A Roadmap for Implementing Minichannels in Refrigeration and Air-Conditioning Systems—Current Status and Future Directions

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The evolution of compact heat exchangers was driven by the need for reducing their size and weight and enhancing individual components and overall system performance. These needs are also becoming relevant in stationary refrigeration and air conditioning applications. The reduction in volume and footprint of the systems is foremost in the minds of urban architects, while refrigerant charge reduction and performance enhancement remain important safety and operating considerations. Minichannels provide a viable solution to address these concerns and are expected to be used widely in large as well as small commercial, industrial, and residential systems. This paper provides a roadmap for their implementation, with a critical review of the current status of our understanding in this field, recommendations for obtaining fundamental performance data and correlations, and an emphasis on innovative designs of minichannel heat exchangers.

INTRODUCTION

Challenges Facing Today’s Refrigeration and Air Conditioning Industry

Refrigeration and air conditioning systems have become an integral part in the design of spaces occupied by people and intended to provide controlled thermal, humidity, cleanliness, and/or other process requirements. As we strive to design these systems, the constraints imposed on the size and/or specific dimensions of the system components are becoming increasingly important, while the operating cost remains a critical factor with high energy prices. The impetus toward improving the energy efficiency of refrigeration and air-conditioning equipment becomes clear when we recognize that this sector accounts for 15 percent of total energy consumption worldwide [1]. Another issue of great environmental concern is the release of large quantities of refrigerants. The risk of accidental leakage, combined with the increased capital investment (energy and resources required to manufacture the refrigerants), makes it desirable to have a low refrigerant inventory in the system.

The size and floor space occupied by refrigeration and air-conditioning equipment continues to be a major concern in urban environment. With soaring real estate prices, reductions in volume and floor space of the system components are very desirable. In modern buildings, small equipment size also translates into smaller passageways for transporting the equipment in and out, and lower reassembly costs. The size restrictions are becoming important even for the window room air-conditioners in metropolitan apartments. The major challenges facing the refrigeration and air conditioning industry can be summarized as follows:

• improving the system COP (coefficient of performance)
• reducing the total refrigerant charge in the system
• reducing the footprint and size of the equipment
• meeting these challenges in a cost-effective manner

The role of heat transfer in the refrigeration industry was addressed in a recent editorial [2]. In a conventional refrigeration/air conditioning unit, the heat exchangers offer significant potential for addressing the above challenges, which are somewhat similar to those faced in other applications, including absorption refrigeration, air liquefaction, automotive
air conditioning, and high flux chip cooling. Drawing from the experience gained in these applications, the potential of minichannel heat exchangers is highlighted and a roadmap for their implementation is presented.

**CHANNEL SIZE CLASSIFICATION**

In our efforts to arrive at a common terminology for smaller diameter channels, several classification schemes based on hydraulic diameters were proposed in literature. A simple channel classification scheme [3] was based on the hydraulic diameter as follows:

- microchannels: 1 μm to 100 μm
- meso-channels: 100 μm to 1 mm
- conventional passages: > 6 mm.

Another approach is to classify the channels according to the specific processes they are being employed for. In two-phase applications, Fukano and Kariyasaki [4] and Serizawa et al. [5] proposed the use of Laplace length scale, \( L = \sqrt{\sigma/\left(D \rho_f - \rho_v\right)} \), for channel size classification based on the relative magnitudes of the surface tension and gravity forces. Kew and Cornwell [6] proposed the ratio of the Laplace length scale to channel hydraulic diameter as the criterion in flow boiling application. The effect of channel size is seen to be different on heat transfer and fluid flow phenomena. For example, in the case of single-phase gas flow, the flow passage dimensions are compared with the mean free path \( \lambda \) of the gas molecules (Knudsen number, \( Kn = \frac{\lambda}{D_h} \)). In adiabatic two-phase flows, the relationship between void fraction and flow Reynolds number provides guidance. Kawahara et al. [7] studied the variation of void fraction with volumetric quality in adiabatic two-phase flow for 50, 100, and 251 μm diameter tubes and noted a marked change in the relationship for the two smaller diameter tubes. The authors suggested that the transition boundary between microchannels and minichannels lies between \( D_h = 100 \mu m \) and \( D_h = 251 \mu m \).

From a practical standpoint, it is useful to consistently employ the same classification scheme while considering the overall behavior of commonly encountered processes, such as single-phase gas and liquid flows, two-phase adiabatic flow, flow boiling, and flow condensation processes. Kandlikar and Grande [8] considered various processes described above and recommended the channel classification shown in the inset to be used for all processes. Kandlikar et al. [9] modified the above classification to replace the channel hydraulic diameter with the minimum channel dimension \( D \) in the original classification scheme.

