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BTO 3.1.2.55 Milestone Report – Report of Literature Review for Maldistribution in Heat Exchangers

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OAK RIDGE NATIONAL LABORATORY
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Energy and Transportation Science Division

BTO 3.1.2.55 MILESTONE REPORT – REPORT OF LITERATURE REVIEW FOR MALDISTRIBUTION IN HEAT EXCHANGERS

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CONTENTS

TABLE OF FIGURES

EXECUTIVE SUMMARY

In this quarter, a comprehensive review of the literature was performed to understand the effects of flow maldistribution and current efforts to improve performance through flow control. Nearly all systems have some degree of maldistribution on either the air-side, refrigerant-side, or both. Air-side maldistribution was found to have a greater impact on heat exchanger performance compared to refrigerant-side, but both sides of the system are important to consider when developing solutions. Up to a maximum of 65% capacity loss was reported in large commercial systems due to air-side maldistribution. Microchannel heat exchangers are becoming more widespread due to their efficiency and size but are even more prone to maldistribution than conventional fin-tube heat exchangers. Current solutions tend to focus on optimizing header design and refrigerant circuitry to improve performance. Several active control methods, such as hybrid control, have been explored to control refrigerant flow in real time, but there is a need for an adaptive yet cost effective system which can optimize performance in changing conditions.

1. IMPACT OF MALDISTRIBUTION AND POTENTIAL ENERGY SAVINGS

Maldistribution in heat exchangers, both air-side and refrigerant-side, has been found to have a significant effect on heat exchanger performance and ultimately, system performance. Heat exchangers are typically designed assuming a uniform distribution of refrigerant in all branches or circuits or with passive features to make the distribution uniform (e.g. distributors with tubes of many different lengths). In practice, a perfect distribution of refrigerant in a real system is rare. Under certain external operating conditions, such as frosting, fouling, damage or blockage, it can even be advantageous to have an imbalance of refrigerant between branches to improve performance. For example, active flow control can be used to mitigate most performance loss due to an imposed air-side maldistribution [1]. Thus, maldistribution is an important area of study when seeking to improve the performance of heat exchangers.

Extensive studies have been performed to quantify the performance drop of heat exchangers due to maldistribution, many of which will be explored in detail in this literature review. Some of these studies also explore methods for mitigating the maldistribution in heat exchangers showing substantial performance recovery under controlled conditions.

Current solutions to address refrigerant maldistribution have been somewhat limited. This includes various electronically-actuated expansion valves (EEVs), smart refrigerant distributers, redesigned headers and tubes, or refrigerant circuitry interweaving and optimization. Complex header redesigns are likely cost prohibitive and not necessarily optimal under all operating conditions. Use of individual valves for each refrigerant circuit is also likely cost-prohibitive. Addressing air-side maldistribution is also challenging, especially regarding frosting and fouling. Improper installation, damage, or blockage can also lead to significant air-side maldistribution. Because of the uncontrollable nature of air-side maldistribution, most of the effort in the literature have been focused on refrigerant-side distribution.

There also has been recent development and interest in microchannel heat exchangers. Microchannels have several advantages, such as higher efficiency and compactness compared to conventional round tubes. This also allows for a smaller refrigerant charge for a similar capacity system. This is especially relevant with the push towards newer environmentally friendly but flammable A2L refrigerants. Microchannels are perhaps even more relevant in the study of maldistribution, as they are more susceptible to maldistribution due to the sudden and significant changes in area from header to channels. Therefore, studying maldistribution in microchannels could result in significant performance improvements and would help to further bring microchannel heat exchangers into mainstream use.

While most of the literature has been focused on understanding the causes and effects of maldistribution, few studies have explored new technologies to control maldistribution. A working solution with the potential to be retrofitted onto existing heat exchangers would have substantial energy savings potential. For example, a smart solution for mitigation of refrigerant maldistribution that is successfully implemented into current residential HVAC systems which results in a 10% improvement in system coefficient of performance (COP) can result in primary energy savings potential of up to 225 TBtu for the 2030 energy market.

2. LITERATURE OVERVIEW

In order to properly address the issue of maldistribution in heat exchangers, a thorough review of the literature is necessary to gain an understanding of the problem and available solutions. Previous studies on maldistribution have explored a wide variety of heat exchangers with varying system capacities, refrigerants, and technologies. This literature review will first give a broad overview of over 40 studies in the area of refrigerant maldistribution. Studies of greater relevance to the project will be examined in more detail. Studies are also grouped based on heat exchanger type in order to gain insight into which system should be directly addressed in this project. Finally, the sum contents of this literature review will aid in the determination of system parameters for the project, which will be chosen based on what is likely to have the greatest impact on industry.

The literature encompasses experimental, numerical, and analytical studies of many different systems of varying capacity and operating parameters. The parameters of interest include equipment type, maldistribution side, refrigerant, heat exchanger dimensions, capacity, nominal pressure range, nominal flow rate range, and the effect of maldistribution on capacity. While not all studies may list all these parameters, they were used to group and compare studies such that the most relevant system could be identified for the project.

The literature contains a nearly even distribution of both experimental and numerical studies. As shown in Figure 1, of the studies examined, 32% contained experimental work only, 36% contained numerical or analytical work only, and 32% contained both experimental and numerical work. Note that even though the numerical-only studies did not involve performing experiments, results were generally verified with experimental data from previous publications. Comparatively more studies focus on refrigerant-side maldistribution compared to air-side maldistribution: 52% of studies examined the refrigerant-side, 25% studies examined the air-side, and 23% studies examined both sides, which is arguably more important when trying to address issues with real commercially-available systems. The most commonly studied refrigerants were R-410a and R-134a, followed by water and other less common fluids. System capacities ranged from as small as 200 W up to as large as 26.4 kW. Pressure and flow rate ranges are dependent on the refrigerant and system capacity and no significant trend was observed. The effect of maldistribution was typically reported as a percent capacity decrease for the HX. The literature indicates that air-side maldistribution can have a much greater impact on capacity compared to refrigerant-side maldistribution; up to a 65% decrease in capacity was reported in the most extreme cases [2]. This result is expected for air cooled systems, as the limiting thermal resistance is always air-side convection and thus any maldistribution will have major impact. A detailed table which includes information from all examined studies is given in APPENDIX A.

