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Influence of liquid receiver on the performance of reversible heat pumps using refrigerant mixtures

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Abstract

A liquid receiver is normally included in reversible vapour compression heat pumps (RHPs) to temporarily store the excess refrigerant charge occurring due to change of operation mode. The presence of a liquid receiver influences the total amount of refrigerant charged into a system, and particularly when using refrigerant mixtures, could affect the system circulation composition. Using a computer simulation, this paper compares the performance of RHPs designed with and without a liquid receiver, when using R407C. It was shown that the presence of a receiver caused an increase in the positive shift in the circulating composition, resulting in improved capacity while reducing the system COP in both heating and cooling modes when compared to a system without a receiver.

Keywords: Heat pump; Modelling; Performance; Receiver; Liquid; R-407C; Transient regime

Influence de l'utilisation d'un réservoir de liquide sur la performance des pompes à chaleur réversibles utilisant des mélanges de frigorigènes

Mots clés : Pompe à chaleur ; Modélisation ; Performance ; Réservoir ; Liquide ; R-407C ; Régime transitoire

1. Introduction

It is generally known that cooling and heating modes of a reversible vapour compression heat pump use different amounts of refrigerant quantity. This situation leads to the presence of excess refrigerant which can occur in any of the two modes, depending on the system design and the internal volumes of the condenser and the evaporator. The excess charge is required to be temporarily stored in a liquid receiver to avoid liquid back up in the condenser that could impair the system performance [1]. When a receiver is present, generally the condenser outlet is considered saturated [2]. When using refrigerant mixtures, a receiver can influence the circulating mixture composition [3] and its presence needs to be considered in estimating the total system charge requirement [4].

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Nomenclature

A	Area (m^2)
C	Concentration (mass fractions)
COP	Coefficient of performance
dT	Temperature difference (°C)
$dTsc$	Degree of subcooling (°C)
HTC	Heat transfer coefficient (W/m^2 K)
HTF	Heat transfer fluid
L	Length of two-phase heat exchanger elements
m	Mass flow rate (kg/s)
M	Mass of refrigerant (kg)
N	Number of two-phase heat exchanger elements
NRV	Non-return valve
P	Pressure (MPa)
Q	Capacity (kW)
RHP	Reversible heat pump
RV	Four-way reversing valve
TEV	Thermostatic expansion valve
Vr	Volume ratio (condenser volume/evaporator volume, refrigerant sides)
X	Composition of vapour phase
Y	Composition of liquid phase
α	Void fraction
ρ	Density (kg/m^3)
ΔM	Hold up mass in two-phase regions (kg)
ΔV	Volume of heat exchanger element (m^3)

Subscripts

2ph	Two-phase region
Cnd	Condenser
Evp	Evaporator
exs	Excess charge (kg)
fwd	Forward mode
i	Refrigerant component number
j	Two-phase element number in a heat exchanger ($j=1$ ……. N)
liq	Liquid phase
LR	Liquid receiver
oil	Lubricant
Ref	Refrigerant
rvs	Reversed mode
sph	Single phase region
sys	System
vap	Vapour phase

The decision whether a receiver is employed in a RHP using refrigerant mixtures relates to both the relative charge requirement (kg) in each mode of the operation and the influence of the refrigerant volume in the receiver on the circulating composition.

Previous works [3,5–8] investigated different aspects of refrigerant charge related issues in refrigeration and heat pump systems. These include how the amount of charge in the receiver varies in relation to the changes in load, and the influences of void faction predictions on the charge estimate in heat exchangers etc. However, there is a lack of investigations into the effects of receiver in mixture systems, particularly in relation to reversible heat pumps. This paper looks at the influences of the receiver on the performance of a water-to-water vapour compression reversible heat pump when using refrigerant mixtures. To address the issue of excess charge occurrence in either forward or reversed mode, systems with different condenser to evaporator volume ratios (Vr) are considered.

