



Performance Analysis on Evaporators in Air Source Heat Pumps: A Comparative Study of Finned-Tube and Microchannel Heat Exchangers with Varying Superheat under Dry Conditions

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Abstract

This study analyzes the thermal performance of evaporators in air source heat pumps, focusing on various operational conditions that affect thermal resistance. The research is conducted on a specialized test rig equipped with a variable adjustable refrigeration circuit within a climate-controlled chamber, allowing for precise simulation of real-world scenarios. This setup enables comprehensive evaluation of heat transfer and frost behavior under controlled conditions, including variations in temperature, humidity, and airflow rates. In the study, two distinct types of heat exchangers are examined: A fin-and-tube heat exchanger and four microchannel heat exchangers. These designs are compared based on their thermal performance, pressure drop characteristics, and susceptibility to frost buildup. By varying the operational parameters, especially the refrigerant heat exchanger outlet superheat, the study aims to understand how this factor influences the rate and pattern of thermal performance and subsequent frost accumulation on the evaporators. The findings of this research offer valuable insights into optimizing the operation of evaporators in air source heat pumps, particularly concerning superheat controlling. Additionally, the benefits and drawbacks of different evaporator types are discussed, and potential optimization pathways for microchannel heat exchangers are outlined to promote their increased implementation in heat pumps.

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Selection and/or peer-review under the responsibility of the organizers of the 15th IEA Heat Pump Conference 2026.

Keywords: Heat Pump; Evaporator; Frosting; Microchannel

1. Introduction

Microchannel heat exchangers (MCHXs) are increasingly tested as evaporators in air-source heat pumps (ASHPs) due to their superior energy efficiency, compact design, and reduced refrigerant charge requirements. However, MCHXs face distinct operational challenges that require innovative solutions to maximize their potential.

One major challenge impacting the performance of MCHX is refrigerant maldistribution, a phenomenon where uneven refrigerant distribution across the microchannels causes inconsistent superheating within the tubes. This significantly reduces heat transfer efficiency and undermines system performance. Zou et al. (2014) [1] highlighted how improper refrigerant distribution in MCHXs with vertical headers can lead to capacity reductions, stressing the importance of optimizing header geometry and flow characteristics. Zanetti et al. [2] analyzed MCHX experimentally and numerically. Due to refrigerant maldistribution at a superheat of 6K they reduced the effective heat transfer surface area by 7% to 14% in their models to account for non-utilizing of surface area. Numerical research by Hu et al. [3] demonstrated the effects of refrigerant maldistribution within

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flat tubes. The study emphasized that improving even refrigerant distribution among microchannels (by geometry change) could enhance overall heat transfer by approximately 30%, due to a reduction of superheated region. Bowers et al. [4] showed maldistribution effects in MCHX for different orientation of the tubes (vertical and horizontal). Liquid refrigerant distribution was improved when the evaporator header was oriented vertically with flow entering at the bottom.

Beyond refrigerant-side maldistribution, the frosting and defrosting behaviour of the evaporator is a crucial point for performance. Several researchers have analysed performance in frosting mode, new type of geometry or surface coating [5–11] to decrease the amount of frost and/or increase the defrosting time. To the authors knowledge only one commercially available air-source heat pump for residential application with a MCHX as evaporator is available on the European market [12].

2. Test Setup

The experiments were carried out on Fraunhofer ISE's heat exchanger test facility [13]. Evaporator prototypes are mounted as single-component samples inside an airtight climate chamber and connected to a refrigeration cycle unit using R290 (see Figure 1 left). The R290 unit has an additional superheater in order to allow two-phase flow exiting the evaporator and calculate its thermodynamic state by energy balance of the superheater. The outlet state of the superheater is superheated vapor. Air is conditioned outside the climate chamber by an air handling unit and supplied to the chambers back wall. Air is then directed through the heat exchanger and returned to the air handling unit. Figure 1 shows the test facility. The key parameters of the two units for air and refrigerant supply are described in Table 1.

