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Refrigerant Charge Reduction in Small Commercial Refrigeration Systems

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ABSTRACT

This paper presents analysis of location of refrigerant inventory and theoretical results of charge reduction in a typical bottle cooler. The original unit was first characterized by determining its capacity, the coefficient of performance (COP), pull-down characteristics and total charge using non-destructive instrumentation. After that the unit was instrumented by insertion of valves to separate main components and allow measurements of the charge in each of them, as well as pressure transducers, mass flow meter and thermocouples. Condenser is found to be the component that retains the most charge. The system was modelled using Engineering Equation Solver (EES) for calculation of performance and charge. Models are fully validated using steady state experimental results. Discussion of the results and direction for the charge reduction is provided.

1. INTRODUCTION

It is clear that charge reduction is a necessity for each and every refrigerant. For HFC's, refrigerant charge reduction decreases direct component of CO₂ effects in life cycle climate performance calculations. Recent trends in the air conditioning and refrigeration industry, and specifically in some commercial and other stationary applications, justifies revisiting the issue because of flammable (like R290, R600a, and similar) and mildly toxic refrigerants (like NH₃). Significant charge reduction, specifically below 50 g of total charge, might open some doors for application of hydrocarbon systems. Also, lower refrigerant charge may lead to lower leakage, and lower leakage means less damage to the environment in respect of ozone depletion, global warming, and so forth. However, problems in operation may occur if charge becomes too low: Small leaks if charge stored in receiver is inadequate may result in insufficient charge and reduced performance; Charge reduction achieved by reducing the hydraulic diameter of pipes too severely may affect efficiency and perhaps capacity due to pressure drop and flashing; Reliability of pumps (if present) are reduced due to possible vapor entrainment and cavitation.

Various methods have been tried to reduce charge in refrigeration systems: Choosing an alternative refrigerant; Pipe sizing; Sizing and loading the liquid receivers; Adopting secondary refrigeration; Selecting proper heat exchangers, compressors, and expansion valves (Poggi *et al.*, 2008). Among those methods, selecting proper heat exchangers is of our interest in this paper. For direct-expansion evaporators and condensers, probably the easiest way to reduce charge is to flatten the tubes. Flattened tubes have a higher internal surface-to-cross-sectional ratio comparing with the original round tubes. By flattening tubes, the heat transfer coefficient and the pressure drop increase simultaneously for a given refrigerant mass flow rate during either boiling (Nasr *et al.*, 2010) or condensing (Wilson *et al.*, 2003). Mass flux will also increase, so that the flow pattern will be changed from stratified flow to annular flow earlier than low mass fluxes in the evaporator. This is beneficial for heat transfer since the annular flow has a higher heat transfer coefficient rather than stratified flow (Nasr *et al.*, 2010). In order to flatten the tubes of heat exchangers without an increase in pressure drop and decrease in heat transfer, it should be combined with some other modifications like increasing the number of parallel channels. Also, for more compact heat exchangers like microchannel heat exchangers, refrigerant amount inside is dramatically reduced to about 10 g/kW if headers are correctly designed (Hrnjak 2010). Extra benefits of using microchannel heat exchangers include smaller size and less weight. In literature, an ammonia chiller with an air-cooled condenser and a plate evaporator was studied by

Hrnjak and Litch (2008). The charge was reduced to 20 g/kW, and the major contribution came from the use of microchannel aluminum tubes.

The aim of the present research is to analyze charge reduction methods and their consequences, adopting flattened condenser tubes in particular. First, a baseline system is characterized by determining its capacity, COP, and optimal charge. Then charge distribution of the baseline is determined by using Quick Closing Valve Technique and Remove and Weigh Technique (Peuker and Hrnjak, 2010). The condenser is found to be the component that retains the most charge. Additionally, a system model is built for performance and charge calculations, which are used to explore various options to reduce charge.

