2006

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ABSTRACT

Refrigerant charge losses in an automotive air-conditioning system accumulated over a long period of time lead to inefficient operation. To identify characteristic steady-state behavior with respect to variable system charges an R134a refrigeration cycle is investigated experimentally and numerically. A closer look is taken at the receiver component, which is supposed to separate vapor from liquid and to balance changing refrigerant demands under different operating conditions. A clear receiver integrated in the experimental setup allows for a visual insight into its phase separation behavior and provides information used in the numerical model. Simulation and experimental results are compared in terms of measurable system variables. The receiver impact on those results as well as on the system's efficiency is discussed.

1. INTRODUCTION

Most cars nowadays found on the roads are, even in areas of prevailing mild conditions, equipped with an air conditioning system. Its role in affecting safety aspects, such as preventing fogged windows or conditions that impair the driver's concentration, has become more important in recent years. On the other hand air-conditioning causes an increased fuel consumption adding to the rising number of auxiliaries. During a vehicle's life cycle, refrigerant losses through leaks may occur over a long period of time leading to inefficient operation. In order to detect a decrease in refrigerant mass at an early stage and prevent damage to the system its characteristic behavior must be known in terms of variables that can be measured in the vehicle.

Simulation tools are a convenient way of reducing time consuming experiments especially for the creation of characteristic maps for a large amount of operating points. In order to be able to rely on numerical models they must first be validated with experimental data. To correctly represent a system's response to charge losses, all relevant effects have to be considered which describe the mass distribution among all cycle components under operation. The following investigations will concentrate on the receiver component and its impact on the system behavior with variable refrigerant charge.

In R134a-systems using a thermostatic expansion valve (TXV), the receiver is usually situated on the high pressure side following the condenser. Suction side accumulators instead are often combined with fixed orifices. Since the first type is increasingly used in automotive applications because of its wider operating range, this arrangement will be subject to the following investigations. In this paper the system is studied by a charge experiment carried out with a clear receiver made of glass that allowed for visually monitoring its internal state and by numerical system simulation, which offers the opportunity to examine component interaction at different levels of detail.

2. EXPERIMENTAL FACILITY

The experiments were carried out with standard parts of an automotive R134a refrigeration system: compressor, condenser, thermostatic expansion valve and evaporator (figure 1a). Both heat exchangers were of parallel flow design with extruded multiport tubes and louver fins. A swash plate reciprocating compressor with variable displacement volume was installed, but operated under full load only. Pipe dimensions as well as those of fittings on the test rig were larger than those of the respective system installed in a vehicle, therefore resulting in a larger total volume (especially of the liquid line) and a larger maximum refrigerant mass.

The receiver-drier which was situated between condenser and expansion device is normally made from aluminum. Its function is to provide an access storage volume in order to allow efficient operation under variable conditions and to separate liquid from vapor in case the condenser outlet is not subcooled under transient conditions. In addition, it contains a desiccant cartridge that removes moisture and particles from the refrigerant. For the investigations presented in this paper the aluminum receiver was replaced by a clear glass vessel of the same internal volume and equipped with a drier cartridge and an outlet pipe similar to those in a regular component (see figure 1b). The receiver was placed on scales to determine its refrigerant load. Rigid feed and outlet connections were replaced by flexible tubes.