(a.) (b.)

Figure 1. (a.) Primary approach of study (b.) Air-side vs. refrigerant-side studies

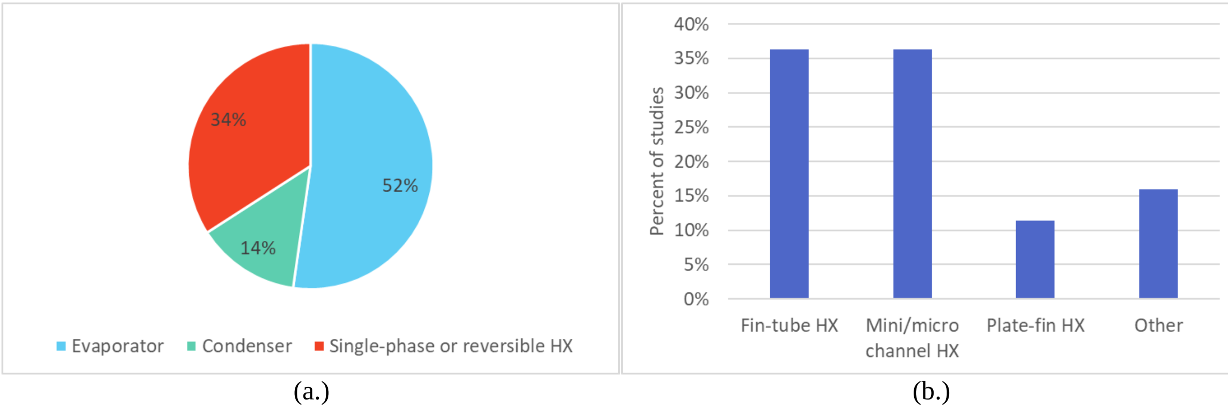

Figure 2. (a.) Number of studies based on HX type (i.e. evaporator, condenser or other) (b.) Number of studies based on HX type

Significantly more studies have been conducted on evaporators as compared to condensers, as illustrated in Figure 2a. This is likely due to the ease of controlling liquid-phase distribution entering the evaporator as opposed to the vapor-phase entering the condenser. In terms of the common HX types that are being studied, Figure 2b shows that more recent work is focused on maldistribution in minichannel and microchannel heat exchangers, which seems to be reflective of industry trends towards higher heat transfer for a given mass flow rate. While many studies have been conducted to quantify the effects of maldistribution, relatively few solutions have been explored to actively mitigate maldistribution. Several studies have focused on the effects of header and distributer parameters and sought to optimize the design of headers to minimize refrigerant-side maldistribution [3, 4]. The studies focused on header design have tended to ignore cases of air-side maldistribution, which can result in situations where even refrigerant distribution is not optimal for performance. Very few active control systems have been studied in the literature. One such active system is hybrid control pioneered at Purdue University [1, 5]. Hybrid control utilizes multiple electronic balancing valves to achieve even exit superheat throughout all circuits of a heat exchanger. Another active system is flow control through electrohydrodynamic (EHD) conduction pumping, which utilizes a coulomb force on a dielectric liquid to produce a net flow [6]. These active control systems are especially attractive due to their ability to redistribute flow in real time to maintain optimal performance of the system.

There are very few active systems being explored in the literature, and thus a cost-effective solution could have a substantial impact. Upstream evaporator control appears to be the most relevant area to address based on existing studies. A smaller capacity system would be better for the testing and development of an active control system. The increasing prevalence of microchannel heat exchangers is also a significant factor to consider when selecting systems to study. Based on these overall trends, preliminary parameters can be chosen for a system to be modelled and experimentally studied.

2.1 EVAPORATOR STUDIES

The literature contains studies on evaporators in many different systems, ranging from HXs in large rooftop units (RTUs) to smaller microchannel heat exchangers used for refrigeration and heat pump systems. These studies not only seek to quantify the effect of maldistribution, but also offer some novel methods of measuring maldistribution (e.g. [7]). These evaporator studies tend to dominate the literature, likely due to ease of control and visualization of the liquid phase entering the evaporator. New methods for control of refrigerant distribution, such as “hybrid control” pioneered by Kim et al. [1] are able to achieve substantial performance recovery due to maldistribution. While many studies focused on conventional finned-tube evaporators, many newer studies are exploring microchannel heat exchangers for their efficiency [7-10].

Groll et al. in Ref. [2] studied maldistribution in three different systems: a large room cooling system (LRCS), a domestic heat pump, and a rooftop unit (RTU) with economizer. These large commercial systems were tested under a variety of operating conditions to see the effect of maldistribution on capacity. Each system was first tested with an EEV which did not allow for individual refrigerant control through HX tubes. This was compared to individual circuit refrigerant flow control using a hybrid control system. Significant improvement in COP and capacity was achieved if air-side maldistribution was applied. They concluded that anywhere from 4% to 26% of capacity could be lost depending on environmental factors. Kærn et al. [11] simulated a vapor compression cycle for a residential air conditioning unit to explore the effects of different types of maldistribution. It was found that air-side maldistribution has a much greater effect on capacity than quality maldistribution and feeder tube bending, with a maximum capacity loss of up to 43.2%. The effect of varying maldistributions on capacity are shown in Figure 3.

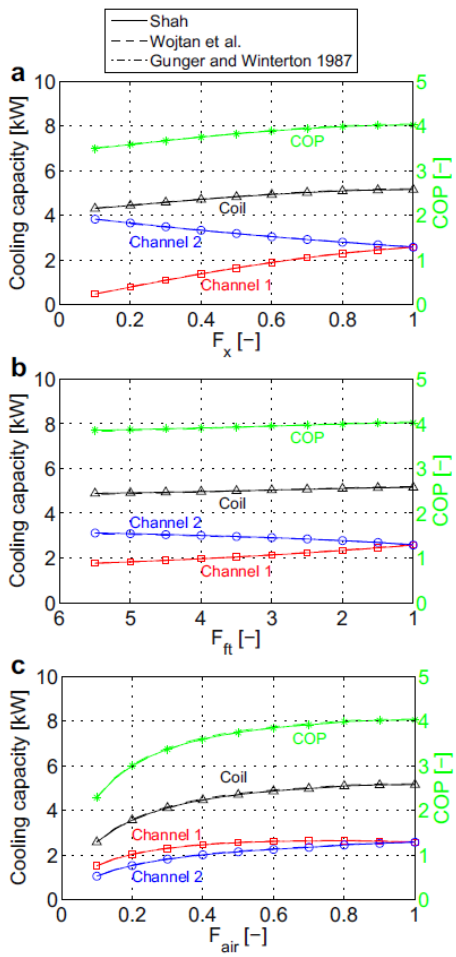

Figure 3. Individual channel and coil capacity and COP vs. a. quality maldistribution, b. feeder tube bending, c. airflow maldistribution [11]