**ADVANTAGES OF MINICHANNEL HEAT EXCHANGERS**

**Minichannel Heat Exchangers: Overview**

A typical minichannel heat exchanger (MCHX) employs a flat tube with triangular or rectangular passages of 1–2.5 mm hydraulic diameter. Still smaller passages are being applied in condenser applications, 25-mm or more wide, and length dependent on the refrigerant and operating conditions. These heat exchangers are generally employed in refrigerant-to-air heat exchangers, with folded louvered fins employed on the air-side. Figure 1 shows a schematic of an MCHX.

Minichannel heat exchangers were studied experimentally by Wang et al. [10]. They used copper plates in which channels of 300 μm to 4.5 mm were etched in brass and stainless steel. The overall dimensions were 296 × 122 × 30 mm, and the highest surface area density achieved was 1500 m²/m³. The resulting heat transfer coefficients on the volumetric basis were as high as 7 MW/m³°C. Although these heat exchangers were intended for biological applications, they demonstrated the heat transfer potential of narrow channels.

Early work of Webb and Jung [11] showed the advantages of folded louver fins and flat tubes over continuous fins and the standard round tubes. The heat transfer coefficient was 90 percent higher for the new design while the pressure drop increased by only 25 percent. The main shortcomings of the round tube design were the inefficient airflow performance in the wake of fins and high thermal contact resistance between the round tubes and the fins. Brazed aluminum condensers employing minichannels for residential application were studied by Webb and Lee [12]. Westphalen et al. [13] identified some additional benefits with these heat exchangers, including all-aluminum construction with improved corrosion resistance, ease of leak repair with
epoxy, reduced air-side resistance due to flat tubes incorporating minichannels, and reduced fan power for condenser and evaporator units.

Utilizing MCHX leads to size reductions and performance improvements in air-conditioning systems. Kim and Bullard [14] compared the performance of two different types of MCHX with the baseline case of finned round-tube condenser and found that for the same Energy Efficiency Ratio (EER), the MCHX designs yielded a 35 percent reduction in refrigerant charge and 35–55 percent reduction in core volume and weight. Kim and Groll [15] compared the performance of MCHX in a unitary split system in an outdoor air conditioning/heat pumping coil. The MCHX had a 23 percent smaller face area and a 32 percent lower refrigerant volume in the core. It resulted in a 1–6 percent improvement in EER, but the defrost cycle and system performance were dependent on the air-side fin density and coil orientation. Further need for research on optimum MCHX configuration for a specific application is clearly brought out from their work.

Rugh et al. [16] presented their experimental results on a minichannel heat exchanger used as an air heater with a very high air-side fin density for heating the passenger compartment of a car. Rectangular minichannels with internal dimensions of $1.52 \, \text{mm} \times 14.4 \, \text{mm}$ were used. The tubes had 0.25-mm ribs that effectively reduced the height to 1 mm. On the air-side, high density louvered fins at 20 fins/cm were provided. The airflow passages were essentially only 0.5 mm wide. Making the fins louvered at an angle of 25° provided further enhancement. The water side heat transfer coefficients were as high as 35,600 W/m²°C. Such high values were made possible by the small passage dimensions combined with flow interruptions, which introduce periodic developing flows as new boundary layers are formed following each interruption in the flow stream.

**Heat Transfer Enhancement with Minichannels and Microchannels**

The heat transfer rate in a given heat exchanger is expressed through the following basic heat transfer equation:

$$ Q = UA \Delta T $$

The overall heat transfer coefficient $U$ depends on the fin efficiencies, individual side heat transfer coefficients and fouling resistances, and the thermal resistance of the separating wall. The flow inside a minichannel is expected to be laminar due to the small channel hydraulic diameter [8]. Although the flow will be under developing conditions, fully developed laminar flow is considered in the analysis presented here to demonstrate the enhancement effects due to channel size reduction. Under the assumption of fully developed laminar flow, the Nusselt number is constant and the heat transfer coefficient in internal flow varies inversely with channel hydraulic diameter as given by the following equation:

$$ h = \frac{Nu \, k}{D_h} $$

The channel size. The pressure gradient (pressure drop per unit length) increases drastically with channel size reduction, but it was also noted that minichannel heat exchangers need to be suitably designed to provide short flow lengths to limit the overall pressure drop [8]. Today’s compact heat exchangers are successfully implemented in various applications, and higher pressure gradients in minichannel heat exchangers simply become a design constraint in arriving at a suitable heat exchanger design.