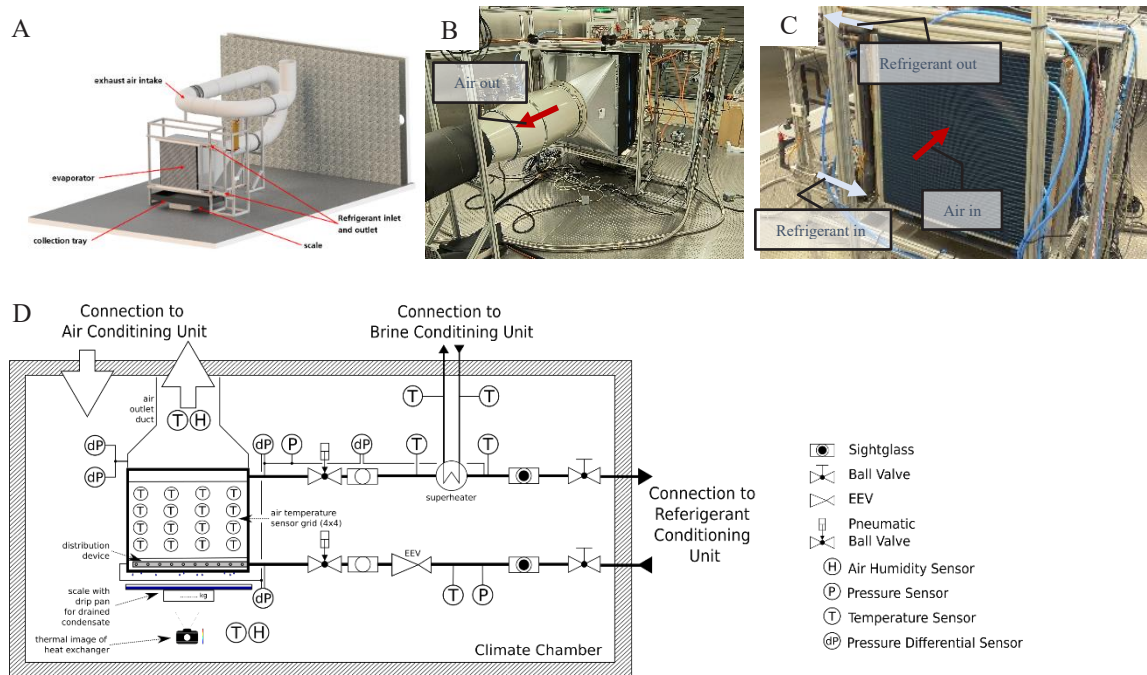


Figure 1. Test facility for heat exchanger performance. A: CAD drawing; B: view from the back on the air outlet side of the evaporator (before insulation phase); C: view from the front on an evaporator at the air inlet side. D: scheme of the sensor positions.

Table 1. Key parameters of air und refrigerant supply units

Parameter	Air side	Refrigerant side
Volumetric flow / Mass flow	500 to 6 000 m ³ /h	Usual operation at 30 g/s. Minimum 1 g/s. Maximum 100 g/s
Inlet temperature	−15 °C to 40 °C	20 °C to 75 °C (before EEV) −20 °C to 20 °C (saturation temperature in evaporator)
Thermal power	0.4 to 30 kW (heating)	0.4 to 40 kW @ 0 °C (evaporation temperature for cooling)

All instrumentation relevant to performance, frosting and defrost analysis is installed and time-synchronously recorded. Sensor information on temperature, pressure, humidity, flow rates are given in Table 2. Visual and infrared (IR) imaging and continuous weighing of defrost water are available as well. All sensors are calibrated for uncertainty analysis. Single-phase conditions (sufficient superheat / subcooling) are ensured at measurement locations to permit unambiguous enthalpy and heat-duty calculations.