2. DESCRIPTION OF THE BASELINE SYSTEM

The environment to test the bottle cooler is a closed chamber with air control vents in the ceiling and outlets at the bottom as required by ASHRAE Standard 72/1983. A fan, installed outside of the chamber, provides air circulation. To regulate the temperature, a PID controlled heater warms the air up before it enters the chamber ceiling. The bottle cooler is basically a vapor compression refrigeration system; condenser is finless round-tube type, evaporator is round-tube with plate fins type, and expansion device is capillary tube. The optimal charge is 260 g of R134a as provided by manufacturer. At first, one heater and two watt meters were installed to the system as received; heater was installed inside the cabinet of the bottle cooler to help stabilize the working cycle and prevent it from on/off cycling, and one watt meter was installed at the electrical supply of the unit (includes the compressor, two heat exchanger fans, lights and control), and the other one at the heater. A data logger was used to collect data from the two watt meters and several T-type thermocouples every two seconds. There are independent on-and-off control of the compressor, two heat exchanger fans, and lights, so from separate experimentation, the power and heat of each heat exchanger fan (W_{cfan} , W_{efan} , Q_{cfan} and Q_{efan}), the light and control ($W_{l\&c}$ and $Q_{l\&c}$), and the overall heat transfer coefficient (UA) between the cabinet and the ambient were obtained. With these, cooling capacity (Q_e) and COP of the system at a certain condition could be determine by applying energy conservation of the cabinet. Calculations are shown in Equations (1) through (5).

$$Q_{evap} = Q_{heater} + Q_{trans} + Q_{efan} + Q_{l\&c} \quad (1)$$

$$Q_{trans} = UA \times (T_{ambient} - T_{cab}) \quad (2)$$

$$W_{system} = W_{cp} + W_{cfan} + W_{efan} + W_{l\&c} \quad (3)$$

$$COP_{cycle} = Q_{evap} / W_{cp} \quad (4)$$

$$COP_{system} = Q_{evap} / W_{system} \quad (5)$$

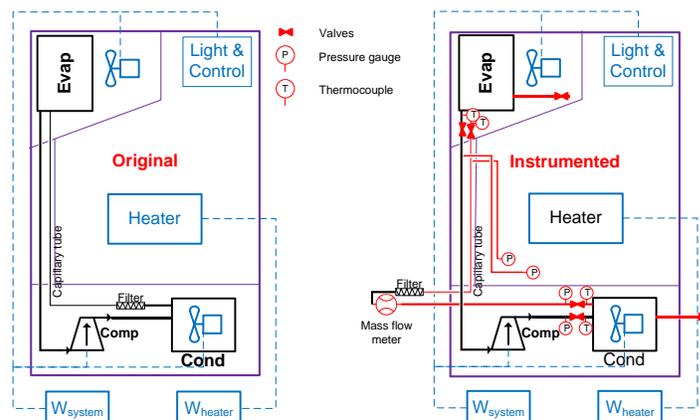


Figure 1 Schematic figure of the original (left) and instrumented (right) unit.

The original system needed to be further instrumented with sensors to validate the model later, so one mass flow meter, four pressure transducers, and more T-type thermocouples were added to the system. Valves were also

inserted as shown in Figure 1, which separate the system into four main parts (condenser, evaporator, compressor and liquid line), and allow measurements of the charge in each of them. The performance (capacity and COP) of the instrumented system was determined by using the same method described above. It was verified that the performance does not vary between the instrumented system and the original one. Based on rough estimation, the optimal charge for the baseline should be 330 g of R134a due to the instrumentation. Also, charge optimization experiments show that the system is insensitive to the charge amount as long as it is in the range of 280 g and 360 g. See Figure 2.

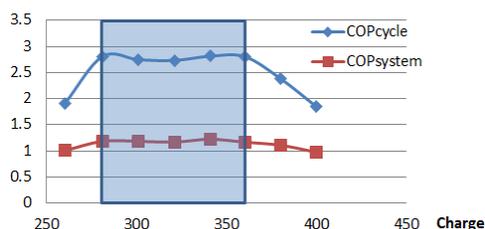


Figure 2 Optimal charge is between 280 - 360 g, corresponding to the highest COP.