The x-axis in each graph represents the measured distribution parameter, where F_x is the ratio of tube quality to inlet quality, F_{ft} is the feeder tube bending parameter used to multiply frictional pressure drop, and F_{air} was the ratio of airflow velocity over one tube to the mean frontal velocity across both tubes. Aganda et al. [12] studied the effect air-side maldistribution on a room air conditioner, resulting in a similar maximum capacity loss of 38%. Bach et al. [5, 13] numerically studied both air and refrigerant maldistribution on a walk-in cooler evaporator and found a maximum capacity loss of up to 22%. This shows there is significant room for improvement and capacity recovery in current commercially available systems.

Many studies have been focused on header and distributer design in order to more evenly distribute refrigerant. Byun and Kim [9, 14] studied the effects of mass flux and inlet/outlet locations for HXs with vertical headers and horizontal minichannel evaporator tubes. They found that thermal degradation by flow maldistribution is larger for a top row-crossing header than for a bottom row-crossing header. Zhang et al. [4] investigated the effect of distributer configuration in plate-fin heat exchangers. They found that improved distributers were very effective in improving flow distribution. By incorporating their improved header, they were able to reduce the flow non-uniformity by up to 57.4%. Payne and Domanski [15] explored the use of smart distributers to mitigate performance loss due to maldistribution. They found that significant performance recovery could be achieved by controlling the exit superheats between circuits. Even for significant coil blockage resulting in significantly increase pressure drop, performance was able to be restored within 2% of the original evenly distributed system. They also found that additional capacity degradation occurs when imposing an excessive superheat due to heat transfer between tubes, likely due to conduction through the fins of the heat exchanger. Habib et al. [3] investigated the effect of various header parameters on the effect of maldistribution. Flow maldistribution was characterized by the standard deviation of flow rate between tubes. Their results showed that Reynolds number was only significant for low flow rate systems. Locating the header inlet at the center resulted in 25-30% reduction in the standard deviation of the flow rate. Increasing the number of inlet nozzles had an insignificant effect but increasing the nozzle diameters resulted in an increase in the standard deviation of flow rate and an increase in pressure drop across the tubes.

While water is not typically used as a refrigerant in commercial systems, some studies like the above are using it to study maldistribution. A unique study by Razlan et al. [16] sought to compare the two-phase flow of refrigerant to an air-water mixture. They found that under the right conditions, an air-water mixture can be used to accurately represent two-phase flow, and thus is a safe and effective method for exploring flow maldistribution. According to Razlan et al. [16], the greatest similarity in flow pattern occurred when the Baker map (i.e. two-phase flow pattern map) parameters of both flows were equal. Another unique study by Brix et al. [8] modelled flow distribution in parallel minichannels using CO_2 as the refrigerant. They used this model to study the effects of uneven inlet quality and air-side maldistribution. They showed that non-uniform airflow leads to significant capacity reduction of the evaporator, while inlet quality seems to have a less significant effect. The results were very similar to those obtained using R134a, further extending the idea of using alternative fluids as the refrigerant when studying maldistribution.

In addition to alternative refrigerants, new methods for measurement of flow maldistribution are being explored. Li and Hrnjak [7] are pioneering an infrared thermography approach to measure liquid distribution in parallel microchannel heat exchangers. Flow through each tube is quantified by relating the liquid mass flow rate and the air-side capacity calculated from measurement of the tube wall temperature. This method was validated against experimental data to prove its accuracy and has the potential to be used for measurements on a wide variety of heat exchangers. Linde presented, in a thesis, a unique experimental setup for flow visualization through a transparent multiport header [10]. The setup was able to visualize the flow of refrigerant going into varying hardware through the transparent header and enabled the study of varying inlet header geometry, tube number, tube pitch, refrigerant, heat load, inlet location and mass flow rate on maldistribution. Flow visualization is typically difficult due to the need for suitable transparent materials but can offer greater insight into the phenomenon of maldistribution.

2.2 CONDENSER STUDIES

There are fewer studies focused specifically on the condenser sections of commercial heat exchangers. This is likely because upstream control of vapor is more difficult, and downstream control of liquid is less effective. These studies tend to report a lesser capacity degradation compared to evaporators, even with substantial air-side maldistribution. The literature was also dominated by numerical studies of condensers, which typically focused only on the refrigerant-side maldistribution.

Chin [17] performed a numerical study of a wavy fin and tube condenser and showed that single-phase and two-phase flow tend to follow similar trends in performance degradation due to maldistribution, with a maximum capacity degradation of up to 9%. Mao et al. [18] performed a numerical study on a multi-louvered fin and flat tube condenser prototype with varying air-side maldistribution and found a maximum capacity decrease of 6% and a 34% increase in pressure drop across the condenser. Chng et al. [19] numerically modelled refrigerant-side maldistribution in a microchannel condenser using a deterioration factor relating to the standard deviation of the maldistribution profile. Their model agrees with experimental evidence of reduction in performance as the standard deviation of maldistribution increased.

Active upstream control of a condenser has not often been explored due to the challenges associated with the vapor phase. Once such control method explored by Feng and Yagoobi [6] utilized electrohydrodynamic (EHD) condition pumping of dielectric fluids. A perforated electrode was used to impose a coulomb force on the dielectric liquid and generate pressure. With this method, the two-phase distribution between two parallel branches was able to be controlled, however, the EHD conduction pump is only able to influence the liquid-phase flow and its effect is also limited by the flowrate across the pump.

2.3 MICROCHANNEL HX STUDIES

More recent studies have begun to explore minichannel and microchannel heat exchangers due to their efficiency and compactness. Their increased efficiency is due to a smaller surface area to volume ratio resulting in greater heat transfer for a given flow rate. However, minichannel and microchannel systems are potentially more susceptible to refrigerant side maldistribution due to the sudden and extreme decrease in area from the header to the channels. Thus, there is potential for significant performance improvements of these systems if maldistribution could be mitigated or controlled. The following paragraphs provide an overview of recent studies involving flow distribution in microchannels.

