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Principles of evaporator coil design for air source cold climate heat pumps using smaller diameter copper tubes and low GWP refrigerants

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ABSTRACT

This paper demonstrates that small diameter round copper tubes in heat exchanger coils reduce refrigerant charge, increase efficiency, and decrease the size of cold climate heat pumps (CCHPs), while allowing for quick reverse-flow defrost cycles. Simulations are run on an I-shape coil with two rows of 5 mm tubes for various internal tube surface enhancements (microfins), fin types and fin densities. The two-row coil is also compared with a two-and-a-half-row coil. Simulations are then run for an L-type, 15 kW evaporator with six rows of copper tubes, varying the diameter, tube spacing, fin density and tube circuitry. Results are compared with respect to weight, refrigerant charge and pressure drop. Finally, a prototype L-shape coil with 54 copper tubes in each of four rows (216 tubes total) was built and tested. The capacity of this CCHP evaporator was measured at 15 kW with an R290 mass flow rate of 200 kg/h and airflow rate 10,400 m^3/h. The test results compared well with simulations.

Keywords: Refrigeration, Carbon Dioxide, Compressors, COP, Evaporators, Energy Efficiency

1. INTRODUCTION

Electrification is essential to decarbonization. Heat pumps allow for coal, natural gas and petroleum to be replaced by renewable energy sources such as solar, wind and hydroelectric. Well-designed heat pumps produce heat energy that is multiple times the energy input to the compressor. In contrast, furnaces and boilers burn fossil fuels at high temperatures and their coefficient of performance is always less than one. This is wasteful of our energy resources and does not allow for decarbonization.

The Twenty First Conference of the Parties within the United Nations Framework Convention on Climate Change announced the Paris Agreement in 2015. Further commitments to reduce greenhouse gas emissions were recorded at subsequent COPs, including COP26 in Glasgow and COP27 in Sharm el-Sheikh.

The Kigali Amendment to the Montreal Protocol is an international agreement to gradually reduce the consumption and production of hydrofluorocarbons. The Kigali Amendment took effect in 2019 and is a legally binding agreement designed to create rights and obligations in international law.

The European Union REPowerEU Plan and the United States Inflation Reduction Act encourage the uptake of heat pumps. In October 2022, the International Energy Agency released its flagship “World Energy Outlook” followed by the release of “The Future of Heat Pumps” in November 2022. The latter predicts that “Heat pumps have the potential to reduce global carbon dioxide emissions by at least 500 million tonnes in 2030.”

In this paper, the focus is on R290 in CCHP evaporators. Copper tube fabricators occupy a unique position in the heat pump supply chain. Interestingly, tube design strongly affects the performance of heat exchangers; and, in turn, high-performance heat exchangers profoundly affect the performance of heat pumps. This interrelationship begins with the tube manufacturers.

Rising sales of heat pumps in Europe and China are inspiring innovations in heat pump design. Heat pumps are widely recognized as a key technology for energy efficiency, electrification, and decarbonization. Ultralow GWP natural refrigerants such as propane (R290) and CO_2 (R744) are practical alternatives to high GWP HFC refrigerants. Cold-climate heat pumps allow for the benefits of heat pumps to be realized at higher latitudes than in the past. An ideal heat pump would operate well in cold climates using an ultralow GWP refrigerant.

A heat-pump design revitalization is taking shape in which all these objectives are achieved, and the heat exchanger design community is playing a leading role in this renaissance. Efficient heat exchangers in eco-friendly heating and cooling appliances can accelerate the planetary transition to “net zero carbon.”

2. HEAT EXCHANGER SIMULATIONS

Tube fabricators offer expertise and experience related to the manufacture and testing ACR tubes, including precision tubes with small diameters and proprietary inner grooves, or microfins. The geometry of the internal surface enhancements along with the operating conditions dictate the inside-the-tube heat transfer coefficient. This coefficient is not generally expressed as single number but rather as a correlation, depending on the type of refrigerant, temperature, pressure, flow rate, quality (phases) and other factors. The correlations are generally shared with heat transfer engineers and developers of heat exchanger simulation software.