3. CHARGE DISTRIBUTION IN THE BASELINE SYSTEM

When steady state was reached (heater is turned on to 358.2 W), four valves were closed at the same time and the system was shut down simultaneously (Quick Closing Valve Technique). By doing so, refrigerant was trapped in each main component (condenser, evaporator, compressor and liquid line). Then in sequence, refrigerant was recovered to a cylinder, which is cooled by liquid nitrogen. Finally, the cylinder was weighed, and the refrigerant weight in each component was calculated (Remove and Weigh Technique). The results are shown in Table 1 and Figure 3. Notice the 'Unknown' part in the table is the charge extracted again from the whole system after long enough time, which is mainly due to the presence of oil inside.

Table 1 Charge distribution in the baseline unit.

Condenser	Evaporator	Compressor	Liquid line	Unknown	Total collected	Actual charge
146.7 (g)	28.3 (g)	20.6 (g)	129.3 (g)	8.1 (g)	333.0 (g)	355.8 (g)

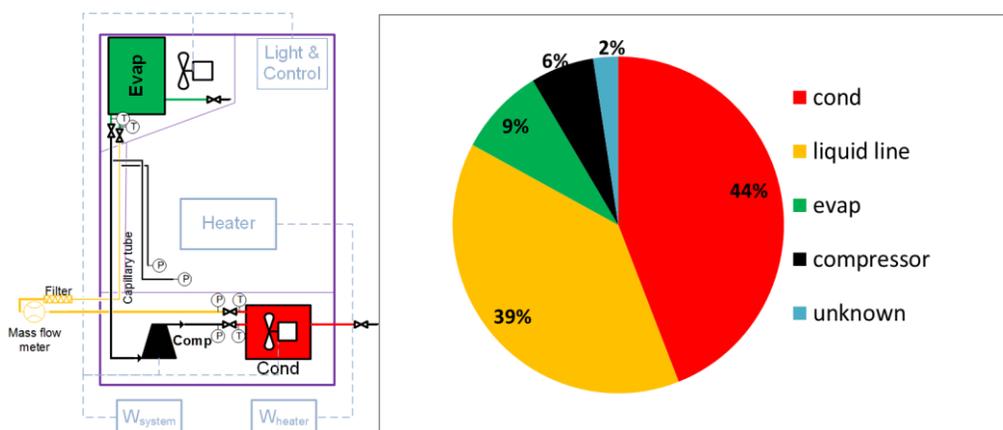


Figure 3 Charge distribution in the baseline unit.

From Table 1 and Figure 3 above, we conclude:

- In total, 333.0 g of refrigerant were collected back from the 355.8 g charged. The error is within 7%.
- Most of the charge (44%) is retained in the condenser.
- Liquid line retains the second largest amount of charge (39%). However, it should be noticed that the installation of a mass flow meter in the liquid line enlarged this part. For the original unit, charge retained in the liquid line would be around 20%.
- Evaporator and compressor retain some charge as well.

Charge in each section of the system depends on its internal volume of its constituting components and the state of refrigerant. Charge is simply mass, which is the product of volume and density, and the higher the internal volume and density, the more charge it is. For the state of refrigerant, three different kinds should be distinguished: superheated, two-phase, and subcooled (states above the critical point are not discussed). Most of the connection pipes contain single-phase refrigerant (either superheated or subcooled); receiver and accumulator always contain two completely separated phases (liquid and vapor); heat exchangers contain two-phase refrigerant due to phase change and single-phase refrigerant. For example, refrigerant in condenser usually would go through all three states – starts with superheated, then two-phase, and last subcooled. Most of the charge is due to the two-phase and subcooled parts with high density of the refrigerant. That is why in the experiment condenser and liquid line are found to be the components that retain the most charge (more than 80% of the total charge). For the evaporator, the inlet state of the refrigerant is two-phase (if isenthalpic expansion is assumed), and the outlet state is superheated vapor; the evaporator only results in 9% of charge retention. For the compressor with connection tubes, the charge is around 6%, which is due to the low density of the vapor state. To conclude, for the charge reduction purpose, only components containing subcooled or two-phase refrigerant and with large internal volume are of main concern.