2. Formulation

The heat pump considered in this study employs a four-way reversing valve for changing from heating to cooling mode and vice versa (Fig. 1). The nominal design heating capacity is 7.5 kW. Two configurations, i.e. with and without a receiver, are studied when using R407C (R32/R125/R134a, 23/25/52% by mass). The system employs a counter flow concentric tube heat exchanger configuration. In the forward design mode, the refrigerant is carried in the outer tube of the condenser and in the inner tube of the evaporator, and this tube configuration reverses (i.e. condenser now has refrigerant in the inner tube and vice versa for the evaporator) in the reversed mode. Compared to the practical systems with similar capacities, the current system uses relatively larger tube diameters[2] in order to minimize refrigerant pressure drops so as to achieve the glide matching situation [9]. Further, the selection of the outer tube diameter takes into account its effects on the reversed mode capacity due to the deterioration of heat transfer performance associated with the reduced refrigerant mass flux.

Water temperatures inlet to the condenser and the evaporator are set at 35 and 14 °C respectively, and the water flow rates are set to achieve glide match in both heat exchangers. For a given set of design operating conditions there will be only one flow rate that maintains a constant dT (i.e. glide match) in the two-phase section of a heat exchanger. Both the degree of subcooling and the TEV superheat in the forward heating mode are set at 5 °C. In the reversed mode, the superheat setting remains the same, while there is no subcooling when a receiver is present. If no receiver is present, the degree of subcooling becomes a variable,

[2] Outer diameter of the inner tube and the inner diameter of the outer tube are 20 and 35 mm respectively.

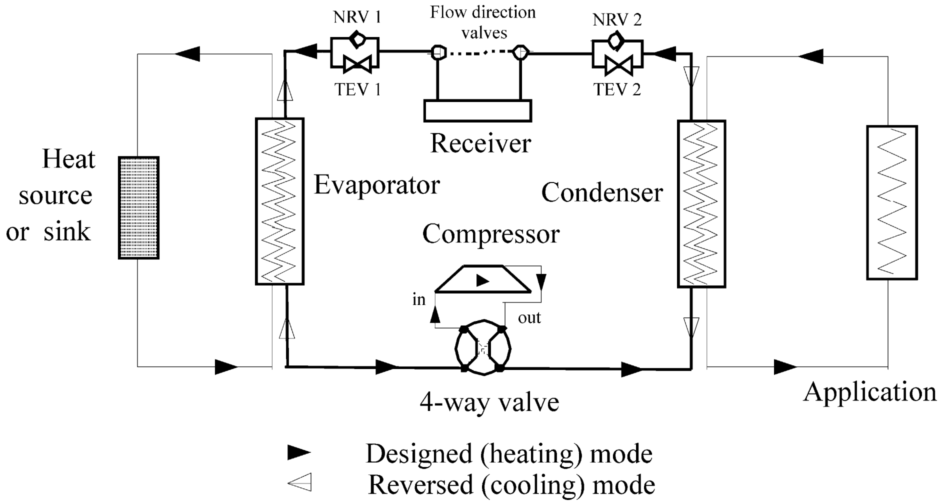

Fig. 1. Schematic of the water-to-water reversible heat pump.

depending on the operating conditions and the system charge.

The main objective of the analysis is to observe how the presence of a liquid receiver influences the compressor sizing, circulating composition and the performance in both forward and reversed modes. For comparison purposes, pure fluid systems (R134a) designed for the same nominal capacity with the same temperature changes of the heat transfer fluid (HTF) as those of the R407C systems are included in the study.

2.1. Charge requirement with/without receiver

For a system without a receiver, the initial design determines the volumes of individual heat transfer zones (single phase and two-phase) and the total volume associated with each heat exchanger, and the volumes of the suction and the liquid lines. Using these volumes and the corresponding densities, the mass of the refrigerant in each single phase zone is estimated. Refrigerant masses in the two-phase zones are estimated using the Humark's void fraction correlation [5] based on the design operating conditions.

If there is a difference in the charge requirement between the forward and the reversed modes, the system will be charged with the larger charge requirement. The excess refrigerant quantity in the other mode, being stored in the receiver, can be estimated using Eq. (1), which adds up the liquid and vapour refrigerant masses present in different parts of the system at steady running. M_{sys} is the total refrigerant charge in the system.

$$M_{exs} = M_{sys} - \left(\sum M_{vap(2ph)} + \sum M_{liq(2ph)} + \sum M_{vap(sph)} + \sum M_{liq(sph)}\right) \tag{1}$$

If a receiver is not present, any excess charge will back up in the condenser, which results in a certain degree of subcooling. However, if too much refrigerant is backed up, there will be a reduction of effective heat transfer surface area, which will give rise to an excessive increase in the condenser pressure and a reduction in the *COP*.