**Heat Exchanger Size Reduction with Minichannels**

Another major advantage offered by minichannels is the reduction in the internal volume and the overall size of the heat exchanger. Neglecting the header and intermediate pass connecting tubes, the surface area available per unit volume of a heat exchanger employing circular tubes of diameter $D$ and length $L$ is given by the following equation:

$$ A/V = \frac{\pi D L}{(\pi/4)D^2L} = 4/D $$

The surface area to volume ratio $A/V$ is an indicator of the compactness of a heat exchanger, and is routinely used in the compact heat exchanger literature [17]. Figure 3 shows the variation of this ratio with the tube diameter. The channel sizes of around 10-mm are commonly employed in today’s stationary refrigeration applications, although it is not uncommon to find even larger channel sizes.
The area to volume ratio for 10-mm diameter channels is found to be 400 m²/m³, while this ratio becomes 4000 m²/m³ and 8000 m²/m³ for 1-mm and 0.5-mm diameter channels, respectively. In other words, for the same internal flow volume, a 1-mm diameter tube provides 10 times larger surface area as compared to a 10-mm diameter tube. This, coupled with a ten-fold improvement in the heat transfer coefficient (as seen from Figure 2) provides a total of 100-fold increase in the individual side $hA/V$ product (heat transfer coefficient times the surface area) for both water and R-134a in single-phase flow.

A comparison based on the heat transfer rate per unit volume of the heat exchanger is illustrative in further recognizing the size benefits of minichannel heat exchangers.

$$\frac{q}{V} = \frac{hA\Delta T}{V} = h\Delta T \frac{A}{V}$$

Equation (4) shows that $q/V$ depends directly on $A/V$, indicating that this ratio would go up inversely with the diameter as seen in Fig. 3. A ten-fold decrease in the channel diameter from 10 mm to 1 mm will yield a ten-fold increase in heat transfer rate per unit volume, in addition to the increase derived from the improvement in the $hA$ product (heat transfer coefficient times the surface area) for both water and R-134a in single-phase flow.

Reduction in Refrigerant Charge

The refrigerant charge is another major consideration from overall cost and environmental viewpoints. The amount of refrigerant in large industrial refrigeration equipment can be in the range of several hundred kilograms. Using minichannels reduces the internal volume of the heat exchangers with a significant reduction in the refrigerant charge. Hrnjak [19] presents a study focusing on charge reduction strategies in ammonia refrigeration systems. He recommends using smaller diameter channels and plate heat exchangers with small gaps. A higher void fraction in the heat exchanger is desirable from this standpoint. Hrnjak reports the following specific refrigerant charge per kW of heat removal rate in ammonia systems:

- air-cooled condensers: 18–159 g/kW
- water-cooled condensers: 23–228 g/kW
- condensers with minichannels: 5–10 g/kW

In ammonia evaporators, Pearson [20] and Ayub [21] provide the following values for typical specific charge in different heat exchanger types:

- shell-and-tube: 1000 g/kW
- plate: 500 g/kW
- gravity-fed plate: 250 g/kW
- spray-type: 54–113 g/kW

In general, specific refrigerant charge is considerably higher in the evaporators as compared to condensers. Ayub [22] provides the values of the refrigerant charge in a DX evaporator employing 15-mm enhanced tubes of around 75 g/kW for a 100 ton evaporator under ARI conditions. Significant refrigerant charge reductions can be obtained by employing minichannel heat exchangers in these systems. Considering the surface area and performance enhancements presented earlier with minichannels, a design goal of 10–20 g/kW for refrigerant charge seems feasible in evaporators. The higher heat transfer rates possible with refrigerant-to-water heat exchangers (as compared to refrigerant-to-air heat exchangers) will result in further reductions in the refrigerant charge.

Hrnjak [19] also suggests increasing the void fraction in condensers and evaporators to reduce the refrigerant charge in a system. Withdrawal of refrigerant in condensers at several locations along its length and use of falling film type evaporators are among several possibilities discussed. Reducing liquid subcooling prior to the expansion valve will provide a higher inlet quality and higher void fraction in the inlet section of an evaporator. However, reducing the liquid subcooling may have an adverse effect on the system COP or capacity. The refrigerant charge penalty is considerably reduced if minichannels are used in the evaporators.