Table 2. Measured quantities and measuring equipment used in the evaporator test setup

Measured quantity / Instrument	Type / Model	Location / Purpose	Accuracy / Notes (rdg: of reading/measurement value)
Inlet air temperature & humidity	Rotronic RMS HCD-S	Air handling unit → climate chamber inlet; used for density calc. and boundary conditions	±0.15 K ±1.3 % RH (at -10 to 10 °C)
Outlet air temperature & humidity	Vaisala HMT120	Downstream of HX after mixing section (~5 m)	±0.4 K ±4.0 % RH (at -40 to 0 °C)
Local air temperature grid	Pt100 Class AA, (4×4 grid)	~20 cm downstream of HX to detect local maldistribution	0.1 K
Air-side differential pressure (low range)	halpstrup walcher PU 0–50 Pa	Across HX for low Δp accuracy	±1.0 % rdg
Air-side differential pressure (high range)	Airflow PTSXR-K 0-250 Pa	Across HX for larger Δp range	±0.3 % rdg
Refrigerant mass flow rate	Micro Motion ELITE 3/8 Zoll	Refrigerant supply unit	± 0.1 % rdg (mass flow) ± 0.5 kg/m ³ rdg (density)
Absolute pressure (refrigerant)	ABB AST266 0-1MPa,	At refrigerant circuit (<20 cm from evaporator inlet/outlet)	± 0.04 % rdg
Differential pressure (refrigerant)	ABB DSH266 0-250kPa	At refrigerant circuit (<20 cm from evaporator inlet/outlet)	± 0.06 % rdg
Refrigerant temperature sensors	Pt1000 Class AA, (4-wire)	At refrigerant inlet/outlet, subcooler lines, superheater	± 0.1 K
Superheater flow meter (secondary)	KROHNE Optiflux 4300	Secondary side of plate superheater to determine heat duty and control steam fraction	± 0.2 % rdg + 1 mm/s
IR camera	FLIR A700	Whole-field thermal mapping of HX face to quantify local temperature and maldistribution	For quantitative thermography
Visible camera	AXIS Q6078-E	Time-resolved video of ice growth and defrost dynamics, local detail views	High-resolution imaging
Defrost water weighing	Mettler Toledo ICS425 Max. 60 kg	Continuous mass trace during defrost cycles; total melted mass	Basic accuracy: ± 0.01 kg Repeatability: ± 0.001 kg
Sight glasses	—	In refrigerant lines (inlet/outlet) to inspect flow regime and confirm single-phase at sensors	Visual check for valid enthalpy measurements

Note: Beside evaporators, the test facility can be used for performance tests on condensers.

3. Tested Heat Exchangers

Different types of microchannel and fin-and-tube heat exchangers have been tested at the described test setup as evaporators with R290 as refrigerant. In this paper a selection of four microchannel heat exchangers (MCHX) and one reference fin-and-tube heat exchanger (FTHX) is used. The geometrical details of the MCHX cannot be published, due to confidentiality. However, the range of parameters is shown in Table 3.

Table 3. Key parameters of heat exchangers

Parameter	Microchannel heat exchanger MCHX	fin-and-tube heat exchanger FTHX
Width (horizontal)	900 mm to 1000 mm	920 mm
Height (vertical)	900 mm to 1000 mm	935 mm
Depth (in air flow direction)	18 mm to 36mm	87 mm
Tube dimensions	1 mm to 1.3 mm (inner height)	1/4" (outer diameter)
Fin pitch	1.45 mm to 1.8 mm fins spacing (17.5 to 14 FPI)	3 mm fin spacing
Material	Aluminum	Copper tubes / aluminum fins

Due to the same frontal area of all heat exchangers a direct comparison of performance of the heat exchangers is reasonable. However, the depth in air-flow direction of the FTHX is more than 4 times the depth of the MCHX. These parameters are essential for the comparison: heat transfer rate, pressure drops, frosting and defrosting behavior.

