualities of the evaporator, compressor displacement, high side pressure. For the R744 HMMWV vehicle system the objective was to achieve maximum cooling capacity. Therefore, the compressor displacement was set to maximum displacement, eliminating one degree of freedom. The results from System #3 showed that keeping equal exit qualities at the evaporators provides best COP. Equal exit qualities at the evaporators result in an equal load distribution between the two evaporators which ensures that both evaporators are operated with high effectiveness. This leaves the charge and the choice of high side pressure as the degrees of freedom. One can think of two iterative approaches to determine the optimum charge and high side pressure. The first approach is to fix a charge and than vary the high side pressure at the given condition. The COP is then plotted versus the high side pressure. Repeating this procedure with a different charge and comparing with the first iteration gives information if the charge should be increased or decreased for the third test. The second approach is to guess a high side pressure and keep this pressure constant while incrementally increase the charge. After the proper charge is determined (the charge which provided the best COP) one varies the high side pressure to check if the initial guess of the high side pressure was too low or too high. Both approaches require roughly the same amount of iterative testing.

The second approach was chosen to optimize the R744 System #4. After the optimum charge was found by iteration, several tests at three different conditions (L35-35-0.1, L43-43-0.1, and L49-49-0.1, see Table 2) were conducted. For each of these ambient conditions, the high side pressure was varied to obtain a curve for COP and capacity. By using the high side pressure corresponding to the maximum COP for each curve, an equation was developed that provided the optimum high side pressure for a given gas cooler air inlet temperature as shown in Figure 4. The expansion valves were then controlled to set the high side pressure according to this equation in addition to keeping the exit qualities of the evaporators equal.

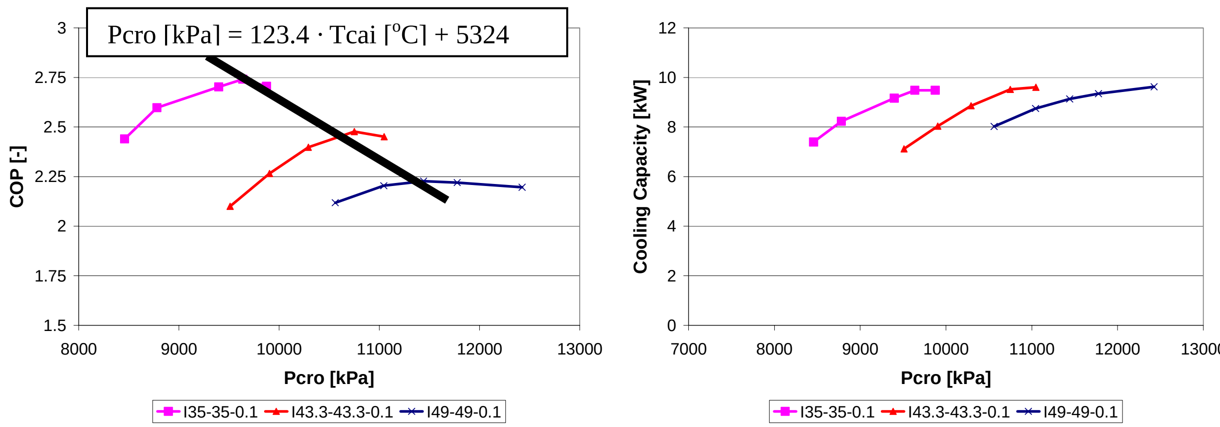


Figure 4: COP and cooling capacity versus gas cooler outlet pressure for System #4

4.4 Comparison between R744 set-up 3 and R134a baseline

Figure 5 shows the total cooling capacities for the test matrix given in Table 2 for Systems #1 and #4 at three compressor speeds. The cooling capacity for the R744 System #4 ranges from 5.3kW to 12.1kW. It can be seen that at idle speed, the R744 system matches the cooling capacity of the R134a baseline system. For higher speeds, the R744 system shows a significant increase in cooling capacity. Compared to the R134a baseline system, the cooling capacity of the R744 system is up to 57% higher. This high improvement is of course a function of the performance of the R134a baseline system.

Figure 6 shows the COPs for the test matrix given in Table 2 for Systems #1 and #4 at three compressor speeds. The COP ranges from 1.2 up to 4.0 for the R744 System #4. Compared to the R134a baseline system, the COP for the R744 system is up to 18% higher. It can be seen that the COP of the R744 System #4 is higher for the lower temperature conditions. However, even for the elevated ambient temperature conditions at 58^{o}C and 64^{o}C, where R744 automobile systems usually show lower performance than R134a systems, the R744 System #4 shows equal COP values.

The main conclusion from the R744 System #4 was that by controlling the expansion valves to keep the exit qualities at the evaporators equal and adjusting the high side pressure it was possible to achieve higher cooling capacities and COPs than the R134a baseline system.