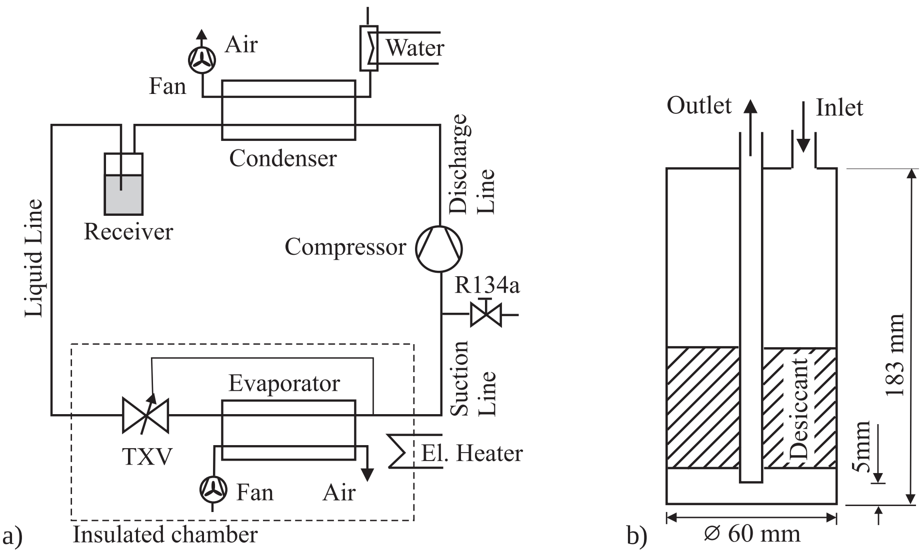


Figure 1: a) schematic of the experimental setup, b) high pressure side receiver

The evaporator was placed inside an insulated chamber, outlet air was heated to a defined temperature before it was fed back to the heat exchanger’s inlet. This air recirculation avoids condensation of water vapor, thus eliminating the influence of icing on the results. Condenser supply air was conditioned using water at constant temperature. Both air flows were driven by fans. Inlet and outlet temperatures were taken at a 4 by 4 measuring grid of K-type thermocouples.

3. NUMERICAL INVESTIGATION

3.1 Simulation Environment

The numerical investigations of charge distribution in the refrigeration cycle were carried out using a commercial simulation environment which is based on the object-oriented modeling language Modelica. Modelica is a free language specification, which allows convenient component-oriented modeling of complex physical systems. The language is developed by the non-profit Modelica Association[1]. Its non-causal approach enables the modeler to write physical equations independent from their numerical solution, which is especially helpful in thermodynamic

[1] http://www.modelica.org

system applications where fluid property transport and hydraulic resistance point into opposite directions or where the flow direction is even undefined at the modeling stage. A Modelica model forms a system of ordinary differential, algebraic and discrete equations (DAE) which is symbolically manipulated (including index reduction) and solved by the simulation environment. For numerical integration in the present case a modified DASSL solver is used. Since Modelica supports an object-oriented model structure, replacement of model parts is conveniently done, such as e.g. changing the level of detail or switching on and off certain model assumptions. Almost all models, including those provided by free or commercial model libraries, are in Modelica source code form which allows for a flexible combination of readily available and own developed parts.

3.2 Component Models

The numerical model of the refrigeration cycle is composed of separate models for each component. Most component models are taken from the AirConditioning Library, a commercial library of Modelica models in the area of automotive air conditioning systems. The Modelica language only supports ordinary differential equations in the time domain. Spatial gradients must therefore be approximated by one dimensional discretization. The models used are based on a finite-volume method following an upwind scheme (Tummescheit, 2002). A homogeneous mass flow is assumed in the two phase region, thus neglecting the relative velocity (or slip) of the two phases. The fluid is assumed to be pure refrigerant without lubricant. Compressor and TXV models are based on characteristic maps, the heat exchangers are described by physical equations and semi-empirical correlations from the open literature. Similar modeling approaches have been used in previous works on CO_2 and R134a refrigeration cycles (Limperich *et. al*, 2003, Pfafferott, 2004). Although the presented simulation task only requires steady state values, they are conveniently achieved by transient simulation in the same way as the experimental results: refrigerant is added stepwise during simulation using a mass flow source connected to one of the refrigerant lines.

3.3 Receiver Model

Under steady state conditions the refrigerant state leaving the receiver will be similar to that of fluid entering the component if pressure drop and heat transfer across the component are negligible. This means that a subcooled state at the condenser outlet can only occur in combination with a receiver completely filled with liquid refrigerant. Since a few degrees of subcooling define an optimum cycle operation as will be shown further below, this will be the normal condition. Only during very fast transients (e.g. sudden and significant change in compressor speed) or in a critically undercharged system, phase separation may actually occur in high pressure side receivers situated downstream of the condenser outlet.

The governing equations that determine the dynamic increase or decrease of refrigerant in the receiver component are the conservation of mass (eq. 1) and energy (eq. 2).

$$V\frac{\mathrm{d}\rho}{\mathrm{d}t} = m_{flow,in} - m_{flow,out} \tag{1}$$

$$V\frac{\mathrm{d}(\rho u)}{\mathrm{d}t} = m_{flow,in}h_{in} - m_{flow,out}h_{out} \tag{2}$$

V is the volume of the receiver, ρ is the mean density of the fluid in the component, m_{flow} is the mass flow rate, u is the specific internal energy of the fluid, h is the specific enthalpy at inlet and outlet, respectively. The changes in potential and kinetic energy are neglected in the energy balance as well as heat dissipation across inlet and outlet areas and heat transfer across the walls. The outflow rate is determined by the downstream hydraulic resistance and the pressure in the receiver. If the receiver inlet properties ($m_{flow,in}$, h_{in}) are determined by the upstream volume, the only remaining unknown is the outlet specific enthalpy h_{out}. Its determination therefore governs the charge transportation in and out of the component with respect to the assumed model. The two basic ideas which represent ideal mixing (MX) and an ideal phase separation (PS), respectively, are as follows.