4. Measurement

The evaluation of performance of the heat exchangers as evaporators is done in dry, wet and frosting conditions. In this paper the focus is on the dry condition (air inlet temperature 7°C and relative humidity 15%). For these conditions the stationary heat transfer rate $\dot{Q}$ (determined on the refrigerant side $\dot{Q}_{\text{ref}}$) and the driving temperature difference (logarithmic temperature difference ΔT_{lm}) between the two fluids are two of the key performance indicators. With these parameters the thermal resistance R_{th} is calculated:

$$R_{\text{th}} = \frac{\Delta T_{\text{lm}}}{\dot{Q}_{\text{ref}}} \quad (1)$$

$$\text{with } \Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}, \quad (2)$$

$$\text{whereas } \Delta T_1 = T_{\text{air,in}} - T_{\text{ref,out}} \text{ and } \Delta T_2 = T_{\text{air,out}} - T_{\text{ref,in}} \quad (3) \text{ and } (4)$$

The inlet and outlet temperatures on the air side ($T_{\text{air,in}}$, $T_{\text{air,out}}$) and refrigerant side ($T_{\text{ref,in}}$, $T_{\text{ref,out}}$) and $\dot{Q}_{\text{ref}}$ are based on the measurement equipment in Table 2. A lower thermal resistance R_{th} of the evaporator is an indicator of a better performance.