Brix et al. [8] as previously mentioned performed a numerical study of a minichannel evaporator using CO_2 as the working fluid and explored the effects of air-side maldistribution and uneven inlet quality on performance. Their results showed that air-flow nonuniformity has a much greater impact on performance. CO_2 had the advantage of being more stable than conventional refrigerants since there is less of a density difference between the liquid and vapor phases, but for a horizontal orientation the results were very similar for those of R134a. Huang et al. [20] developed a co-simulation approach for modeling flow maldistribution in the header of a microchannel heat exchanger. The effect of gravity and air-side maldistribution was analyzed and compared to experimental data for an automotive condenser. The co-simulation approach resulted in a heat load within 1% of experimental data. Zou, Li and Hrnjak [21] studied the effect of polyalkylene glycol (PAG) oil added to R134a on distribution in a microchannel heat exchanger header. They found that with a small addition of oil (0.5%) the distribution worsened, but with increasing amounts of oil (2.5% and 4.7%) the distribution improved. Zou, Tuo, and Hrnjak [22] also studied flow through a transparent header in a microchannel heat exchanger. A microchannel evaporator model using experimental results numerically evaluated heat exchanger performance. A single pass heat exchanger resulted in a capacity degradation of up to 40% compared to a uniform refrigerant distribution. The maldistribution is even more significant in a two-pass heat exchanger, and thus its capacity is even lower than the single pass case. Zuo and Hrnjak [23] explored an outdoor reversible microchannel heat exchanger and examined the effects of inlet conditions, header geometries, and fluid properties on two-phase flow maldistribution in the header. A transparent header was used for visualization to gain insight into which flow regimes contribute to maldistribution. They concluded that the size of the churn flow region most affects the flow distribution and the best distribution occurs at high mass flux when the churn flow region immerses all microchannel inlets. The previously mentioned properties affect the size of the churn flow region and thus influence flow distribution.

2.4 ALTERNATIVE FLOW CONTROL TECHNOLOGIES

While the literature universally agrees that flow maldistribution has serious effects on the performance of heat exchangers and is found to occur in most commercial heat exchangers, there are a limited number of solutions being actively explored. Passive methods for control of flow maldistribution typically involve optimizing the headers of heat exchangers, but this lacks the robustness to account for external and environmental factors which may lead to maldistribution. In a real system, environmental factors can significantly affect the air-side of a heat exchanger, through frosting, fouling, or blockage. In these cases, it can be advantageous to increase the refrigerant flow rate in specific circuits in order to maximize performance. This is only possible with active control methods, of which there are few being examined in the literature.

The passive methods of improving refrigerant flow maldistribution typically use flowrate as a control parameter. Achieving equal flowrate through all tubes is ideal when the system is exposed to a uniform airflow. Several studies in the literature focus on header redesign and optimization. Byun and Kim [14] studied the effects of inlet location, outlet location, and mass flux on flow distribution and capacity for parallel heat exchangers with vertical headers. They found that flow distribution was improved using a top inlet configuration and top or bottom outlet configuration due to gravitational effects forcing more liquid flow to the bottom tubes of the heat exchanger. As previously mentioned, Byun and Kim [9] also explored the differences between top and bottom row-crossing headers in a similar study. They showed that thermal degradation by flow maldistribution was larger for the top row-crossing header configuration. Habib et al. [3] studied the influence of inlet nozzle parameters, such as location, diameter, number of nozzles, and inlet flow Reynold's number on flow maldistribution. Jiao et al. [24] experimented with implementation of a second header to mitigate flow maldistribution. They found that flow became more uniform when the inlet and outlet diameter ratios for both headers were equal and showed that performance of a plate-fin heat exchanger could be improved with optimal header design. Mohan et al. [25] performed a numerical parametric study of varying channel diameter on flow maldistribution and found that a more uniform distribution could be achieved by varying the diameters in individual channels. This is similar to a numerical study by Said et al. [26] in which vena-contracta of tube inlets were normalized to evenly distribute flow. In the previously mentioned study by Zou et al. [21], a passive technique of adding varying amounts of PAG oil to refrigerant was found to improve distribution. Yashar et al. [27] numerically optimized the circuitry of a 7.5-ton commercial R-410a rooftop unit and were able to achieve marginal performance improvements. While many of these studies of passive methods of improving performance through flow distribution were able to achieve significant recovery, they lack the capacity to adapt to changing conditions to maintain optimal performance.

Active flow control systems seek to address the need for changing operating conditions, but relatively few active control methods are being explored. The two most prominently being explored in the literature are hybrid control and electrohydrodynamic (EHD) conduction pumping. Hybrid control is a method pioneered by Kim et al. [1] involving small balancing valves on individual circuits in a heat exchanger. They were able to achieve substantial performance recovery by controlling and normalizing the exit superheat of each circuit in the evaporator. Through numerical simulations, they were able to impose non-uniform airflow over the tubes and still recover to near 100% of original capacity. Additionally, they compared the effect of upstream control versus downstream control and found that upstream control was more effective in all cases of air-side maldistribution. Figure 4 below shows the condensed results of Kim et al. [1], where the airflow maldistribution factor is defined as the difference in airflow between circuits 2 and 1 divided by the airflow in circuit 2.