4. MODEL DESCRIPTION

Jin and Hrnjak (2013) experimentally measured refrigerant and oil charge in automotive AC systems. They have also developed a model and validated it by their experiments (Jin and Hrnjak, 2014). Here a verified steady state model was built to predict the performance and charge of the baseline system. The model includes four main procedures, the condenser, evaporator, compressor, and connection tube models.

4.1 Heat Exchanger Model

Effectiveness - Number of Transfer Unit (ϵ -NTU) method and Finite Volume Method are adopted in the heat exchanger model. The condenser is divided into 416 finite volumes and the evaporator is divided into 266 finite volumes. Within each volume, the outlet refrigerant state could be calculated if the inlet state and mass flow rate are given. We use the outputs of this volume as the inputs of our next volume. This way, heat transfer, pressure drop and charge can be calculated in every volume. The correlations chosen are based on the heat exchanger type. See Table 2. And the same principles apply to the connection tube model as well.

Table 2 Correlations used in the condenser and evaporator models.

		Condenser (Round tube)		Evaporator (Round tube w/ plate fins)
Air side	1-phase	HTC	Churchill and Bernstein (1977)	Kim <i>et al.</i> (1999)
	1-phase	HTC	Gnielinski (1976)	Same
Refrigerant side	1-phase	DP/f	Churchill (1977)	Same
		HTC	Cavallini <i>et al.</i> (2006)	Gungor and Winterton (1986)
	2-phase	DP	Friedel (1979)	Same
		α	Homogeneous, Zivi (1964), Rouhani and Axelsson (1970), Graham <i>et al.</i> (1999), ...	

4.2 Compressor Model

For compressor model, the volumetric efficiency and the isentropic efficiency are obtained by applying polynomial regression from the data of specification.

$$\eta_{isentropic} = 0.53226 - 0.00193 \times (P_{cpro}/P_{cpri}) - 3.32428 \times 10^{-4} \times (P_{cpro}/P_{cpri})^2 \quad (6)$$

$$\eta_{volumetric} = 0.80172 - 0.01657 \times (P_{cpro}/P_{cpri}) - 9.31741 \times 10^{-5} \times (P_{cpro}/P_{cpri})^2 \quad (7)$$

The compressor model is built based on equations (6) and (7). The inputs to the model are P_{cpri} , h_{cpri} and P_{cpro} , and the outputs are \dot{m}_r and h_{cpro} . In the model, heat dissipation from the compressor shell is considered. The compressor shell temperature is estimated to be the mean temperature of the suction and discharge temperatures, and the overall heat transfer coefficient (UA) between the ambient and the compressor shell is set to be 24 W/K based on experimental data.

4.3 Charge Model

For single phase (either superheated or subcooled) fluid, charge is calculated using $M=\rho V$, where density ρ is determined by the mean temperature and pressure of the inlet and outlet of the volume. For two-phase, charge is calculated by $M=[\rho_v\alpha+\rho_l(1-\alpha)]V$, where α is the void fraction, which varies according to different void fraction correlations.

4.4 Void fraction correlations

Several void fraction correlations are tested, including Homogeneous, Zivi (1964), Armand (1946), Lockhart and Martinelli (1949), Rouhani and Axelsson (1970), Niño *et al.* (2002), and Graham *et al.* (1999). Homogeneous, Zivi (1964), Armand (1946), and Lockhart and Martinelli (1949) correlations show a fixed relationship between quality and void fraction. For some other correlations, void fraction is not only a function of quality, but also a function of mass flux, such as Rouhani and Axelsson (1970), Niño *et al.* (2002) and Graham *et al.* (1999) correlations. The higher the mass flux, the higher the void fraction would be. For Graham *et al.* (1999) correlation, void fraction is also a function of internal diameter (void fraction increases as internal diameter decreases). Rouhani and Axelsson (1970) correlation has been recommended as having an accurate predictive capability compared with other correlations (Woldesemayat and Ghajar, 2007). Also, Graham *et al.* (1999) correlation is recommended because it accounts for the effects of tube diameter on void fraction. As the tube diameter decreases, the mass flux would increase for a given mass flow rate, and flow region might change, which in turn affects void fraction. Related work by Hrnjak (2010) shows higher mass fluxes result in higher void fraction.