When charging a RHP, the presence of a liquid receiver needs to be considered in deciding the total refrigerant requirement. In practice, for single mode systems, the liquid receiver contains about one-sixth of the system charge during normal running [4], where this refrigerant acts as a vapour seal between the incoming and outgoing refrigerant streams. Similarly for a reversible system the additional charge (about one-sixth of total) is needed to perform the same function when the system runs on the mode where excess charge does not occur. The added refrigerant represents the minimum amount of refrigerant staying in the receiver at the design conditions. At any other time the liquid level in the receiver will increase, varying with the load.

In this study it is assumed that when a receiver is present the condenser outlet is saturated (free liquid draining from condenser to receiver). Further, it is assumed that there is no appreciable pressure drop in the liquid line, and the refrigerant in the receiver will also be saturated at the condensing pressure. During the discussion '*heating*' refers to the forward (design) mode while '*cooling*' refers to the reversed mode operation.

2.2. Concentration shift

Experiments [10–12] have shown that there is a difference in the charged and circulating concentrations of refrigerant(s) when using mixtures. This is caused by the differential hold up of individual refrigerant components in the liquid and vapour phases in various system components where the two phases coexist [12]. These refer to the two-phase sections of the condenser and the evaporator, and the receiver. For a given refrigerant mixture, the magnitude of concentration shift varies with operating conditions and the system design [11].

At a given vapour quality, the vapour phase is normally enriched with the lower boiling components of the mixture, while the liquid phase retains more of the less volatile components. The difference between the liquid and vapour compositions progressively changes with vapour quality. That means, within the two-phase section of a heat exchanger, certain amounts of individual refrigerant components are retained, which causes the change in circulating composition [12]. To estimate the two-phase refrigerant hold up in the heat exchanger, an element by element analysis is required (Fig. 2).

The total hold up of liquid and vapour in the two-phase section are estimated using Eqs. (2) and (3), respectively [12].

$$M_{liq} = \sum_{j=1}^{N} AL_j(1-\alpha_j)\rho_{\text{liq}(j)} \tag{2}$$

$$M_{vap} = \sum_{j=1}^{N} AL_j\alpha_j\rho_{liq(j)} \tag{3}$$

To estimate the hold up of the ith component in the heat exchanger, say R32, Eq. (4) is used.

$$\Delta M_{(\text{i})2\text{ph}} = \sum_{j=1}^{N}\Delta V_{(j)}\alpha_{(j)}\rho_{\text{vap}}X_{(i)} + \sum_{j=1}^{N}\Delta V_{(j)}(1-\alpha_{(j)})\rho_{\text{liq}}Y_{(i)} \tag{4}$$

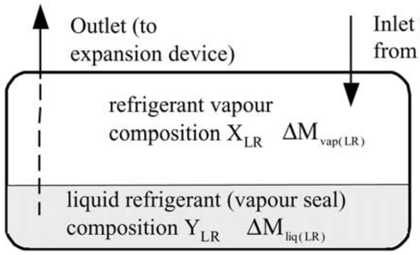

Fig. 3. Model for concentration estimates associated with the liquid receiver.

Assuming at equilibrium, the hold up in the receiver is obtained, based on its volume and the amount of liquid and vapour it contains, which are of different compositions. The amounts of the individual mixture components are estimated using the corresponding phase compositions (X_{LR} and Y_{LR}) and refrigerant masses ($\Delta M_{\text{vap(LR)}}$ and $\Delta M_{\text{liq(LR)}}$) of the vapour and the liquid (Fig. 3).