CURRENT STATUS OF RESEARCH ON MINICHANNEL CONDENSERS AND EVAPORATORS

This section reviews the fundamental information available on condensation and boiling modes of heat transfer in minichannels.
Minichannel Condensers

Minichannels as small as 0.5 mm in hydraulic diameter have been successfully employed in condensers for many years, and the heat transfer, pressure drop, two-phase flow patterns, and system configuration in minichannel condensers have been studied extensively in literature. Garimella [22a] provides an exhaustive literature survey on these topics; readers are referred to this source for an in-depth review of the condensation phenomena in minichannels. A brief overview of the current status is presented here. The multi-port condenser geometries employed by Yang and Webb [23], Koyama et al. [24, 25] and Garimella [22a] are shown in Figures 4–6. Yang and Webb [23] conducted one of the first studies utilizing a flat rectangular tube with four rectangular minichannels nominally 2 mm × 3.5 mm with and without microfins. One of the interesting outcomes of their study was the effect of microfins on condensation heat transfer in small diameter tubes. The entire subcooled single-phase data, for both plain and microfin tubes in the Re range from 4,000 to 21,000 was correlated using Petukhov’s single-phase correlation. This indicates that the microfin effects in single-phase flow are well accounted for by using the hydraulic diameter. During condensation experiments, the microfins were flooded in the low quality region and the heat transfer was comparable to plain tubes using the hydraulic diameter. In the high quality region, the microfins showed a definite enhancement, indicating the effect of microfins on the thin annular liquid film flow.

The work of Koyama et al. [24, 25] is instructive in explaining the geometry effect on condensing flows in minichannels. For their internally finned channels, the heat transfer performance was similar to the plain channels after accounting for the area enhancement. This work, coupled with Yang and Webb’s [23] observations, indicates that the microfins are effective in the high-quality region, and the fins in general provide a heat transfer enhancement that can be attributed to the increase in surface area.
In predicting the pressure drop, a number of models are available in literature as summarized by Garimella [22]. The homogeneous flow model is seen to be promising in a large number of cases. Another correlation that is widely used in calculating pressure drop is by Chisholm [26].

Pressure drop correlation by Chisholm [26]:

\[ \Phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \]  
(5)

or

\[ \Phi_G^2 = 1 + C X + X^2 \]  
(6)

where \( \Phi_L \) and \( \Phi_G \) are the ratios of the two-phase to respective liquid or vapor single-phase pressure drops. The values of the constant \( C \) for different two-phase flow conditions are given in Table 1. Mishima and Hibiki [27] modified the Chisholm constant \( C \) as follows for small diameter tubes.

For circular tubes,

\[ C = 21(1 - e^{-33D_L}) \]  
(7)

For non-circular geometries,

\[ C = 21(1 - e^{-319D_b}) \]  
(8)

Some of the available condensation data show that this correlation overpredicts the pressure drop data. For example, Koyama et al. [24] found that the constant \( C \) varied between 5 and 21, depending on the flow conditions. English and Kandlikar [28] conducted experiments with air-water flow at lower flow rates, corresponding to the laminar-laminar case in Table 1, and found that using a value of \( C = 5 \) from Table 1—for this case, in Eq. (8)—yielded very good agreement. Instead of using a constant value of \( C = 21 \) in Eqs. (7) and (8), English and Kandlikar recommend a value of \( C \) from Table 1 depending on the laminar/turbulent flow conditions in individual phases.

Heat transfer during condensation in minichannels has been studied extensively, and a number of correlations are available in literature for specific geometries. Some of the more successful ones are by Akers et al. [29], Yang and Webb [30], Yan and Lin [31], Wang et al. [32], Cavallini et al. [33], Koyama et al. [24], Wang and Rose [34, 35], and Bandhauer et al. [36].

The model proposed by Shah [37] for condensation in large diameter tubes is simple to use and is suggested for preliminary estimates. The Shah correlation is given by:

\[ \frac{h_{TP}}{h_{LO}} = (1 - x)^{0.8} + \frac{3.8x^{0.76}(1 - x)0.04}{(P/P_{crit})^{0.38}} \]  
(9)

where

\[ h_{LO} = 0.023 \left( \frac{k_L}{D} \right) \left( \frac{GD}{\mu_L} \right)^{0.8} \left( \frac{L}{\Pr} \right)^{0.4} \]  
(10)

More accurate methods specifically designed for minichannels smaller than 1 mm in diameter have been proposed in literature. The analytical approach by Wang and Rose [34, 35] and the empirical method of Bandhauer et al. [36] are considerably more complex, but are recommended for better accuracy. Minichannels with internal microfins are used to enhance the heat transfer performance during condensation. A model by Yang and Webb [30] takes into account the surface tension forces explicitly in minichannels with internal microfins and is recommended for such channels.