4.5 System model

See Figure 4. The system model is identical to solving 14 unknowns - 7 points with 2 properties each (pressure and enthalpy). In order to solve the 14 unknowns, we need 14 equations. For the compressor model, if we know $P_{c,pr}$, $h_{c,pr}$ and $P_{c,pro}$, we could get the mass flow rate \dot{m}_r and $h_{c,pro}$, so it is counted as 1 equation. For two heat exchangers and three connection tubes (See points 2-3, 4-5, and 7-1 in Figure 4), if P_{inlet} , h_{inlet} are known, P_{outlet} and h_{outlet} could be calculated, so each one is counted as 2 equations and we would have 10 equations. The capillary tube is assumed to be isenthalpic, which is counted as 1 equation. In total, we have 12 equations, and we need 2 more. By adding two more parameters (the superheated degree at the compressor inlet and the subcooled degree at the condenser outlet), the system model can be solved.

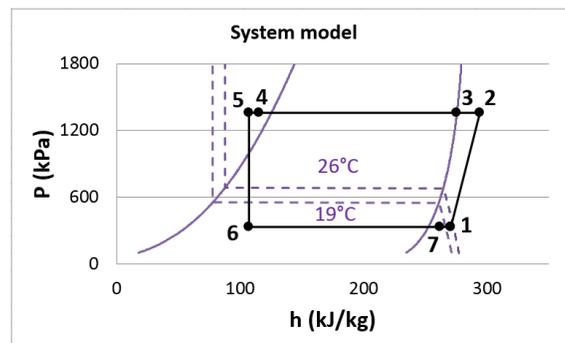


Figure 4 System model at the condition of $T_{ambient}=26\text{ }^{\circ}\text{C}$, $P_{atm}=100\text{ kPa}$, and $W_{heater}=358.2\text{ W}$.

4.6 Charge model validation

At a specific condition ($T_{ambient}=26\text{ }^{\circ}\text{C}$, $P_{atm}=100\text{ kPa}$, and $W_{heater}=358.2\text{ W}$), the experimental cycle is determined by data from measurements. All the points are determined by pressure and temperature (averaged values), except for the point 6 (the inlet of the evaporator), which is determined by its enthalpy and pressure. The experimental cycle matches well with the system model.

For the charge model, Rouhani and Axelsson (1970) and Graham *et al.* (1999) void fraction correlations are chosen because of their accurate predictions. See Table 3. The error of charge between the experiment and the model might come from:

- Experimental error. The experiment of determining charge distribution was conducted by three people altogether to close the valves and shut down the system power at the same time. Operation by people might cause error. So does the extraction process.

- Model imperfection. There are ideal assumptions about the model, so the model itself has errors. For example, the mass flow rate calculated from the compressor model (2.5 g/s) slightly deviates from the one measured by mass flow meter (2.4 g/s).

Table 3 Charge model results compared with experimental data.

	Homogeneous	Error	Zivi	Error	Rouhani & Axelsson	Error	Graham <i>et al.</i>	Error	Experiments
	(g)	(-)	(g)	(-)	(g)	(-)	(g)	(-)	(g)
Condenser	120.4	-17.9%	158.1	7.8%	171.3	16.7%	165.2	12.6%	146.7
Evaporator	7.0	-75.3%	15.5	-45.2%	24.1	-14.8%	23.0	-18.7%	28.3

5. PREDICTION OF CHARGE REDUCTION BY MODEL

5.1 Select smaller round tube of the condenser

First, we consider using a smaller diameter of the condenser tube. If so, the internal volume would decrease correspondingly, as would the charge. However, pressure drop of the condenser would increase and the heat transfer would drop due to a smaller heat transfer area between the refrigerant and air compared with the original. See the results predicted by the model below.