Simulations now are commonplace in practically every branch of science and every field of engineering design, including thermal physics and the design of heat exchangers. The mathematics has been available for decades. Now computing power has caught up with the mathematics, making powerful simulation software widely available to product design engineers. Simulation tools for design engineers have also become easy to use with user-friendly menu driven interfaces and 3D visualizations of results.

Heat pump design is critically assisted by digital simulations of heat transfer in round tube plate fin (RTPF) heat exchangers. Advances in computational power as well as accurate tube and fin correlations now allow the airflow around tubes and fins as well as refrigerant flow inside tubes to be simulated with a high degree of accuracy.

Lui et al. (2004) outlined the core equations of tube-fin heat exchanger simulation software and provided references to earlier work. Early simulations focused on heat exchangers made from small diameter copper tubes. These early studies were driven by the desire to reduce materials usage in the high-volume manufacture of air conditioners. Subsequently, laboratory work was undertaken to measure the heat transfer coefficients (HTCs) inside smaller diameter copper tubes. For example, laboratory measurements of condensing R410A refrigerant were made inside smooth copper tubes with small diameters (Ding et al., 2010a) and microfin tubes (Ding et al., 2010b). Simulations were also developed for tube circuitry (Ding et al., 2010c) and fin efficiencies (Fang et al., 2010).

In recent years, the International Copper Association and Shanghai Jiao Tong University introduced HXSim™ software, a user-friendly simulation software program that can directly model many types of copper tube heat exchangers. Simulation results can be compared to other software that may include simpler or less accurate methods. Some examples of using this software to model smaller diameter copper tubes in RTPF heat exchangers have been presented by Ding (2019). Song et al. (2022) described a design methodology and simulations for refrigerated display cabinets with R404A refrigerant. Shabtay et al. (2021) described simulations for outdoor condenser units with copper-tube outer diameters of 7 mm vs. 5 mm in a nominal 3500 W split AC system with R32 refrigerant; and they compared cooling-cabinet condensers with tube diameters of 9.52 mm vs. 5 mm using R290 refrigerant and similar capacity.

Table 1 compares the results of simulations with testing of actual heat exchangers. Conditions for both the tests and simulations were as follows: Evaporation temperatures of 5 °C, 10 °C and 15 °C; inlet dryness of 0.1, 0.2 and 0.3; outlet superheat, 2 K; oil content of the refrigerant, 0.5 percent; and air inlet temperatures of 27 °C/19 °C (dry bulb/wet bulb).

Table 1. Comparison of simulation results with experimental results for a typical heat exchanger

Evaporation Temperature (°C)	Vapor Quality (Dryness)	Heat Transfer Test (W)	Heat Transfer Simulation (W)	Deviation (%)	Pressure Drop Test (kPa)	Pressure Drop Simulation (kPa)	Deviation (%)
5	0.1	1112	1198	7	6.26	5.02	20
10	0.2	976	1023	5	4.71	4.15	12
15	0.3	754	799	6	3.73	3.27	13
5	0.1	1131	1225	8	7.44	6.78	9
10	0.2	1008	1053	4	6.24	5.15	18
15	0.3	812	846	4	4.82	4.06	16

For the heat exchange capacity, the deviations were between four to eight percent with an average deviation of 5.6 percent; for airside pressure drop, deviations were between nine and twenty percent with an average deviation of 14.4 percent. Accuracy is best when true correlations are used in the simulations for both the refrigerant inside the tube and the airflow outside the tubes (Nasuta et al., 2018) and (Sarpotdar et al., 2018).

The deviations between the modelling and testing are typical of simulations performed for evaporators at low temperatures. Boiling evaporation is a two-phase process: Small changes in the temperature or refrigeration flow rate can lead to large deviations between the predicted performance and actual performance. Although Table 1 is not for propane refrigerant, the deviations at low temperatures are typical.

3. VARIATION OF TUBE DIAMETER AND MICROFINS

There has been a steady advancement of copper tube technology in the past two decades. Copper tubes with outer diameters of 5 mm with internally enhanced surfaces now are commonly used for air-conditioning and refrigeration applications (Cotton et al., 2019). Simulations demonstrate why the technology advanced in the direction of smaller diameter tubes as well as the advantages of enhancing the inside surfaces of the tubes.