Minichannel Evaporators

The implementation of minichannels in evaporators lagged behind the condenser application mainly because of the instability concerns [38–41]. Recent attention to these problems by researchers worldwide has resulted in some practical solutions that are expected to pave the way for widespread implementation of minichannel evaporators in various applications.

The flow boiling phenomenon in narrow channels was studied using high-speed imaging techniques by Steinke and Kandlikar [42] and Kandlikar and Balasubramanian [43]. Figure 7 shows a high-speed image obtained at 2000 fps of water boiling inside a set of six parallel microchannels, each of 1054 × 197 \( \mu \)m rectangular cross-section. Although the overall flow direction is from left to right, instantaneous directions of liquid vapor interfaces in adjacent channels are seen to be moving in both directions. The resulting reverse flow in some of the channels is detrimental to the operation of an evaporator. It also leads to backflow of vapor into the inlet manifold, causing flow maldistribution and instabilities. The flow patterns in minichannels are dominated by surface tension effects, and significant differences can be expected in diabatic flow patterns due to nucleating bubbles.

The basic mechanisms leading to unstable operation during flow boiling are discussed by Steinke and Kandlikar [42]. Flow
instabilities were found to be the result of rapid growth of vapor bubbles nucleating near the inlet region of the microchannels and minichannels. Kosar et al. [44, 45], and Kandlikar et al. [9] introduced inlet restrictors and artificial nucleation cavities to suppress the instabilities. This method seems to have promise, and further research is warranted in this area.

Another topic that is of great importance is the predictive ability for heat transfer and pressure drop in minichannels. Qu and Mudawar [46] and Jacobi and Thome [47] proposed flow pattern based models to predict the heat transfer. The original flow boiling correlation by Kandlikar [48] was very successful in predicting the heat transfer coefficient and its trends with quality for water and refrigerants. The same correlation is extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing that the all-liquid flow heat transfer coefficient needed in the correlation may be extended to minichannels, mainly by recognizing.

The single-phase all-liquid flow heat transfer coefficient is given by

\[ h = \frac{\text{Nu} k}{D h} \]

where \( h \) is the heat transfer coefficient, \( \text{Nu} \) is the Nusselt number, \( k \) is the thermal conductivity, and \( D h \) is the hydraulic diameter.

The friction factor \( f \) is given by

\[ f = \frac{1.58 \ln(Re) - 3.28} \]

where \( Re \) is the Reynolds number.

\[ Re = \frac{\rho v D}{\mu} \]

Table 2: Recommended \( F_F \) (fluid-surface parameter) values in flow boiling correlation by Kandlikar [48].

<table>
<thead>
<tr>
<th>Fluid</th>
<th>( F_F )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>1.00</td>
</tr>
<tr>
<td>R-11</td>
<td>1.30</td>
</tr>
<tr>
<td>R-12</td>
<td>1.50</td>
</tr>
<tr>
<td>R-13B1</td>
<td>1.31</td>
</tr>
<tr>
<td>R-22</td>
<td>2.20</td>
</tr>
<tr>
<td>R-113</td>
<td>1.30</td>
</tr>
<tr>
<td>R-114</td>
<td>1.24</td>
</tr>
<tr>
<td>R-134a</td>
<td>1.63</td>
</tr>
<tr>
<td>R-152a</td>
<td>1.10</td>
</tr>
<tr>
<td>R-32/R-132</td>
<td>3.30</td>
</tr>
<tr>
<td>R-141b</td>
<td>1.80</td>
</tr>
<tr>
<td>R-124</td>
<td>1.00</td>
</tr>
<tr>
<td>Kerosene</td>
<td>0.488</td>
</tr>
</tbody>
</table>

\( F_F = 1 \) for stainless steel tubes Any refrigerant

For \( 3000 \leq Re_L \leq 10^4 \)

\[ h_{LO} = \frac{\text{Nu} k}{D h} \]

for \( 410 \leq Re_L \leq 1600 \)

The friction factor \( f \) in Eqs. (14) and (15) is given by

\[ f = \frac{1.58 \ln(Re_L) - 3.28} \]

In the transition region between Reynolds numbers of 1600 and 3000, a linear interpolation is suggested for \( h_{LO} \). The fluid-dependent factors \( F_F \) for various refrigerants on copper or aluminium are listed in Table 2. For stainless-steel tubes, a value of 1.0 is recommended for all refrigerants.