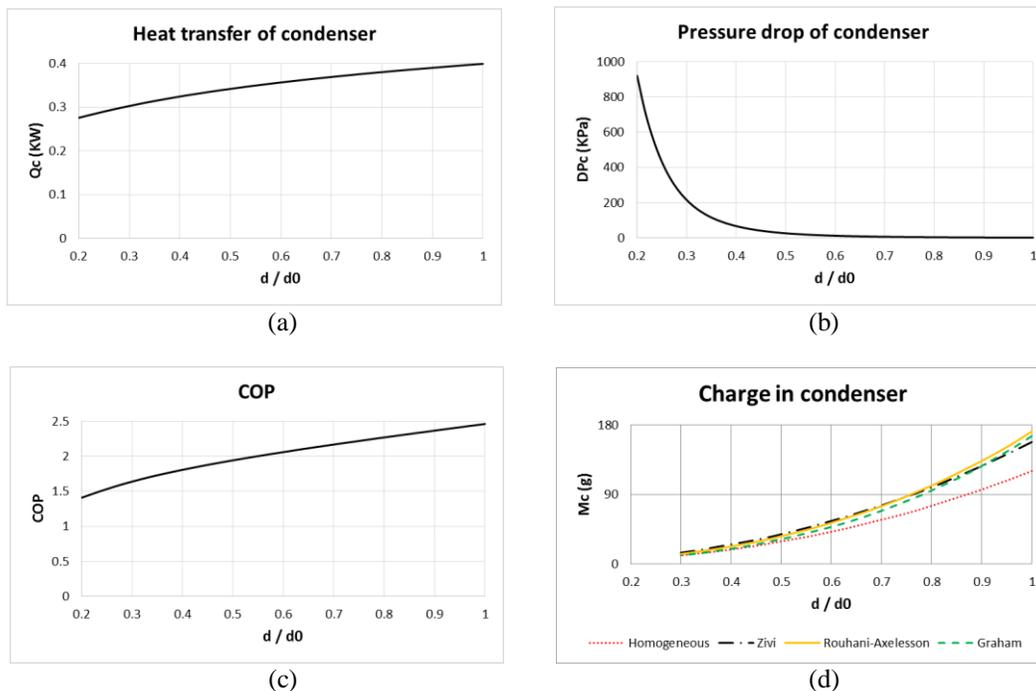


Figure 5 As the diameter of the condenser tube decreases, (a) heat transfer decreases, (b) pressure drop increases, (c) COP decreases, and (d) refrigerant amount decreases.

See Figure 5(a), heat transfer of the condenser decreases as the diameter of the round tube decreases, which is mainly due to a smaller heat transfer area. See Figure 5(b), as the diameter of the round tube decreases, the pressure drop increases slowly at first. Then as it continues to decrease (less than half of the original diameter), the pressure drop begins to increase dramatically, requiring more and more compressor power, which leads to a lower COP as shown in Figure 5(c). See Figure 5(d), Four void fraction correlations are chosen to calculate the charge in the condenser; Homogenous correlation predicts the lowest charge, while Zivi (1964), Graham *et al.* (1999) and Rouhani and Axelsson (1970) predicts closely.

5.2 Flattened tube of condenser

The original round tube has an internal diameter of 6.922 mm, an external diameter of 7.938 mm (5/16 inch), and a cross section area of 37.63 mm². The condenser tubes are modelled to be successively flattened into an oblong shape without change of the tube thickness ($\delta = 0.5080$ mm) and inner perimeter (21.75 mm). Air side heat transfer coefficient correlation is changed to forced convection perpendicular to non-circular tubes (Jakob, 1949). Model results are shown in Figure 6.