Table 2 shows the differences in heat exchanger properties that occur as tube diameter is varied from 9.52 mm to 5 mm, while maintaining the same output. Inputs were adjusted to produce an output of about 14.4 kW for a constant flow rate of about 3.3 kg/min. In this initial study, the number of smaller diameter tubes (5 mm and 7 mm) was increased from 164 to 228 and coil depth was decreased to accommodate the refrigerant flow rate, which was held approximately constant for all four coils.

Table 2. Effects of reducing tube diameter on heat exchanger design

	5 mm	7 mm	7.94 mm	9.52 mm
Fins per Inch (FPI)	14	13	16	17
Number of Tubes	228	228	164	164
Coil Depth (mm)	50.8	66	65	88
Airside Pressure Drop (Pa)	50	62	68	73
Air Velocity (m/s)	2.45	2.45	2.42	2.42
Fin Weight (kg)	13.44	16.23	19.36	19.49
Internal Volume (liter)	3.99	8.58	8.50	12.40
Refrigerant Side Pressure Drop (kPa)	4.1	4.7	4.4	3.6
Refrigerant Flow (kg/min)	3.31	3.31	3.31	3.36
Refrigerant Velocity (kg/min)	5.92	5.51	5.74	4.96
Refrigerant Charge (kg)	0.52	1.13	1.10	1.64
Output (kW)	14.4	14.35	14.35	14.4

As the tube diameter decreases, while maintaining the same output, there is a decrease in the refrigerant charge, copper tube mass, aluminum fin mass and the size of the heat exchanger. This remarkable phenomenon has been long known and is already successfully employed in the manufacture of residential air conditioners and light commercial refrigerant equipment. For this reason, most of the rest of this paper focuses on 5 mm copper tubes. A simple I-block heat exchanger with two rows of copper tubes was used as a baseline coil design. Fig. 1 shows two- and three-dimensional renditions of this simple coil as generated by the simulation software.

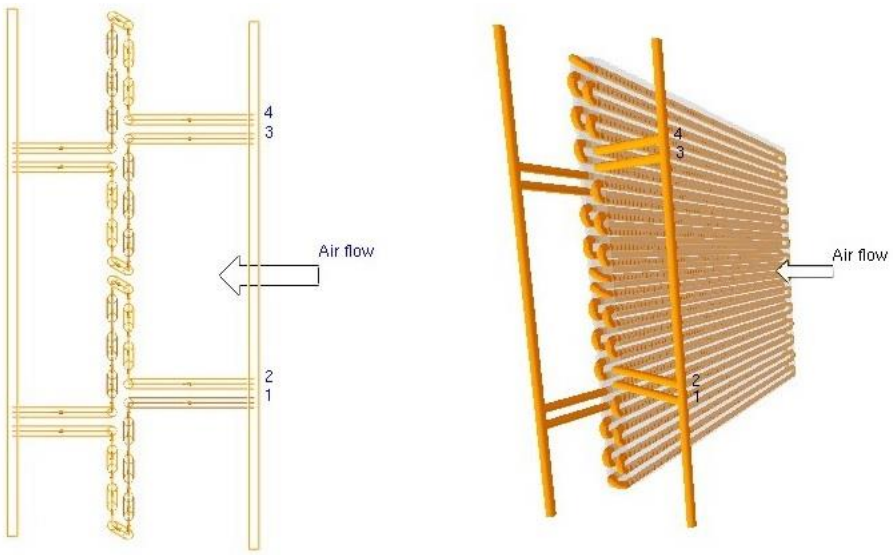

Figure 1: Two- and three-dimensional renditions of a two-row heat exchanger in HXSim.