By excluding the amount of each refrigerant component held from the total charged mass of the component, the circulating composition within the system circuit can be obtained [12] as given in Eq. (5).

$$C_{(i)} = \frac{M_{(i)}\sum\Delta M_{(i)}}{M_{\text{sys}} - \Delta M_{\text{sys}}} \tag{5}$$

where $\Delta M_{\text{sys}} = \Delta M_{2\text{ph}} + \Delta M_{\text{oil}} + \Delta M_{\text{LR}}$ $\Delta M_{\text{LR}} = \Delta M_{\text{vap(LR)}} + \Delta M_{\text{liq(LR)}}$ $\Delta M_{(i)} = \Delta M_{(i)2\text{ph}} + \Delta M_{\text{LR}}$ for systems with a receiver $\Delta M_{(i)} = \Delta M_{(i)2\text{ph}}$ for systems without a receiver

In obtaining Eq. (5), based on results of Kruse and Wieschollek [10], it is considered that the circulating refrigerant concentration throughout the circuit (e.g. at the compressor inlet or the expansion valve inlet), except in hold up sections, is the same. The contributions of differential solubility of oil and the effect of

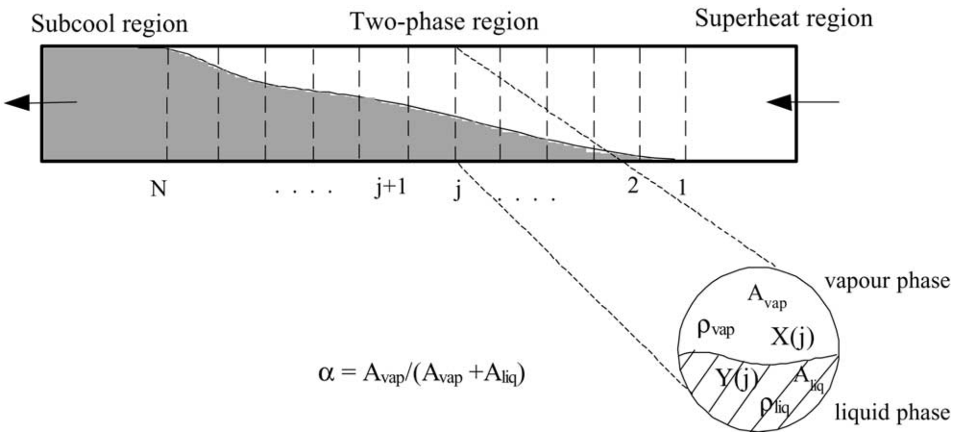

Fig. 2. Model for element by element analysis of the 2 phase section of condenser.

crankcase oil on the composition shift are negligible compared to the two-phase hold ups [3,12], i.e. $\Delta M_{oil} \approx 0$.

2.3. Analysis

Steady state performance of mixture RHP systems that do not include a receiver has been investigated computationally, using a distributed parameter heat exchanger model [9], for a range of Vr without considering the effects of the composition shift. Both the condenser and the evaporator use a phase-wise calculation approach.

Single-phase sections are considered as single elements of average fluid properties and the estimation of the required thermal quantities are based on NTU-effectiveness approach. For the two-phase frictional pressure drop, a two-phase multiplier proposed by Friedel [13] is used. A reciprocating compressor is employed in the study and the compression is represented by a polytropic process. The term 'volume ratio' denotes the ratio of the refrigerant side volume of the condenser to that of the evaporator in the heating mode. The sources of correlations used in [9] are listed in Table 1. The required refrigerant properties are estimated by internally calling up the property subroutines of NIST database 23, commonly known as REFPROP [17].

The present study stems from the above simulation [9], however, incorporates the issues of excess charge and concentration shift in relation to the presence of a liquid receiver. The solution algorithm, (Fig. 4), has two basic sections; the first part considering the thermodynamic conditions of the system, as given in [9], and the second handling the calculations involved the excess charge and mixture concentrations. To estimate the system charge, first the charge requirements in both the design and the reversed modes are determined. Difference between the two quantities represents the excess charge; a decision whether a given system needs a receiver is made. If a receiver is present, the estimated value is increased by about one-sixth (or about 16%) to represent the total system charge.