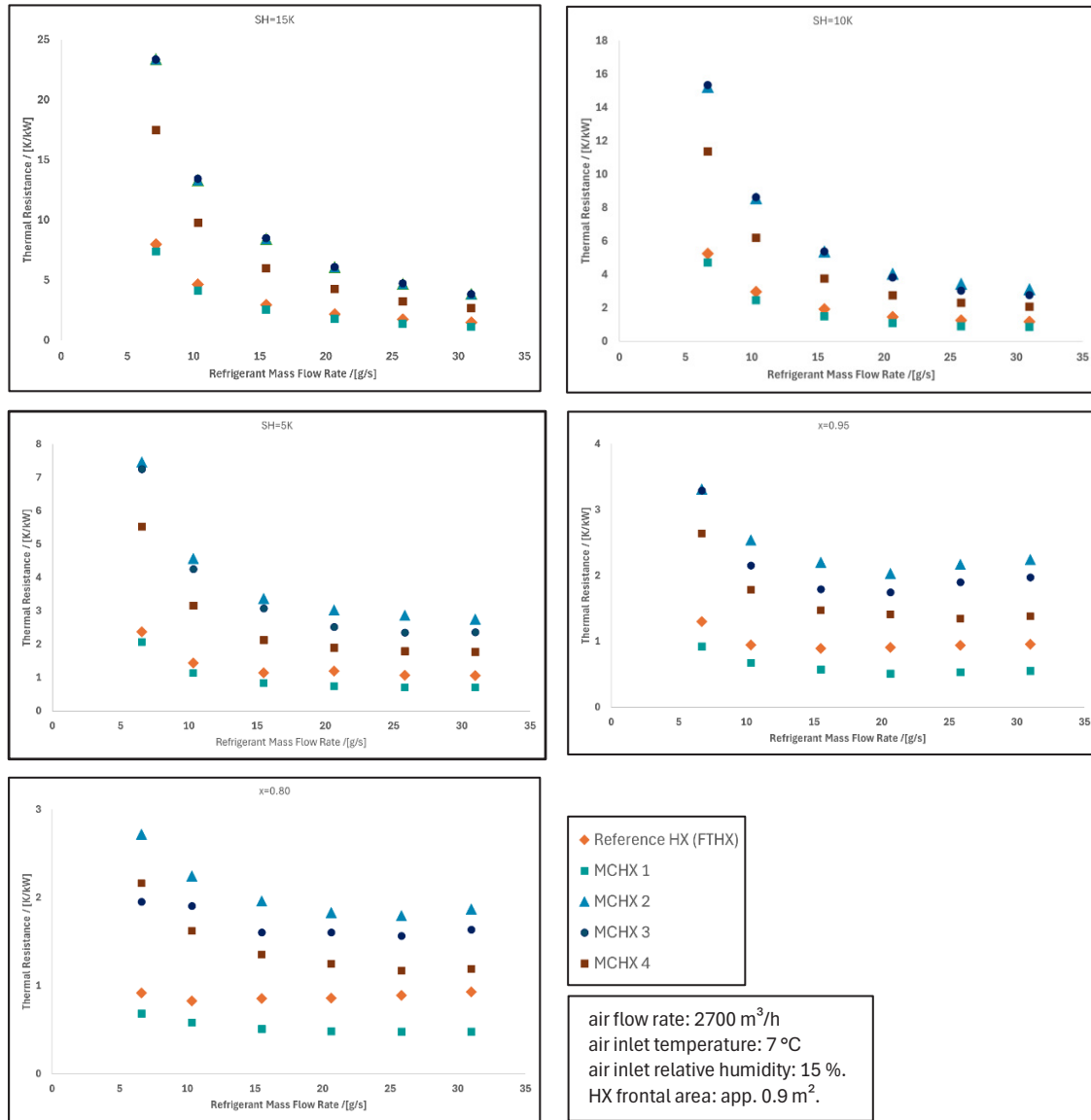


Figure 2. Thermal resistance of tested heat exchangers for different refrigerant mass flow rates for dry operation point with super heat SH equal to 15 K, 10 K and 5 K and vapor quality x equal to 0.95 and 0.8.

One important operation parameter in the refrigerant circuit, that can be controlled in operation, is the superheat (SH) at the evaporator outlet. A common value in heat pumps is 5 K to 10 K. Lower values for SH increase the risk of liquid refrigerant in the compressor and a possible damage of the compressor. However, adaptations on the compressor or an internal heat exchanger can allow operation with close to zero SH and thus possible two-phase refrigerant at the evaporator outlet. For all types of heat exchangers such an operation might be favorable due to less risk of maldistribution and less frosting. The two-phase flow is described by the vapor quality x , whereas x equals the mass of the vapor (gas) divided by the total refrigerant mass. The outlet vapor quality is calculated from energy balance in the “brine heated superheater” from flow rates of brine and R290, inlet and outlet temperatures of the brine and R290 outlet temperature and pressure. For MCHX the positive effect could be stronger than for fin-and-tube type heat exchangers. Fin-and-tube HX have usually several passes on the refrigerant side, such that the air must pass several refrigerant pipes with different states of the refrigerant. The MCHXs tested at Fraunhofer have one pass on the refrigerant side, such that in some regions of the heat exchanger the air is interacting only with superheated refrigerant (if $SH > 0$). The potential of air is thus not used adequately in these regions. Figure 2 shows the thermal resistance for different values of superheat and vapor quality for the tested heat exchangers.

For all HX the thermal resistance decreases with increasing refrigerant mass flow rates (with some exemptions for vapor quality of 0.95 and 0.8). This reduction in R_{th} is due to a higher heat transfer coefficient because of less area needed for superheating.

For all HX the thermal resistance strongly decreases with decreasing SH due to higher refrigerant side heat transfer coefficients for the two-phase flow compared to the one-phase superheated flow. Figure 3 shows the relative thermal resistance for different outlet states with information from Figure 2 for 15 g/s refrigerant mass flow rate. The plot is normalized at 5 K SH for each heat exchanger. The investigated MCHXs show much higher reduction in thermal resistance than the reference FTHX when reducing the vapor quality. On the other hand, the MCHX shows greater sensitivity of thermal resistance to increases in SH compared with the FTHX. Maldistribution was no major issue for all heat exchangers due to a well-engineered distribution device in the inlet header of the MCHXs.