Next, a study was made of the effects of internal surface enhancements (microfins or grooves) and wall thickness in the case of 5 mm tubes, using the correlations available for four types of enhancements and three wall thicknesses for smooth tubes. Table 3 shows the variation in heat exchange capacity as the tube thickness and internal surface enhancement were varied. These simulation results suggest that tube wall thickness has little effect on the capacity for the smooth tubes. They also show a difference between smooth tubes and microfinned tubes. As mentioned above, these differences must be interpreted with caution for evaporators operating at low ambient temperatures. The inside-the-tube heat transfer coefficient (HTC) is dependent upon the phase of the refrigerant and the pressure of the vapor as it travels through the tubes.

Table 3. Effect of inside tube enhancement on capacity

Inside Tube Enhancement (HXSim ID)	Tube Outer Diameter (mm)	Tube Wall Thickness (mm)	Microfin Height (mm)	Capacity (kW)	Airside Pressure Drop (kPa)
4	5	0.2	0.15	3.438	25.7
5	5	0.21	0.14	3.367	25.7
6	5	0.23	0.12	3.486	25.7
7	5	0.25	0.15	3.407	25.7
8	5	0.23	0	2.609	25.7
9	5	0.28	0	2.688	25.7
10	5	0.3	0	2.724	25.7

For a larger six-row heat exchanger, simulations on Lordan's proprietary software do not show much change in the performance of the evaporator at low ambient temperatures. All else being equal, the heat exchange capacity improved by less than seven percent, increasing from 5.03 kW for smooth tubes to only 5.36 kW for inner-grooved tubes.

While it is possible to insert additional microfin tube geometries in the HXSim software, accurate correlations are required for accurate simulations. It should also be noted that the advantage of the microfins differs according to the application. Microfins have different effects for boiling (evaporation) compared to nucleation (condensation).

4. VARIATION OF FIN DENSITY AND FIN TYPE

Fin pitch is an important factor in the design of heat exchangers. “Fin pitch” can be defined as the distance between the centerlines of the fins or inversely as the number of fins per unit length. Fin pitch is commonly expressed in terms of fin density, and fins per inch (FPI) is commonly used. For example, a fin density of 25.4 FPI is equivalent to a fin pitch of 1 mm. Considering that fin thicknesses are typically less than 0.3 mm, a fin density greater than 20 FPI is possible, depending on the application; however, for CCHPs where the evaporator may be subject to ice and snow, high fin density is not recommended.

Table 4 shows simulation results for varying fin density. The simulations were run on the same two-row, I-block coil that was used to compare microfins. The results clearly show that fin density affects airside pressure drop and capacity in the same direction. As the fin density increases, the airside pressure drop increases and so too does the capacity. As the fin density decreases, the airside pressure drop decreases and so too the capacity decreases.

Table 4. Variation of capacity and airside pressure drop with fin density

Fin Pitch (mm)	Fin Density (FPI)	Capacity (W)	Pressure Drop (kPa)
2.0	12.7	2437	10.5
1.9	13.4	2459	11.5
1.8	14.1	2623	12.7
1.7	15.0	2752	14.0
1.6	15.9	2905	15.6
1.5	17.0	3068	17.4
1.4	18.1	3247	19.7
1.3	19.5	3288	22.4
1.2	21.2	3361	25.7
1.1	23.1	3494	29.9
1.0	25.4	3506	35.3

Refrigeration equipment tends to use a low fin density, which is advantageous regarding “freeze up” and icing. The same applies also to CCHPs. Smaller diameter tubes and low fin density are also preferred for reverse-flow defrost cycles.

Often, it is desirable to create a fin die for a brand-new fin pattern. In these cases, simulation software would offer cost savings if the performance could be reasonably predicted, without having to build actual heat exchangers. The use of simulation software can reduce time to market and costs to build prototypes.

HXSim simulations were run on the same 19.05 mm x 16.5 mm hole pattern for wavy and louver fin types. Correlations for a slit fin were not available with this pattern. Fin type was varied using the same air flow. Heat exchange capacity increased from 3.36 kW for the wavy fin type to 3.53 kW for the louver fin type; however, the airside pressure drop more than doubled from 25.7 Pa to 54 Pa.

As in the simulations of fin density, fin type has a significant effect on capacity and airside pressure drop in the same directions. Simulations on louver fins and wavy fins for the same output capacity and airflow with all else being equal showed a larger pressure drop for the louver fin compared to the wavy fin.