Table 1
Correlations used in [9]

Correlation	Reference
Void fraction correlation	Humark [5]
Pressure drops, two-phase multiplier	Friedel [13]
HTC, single phase refrigerant	Petukhov [14]
HTC, condensing	Bivens and Yokozeki [15]
HTC, evaporation	Collier and Thome [16]

3. Results and discussion

3.1. Influence of receiver on system design and the circulating composition

Table 2 presents the system parameters when a RHP is designed at a $Vr = 1$, with and without a receiver, for a nominal heating capacity of 7.5 kW. The results show that, when designed for R407C, the presence of a receiver causes a 12.5% drop in COP and about 15% increase in compressor displacement rate compared to a RHP without a receiver. The estimated charge requirement, without including the added 16%, is lower when a receiver is present (3.28 kg, as given within brackets). This is due to the absence of the liquid volume associated with the subcooled section when a receiver is present. The actual total system charge is however higher, i.e. 3.8 kg that includes the one-sixth added for the receiver.

The circulating concentrations show a positive shift, i.e. enriched with lower boiling components (R32 and R125) when compared with the original charged composition. This is due to the hold up of less volatile component (R134a) in the two-phase sections of the heat exchangers. These trends are similar to those observed by [3].

When a receiver is present, however, higher magnitudes of enrichment are observed. This is due to the additional hold up in the receiver and an increase in the hold up in the condenser (due to no subcooled section). At this instance the receiver holds about 0.43 kg of liquid $\left(\Delta M_{\mathrm{liq(LR)}}\right)$ and about 0.1 kg of vapour $\left(\Delta M_{\mathrm{vap(LR)}}\right)$ refrigerant. The total hold up in the receiver is almost the same as the refrigerant(s) held up in the evaporator, which is about 0.6 kg. The vapour in the receiver contains about 70% of R32 + R125, while the fraction of R134a in liquid is close to 60%. Of the total composition shift, about 20% is caused by the receiver; the rest is caused by the differential hold ups in the condenser and evaporator.

For the R134a systems it appears that the presence of the receiver causes only small changes in performance, and the charge requirements are roughly the same when excluding the additional one-sixth due to the receiver (i.e. 2.87 and 2.89 kg with and without a receiver respectively).

Without any subcooling it is necessary to increase the refrigerant mass flow rate to maintain a given capacity. The data show that the increase in refrigerant flow rates when a receiver is present is about 18% for R407C and 5% for R134a. This results in a larger compressor size required and the corresponding drop in COP.

Variations in circulating concentration (relative to the original concentration) in the heating mode and the running heating capacities (calculated based on the circulating compositions) are presented, for different Vr, in the Fig. 5a and b respectively. Both the capacity and the