Ideal mixing (MX):

$$h_{out} = h \tag{3}$$

The specific enthalpy at the outlet equals the mean specific enthalpy in the component *h*. This assumption is also valid for pipe flow at steady state.

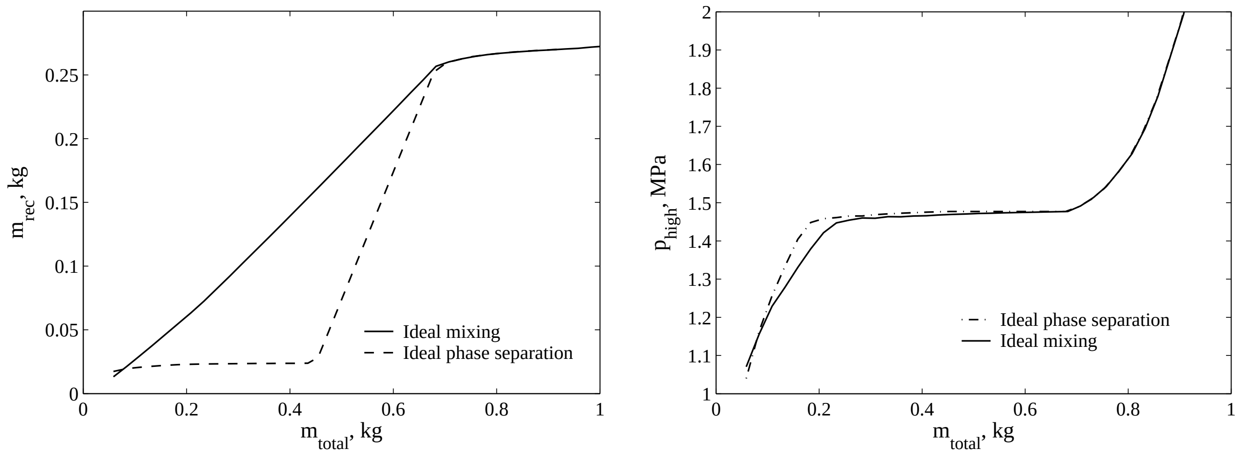
Ideal phase separation (PS):

$$l = \frac{(1-x)\rho_{vap}}{x\rho_{liq} + (1-x)\rho_{vap}} H \tag{4}$$

$$h_{out} = \begin{cases} h_{vap} & \text{if } 0 < l < H_{out} \\ h_{liq} & \text{if } H_{out} \le l < H \\ h & \text{if } l = 0 \text{ or } l = H \end{cases} \tag{5}$$

The outlet enthalpy is determined by the liquid level *l* in the component. The steam quality *x* in the receiver and the densities at saturation, ρ_{vap} and ρ_{liq} are used to compute the liquid level. If this is below the outlet H_{out}, saturated vapor is leaving the component, if the level is above H_{out}, the outflow is saturated liquid. If no level is present ($l = 0$, $l = H$), the model is equal to the ideal mixing assumption.

Figure 2 presents the difference of the two models in a charge simulation with a similar component configuration as the one used in the experiments, but a smaller total volume. Refrigerant was added in 50 g steps, plotted values were taken after steady state was reached at every charge increase. Both approaches have in common, that the mass in the receiver rises almost linearly with increased total refrigerant mass. Since the MX receiver model behaves like any other volume in the liquid line (pipes etc.) the added refrigerant is evenly distributed among all liquid line components and its steam quality equals that of the condenser outlet. The PS receiver state instead remains close to vapor density with a level at the outlet height (5 mm in this case) as long as the remaining liquid line volume is in the two-phase region. As soon as the pipe volume before and after the receiver as well as the condenser outlet reaches the saturation curve, saturated liquid leaves the receiver independent from the liquid level. The level rises with added refrigerant which is now all charged to the receiver until that is completely filled. The condenser outlet state, including the pressure, remains constant during this time. Similar results for both models are achieved as soon as the complete liquid line is filled with liquid refrigerant, the condenser starts subcooling and the discharge pressure increases drastically due to a reduced total compressibility. In short, the main difference between phase separation and ideal mixing in a high pressure side volume is the ability to decouple outlet steam quality from the actual component state in the first.



a) b)

Figure 2: Mass accumulation (a) and discharge pressure (b) vs. the total charge in a system simulated with different receiver models.