Louvered fins and slit fins are rarely used for CCHPs. One reason why wavy fins are preferred over louver fins or split fins for CCHPs is that freeze up is quicker for the slit and louvered fins and defrosting becomes an issue (Martin, 2013). For an outdoor evaporator of a CCHP, there are therefore not many advantages gained from louvered or slit fins.

5. ADDITIONAL HALF ROW

In another simulation study, a half-row was added to the I-block design, to demonstrate the effect of adding rows on airside pressure drop and heat exchange capacity. Our study of the 5 mm heat exchanger compares two schemes: a two-row heat-exchanger (Fig. 1); and a "2.5 row" heat exchanger as illustrated in Fig. 2.

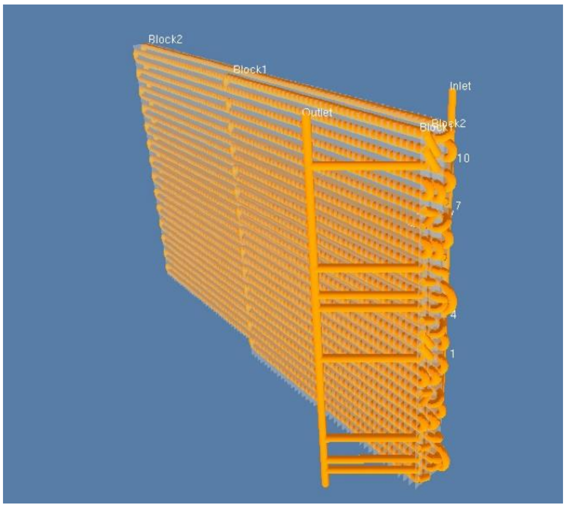

Figure 2: A half row was added to the I-block to create a 2.5 row heat exchanger.

The fin type is wavy. The length of the heat exchanger is 774 mm. Each row has 28 holes but the tubes in the third row are shortened to 400 mm: hence the "half row" in the "2.5 row" scheme. Table 5 shows that the additional "half row" increased the heat exchanger capacity from 3361 W to 3633 W.

Table 5. The effect on performance reduced tube diameter and adding a "half row"

	7 mm Original Prototype	5 mm (2 rows)	5 mm (2.5 rows)
Heat Exchange Capacity (W)	3617	3361	3633
Refrigerant Pressure Drop (kPa)	36.6	69.0	62.9
Heat Exchanger Inlet Temperature (^oC)	3.6	3.6	3.3
Inlet Dryness of Refrigerant (kg/kg)	0.19	0.19	0.16
Heat Exchanger Outlet Temperature (^oC)	2.7	– 1.0	2.4
Average Heat Transfer Coefficient of Refrigerant Side (W/m^2K)	4718	7956	5907
Airside Heat Transfer Coefficient (W/m^2K)	84.5	79.4	88.2
Air Flow (m^3/h)	1900	1884	1884/968

Care must be taken to properly model airflow when half rows are inserted in the heat exchanger. The airside pressure drop would vary through different areas of the heat exchanger and the airflow is not even across the face of the heat exchanger.

6. VARIATION OF TUBE CIRCUITRY FOR A CCHP EVAPORATOR

Armed with a profound knowledge of the effects of tube diameter, fin type and fin pitch as well as the number of rows, the next step in the design of a CCHP is the specification of tube circuitry. As an example, the design objective might be a fixed capacity of 15 kW for heating water to a temperature to 35 °C with other conditions as follows: ambient air temperature 7 °C dry bulb and 6 °C wet bulb; evaporation temperature 2 °C; superheat (SH) 2 K; and air speed 2 m/s (8000 m^3/h). Moreover, all tubes could be specified as internally enhanced with microfins, according to the advantages previously mentioned. Six rows of tubes were used in the circuit designs. Simulations were run for various circuits and fin hole patterns, and the results are tabulated in Table 6. These results could then be compared with respect to copper mass, aluminium mass, internal volume, refrigerant charge, and pressure drop. The most promising circuitry results in the lowest copper mass although it does not necessarily result in the lowest aluminum fin mass, for example.

