Postprint: A Feasibility Study on Double-Row Liquid-Distribution Condensers for R410A Air Conditioning Systems

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Abstract

This study employs an R410A household split-type air conditioner as the prototype, constructs a liquid-separated condenser air conditioning system using a novel double-row liquid-separated condenser, and investigates its system matching and comprehensive performance. Utilizing a double-row liquid-separated condenser with theoretically optimized tube arrangement, this work experimentally examines the performance variation patterns of the liquid-separated condenser air conditioning system under capillary tube lengths of 300 mm–800 mm and refrigerant charges of 950 g–1350 g, and compares them with the prototype system. The results demonstrate that under nominal cooling conditions, compared with the prototype system, the optimal liquid-separated system exhibits 5.1% higher energy efficiency ratio (EER) and 4.2% higher cooling capacity, requires 16.7% less capillary tube length, while having 8.6% higher refrigerant charge.

Full Text

Preamble

Investigation on a R410A Air Conditioning System with Substitution of Double-Row Liquid-Vapor Separation Condenser

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Abstract

This study employs a R410A household split-type air conditioner as the prototype to construct a liquid-vapor separation condenser (LSC) air conditioning system using a novel double-row liquid-vapor separation condenser (D-LSC), investigating its system matching and comprehensive performance. Using a double-row liquid-vapor separation condenser with a theoretically optimized tube pass arrangement, the performance variation of the LSC air conditioning system was experimentally studied under capillary tube lengths of 300 mm to 800 mm and refrigerant charge amounts of 950 g to 1350 g, with comparisons made against the prototype system. The results demonstrate that under nominal refrigeration conditions, the optimal LSC system achieves an energy efficiency ratio (EER) and cooling capacity 5.1% and 4.2% higher than the prototype system, respectively, while requiring 16.7% less capillary tube length and 8.6% more refrigerant charge.

Keywords: system performance; double-row liquid-vapor separation condenser; cooling capacity; EER

Introduction

The air conditioning industry represents a traditionally high energy consumption sector. As one of the two major heat exchangers in air conditioning systems, improving the thermal performance of condensers constitutes an important pathway for system energy savings. Numerous scholars have investigated the influence of tube pass arrangement on condenser thermal performance [1,2,3], finding that reasonable tube pass numbers and arrangements can significantly reduce internal flow pressure drop and effectively enhance heat transfer performance [4]. When the aspect ratio of condensers or refrigerant flow rate increases, the flow uniformity within the condenser improves [5]. Further research on the impact of condenser structure on air conditioning system performance has revealed that optimizing condenser flow arrangement can effectively improve overall system performance [6]. Additionally, Wang Yifei [7] and Illán-Gómez [8] conducted studies on microchannel condensers applied to household air conditioning systems, demonstrating that microchannel condensers can improve system COP.

Beyond enhancing condenser performance, refrigerant charge amount and throttling device structure have also been investigated [9,10]. Experimental findings indicate that insufficient charge is more detrimental to cooling capacity and COP than overcharge [11], while different capillary tube structures can produce similar effects on refrigeration systems through diameter and length adjustments [12].

Peng Xiaofeng [13] proposed the liquid-vapor separation condenser based on vapor-liquid separation heat transfer enhancement mechanisms, subsequently leading to in-depth research on single-row liquid-vapor separation condensers [14,15]. The structure of double-row liquid-vapor separation condensers differs from single-row versions, with varying heat loads between front and rear tube rows and refrigerant mixing, vapor-liquid separation, and redistribution processes occurring in the headers between rows. The overall thermal characteristics have not received dedicated study, and research on their comprehensive performance in air conditioning systems remains blank. This paper substitutes a double-row liquid-vapor separation condenser into a conventional air conditioning system to investigate its impact on comprehensive system performance.

1.1 Two Types of Condensers

[Figure 1: see original paper] shows the novel double-row liquid-vapor separation condenser (D-LSC). The D-LSC consists of two rows of finned tubes, several "Y" -shaped three-way connecting tubes, and a pair of collection headers. The front and rear tube rows are connected at both ends by "Y" -shaped three-way tubes before entering the same header. Perforated baffles called vapor-liquid separators are installed at appropriate positions in the header at the junction between adjacent tube passes, with hole diameters of 1.0 mm to 2.0 mm. These baffles divide the condenser into several passes with different numbers of heat transfer tubes. Except for the baffles at the first pass and subcooled pass inlets, all other baffles have small holes drilled in them. The vapor-liquid mixture undergoes separation on the perforated baffles; due to the pressure difference across the baffle and the density difference between vapor and liquid phases, condensate drains downward along the header. Finally, capillary force creates a liquid seal on the baffle while vapor-phase refrigerant enters the next pass to continue the condensation process. This maintains high dryness fraction condensation in the D-LSC tube sections, thereby enhancing heat transfer. This study employs the segmented calculation method proposed by Hua [10] to obtain the optimal tube pass arrangement for a 12-pass D-LSC (48 tubes) based on the prototype system condenser structure, with tube numbers along the flow path of 8-6-6-4-4-4-4-4-2-2-2-2. Since no accurate theoretical calculation method exists for baffle hole design, the optimal D-LSC was selected from five empirically designed units through experimentation.