The real receiver component consists of elements featuring both model ideas presented above. To estimate their respective relevance, photos of a clear receiver under operation at different system charges were taken (Figure 3).

Three observations were made. First, a liquid level is present at all stages with a small foam region if the level is around the outlet pipe or above the drier cartridge. Second, regions behaving ideally mixed at steady state include desiccant material and outlet pipe. Third, the desiccant material behaves like a flow barrier for the incoming two-phase flow, thus resulting in a semi-mixed region just above it, if the liquid level is further below. The receiver model used in simulations discussed further below consists of one mixed component and one phase separator with a volume ratio of V_MX/V_PS = 0.5. In steady state investigations the position of the two elements with respect to each other is irrelevant as long as the impact of pressure drop due to friction and sudden expansion or contraction is neglected. In the present model the hydraulic resistance was concentrated at the outlet of the receiver. Heat transfer from the fluid to the ambient was also neglected for the present, which would counterbalance evaporation resulting from a pressure drop along the flow path.

Figure 3: Mass flow through a clear high pressure side receiver under steady state operation and constant boundary conditions. Total system charge increases from left to right: 400g, 700g and 1200g.

3. EXPERIMENTAL AND SIMULATION RESULTS

3.1 Charge Experiments

The system as described in section 2 was charged with refrigerant through a suction line valve at 100g steps, starting with 200g after evacuation, while all other boundary conditions were kept constant. Steady-state measurements were taken after 10 minutes for each charge increase. Table 1 lists the boundary conditions of the presented experiment. The compressor speed was held at a low (idle) level to reduce compressor stress at low charges when lubrication is assumed to be insufficient.

Table 1: Experiment boundary conditions

Compressor		Condenser Air			Evaporator Air	
n, s^{-1}	V_{displ}, cm^3	c, m/s	m_{flow}, kg/s	T, K	m_{flow}, kg/s	T, K
16.6	168	2.04	0.67	313.6	0.130	314.2

The results were compared to those of simulation experiments with similar boundary conditions and component geometry using the receiver model described in the previous section. The refrigerant mass in the receiver was determined from the recorded weight at different total charges less the effective weight posed on the scales by momentum forces resulting from the fluid U-turn. Both, experimental and simulation results show three stages of charge increase (figure 4).