Table 6. Various designs of 15 kw evaporator with six rows for CCHP

Design number	1	2	3	4	5	6
Tube diameter (mm)	5	5	7	7	8	9.52
Rows deep	6	6	6	6	6	6
Tubes per row	64	72	36	48	24	18
Total tubes	384	432	216	288	144	108
Column space (mm)	16 x	19.05 x	25 x	19.05 x	25.4 x	25.4 x
Row space (mm)	13.8	12.7	12.5	16.5	15.87	22
Fin density (FPI)	11	9	16	9	12	13
Fin type	Wavy	Wavy	Wavy	Wavy	Wavy	Wavy
Simulation Results						
Pressure drop, Airside (Pa)	58	49	81	56	64	70
Pressure drop, Refrigerant side (kPa)	4.2	2.3	2.0	1.3	2.9	3.0
Copper mass (kg)	16.32	22.8	20.99	27.30	23.74	32.71
Aluminum mass (kg)	14.92	11.66	21.36	15.88	20.35	30.55
Internal volume (liter)	5.66	7.67	9.23	12.1	12.1	17.6
R290 charge	0.794	1.072	1.551	1.94	2.132	3.427
Number of circuits	32	36	18	24	24	18
Distributor tube diameter (mm)	4	4	4.76 (3/16”)	4.76 (3/16”)	4.76 (3/16”)	4.76 (3/16”)
Tubes per circuit	12	12	12	12	6	6

7. CONCLUSION

Fig. 3 shows a three-dimensional simulation result for an evaporator built for a cold climate heat pump, using all the know-how described in this paper. The evaporator provides 15 kW heat exchange capacity, which can be used in a system that provides 20 kW of heating, enough for a whole house. Fig. 3 shows the actual evaporator on the factory floor prior to connection of the distributor lines to the 54 circuits.

The low-GWP heating system is entirely outdoors, allowing for a higher charge limit of propane to be used under certain conditions. The ambient heat collected from this evaporator is used in the system to heat water in an indoor tank or an insulated tank. That 35 °C water can then be circulated indoors for heating spaces, typically by forcing air through an air handler or a wall unit (i.e., a split system with a coil and fan).