[Figure 2: see original paper] shows the prototype air conditioning system' s double-row serpentine condenser (D-SC), with the same number of heat transfer tubes and total heat transfer area as the D-LSC. Refrigerant enters the condenser through four parallel circuits from the rear row based on the windward side and exits from the front row, with "U" -shaped elbows connecting tubes between row ends. The structural dimensions of both condensers are listed in Table 1 .

Table 1 Geometric Parameters of the Two Condensers

Parameter	Value
Finned tube inner diameter /mm	6.7
Finned tube outer diameter /mm	7.3
Header length (D-LSC) /mm	506

Parameter	Value
Circuit length (D-LSC) /m	9.576
Hole diameter/Number of holes in 2nd baffle /mm	(data missing)
Hole diameter/Number of holes in 3rd baffle /mm	(data missing)
Hole diameter/Number of holes in 4th baffle /mm	(data missing)
Hole diameter/Number of holes in 5th baffle /mm	(data missing)
Hole diameter/Number of holes in 6th baffle /mm	(data missing)
Hole diameter/Number of holes in 7th baffle /mm	(data missing)

Note: The original table contained incomplete data. The hole parameters for baffles 2-7 were empirically determined as: 1.5/3, 1.5/4, 1.5/4 & 1/1, 2/4 & 1/2, and 1.5/3 (diameter in mm/number of holes).

To obtain the condenser' s heat transfer performance, T-type thermocouples were installed at both ends of all heat transfer tubes to measure adiabatic wall temperatures, while 24 T-type thermocouples were placed between front and rear rows and at the rear row air outlet to measure inter-row air temperatures. [Figure 2: see original paper] shows the thermocouple installation points on the condenser.

1.2 Air Conditioning Unit Configuration

This experiment utilized a 1.5HP split-type fixed-frequency household air conditioning unit from a domestic manufacturer as the prototype, shown in [Figure 3: see original paper]. The unit employs a scroll compressor, tube-fin evaporator, tube-fin serpentine condenser, cross-flow indoor fan, axial outdoor fan, and capillary tube as the throttling device. The main component configurations are listed in Table 2 . The only structural difference between the liquid-vapor separation condenser system (D-LSC system) and prototype system (D-SC system) lies at both ends of the condenser tube section. Both systems underwent experimental matching calibration to determine the capillary tube length and refrigerant charge amount that yield maximum EER.

Table 2 Configuration of the Original Air Conditioning Unit

Component	Specification
Compressor	ASD102CDPA8JT (Highly)
Condenser	Tube-fin serpentine type
Evaporator	Tube-fin type
Indoor fan	Cross-flow /W
Outdoor fan	Axial /W
Rated power	1100 (RP)

Note: The original table contained incomplete parameter data.

1.3 Experimental System

This study employed a standard enthalpy difference experimental system to test both air conditioning units, as shown in [Figure 4: see original paper]. The system consists of indoor and outdoor chambers with constant temperature and humidity capabilities, each equipped with refrigeration units, heaters, and humidifiers for temperature and humidity control. The dry- and wet-bulb temperatures of the indoor and outdoor chamber environments were collected using air samplers and measured with platinum resistance thermometers. Enthalpy differences before and after heat exchange were measured in the air duct, which includes a flow equalizing box, differential pressure gauge, nozzle flow meter, and induced draft fan.

Air in the initial state enters the duct after heat exchange with the evaporator, mixes thoroughly in the flow equalizing box, then has its dry- and wet-bulb temperatures measured by platinum resistance thermometers. A nozzle flow meter measures air volume, which is finally discharged through the induced draft fan. The experimental environment regulation and parameter measurement processes are controlled and implemented by an automatic control system. Electrical signals from various parameters were collected by an MX-100 data acquisition unit and transmitted to a PC terminal for data processing and export. The measurement instruments and their accuracies are listed in Table 3 .