MINICHANNELS IN ADVANCED REFRIGERATION SYSTEMS

Minichannels in CO\(_2\) and Ammonia Systems

Although CO\(_2\) was one of the earliest refrigerants used, it was replaced by the advent of man-made refrigerants. It was not until 1993 that the interest in transcritical CO\(_2\) was revived after the work of Lorentzen and Pettersen [50] and Lorentzen [51]. Minichannels are well suited for containing the high pressures encountered in the CO\(_2\) systems. Pettersen et al. [52] developed new designs of MCHX with 0.8-mm diameter tubes placed together forming a 42-mm wide flat plate arrangement (similar to Figures 4 or 5). Cross-conduction of heat between different plates is of greater concern in a gas cooler than in a condensing application. In the design of a gas cooler, the use of single-phase heat transfer correlations incorporating variable property effects was recommended because of the large variations in the properties of transcritical CO\(_2\) along the length of the heat exchanger.

CO\(_2\) systems are also being considered for residential heat pump application. Richter et al. [53] compared the performance of an R404A system with a CO\(_2\) system designed to fit the
same chassis. Garimella and Bansal [54] and Zhao and Ohadi [55] developed gas coolers for automotive CO₂ systems. Flow boiling in minichannels and overall system performance were studied by a number of investigators, including Kim and Bullard [56], Pettersen [57], and Elbel and Hrnjak [58], who showed that flash gas bypass around the evaporator resulted in an increase in capacity and a 7–9 percent improvement in the COP. A review of the flow boiling heat transfer with CO₂ is presented in a recent article by Thome and Ribatski [59]. The backflow described earlier is also seen to be present, and needs to be addressed in developing commercial CO₂ systems.

Worldwide research in CO₂ systems indicates that these are strong contenders in mobile as well as stationary refrigeration applications. The use of minichannels enables these systems to handle high pressures while maintaining a compact design. However, these systems are still in experimental stages, and further research is needed before considering their practical implementation. Pearson [60] provides a very succinct overview of the historical development of the CO₂ systems and identifies compressor technology as one of the major challenges. Use of minichannel heat exchangers is recognized as important in developing leak-proof, low-charge CO₂ systems.

Large ammonia systems, however, are currently in operation for industrial and commercial refrigeration applications. As discussed in the earlier sections, using minichannels reduces the component sizes and the amount of ammonia charge in the system. It is therefore reasonable to expect widespread use and acceptance of minichannels in ammonia systems in the coming decade.

**Minichannels in Absorption System Components**

The high surface area-to-volume ratio provided by narrow channels is also attractive for mass transfer applications. In fact, some of the very high mass transfer rates within a compact geometry are accomplished in natural systems such as the kidney, lung, and brain. The absorber in an absorption refrigeration system provides a unique opportunity for utilizing minichannels. Garimella [61] provides a comprehensive overview of the heat exchangers in an absorption system that could potentially benefit with the use of minichannels. He developed an absorber utilizing short lengths of minichannels in vertical stacks with intermediate distribution trays to promote uniform flow distribution and wetting, the major areas of concern. Meacham and Garimella [62] presented a model to predict the local variation in heat and mass transfer coefficients in an ammonia-water absorber utilizing minichannel absorber passages. The compact design offers several unique features, such as the ease of modifying pass arrangement, and can be utilized in other heat and mass transfer components in the system—the desorber, condenser, and evaporator. Employing the same heat exchanger geometry in the entire system makes it easier from a maintenance viewpoint. The high cost of system components utilizing minichannel tubes can be brought down considerably through the mass production of these components.

**RESEARCH NEEDS AND ECONOMIC CONSIDERATIONS**

**Secondary Refrigerant and Water-Cooled Systems**

In order to fully realize the benefits of minichannel passages for refrigerant system size reduction, the air streams in the condenser and evaporator need to be replaced with water or a secondary refrigerant providing a better match of thermal resistances on the two sides of the respective heat exchangers. Using water for condenser heat removal and water or secondary refrigerant in the evaporator is highly desirable. The resulting size reductions in the heat exchangers will reduce the refrigerant charge further. Pumping power will be lower to move the liquid streams as compared to the air streams. The lower temperature differences in the heat exchangers due to higher overall heat transfer coefficients will also increase the system COP. The design issues and related research needs are well documented in the literature (e.g., [63]). The major issues include selection of a proper secondary coolant, component (solenoid valves and actuators) failures due to continuous low temperature operation, corrosion, and bacterial activity.

**Condenser Design Data**

Available research on condensation indicates that minichannels can be effectively utilized in commercial and industrial refrigeration systems. Further research is needed, however, to provide specific design information related to heat transfer and pressure drop in different channel geometries. The effectiveness of enhancement features such as internal fins and microfins needs to be studied further. Design information regarding large capacity condensers with minichannels is also needed.