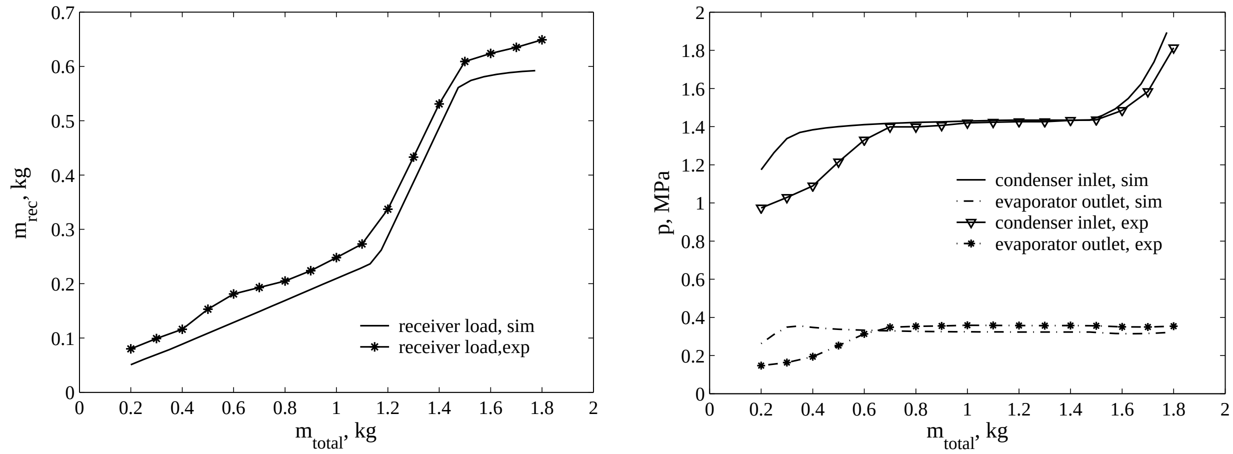
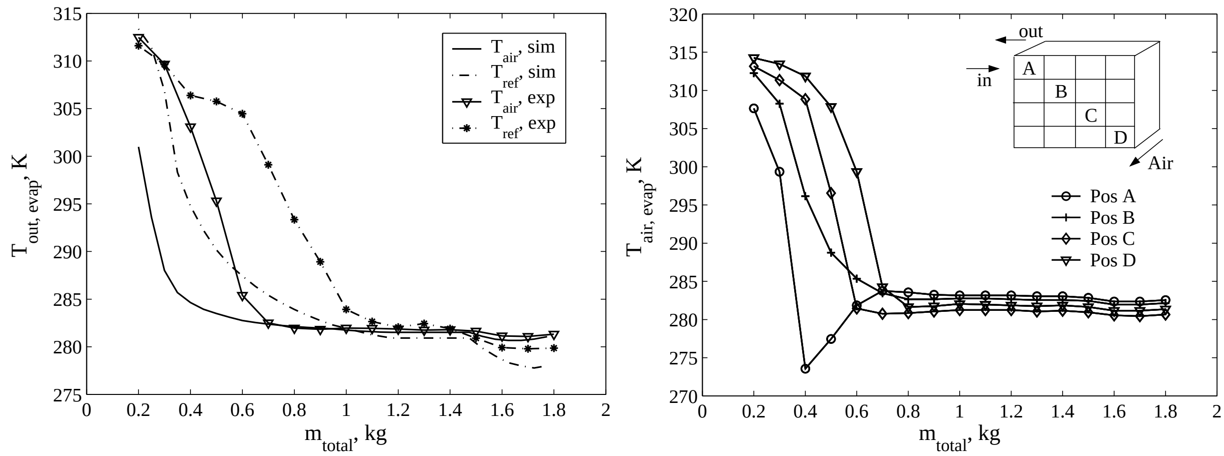


Figure 4: Refrigerant mass in the receiver and system pressures vs. total system charge, simulation and experiment

The first stage corresponds to an increased inlet and outlet density and a mass accumulation in parts of the receiver with mixed flow, including connections that have an impact on the scales. A larger slope is found at stage two, when the liquid level in the receiver rises at constant saturated inlet and outlet properties. Stage three represents a subcooled state in the receiver as well as at the inlet and outlet with a slight density increase with pressure. The absolute deviation between simulation and experiment may result from errors in the estimated receiver volume as well as from additional forces on the scales under operation. While the mass storage in the receiver is in good agreement with the experimental results, the simulated discharge pressure reaches the saturation plateau at charges too low. The experiment instead shows a moderate rise at the beginning with an increasing slope before reaching the plateau. At moderate and optimal charges the pressure results show a good agreement.

At low total charges the simulation seems to disregard additional mass storage outside the receiver. One reason may be the homogeneous flow model which neglects the relative velocity in the two-phase region. A great impact is also expected to originate from the compressor oil still present in the cycle after evacuation, which is not accounted for in the numerical model. Since the setup did not include an oil separator and lubricant retention is hard to predict from a theoretical point of view, the amount of oil present in the system at experiment start is unknown. An oil film was observed at the receiver bottom at the beginning of the experiment. An increasing amount of refrigerant dissolves in the oil with rising system pressure and therefore does not participate in the refrigeration process at that stage. In addition one has to consider that the semi-empirical compressor and TXV models are not validated in the low charge region.



a) b)

Figure 5: a) air and refrigerant outlet temperatures at the evaporator vs. total system charge. b) measured evaporator air outlet temperatures at different sensor positions vs. total system charge

Following from the differences in mass distribution at low charges, the deviation between simulation and experimental results in terms of the temperatures at the evaporator outlet is significant (figure 5a) but converges at moderate charges. If the air outlet temperature is used to determine an undercharged system, the position of the sensor may be essential to capture the expected characteristics, as demonstrated in figure 5b. The start of the two phase region in the evaporator is situated close to the inlet if significantly undercharged and moves toward the exit with increasing system charge. As a result, air temperatures at different evaporator outlet positions, may even show opposite characteristics at significant charge losses. Further investigations must show if a reliable detection of refrigerant loss in a wide range of boundary conditions may be possible using two temperature sensors at different positions.