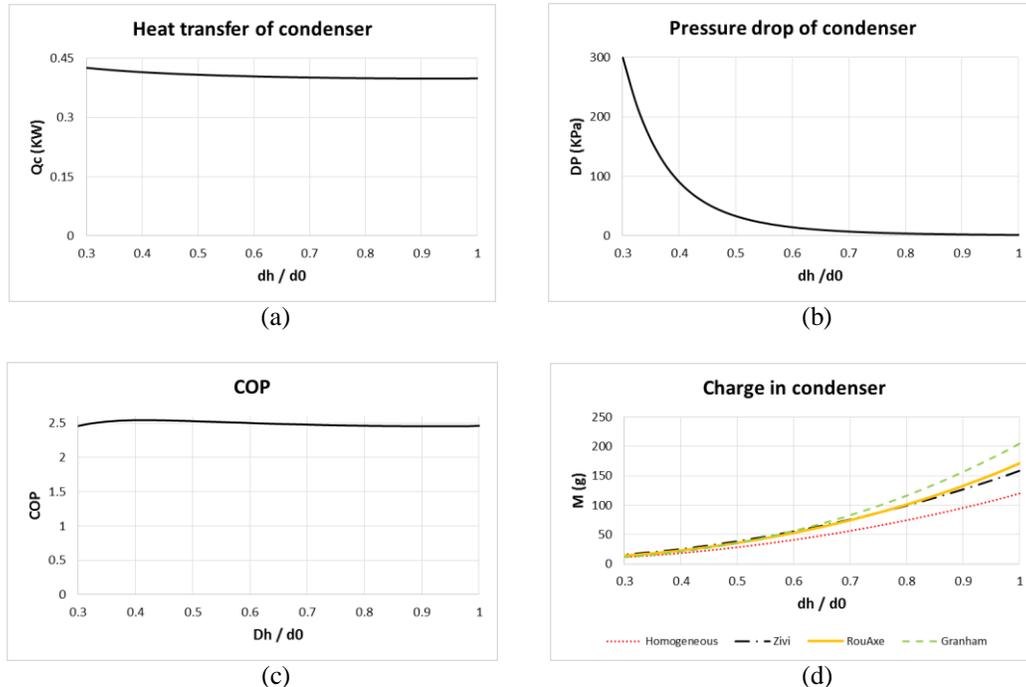


Figure 6 As the condenser tube is being flattened, (a) heat transfer increases, (b) pressure drop increases, (c) COP first increases, then decreases, and (e) refrigerant amount decreases.

See Figure 6(a), heat transfer of the condenser increases a little, because the air side heat transfer coefficient increases due to a late separation of boundary layer and a smaller pressure drop. The refrigerant side heat transfer coefficient and the heat transfer area do not change much. See Figure 5(b), as the tube is being flattened, the pressure drop increases slowly at first and then increases dramatically, requiring more and more compressor power. See Figure (c), an increase of heat transfer in condenser and evaporator dominates at first, which causes an increase in COP; after the COP reaches a peak, it begin to drop, which is due to the increase of pressure drop. See Figure 5(d), Four void fraction correlations are chosen to calculate the charge in the condenser; Homogenous correlation predicts the lowest charge, while Zivi (1964), Graham *et al.* (1999) and Rouhani and Axelsson (1970) predicts closely. Compare with smaller tubes with the same cross-sectional area, the flattened tubes are more beneficial for heat transfer and COP.

CONCLUSIONS

- Most of the charge is retained in the condenser and liquid line, while a small portion of charge is retained in the evaporator and compressor. Condenser contains two-phase and subcooled refrigerant, and the liquid line contains subcooled refrigerant, which lead to a large amount of charge retention.
- Based on the model prediction, flattening the finless-round-tube of the heat exchanger to some proper extents is a simple way to reduce charge without penalizing the system performance.

NOMENCLATURE

COP	coefficient of performance	(–)
d	tube diameter	m
d ₀	original tube diameter	m
D _h	hydraulic diameter	m
DP	pressure difference / pressure drop	kPa
f	friction factor	(–)
h	specific enthalpy	kJ/kg
HTC	heat transfer coefficient	W / (m ² -K)
M	charge	g
<i>m</i>	mass flow rate	g/s
P	pressure	kPa
Q	heat transfer	W
T	temperature	°C
UA	overall heat transfer coefficient	W / K
V	volume	m ³
W	power	W

Subscript

atm	atmosphere
cab	cabinet
cp	compressor
cfan	condenser fan
evap	evaporator
efan	evaporator fan
l&c	light and control
i	inlet or inner
o	outlet or outer
r	refrigerant or refrigerant side
trans	transfer between the cabinet and the ambient

Greek Letter

α	void fraction	
δ	tube thickness	mm
η	efficiency	(–)
ρ	density	kg/m ³

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