**Flow Boiling Stabilization**

Minichannel flow passages are not as commonly employed in the evaporator application as in the condenser applications because of the flow instabilities during the boiling process. Further research and testing in this area is needed to achieve stable operation, especially with refrigerants, ammonia, and CO₂.

**Water-to-Refrigerant Minichannel Heat Exchangers**

The available minichannel heat exchanger configurations for water-to-refrigerant heat transfer are somewhat limited. Although some shell-and-tube heat exchangers are developed with 1-mm ID tubes, the manufacturing costs are very high due to tighter tolerances needed for the cover plates and baffles. The plate heat exchangers with smaller hydraulic diameter passages are perhaps ideal among the available choices. Plate-fin heat
exchangers with evaporating and condensing streams are commonly employed in air-liquefaction industry. This arrangement is not suited for the current applications, as the serrated fin design will result in rather large pressure drops that may be detrimental to the overall system COP. On the other hand, the extruded tubes shown in Figures 4–6 are ideally suited for the condensing or evaporating refrigerant streams. New heat exchanger developments specifically addressing the system requirements are needed. For example, Neksa et al. [64] propose a fan-less gas cooler that employs 1.1-mm ID tubes attached to vertical plates. Rane and Tandale [65] employed a tube-tube heat exchanger (TTHE) arrangement in which minichannels are placed side by side and bonded thermally using a thermal bonding material for effective heat transfer. Hot and cold streams flow through the alternate minichannels, which are bent periodically along the flow length to enhance heat transfer.

Fundamental understanding of the transport processes is another area of great importance. For example, use of superhydrophobic walls with transverse ribs was shown to reduce the pressure drop during liquid flow [66]. However, the combined effect of such methods on heat and momentum transfer needs to be investigated.

The minichannel heat exchangers that are currently being used in refrigeration applications are mainly derived as extensions of automotive heat exchangers. The flat tubes containing minichannels are assembled in a plate-fin type arrangement to serve as condensers or evaporators. These geometries are particularly suited for air-to-refrigerant heat transfer. In the case of secondary refrigerant or water-cooled condenser applications, heat exchangers are needed to handle the refrigerant-to-water or refrigerant-to-secondary refrigerant streams. A new arrangement is presented in Figure 8 that shows a unit cell with minichannel flow passages on both refrigerant and water sides. The minichannels are incorporated within flattened plates. Water or secondary refrigerant flows in the narrow passages formed between the two adjacent plates. The heat transfer coefficient of the water is high due to the flow in narrow passages. Further enhancements can be achieved by introducing microfins or grooves forming a cross-hatch or cross-rib pattern on the outside surfaces of the plates, similar to that employed in plate heat exchangers. The passage dimensions, width of the plate, groove arrangement, and other geometrical parameters need to be optimized for specific applications. Further research on thermal performance and manufacturing aspects of these heat exchangers is recommended.

Economic Considerations

The economic advantages offered by minichannel heat exchangers include the following:
- use of lightweight and low-cost aluminum,
- mass-production capability through advances in brazing technology,
- cost savings through performance enhancement,
- cost savings through low refrigeration charge, and
- cost savings associated with lower space requirements

Detailed analyses of cost savings through refrigerant charge reductions or process variable optimizations are presented in the literature by a number of investigators. For example, Walker and Baxter [67] showed that the operating costs are reduced by 5–15 percent by using a low-charge, distributed, secondary loop, water-cooled evaporative condenser. Using minichannels is expected to provide further savings. Selbas et al. [68] and Mishra et al. [69] conducted thermo-economic optimization of vapor compression and vapor-absorption refrigeration systems by varying the design variables that affect heat transfer rate, pressure drop, and cost factors. It is recommended that a detailed economic analysis be performed (using computerized techniques, such as by Usta and Ileri [70]) to show the potential benefits associated with minichannels in large-scale commercial refrigeration and air conditioning systems. Although it is difficult to assign the heat exchanger costs in such economic analyses, it is expected that significant cost reductions in minichannel heat exchangers will be realized as they become more widely used. The use of aluminum and advances in brazing technology offer significant benefits in producing minichannel heat exchangers.

ROADMAP FOR MINICHANNEL IMPLEMENTATION

Based on the literature review presented in this paper, it can be concluded that minichannels provide a great opportunity to reduce the overall size and cost and improve the performance of the heat exchangers and the entire system. Further research and development work is needed for commercial utilization of this technology. At the same time, innovations are needed in implementing the minichannels. Some of the opportunities in arriving at energy-efficient and environmentally friendly systems are outlined by Garimella [71].