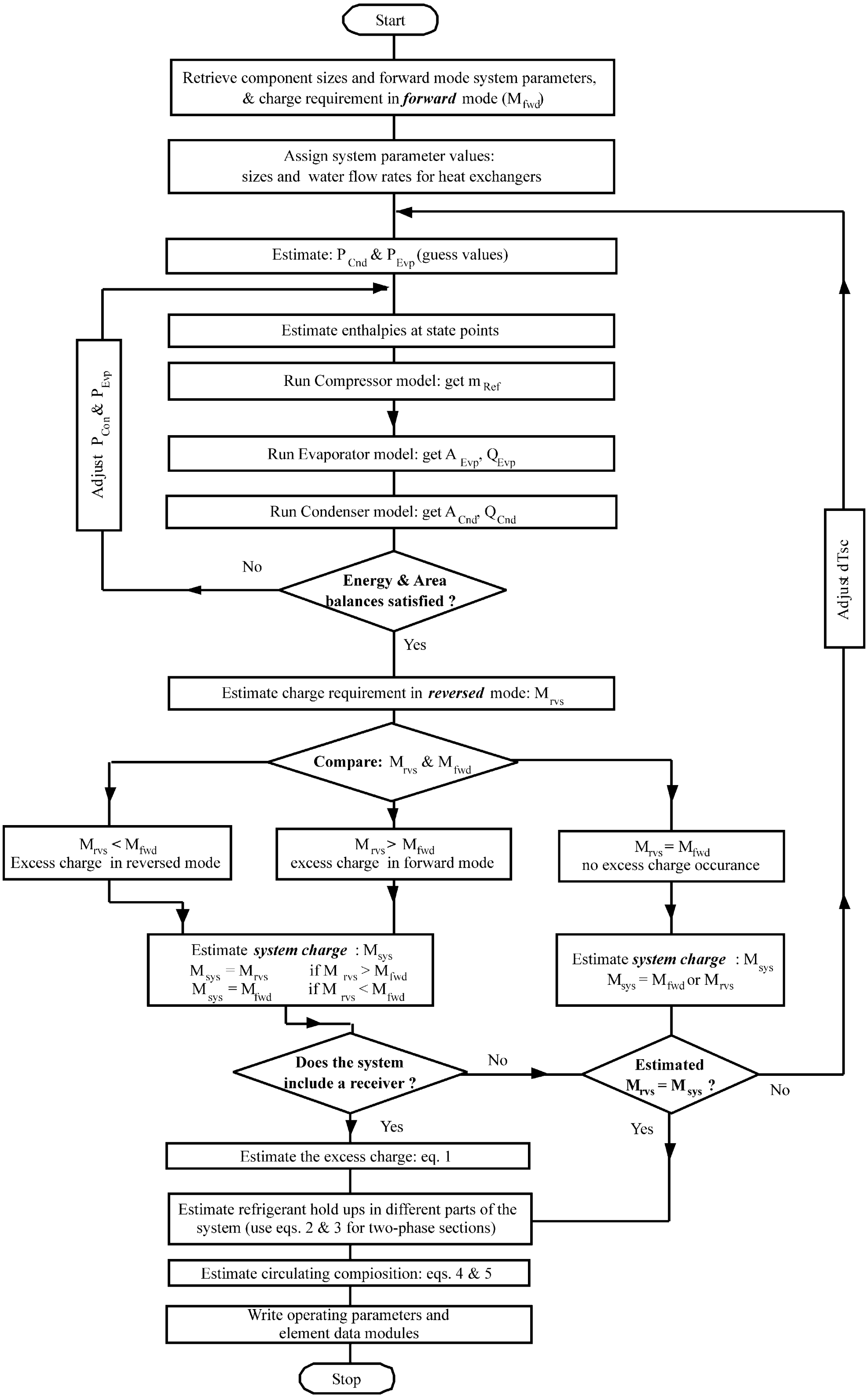

Fig. 4. Solution algorithm for predicting RHP performance with and without a liquid receiver.

Table 2
RHP designs with and without a receiver, $Vr = 1$

Parameter		R407C system		R134a system	
		No receiver	With a receiver	No receiver	With a receiver
COP		6.48	5.66	5.95	5.76
Compressor size (m^3/h)		6.10	6.99	10.07	10.42
System charge (kg)		3.55	(3.28) 3.80	2.89	(2.87) 3.36
Condenser area ($m^2 \times 10^2$)		0.33	0.33	0.25	0.26
Condenser volume ($m^3 \times 10^2$)		0.34	0.36	0.27	0.28
Evaporator area ($m^2 \times 10^2$)		0.72	0.70	0.56	0.57
Evaporator volume ($m^3 \times 10^2$)		0.35	0.34	0.27	0.27
Water flow rate (kg/s)					
Condenser		0.33	0.33	0.32	0.32
Evaporator		0.31	0.30	0.31	0.31
	Original composition	*Circulating composition*			
R32	0.230	0.301	0.365		
R125	0.250	0.305	0.362		
R134a	0.520	0.394	0.273		

concentration shift decrease with Vr, and the capacities are generally higher when a receiver is present, for all the Vr considered. In addition, as a result of the positive shift, the actual running capacities are larger than the nominal design value (7.5 kW).

3.2. Heat exchanger volume ratio and requirement for a receiver

To decide whether a liquid receiver is needed, the charge distributions within the system in both forward and reversed modes need to be assessed. Further different Vr can be analyzed to represent different systems, depending on the design and temperature requirements, and the load pattern. Systems designed for the same heating capacity will normally have different cooling capacities and COPs when reversed depending on the Vr.