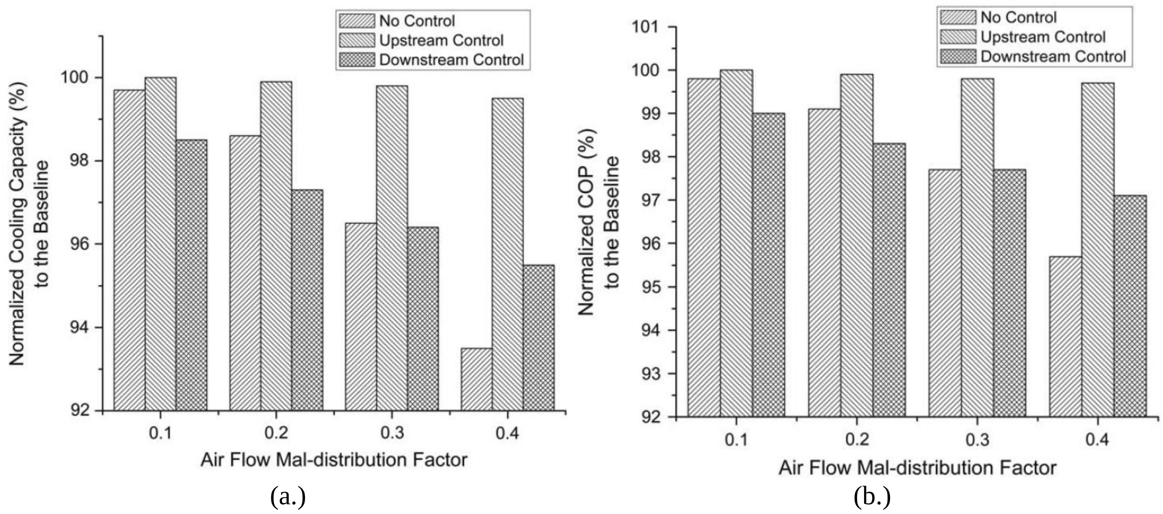

Figure 4. (a.) Reduction in cooling capacity compared with uniform air flow as a function of air flow maldistribution factor (b.) Reduction in system COP compared with uniform air flow as a function of air flow maldistribution factor [1]

Hybrid control has been explored in several other studies with positive results. Bach et al. [5, 13] found that active hybrid control of refrigerant flow showed better performance recovery compared to passive interleaved circuitry. Groll et al. [2] examined 3 different systems and compared the effect of a single electronic expansion valve compared to individual circuit hybrid control. They found that hybrid control could achieve significant performance recovery up to 65% in a 5-ton system. Exit superheat was the parameter being normalized to achieve performance recovery in these studies and is likely to be relevant for any other future active control methods.

Another technology for active flow control currently being explored is electrohydrodynamic (EHD) conduction pumping. EHD conduction pumping is a phenomenon that uses an applied electric field to impart a coulomb force on a dielectric liquid through its naturally occurring electrolytic impurities [6]. More recent studies such as Yang et al. [28] have begun to bring EHD conduction pumping to smaller scale system and have the potential to be implemented into microchannel heat exchangers. EHD conduction pumping has been shown to be an effective flow control technology with the capability to generate over 1 kPa of pressure with minimal power consumption. EHD conduction pumping has some limitations which prevent widespread adoption. The working fluid in the system must be a strong dielectric fluid, so this is particularly appealing to electronics cooling. Additionally, the pressure generation is highly dependent on the flowrate with lower pressure generated at a higher flowrate. The mechanism is also only able to influence the liquid phase and thus is only able to be implemented under specific conditions. While these active control methods are currently being explored, each has various limitations or barriers to widespread implementation. Therefore, there is still a need for development of low cost and effective active flow control systems to mitigate the performance losses caused by maldistribution.

3. CONCLUSIONS

Maldistribution is present to some degree in all systems whether air-side or refrigerant-side. Based on the literature examined, evaporators have the greatest potential for capacity improvement through mitigation of maldistribution. An increasing number of recent studies on microchannel heat exchangers aligns with industry trend due to their increased efficiency, compactness, and lower charger volume with adoption of A2L refrigerants. Maldistribution also tends to be more prevalent in microchannel heat exchangers due to the substantial reduction in area from header to channel. Thus, the study of maldistribution in microchannel heat exchangers appears to have the greatest potential impact.

Many heat exchangers are designed assuming uniform refrigerant and air distribution but in a real system there is always some amount of maldistribution present. When comparing air-side and refrigerant-side maldistribution, it is obvious that air-side maldistribution will have a greater impact on capacity. This is because the limiting thermal resistance is always air-side convection, and any reduction in airflow over part of the heat exchanger will have substantial impact on the capacity of those affected areas. While the air-side is primarily the limiting factor, it is often impractical to precisely control the distribution of air, and therefore most control methods focus on refrigerant distribution.

Passive solutions to refrigerant maldistribution, such as optimized header design, can result in an even flowrate between circuits in a heat exchanger. However, these passive solutions cannot account for changing environmental conditions surrounding the system. This could include improper installation, damage, or blockage due to frost, fouling, or debris. In these cases of imposed air-side maldistribution, optimal performance would inherently require an uneven balance of refrigerant between tubes. In order to be able to compensate for these situations, and active flow control system is required. Based on this literature review, very few active control methods are currently being explored. Thus, there exists a need for a low-cost and adaptive system which could be implemented into current and future heat exchangers to optimize performance in all situations.

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APPENDIX A. TABLE OF REFERENCES

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Reference name & year	Experimental?	Numerical/analytical?	Equipment type	Air-side?	Refrigerant-side?	Refrigerant	HX type and general dimensions	Nominal HX capacity	Nominal refrigerant pressure range	Nominal refrigerant flow rate range	Maldistribution effect on HX capacity	Notes
Aganda et al., 2000 [12]	×		Room Air Conditioner	×		R-22	Evaporator, 7 circuits with 5 rows of tubes, 9.525 mm ID copper tubing with aluminum plate fins	Not stated	598 kPa	0.0159 kg/s	Up to 38%	Air maldistribution led to reduced refrigerant flow and up to 38% HX capacity loss
Bach et al., 2013 [5]	×		3-ton walk-in cooler refrigeration system and 5-ton domestic HP			R-404a, R-410a	Fin-tube evaporator	10.6 - 17.6 kW	400 - 2500 kPa	Not stated	Up to 30%	Compared flow control using TXV, EXV and hybrid control by matching evaporator exit superheats
Bach et al., 2014 [13]		×	3-ton walk-in cooler refrigeration system	×	×	R404a	Fin-tube evaporator, 8 circuit (partial simulation)	10.6 kW (partial)	450 kPa	Not stated	Up to 28%	Active (hybrid control) mitigation of refrigerant and air maldistribution showed better performance recovery compared to passive (interleaved circuitry)
Brix et al., 2010 [8]		×	General - two parallel minichannels	×	×	CO_2	Minichannel evaporator, 2 tubes, 11 ports, 0.8 x 1.2 mm	300 W	4200 kPa	2 g/s	Up to 40%	Air flow non-uniformity induced significant refrigerant maldistribution and capacity degradation, more-so than uneven inlet quality