3.2 Maximum Coefficient of Performance

The previous results showed a decrease in evaporator air outlet temperature at a few degrees of condenser subcooling due to an improved exploitation of the two-phase region. At the same time the entropy production in the expansion valve and the mass flow rate decrease. Accordingly, the Coefficient of Performance (COP), defined as the ratio of evaporator cooling load and compressor power at the shaft, rises at that stage before decreasing again due to a fast pressure increase, indicating an optimum refrigerant charge. In steady state operation condenser subcooling only occurs in conjunction with a subcooled downstream liquid line, including the receiver if it is situated downstream of the condenser. A receiver placed between the condensing and the subcooling part of the heat exchanger allows subcooling while still being in a two-phase state itself. The COP in the plateau region at lower charges is therefore larger. This type of condenser with integrated receiver-drier has become widely used mainly for packaging and production reasons (Ravikumar *et.al.*, 2005). The same maximum COP is still found at the same charge, but the integrated receiver system features a plateau COP closer to this extremum as can be seen from simulation results of a virtual system in figure 6. The location of the optimum slightly varies with respect to boundary conditions such as air inlet temperature and mass flow rate as well as compressor speed, the average COP under variable conditions may therefore be higher for an integrated receiver system especially at slightly undercharged conditions. But it must be noted, that the extremum may be less significant and differences between both configurations may be smaller than those presented in figure 6, and are also influenced by heat exchanger design and liquid line volume.

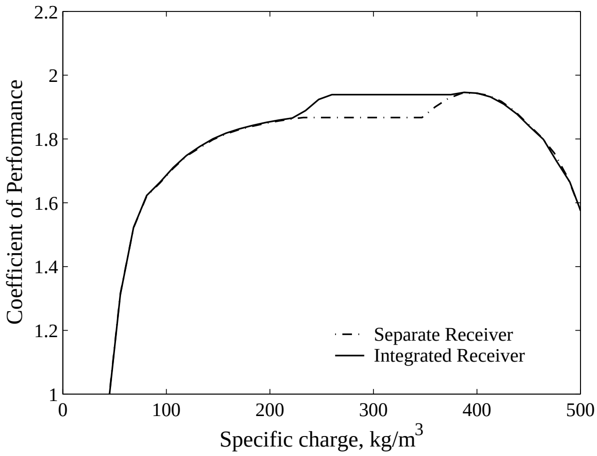


Figure 6: Simulation results of the Coefficient of Performance indicate an optimum refrigerant charge and dependence on receiver position

4. CONCLUSIONS

- Substitution of the standard high-pressure side receiver for a version made from a transparent material provides a visual insight into the mass storage behavior of the component in an R134a automotive refrigeration cycle. Phase separation was observed in the component as long as its overall state was in the two-phase region.

- The degree of phase separation in the receiver at low system charges determines the amount of refrigerant stored in the component influencing the system behavior, e.g. in terms of the discharge pressure. Comparison of experimental and numerical results shows a good agreement for the selected receiver model with respect to the mass stored in the component and properties at normal and optimal operating charges. However, it indicates that the overall model shows inaccurate prediction of system behavior at low total charges. The impact of model assumptions such as a homogeneous two-phase flow and a negligible compressor oil influence, as well as characteristic behavior of compressor and valve at low charges must be further investigated in a model analysis.

- System simulation with Modelica can be used for a convenient analysis of several system architectures with respect to an optimum refrigerant charge. Since a receiver integrated in the condenser component allows for a larger variation of total system charge while retaining a subcooled state at the condenser outlet, the average Coefficient of Performance over a period of charge decrease was shown to be higher with this configuration.

NOMENCLATURE

A	area	(m^2)
c	velocity	(m/s)
h	specific enthalpy	(J/kg)
l	liquid level	(m)
m	mass	(kg)
m_{flow}	mass flow rate	(kg/s)
n	compressor speed	(s^{-1})
p	pressure	(MPa)
T	temperature	(K)
t	time	(s)
ρ	density	(kg/m^3)
u	specific internal energy	(J/kg)
V	volume	(m^3)
x	steam quality	(kg/kg)

Subscripts

evap	evaporator
in	at inlet
liq	liquid phase
out	at outlet
rec	receiver
vap	vapor phase

Abbreviations

COP	Coefficient of Performance
MX	ideal mixing model
PS	phase separation model
TXV	thermostatic expansion valve
sim	simulation
exp	experiment

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