Fig. 6 presents the charge requirement relative to that in heating mode, and the cooling performance of three R407C systems ($Vr = 0.7$, 1.0 and 2.1) designed without a receiver. Of the three systems, the one with a smaller condenser volume than that of the evaporator, i.e. $Vr = 0.7$, requires about 40% more refrigerant in cooling mode than during heating. Therefore, the system charge requirement has to be based on the cooling mode operation, so that a receiver is needed mainly to store the excess charge during heating; on the other hand, the receiver stores the excess charge in the cooling mode for $Vr = 2.1$. For $Vr = 1$, the charge requirements in both modes are of similar values, and further the system can operate without a receiver. Similar trends are also observed with R134a systems with different Vr. The observations suggest that systems with Vr of 1 or lower can be considered for the applications that need cooling for a longer period (energy efficiency consideration), while a design with $Vr \sim 2$ is a good candidate for the applications that need a larger cooling capacity for a shorter duration.

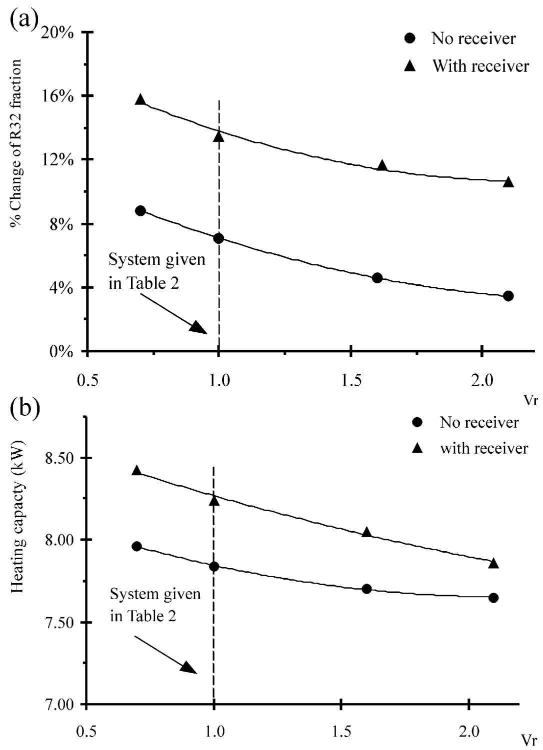

Fig. 5. (a) Percentage change of R32 mass fraction in the circulating composition at different Vr (heating mode) (b) Running heating capacity at different Vr.

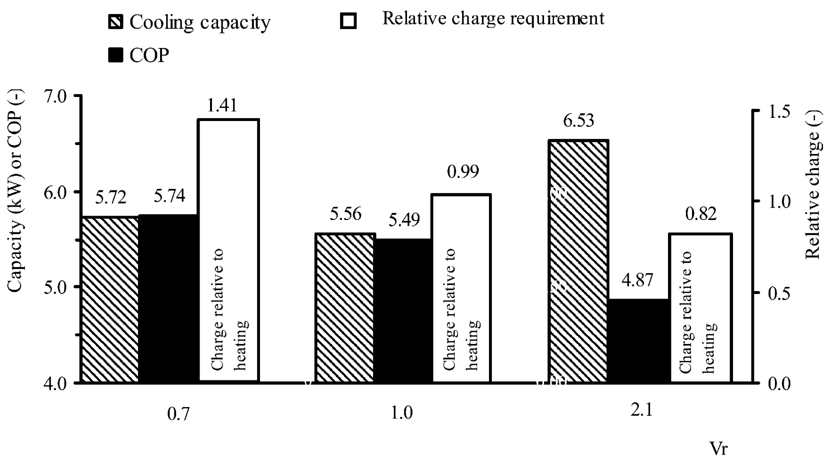

Fig. 6. Performance and charge requirement during cooling: R407C systems without a receiver.

Table 3
Charge distribution and circulating composition for R407C systems in cooling mode

Parameter	Volume ratio, *Vr*					
	0.7		1.0		2.1	
	No receiver	With a receiver	No receiver	With a receiver	No receiver	With a receiver
Total charge (kg)	5.86	6.02	3.55	3.80	3.08	3.36
Charge in receiver (kg)	–	1.08	–	0.61	–	1.34
Change of mass fractions in circulating composition						
R32	0.336 (+10.6%)	0.353 (+12.3%)	0.315 (+8.5%)	0.368 (+13.8%)	0.272 (+4.2%)	0.384 (+15.4%)
R125	0.334 (+8.4%)	0.360 (+11.0%)	0.321 (+7.1%)	0.370 (+12.0%)	0.285 (+3.5%)	0.388 (+13.8%)
R134a	0.330 (−19.0%)	0.287 (−23.3%)	0.364 (−15.6%)	0.262 (−25.8%)	0.443 (−7.7%)	0.228 (−29.2%)