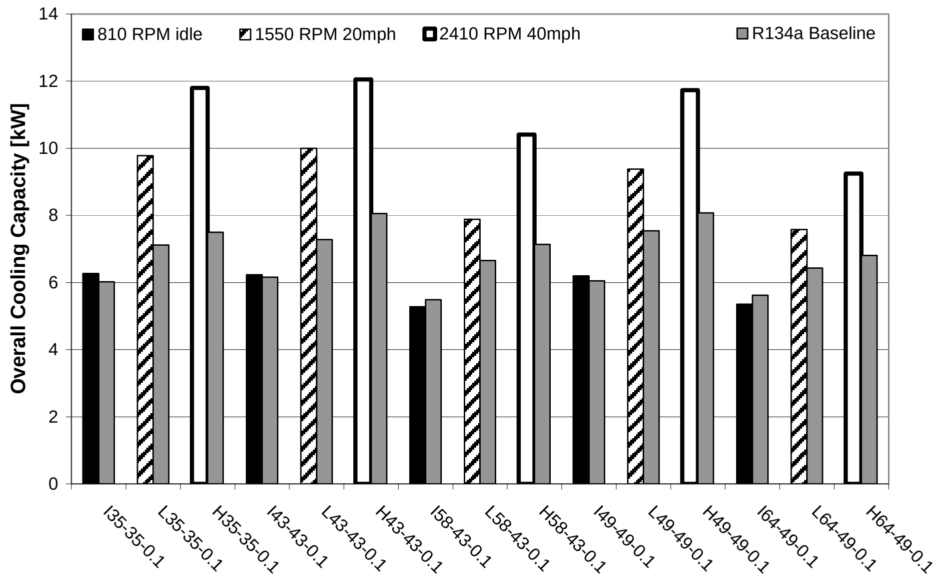


Figure 5: Overall cooling capacity comparison for System #1 (R134a Baseline) and System #4 (R744)

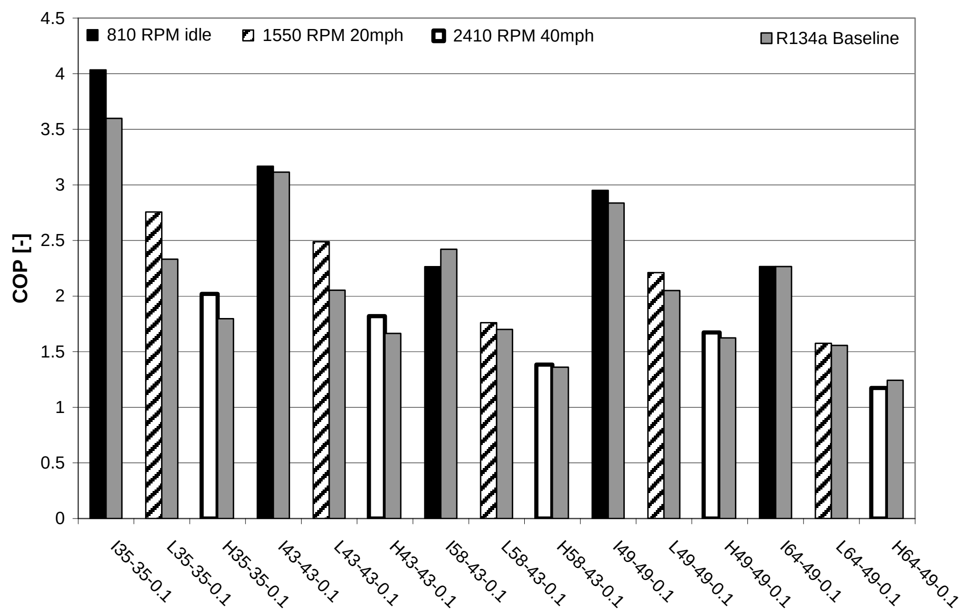


Figure 6: COP comparison between System #1 (R134a Baseline) and System #4 (R744)

6. CONCLUSIONS

The R744 System #2 had two orifice tubes. This system showed an increased cooling capacity between 5% and 35% compared to the R134a baseline System #1. However, the COP was significantly lower by 54% to 64%. System #2 also revealed the possibility of instabilities occurring. One fixed orifice tube was replaced by an EEV for R744 System #3. This made it possible to avoid instabilities and to keep the exit qualities equal. This system showed a 67% higher cooling capacity than the R134a system compared at the charge condition. However, the COP was still lower by 30%. When compared at equal cooling capacities System #3 showed a 14% higher COP than the R134a baseline system. The main conclusion from R744 System #3 was that by controlling one expansion device either higher cooling capacity or higher COP could be achieved but not both at the same time. A comparison experiment showed that an additional increase in COP could be achieved when reducing the compressor speed instead of the compressor displacement. The final R744 System #4 had two EEVs which made it possible to control the high side pressure while keeping the exit qualities of the evaporators equal. A high side pressure equation was determined to maximize COP based on one of the approaches presented in this paper. System #4 showed higher cooling capacity and higher COP compared to the R134a system. The cooling capacity was increased up to 57% and the COP up to 18%.

In summary, it was demonstrated that the HMMWV R744 breadboard System #4 can provide higher cooling capacity and higher COP compared to the HMMWV R134a breadboard system built from currently used HMMWV vehicle components.

NOMENCLATURE

COP	coefficient of performance	(-)
T	temperature	(^{o}C)
Tcai	air inlet temperature to gas cooler	(^{o}C)
Tdpei	dew point temperature at evaporator air inlet	(^{o}C)
RH	relative humidity	(%)
RPM	revolutions per minute	(1/min)
V	volumetric air flow rate	(m^3/s)
UA	overall heat transfer coefficient times area	(W/K)

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