According to GB/T 7725-2004, the outdoor chamber environment was set to a dry-bulb temperature of 35°C and wet-bulb temperature of 24°C, while the indoor chamber environment was set to 27°C and 19°C, respectively. Test air conditioning units were installed with capillary tubes ranging from 300 mm to 800 mm in 100 mm increments for sequential testing. After removing non-condensable gases from the unit, refrigerant was charged into the system through a charging tube maintained horizontally at both connection ends. A ferrule fine-tuning valve (model: SS-VJ10-ML6-FM1) precisely controlled the flow rate, with both charging tube and fine-tuning valve fixed on a steel bracket. Refrigerant R410A was charged in 50 g increments from 950 g to 1350 g. The MX-100 data acquisition unit collected experimental data every 10 s. When the test unit operated continuously and stably for 45 min in the standard environment, the PC terminal output and stored data. This experiment used maximum EER as the matching criterion.

Table 3 Uncertainty of Measuring Apparatus

Instrument	Model	Accuracy
Platinum resistance thermometer	TR/02010 Pt100	±0.1 K
Pressure sensor	PT517Z	±0.5%
Nozzle flow meter	WX-BP	±0.1%
Thermocouple	-	±0.5 K
Electronic scale	ZW3433B	±0.5%
Precision balance	BH-30	(data missing)

1.4 Experimental Error Analysis

The total experimental error in this study consists of refrigerant charging operation error and system measurement transmission error, which are independent and therefore additive.

The absolute operation deviation for refrigerant charging in the experiment is ±2 g. The maximum relative operation error for charge amount is ±0.21%.

System measurement error follows Type B uncertainty calculation methods, with measured values following a uniform distribution. The relative uncertainty of air volume flow rate is ±0.06%.

The uncertainty of cooling capacity is (air specific volume and humidity ratio obtained from tables and treated as constants): The maximum relative uncertainty of cooling capacity (including charging operation error) is ±2.55%.

The total power consumption of the refrigeration system equals system electrical power. Its relative uncertainty is: The maximum uncertainty of total power consumption (including charging operation error) is ±0.50%.

The system EER uncertainty is calculated as: The maximum relative uncertainty of system EER (including charging operation error) is ±2.79%.

2 Data Processing Methods

The air conditioning system cooling capacity is: where Ga is air flow rate, m³h ¹; ha,i and ha,o are air inlet and outlet specific enthalpy, Jkg ¹; Va is air specific volume, m³kg ¹; Wa is humidity ratio per unit mass of air, kg(water vapor) · kg ¹(dry air).

The condenser heat load is: where Ptot, Pfan,eva, and Pfan,con are total unit power consumption, indoor fan power, and outdoor fan power, respectively, W.

The heat load ratio between front and rear tube rows of the condenser ( ): where Tai, Tam, and Tao represent initial air temperature, inter-row air temperature, and outlet air temperature, K; subscripts row 1 and row 2 denote front and rear rows.

The refrigeration system EER is: EER Q P

The total condenser pressure drop is: P P P where P1 and P2 are condenser inlet and outlet pressures, respectively.

3 Results Analysis

[Figure 5: see original paper] shows the variation of D-SC system EER with different refrigerant charge amounts and capillary tube lengths. Under experimental conditions, the optimal system EER is 3.16, corresponding to a refrigerant charge of 1050 g and capillary tube length of 600 mm. System EER decreases significantly with both insufficient and excessive charge. With a 600 mm capillary tube, system EER at 950 g charge is 7.9% lower than at 1050 g, while at 1350 g charge it is 9.8% lower than at 1050 g. When charge is insufficient, excessive superheat at the compressor outlet reduces compression efficiency, while inadequate flow rapidly decreases cooling capacity. With excessive charge, increased refrigerant accumulation in the evaporator and condenser raises flow resistance, increasing compressor power consumption while cooling capacity increases insignificantly, resulting in obvious EER decline. Refrigerant charge amount affects EER more than capillary tube length. With insufficient charge, overall system EER is lower, and shorter capillary tubes yield higher overall EER. With clearly excessive charge, system EER also declines significantly, and shorter capillary tubes produce lower EER. This occurs because insufficient charge with shorter capillary tubes increases refrigerant mass flow and cooling capacity, while excessive charge with shorter capillary tubes increases mass flow and compressor power consumption, raising evaporator and condenser pressure drop and reducing their heat transfer, thus lowering system EER.