3.3. Influence of receiver on circulating composition and performance in cooling mode

When three systems like Fig. 6 are designed with a receiver, the circulation concentrations in cooling mode and the corresponding changes of compositions (% values within brackets) relative to that of the original R407C are shown in Table 3. At $Vr = 0.7$, only about 18% (1.08 kg) of the total charge stays in the receiver, so that a significant portion of the total charge is distributed between the two heat exchangers during cooling. It is noted that with a receiver, the composition shift increases with increasing Vr, whereas an opposite trend is observed when a receiver is absent. As a result, at a higher Vr (=2.1), the differences between the circulation concentrations for systems with and without a receiver are significantly higher compared to the systems with lower Vr (=0.7 and 1.0), as a significant proportion of the system charge now stays in the receiver.

Fig. 7 presents the relative cooling performance of the three R407C systems designed with a receiver compared to the systems designed without a receiver. With a receiver, it is clear that the cooling COP drops by about 10% to around 20% whereas the capacities show certain improvements. The drops in COP are mainly due to the use of a relatively large compressor in each system, (17, 15 and 12% increase in compressor size at Vr of 0.7, 1.0 and 2.1 respectively) and the increase in pressure ratio with Vr (from 3.1 at $Vr = 0.7$ to 3.7 at $Vr = 2.1$). The increase in capacity is due to the combined effect of using a larger compressor and the increase in the amount of R32 and R125 in the circulating mixture.

For a pure fluid system the analysis shows that the influence of the receiver on the performance is negligible

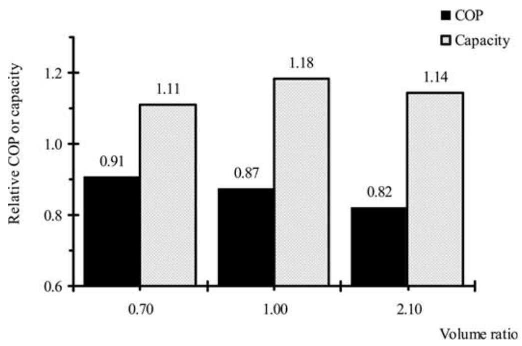

Fig. 7. Relative cooling capacity and *COP*: R407C systems with a receiver.

in the above context as the corresponding changes in compressor size are around 4%. Therefore, for pure fluid systems, the decision of including a liquid receiver is mainly based on the difference between relative charge requirements in heating and cooling modes.

The simulated *COP* values very much depend on the settings of the inlet temperatures of the HTFs, and the temperature differences between the HTFs and the refrigerant in the heat exchangers. Although the presented values are considered on the high side when compared to practical figures, the work mainly tried to obtain a relative comparison between systems of different *Vr*, with and without a liquid receiver, and using pure and mixture refrigerants. Further, the tube diameters too influence the system performance, e.g. a larger outer tube diameter increases the system charge and the composition shift, and deteriorates the cooling performances. However, simulations with different tube diameters show that the trends of performance and composition shift with *Vr* remain unchanged.

There was no accompanying experiment that validates the presented simulation. However, a limited validation on the predictions of the model was obtained using performance (experimental) data of Payne et al., [18] and composition shift data of Chen and Kruse [3,12]. The performance and composition change predictions agreed within 15 and 20% accuracy respectively with the data.

4. Conclusions

The influences of a receiver on the compressor sizing, circulating composition and performance of a water-to-water vapour compression reversible heat pump were investigated using a computer simulation. Depending on the design, excess charge can occur either in the heating or the cooling mode, and the amount of excess charge is a function of the heat exchanger volume ratio. The use of a receiver in mixture RHPs enhances the positive shift of circulating concentration, and systems with larger *Vr* show relatively higher shifts in the reversed mode. Overall, the presence of receiver improves both heating and cooling capacities (but not the *COP*) of mixture RHPs, regardless of the volume ratio, due to the increased shift and the use of relatively larger compressors, when compared with a system without one.

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