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Byun and Kim, 2011 [14]	×		General - parallel flow HX for typical residential AC		×	R-410a	Minichannel evaporator, vertical headers, 18 horizontal flat tubes, hydraulic diameter: 1.32 mm, length: 780 mm	1 kW	1100 kPa	50 - 70 kg/m^2 s	Up to 13%	Studied effects of inlet/outlet location on vertical headers and mass flux on flow distribution and capacity
Byun and Kim, 2016 [9]	×		General - two row/four pass parallel minichannel HX		×	R-410a	Minichannel evaporator, 2 row, 4 pass, horizontal headers, 18 vertical flat tubes, hydraulic diameter: 1.19 mm, length: 910 mm	1 kW	1100 kPa	70 - 130 kg/m^2 s	Up to 28%	Similar setup as above; thermal degradation by flow maldistribution is larger for top row-crossing header than for bottom config.
Chin and Raghavan, 2011 [29]		×	General - simulated crossflow HX	×		Water	Fin-tube cross flow HX, 2 pass with 5 tubes	Not stated	Not stated	4.2 kg/s	Not stated	The first two statistical moments of the velocity distribution (i.e. mean and standard deviation) were found to have the highest effect on HX performance degradation
Chin, 2017 [17]		×	General - simulated crossflow HX		×	R-22	Fin-tube cross flow condenser, 3 rows, 10 circuits, OD: 9.52 mm, length: 2000 mm	Not stated	1950 kPa	20 - 80 kg/h	Up to 9%	As above, thermal degradation of the HX is strongly dependent on statistical moments of the flow profile and changing vapor quality along the flow direction

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Chng et al., 2017 [19]	×	×	General - microchannel HX		×	R-32	Microchannel condenser, 10 tubes, hyd. diameter: 0.83 mm, length: 1700 mm	0.41 - 3.2 kW	Not stated	14.29 - 43.05 kg/hr	Up to 8%	As above, thermal degradation of the HX is strongly dependent on statistical moments of the flow profile. Superheat and subcooling effects were also studied.
Choi et al., 2003 [30]	×		General - fin-tube evaporator	×	×	R-22	Fin-tube evaporator, 3 rows, 3 parallel circuits, 54 tubes, OD: 9.53 mm	7 kW	1560 kPa	Not stated	Up to 30%	Individual exp. valves used for superheat control of each circuit. Capacity degradation was high even at uniform superheat when air flow was maldistributed.
Elgowainy, 2003 [31]		×	Residential heat pump	×		Not stated	Fin-tube evaporator/condenser, 1 row, 5 tubes, OD: 10 mm	Not stated	Not stated	Not stated	Not stated	Mass-averaged pressure drop was predicted to be 9% greater for non-uniform vs. uniform air flow. Similarly, area-averaged air-side heat transfer coefficient was predicted to be 1.5% less for non-uniform vs. uniform flow.
Feng and Yagoobi, 2005 [6]	×		General - two-phase flow control using electrohydrodynamics		×	HCFC-123	Condenser, 2 branches, ID: 10.2 mm, length: 1.5 m	Not stated	100 kPa	50 - 100 kg/m^2 s	Not stated	Used EHD conduction pumping to control two-phase flow distribution between

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												two HX branches with varying inlet quality
Groll et al. (CEC report), 2011 [2]	×		Large room cooling system	×	×	R-404a	Fin-tube evaporator, 8 circuits	10.6 kW	400 - 2000 kPa	80 g/s	Up to 29%	Each system was first tested with an EEV which did not allow for individual refrigerant control through HX tubes. This was compared to individual circuit refrigerant flow control using a hybrid control system. Significant improvement in COP and capacity was achieved if air-side maldistribution was applied.
	×		Domestic heat pump	×	×	R-410a	Fin-tube evaporator/condenser, 9 circuits (4 with 10 tubes and 5 with 8 tubes)	17.6 kW	800 - 2500 kPa	28 - 80 g/s	Up to 26%	
	×		Rooftop air-conditioner	×	×	R-410a	Fin-tube evaporator, 6 circuits, 14 tubes	14.0 kW	800 - 4000 kPa	40 - 80 g/s	Up to 65%	
Habib et al., 2014 [3]	×		General - parallel tubes to simulate industrial air-cooled HX		×	Water	Single-phase HX, 16 tubes	Not stated	Not stated	2.84 - 5.93 kg/s	Not stated	Study focused on adjusting inlet to header parameters to see effect on maldistribution

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Huang et al., 2014 [20]		×	General - microchannel HX	×	×	R-134a, water	Microchannel HX, 10 tubes, 2 x 30 mm, 13 ports (1 x 1 mm), length: 180 mm	3.5 - 7.8 kW	101 - 1470 kPa	21 g/s	Not stated	Co-simulation approach combined detailed header CFD with effectiveness-based finite volume tube-side heat transfer and refrigerant flow modeling tool.
Inamdar et al., 2016 [32]		×	General - fin-tube HX	×		Water	Fin-tube HX, model based on dimensions from multiple other studies	Not stated	101 kPa	Not stated	Not stated	Model of the deposition of particles on the air-side surfaces of fin-tube HXs
Jiao et al., 2003 [24]	×		General - plate-fin HX		×	Water	Plate-fin HX, 1100 cold-flow and 1000 hot-flow micropassages, fin dimensions: 6.5 x 2 x 0.3 mm^3	Not stated	101 kPa	Mean velocity: 0.3 - 0.4 m/s	Not stated	Second header added to plate-fin HX to aid in flow distribution; effects of inlet pipe, first header and second header diameters studied on flow distribution.
Kærn et al., 2009 [33]		×	Room air-conditioner	×	×	R-410a	Fin-tube evaporator, 2 tubes, ID: 7.6 mm, OD: 9.6 mm, length: 7m	Not stated	1000 - 3000 kPa	100 - 600 kg/m^2 s	Up to 38%	Study shows that air-flow maldistribution has a more significant effect than a malfunctioning distributer; however most loss can be recovered by controlling individual tube superheats