[Figure 6: see original paper] shows the variation of D-SC system cooling capacity with different charge amounts and capillary tubes. The cooling capacity corresponding to the highest system EER is 3.403 kW, while the maximum cooling capacity achieved is 3.547 kW, corresponding to a refrigerant charge of 1200 g and capillary tube length of 600 mm. The charge amount for maximum cooling capacity is 14.3% higher than that for maximum EER, while the cooling capacity is 4.2% higher. At this point, compression efficiency is lower than at maximum EER, but increased refrigerant flow raises system flow resistance and heat transfer in both evaporator and condenser, achieving maximum cooling capacity but with reduced EER. With clearly excessive refrigerant (1250 g to 1350 g), system cooling capacity tends to decrease because excess refrigerant accumulates in local condenser and evaporator tubes, thickening the liquid film, increasing flow resistance, and reducing heat transfer efficiency. Capillary tube length also affects cooling capacity. The maximum cooling capacities corresponding to different capillary tubes differ by up to 2.8%, with different required charge amounts. Reducing capillary tube length increases refrigerant mass flow, but shortened throttling raises post-throttling refrigerant temperature. Under the competing effects of these two factors, different capillary tubes produce noticeably different cooling capacities.

[Figure 7: see original paper] shows the variation of D-LSC system EER with different charge amounts and capillary tubes. The maximum system EER is 3.32, corresponding to a refrigerant charge of 1150 g and capillary tube length of 500 mm. System EER continuously decreases when deviating from the optimal charge amount under different capillary tubes. Compared with [Figure 5: see original paper], the optimally matched D-LSC system EER is 5.1% higher than the optimally matched D-SC system. The D-LSC system optimal EER corresponds to a charge amount 4.7%-9.4% higher than the D-SC system, yet the D-LSC system demonstrates better adaptability to refrigerant overcharge. At 1350 g charge, D-LSC system EER is 4.5%-8.6% lower than its maximum, while D-SC system EER is 5.9%-10.8% lower. The optimal D-LSC system requires a capillary tube length 16.7% shorter than the D-SC system. According to Zhong [11], when refrigerant mass flow inside the tube is low, the heat transfer coefficient in a liquid-vapor separation condenser is lower than in a serpentine condenser, but the heat transfer enhancement effect becomes prominent at higher mass flow rates.

To compare the relative impact of capillary tube length and refrigerant charge amount on D-LSC system performance, the optimal D-LSC values were individually changed to those from the D-SC system. When the optimal D-LSC system charge deviates to 1050 g, system EER is 4.1% lower than the optimum. When the optimal D-LSC capillary tube length deviates to 600 mm, system EER is 3.7% lower than the optimum. Therefore, charge amount affects D-LSC system performance more significantly than capillary tube length.

[Figure 8: see original paper] shows the variation of D-LSC system cooling capacity with different charge amounts and capillary tubes. The D-LSC system achieves a cooling capacity of 3.478 kW at optimal EER, while the maximum cooling capacity is 3.696 kW, corresponding to a refrigerant charge of 1250 g and capillary tube length of 500 mm. The maximum cooling capacity is 6.3% higher than the cooling capacity at optimal EER, while the charge amount is 8.7% higher. Since liquid-vapor separation condensers exhibit more pronounced heat transfer enhancement at high cross-sectional flow velocities, increasing refrigerant charge by the same amount from the optimal EER point yields greater cooling capacity increase in the D-LSC system than in the D-SC system. With insufficient refrigerant charge, D-LSC system cooling capacity decreases rapidly, while with excessive charge (approximately >1250 g), cooling capacity decreases slightly. Compared to maximum cooling capacity, at 1350 g charge, system cooling capacity decreases by only 1.9%-2.8%, indicating lower sensitivity of D-LSC system cooling capacity to refrigerant overcharge. The maximum difference in cooling capacity among different capillary tubes is 4.8%, demonstrating that capillary tube length significantly affects D-LSC system performance.

[Figure 9: see original paper] shows the variation of condenser tube-side pressure drop with refrigerant charge amount for D-LSC and D-SC systems under optimal capillary tubes. Both D-LSC and D-SC system condensing pressure drops increase with charge amount, but stabilize or even decrease slightly with excessive charge (approximately >1250 g). As charge amount initially increases, refrigerant mass flow and condensing temperature continuously rise, significantly increasing condenser pressure drop. With excessive charge, liquid refrigerant accumulates locally in the condenser and evaporator, stabilizing system refrigerant flow. Meanwhile, continuously increasing condensing pressure reduces refrigerant viscosity and specific volume, resulting in 平缓 or slightly decreasing pressure drop. Refrigerant pressure drop in the D-LSC is significantly lower than in the D-SC, with the former being 32.0%-38.5% lower than the latter. As Zhong [15] noted, continuous liquid removal through vapor-liquid separation increases the relative flow cross-section for the vapor phase, weakening relative flow between vapor and liquid phases and reducing interfacial shear stress. Additionally, reasonable tube pass arrangement creates more balanced refrigerant velocity between passes, which are the main reasons for the low pressure drop in liquid-vapor separation condensers.

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Note: Figure translations are in progress. See original paper for figures.