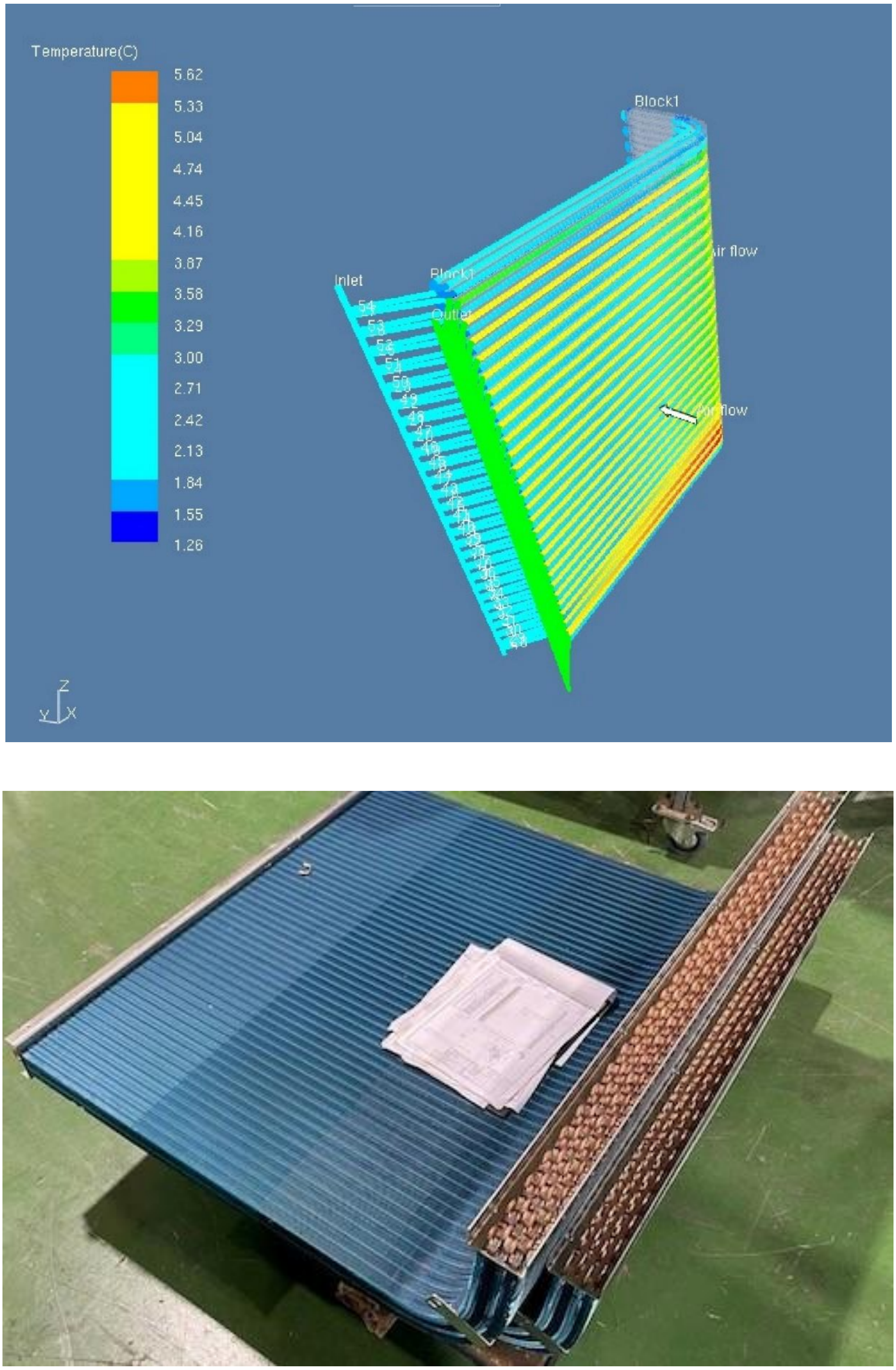

Figure 3: Three-dimensional simulation results with temperature map for 15 kW evaporator (top) and photograph of the actual evaporator on factory floor prior to brazing of the inlet and outlet distributor lines (bottom).

More and more, the advantages of air source heat pumps (ASHPs) for heating residential and commercial buildings are of great interest to policymakers, OEMs and the general public. Furthermore, the practicality of CCHPs is garnering attention around the globe. ASHPs for hot water and space heating are more efficient than boilers or furnaces. They also allow for electrification of residential heating, as well as commercial and industrial heating, enabling the use of alternative energy and reducing dependence on fossil fuels.

The vapor compression cycle (VCC) is central to a major paradigm shift in heating, and RTPF heat exchangers are at the heart of the VCC. Improvements in CCHPs are taking place at an astonishing pace as OEMs seek to become the first to market with superior products. There is much room for improvement using creative design and attention to fundamentals.

In this paper, the advantages of smaller diameter copper tubes were demonstrated along with a summary of the performance improvements possible using various combinations of tube circuitry, fin density and microfins. Simulations provide reasonable accuracy, although it is noted that simulations could always be improved with more accurate correlations for two phase flow of propane at low temperatures, especially with regards to the surface enhancements on the inside of smaller diameter copper tubes.

The research in this paper demonstrates the many advances in tube technology and simulation software that are driving breakthroughs in ASHPs and CCHPs. A profusion of new ASHP products for efficient electric heating will be introduced into global markets in next decade. CCHPs will deliver the benefits of heat pumps to regions of the world where ASHPs were impractical in the past and they will do so with ultralow GWP refrigerants. Many of these same regions also will adopt the VCC for the first time to cope with hotter summers. Further research and testing will provide even greater improvements in the heat exchanger design, leading to even less dependence on burning fossil fuels to keep the inhabitants of our planet warm in the winter and cool in the summer.

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NOMENCLATURE

ASHP	air source heat pump	CCHP	cold climate heat pump
FPI	fins per inch	GWP	global warming potential
RTPF	round tube, plate fin	VCC	vapor compression cycle

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