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Kærn et al., 2011 [11]		×	Room air-conditioner	×	×	R-410a	Fin-tube evaporator, 2 row, 2 passes, 18 tubes, ID: 7.6mm, OD: 9.6 mm, length: 444.5 mm	8.8 kW	1000 - 3000 kPa	0.06 kg/s	Up to 43.2%	RAC system model shows non-uniform airflow significantly reduced capacity (43%); differing liq/vap distribution in distributor had smaller impact (13%) and feeder tube bending was even less (4%)
Kim et al., 2009 [1]		×	Residential heat pump	×	×	R-410a	Fin-tube evaporator, 3 row, 5 parallel circuits, 17 tubes each	10.5 kW	1000 - 3000 kPa	Not stated	Up to 6%	Hybrid method for optimizing refrigerant distribution in evaporators was studied involving small balancing valves in reach refrigerant circuit along with primary expansion device; benefits of controlling the superheat of the individual tubes using upstream and downstream control valves were investigated.
Lee et al., 2018 [34]	×	×	General - fin-tube HX	×		R-410a	Fin-tube condenser, 18 tubes per bank in 3 banks, OD: 10.5mm , thickness: 0.55 mm, length: 0.46 m	8.5 kW	1000 - 3000 kPa	23 g/s	Up to 11%	Integrated CFD-segmented HX model used to study effect of geometric parameters and air flow rate on air flow maldistribution;

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												CFD model validated with PIV measurements; parameters varied in study include HX angle, HX depth, HX height, air flow rate and fin type.
Li and Hrnjak, 2015 [7]		×	General - microchannel HX		×	Not stated	Microchannel evaporator	2.25 - 3.6 kW	Not stated	Not stated	Not stated	Non-contacting, non-intrusive, Infrared Thermography approach to quantify liquid refrigerant distribution; validated against experimental data.
Linde (MSc thesis), 2005 [10]	×		General - microchannel HX		×	R-134a	Microchannel evaporator, 30 channels, width: 18 mm, length: 1 m, 5 ports (2.86 x 1.32 mm)	5-10 kW	300 - 700 kPa	80 g/s	Not stated	Experimental setup to visualize flow through a multiport header and two-phase refrigerant distribution in microchannel heat exchanger; ability to vary inlet header geometry and study influence of tube number, tube pitch, refrigerant, heat load, inlet location and mass flow rate on maldistribution.

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Mao et al., 2013 [18]	×	×	General - minichannel HX	×	×	R-22 & R-134a	Minichannel condenser, 2-6 tube passes, tube height: 21 mm, tube width: 16 mm, 10 ports (1 x 1.25 mm)	7.7 - 9.5 kW	900 - 2800 kPa	0.038 - 0.051 kg/s	Up to 6%	The effects of various 2D air flow maldistribution profiles on HX effectiveness, capacity and pressure drop were investigated; the maximum capacity reduction and pressure drop increment were 6% and 34%, respectively.
Mohammadi and Malayeri, 2013 [35]		×	General - shell and tube HX		×	Crude oil	Single-phase shell-tube HX, shell ID: 838.2 mm, tube ID: 20.10 mm, tube OD: 25.4mm, tube length: 2000 mm	Not stated	Not stated	Mean velocity: 1.2 - 3.8 m/s	Not stated but maximum flow velocity deviation of 25%	For turbulent, single-phase flow through the shell-tube HX, flow maldistribution was found to be a function of tube number; increased tube number increased distribution uniformity.
Mohan et al., 2014 [25]	×	×	General - crossflow automotive HX		×	Water	Single-phase cross-flow automobile HX, channel ID: 8.57 mm	Not stated	101 kPa	0.035 kg/s	Not stated	Parametric study of effect of channel diameter on HX flow distribution and pressure drop; changing individual channel diameters is more effective for flow control distribution compared to changing all channel diameters.

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Ozawa et al., 1989 [36]	×	×	General - two parallel channels with air-water flow simulating flow boiling		×	Air-water mixture	Parallel tube channel HX, two tubes, ID: 3.1 mm, length: 3.1 m	Not stated	101 kPa	Total volumetric flux: 0.374 m/s	Not stated	Effect of pressure drop oscillations in two-phase flow through parallel channels was studied; analysis in paper can be used to estimate characteristics of non-uniform flow distribution and oscillation
Payne and Domanski (NIST report), 2002 [15]	×	×	General - fin-tube HX	×	×	R-22	Fin-tube evaporators, ID: 9.53 mm, round tubes	3-8 kW	600 kPa	100 kg/h	Up to 43%	Effect of smart refrigerant distributors and non-uniform air flow on HX capacity was studied; capacity degradation due to maldistribution could effectively be restored by controlling refrigerant superheat at evaporator outlets.
Razlan et al., 2018 [16]	×		Heat pump system		×	R-134a, air-water mixture	Minichannel evaporator, multi-pass, upward flow minichannels, 1.7 x 20 mm with 17 ports, each port 0.5 x 0.8 mm	Not stated	200 - 1000 kPa	4.15 - 8.5 kg/h	Not stated	Comparison of flow patterns of two-phase refrigerant to air-water mixture; under the right conditions, air-water can be used to accurately represent two phase flow; equal Baker map parameters resulted in the greatest

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												similarity in flow pattern.
Rossetti et al., 2015 [37]		×	Open display cabinet	×		Not stated	Fin-tube evaporator, 10 rows, 2 tube	Not stated	Not stated	Not stated	Up to 10%	Thermofluid model utilized thermal equivalence model for the evaporator fins, reducing computational cost and 1.2 million nodes, a factor of 10 lower compared to similar models in the literature.
Said et al., 2014 [26]		×	General - header and tube HX		×	Industrial oil	Single-phase fin-tube HX, 9 tubes, ID: 25 mm, length: 400 mm	Not stated	600 Pa	0.25 kg/s	Not stated but up to 12x flow rate deviation	Study on normalizing inlet vena-contracta to evenly distribute flow through HX header; two approaches involving reduced and increased inlet diameters resulted in balance between flow maldistribution mitigation and pressure drop.
Shojaeefard et al., 2017 [38]	×	×	General - minichannel HX		×	R-134a	Minichannel condenser, 4 passes, 31 tubes, 7 ports, tube width: 10 mm, tube length: 546 mm	2.34 - 13.69 kW	100 - 3500 kPa	0 - 600 kg/h	Up to 4%	Hybrid simulation method involving simultaneous CFD model for flow through header and FE model for flow through