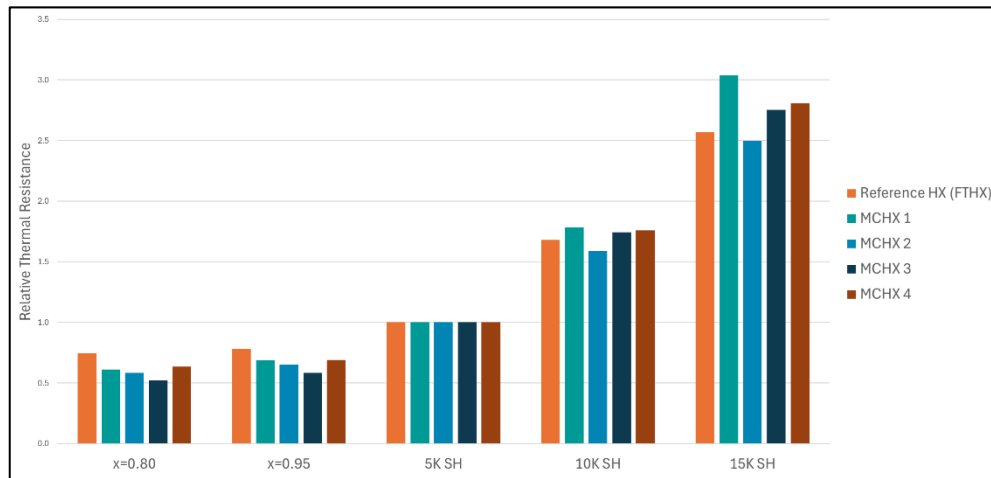


Figure 3. Relative thermal resistance R_{th} of tested HX. All values of R_{th} are normalized with the 5 K SH values of R_{th} . Refrigerant mass flow rate equal to 15 g/s

For the frosting condition, the evaluation of performance is multifaceted. Instead of steady state condition (as for the dry conditions), the evaluation must now take a dynamic behavior into account. The time between end of defrost and beginning of next defrost (frosting period duration) has been analyzed for all HX, with different criteria to start the defrost (upper threshold for air side pressure drop, maximum difference between air inlet and refrigerant saturation temperature, lower threshold for heat transfer rate). However, for a fair comparison the HXs must have similar thermal resistance, otherwise the value for the frosting period duration is not meaningful. For the performed tests this is not or only to a limited extent the case.

5. Conclusion

This study examines the performance of fin-and-tube (FTHX) and microchannel (MCHX) heat exchangers as evaporators in air-source heat pumps, focusing on their thermal characteristics, under varied operating

conditions. Using a specialized test facility with precise climate control and advanced measurement tools, the authors assess the impact of parameters such as refrigerant outlet superheat and vapor quality on heat transfer efficiency. Results highlight significant differences between FTHX and MCHX designs, with MCHX showing much stronger decrease in thermal resistance (increase in heat transfer) when lowering the superheat in dry conditions. For one of the tested evaporators (MCHX 1) thermal resistance had similar values for a superheat of 5 K and higher, but for zero superheat with outlet vapor quality equal to 0.95 and 0.8 the MCHX 1 had a 35 % and 40 % reduced thermal resistance, respectively. Thus, the efficient use of MCHX as evaporators in heat pumps can be pushed forward when refrigerant circuit design allows zero superheat. Meanwhile frosting and defrosting behaviour and geometrical changes for better longer frosting periods and better condensate drainage are necessary steps for a wider market introduction. In general approaches to increase frosting periods often result in increasing thermal resistances.

Acknowledgements

The authors would like to thank the Federal Ministry for Economic Affairs and Energy (BMWE) and the Project Management Agency Jülich (PTJ) for their support of the project QUEEN-HP MEGA, funding number FKZ 03EN4054A and 03EN4054B.

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