Table 3 presents a roadmap showing the implementation status and specific research needs in residential, commercial, and industrial refrigeration and air conditioning systems. It is

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Figure 8  Refrigerant-to-water heat exchanger employing enhanced microchannel passages on both sides for refrigeration chiller or condenser application.
Table 3 Roadmap for implementing minichannels in refrigeration and air-conditioning applications

<table>
<thead>
<tr>
<th>Application</th>
<th>Implementation status</th>
<th>Research needs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Residential room air-conditioners</td>
<td>I1: Evaporator technology not ready</td>
<td>R1: Fundamental stability concerns during boiling need to be addressed.</td>
</tr>
<tr>
<td></td>
<td></td>
<td><em>Fundamental research as well as product design and development needed.</em></td>
</tr>
<tr>
<td>Commercial unitary air-conditioners</td>
<td>I1</td>
<td>R1–R3</td>
</tr>
<tr>
<td></td>
<td>I3: Refrigerant-to-water condenser development needed.</td>
<td>R4: Design and development of water-to-refrigerant condensers with minichannels.</td>
</tr>
<tr>
<td></td>
<td>Significant efforts needed.</td>
<td><em>Fundamental research as well as product design and development needed.</em></td>
</tr>
<tr>
<td>Industrial and commercial refrigeration, a/c</td>
<td>I1, I3</td>
<td>R1–R4</td>
</tr>
<tr>
<td></td>
<td>I4: Use water-to-air minichannel HX (plate-fin arrangement), technology available from compact HX and air-liquefaction industry.</td>
<td>R5: Design and development of water-to-refrigerant evaporators with minichannels.</td>
</tr>
<tr>
<td></td>
<td>I5: Refrigerant-to-water (or secondary refrigerant) evaporator development needed.</td>
<td><em>Fundamental research as well as product design and development needed.</em></td>
</tr>
<tr>
<td>Absorption refrigeration, CO₂, and ammonia systems</td>
<td>I1, I3, I5</td>
<td>R1–R5</td>
</tr>
<tr>
<td></td>
<td>Major efforts needed.</td>
<td><em>Fundamental research as well as product design and development needed.</em></td>
</tr>
</tbody>
</table>

*Note: Explanations for terms I1–I5 and R1–R5 appear at their first appearance in the table.*

expected that this roadmap will guide us in undertaking targeted research in academia and industry to realize the overall energy savings that are possible through the implementation of minichannel heat exchangers.

**CONCLUSIONS**

A comprehensive overview of the current status of minichannel research as related to refrigeration and air conditioning application is presented. As the minichannels are defined as channels in the diameter ranges of 200 μm–3 mm, most of the refrigeration heat exchangers can be classified as minichannel heat exchangers (abbreviated as MCHX).

The surface area and heat transfer coefficient enhancements associated with minichannel flow can be exploited effectively to make the refrigeration equipment smaller in size. Significant refrigerant charge reductions are also possible due to the higher surface area-to-volume ratio for the minichannels. Additional advantages include capital cost reductions, reduced environmental impact due to lower refrigerant inventory, and possible improvements in COP of the system.

The minichannel heat exchangers currently being considered for refrigeration applications are modeled after the existing heat exchangers in automotive applications. A new initiative is recommended to develop new types of heat exchangers that fully exploit the advantages of minichannel flow passages on both sides of refrigerant-to-water or refrigerant-to-secondary refrigerant heat exchangers. Such balanced coupling will enable even more compact units with associated savings in handling costs of the fluid streams as well. As an example, a heat exchanger geometry that is suitable for water-to-refrigerant heat transfer is presented in Figure 8.

A roadmap for the implementation of minichannels in refrigeration and air-conditioning application is presented, and associated research and developmental efforts needed in academia and industry are highlighted.

**NOMENCLATURE**

- $A$: surface area, m²
- $Bo$: boiling number, $= q/(Gh_{HX})$
- $C$: constant in Eqs. (5) and (6)
- $C_p$: specific heat, J/kg°C
- $Co$: convection number, $= [(1 - x)/x]^{0.8}[p_ℓ/p_L]^{0.5}$
- $D$: minimum dimension of a channel, or diameter, m
- $D_h$: hydraulic diameter, m
- $dP/dx$: pressure gradient, Pa/m
- $f$: friction factor
- $F_{Fl}$: fluid-surface dependent parameter in Kandlikar correlation
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REFERENCES


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