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												channels; effects of tube protrusion depth, inlet tube location, inlet tube diameter and combinations on flow maldistribution were studied.
Tuo and Hrnjak, 2013 [39]	×	×	Automotive air-conditioner		×	R-134a	Microchannel evaporator, 1 pass, 34 tubes, 19 ports, hydraulic diameter: 0.8 mm	Not stated	299 - 387 kPa	10 - 30 g/s	Up to 25%	Study on effect of header pressure drop on flow maldistribution; header-induced flow maldistribution still exists when quality induced maldistribution is eliminated; limiting outlet header pressure drop by adjusting outlet header diameter significantly limited HX capacity degradation.
Vist and Pettersen, 2004 [40]	×		Automotive air-conditioner		×	R-134a	Minichannel evaporator, 10 parallel circular tubes, ID: 4 mm, length: 0.9 m	5 kW	690 - 710 kPa	0.025 - 0.042 kg/s	Not stated	Effects of inlet vapor quality, heat load on individual tubes, manifold diameter, manifold inlet tube length and HX orientation on flow maldistribution were studied.

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Wen et al., 2006 [41]	×		General - Plate-fin HX	×		Air	Single-phase plate-fin HX, inlet baffles 600 x 260 mm, hole ID: 10 - 30 mm	Not stated	400 Pa drop across baffle	Mean velocity: 4 m/s	Not stated but maximum flow velocity ratio of 23.2	PIV was used to study the flow distribution resulting from inlet baffles of varying geometry; by optimizing the geometry, the ratio of max to min velocity through header holes was reduced from 23.2 to 1.76.
Yaïci et al., 2016 [42]		×	General - plate-fin HX	×		Air/water	Single-phase plate-fin HX, 4 row, 4 tubes, OD: 9.97 mm	Not stated	Not stated	Mean velocity: 4 m/s	Not stated	3D CFD used to study the effect of inlet air flow maldistribution on design and thermal-hydraulic performance of HX; parameters included Reynolds number, longitudinal/transversal tube pitch and fin pitch; Up to 67% deviation in friction factor and Colburn factor was determined for non-uniform vs. uniform flow with varying fin pitch, for example.
Yang, Wen et al., 2017 [43]		×	General - plate-fin HX	×		Air	Single-phase plate-fin HX, punched baffle, hole ID: 10 - 15 mm	Not stated	0.5 - 12 kPa drop across HX	0.002 kg/s	Up to 46.8%	CFD used to obtain flow distribution in core of plate-fin HX;

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												by adjusting geometry, HX effectiveness, maldistribution parameter and pressure drop are significantly affected for non-uniform flow vs. uniform flow.
Yang, Talmor et al., 2017 [28]	×	×	General - flow control using electrohydrodynamics		×	Novec 7600	Single-phase parallel minichannels, 3 tubes, ID: 1 mm, length: 635 mm	Not stated	101 kPa	4-7 mL/min	Not stated	Electrohydrodynamics used to redistribute artificially induced maldistribution in 3 parallel tubes; flow rates directed to specific tubes with precise control; suitable for small-scale, highly branched thermal control systems.
Yashar et al., 2015 [27]	×	×	Rooftop air-conditioner	×		R-410a	Fin-tube evaporator, 144 tubes, 4 rows, 16 circuits, OD: 9.52 mm	26.4 kW	500 - 3500 kPa	Not stated	Up to 2.2%	Effect of refrigerant circuit optimization was studied; results show that success of the proposed optimization concept depends strongly on the availability of simple but accurate measurement of air-side velocity profile

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												and improved CFD models.
Zhang et al., 2015 [4]	×		General - plate-fin HX		×	Air	Single-phase plate-fin HX, 30 channels, 6.5 x 2 mm channels	Not stated	Not stated	Mean velocity: 4 m/s	Not stated	Effect on distributor inlet angle and configuration on flow distribution was studied; thermal performance change due to flow maldistribution due to different distributors was investigated; the degree of temperature non-uniformity was reduced from 1.078 to 0.712 using the improved distributor.
Zou et al. (ASHRAE NY-14-C010), 2014 [21]	×	×	General - microchannel HX		×	R-134a	Microchannel evaporator, hydraulic diameter: 0.5 mm, header ID: 15.44 mm	0.2 - 0.65 kW	Not stated	4.19 g/s	Up to 48%	Oil added to refrigerant to study effect on two-phase refrigerant flow in vertical header; small oil circulation rate (OCR) made distribution worse, but this effect reduced with higher OCR; visualization showed greater amount of oil enhanced mixing between phases, improved distribution

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												and reduced capacity degradation.
Zou and Hrnjak (ACRC report), 2014 [23]	×	×	General - reversible microchannel HX		×	R-134a, R-410a, R-32, R245fa	Microchannel HX, outdoor reversible, vertical header, hydraulic diameter: 0.5 mm, header ID: 15.44 mm	Various	500 - 4000 kPa	2.14 to 6.25 g/s	Up to 30%	Along with detailed experimental measurements and modeling, flow visualization in transparent headers was performed; effects of inlet conditions, header geometry, fluid properties and refrigerant distribution among microchannel tubes were investigated; refrigerant distribution was found to be related to size of churn flow in header (also affected by fluid properties); influence of oil mixture was also studied; based on experimental results, a distribution function was derived to model refrigerant distribution and predict HX capacity degradation.
Zou et al. (ASHRAE	×	×	General - microchannel HX		×	R-410a	Microchannel evaporator, hydraulic	0.3 - 1.3 kW	1000 kPa	2.14 to 6.25 g/s	Up to 40%	Maldistribution effects in multi-pass

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NY-14-037), 2014 [22]							diameter: 0.5 mm, header ID: 15.44 mm					evaporator with upward flow header; flow regimes affected by inlet conditions and header geometry; capacity degradation due to non-uniform flow in one pass of HX was quantified. Comparison between model and experimental results showed maldistribution was more significant in two-pass HX.
Zou et al., 2014 [44]	×	×	General - microchannel HX		×	R-410a & R-134a	Microchannel evaporator, hydraulic diameter: 0.5 mm, header ID: 15.44 mm	0.3 - 1.3 kW	400 - 1000 kPa	2.14 to 6.25 g/s	Up to 50%	Same as above, including R-134a refrigerant and additional measurements