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Simulation of the effects of copper tube diameter on refrigerant charge reduction in split AC systems and refrigerated cabinets

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ABSTRACT

Heat exchanger design software accurately simulates the performance of heat exchangers, allowing a wide range of design parameters to be explored without building prototypes. In this paper, basic principles are reviewed using HXSim software from the International Copper Association. The first case study compares 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. The second compares cooling-cabinet condensers with tube diameters of 9.52 mm vs. 5 mm using R290 refrigerant and similar capacity. Detailed graphical and tabular input and output reveal the ease and accuracy of simulations for a wide range of design parameters and performance objectives. The International Copper Association offers software to qualified designers and OEMs and will assist in the development of heat exchangers optimized for efficiency, refrigerant charge and materials usage.

Keywords: Heat Exchanger, Condenser, Evaporator, Copper, Simulation, HXSim, MicroGroove, Microfins, R32, R290

1. INTRODUCTION

Coil design has yielded to fast computer-based numerical calculations. The equations governing heat transfer and mass flow can now be employed in software that calculates the performance of tube-fin heat exchanger coils to a high degree of accuracy. What's more, a wide range of parameters can be varied with ease in the latest generation of coil simulation software. Three-dimensional graphical representations of coil design inputs and 3D output of performance results are now routine.

Ding (2019) describes the features of the latest generation of heat exchanger simulation software in a user guide. The user enters design parameters such as coil block type, tube spacing, fin spacing and so on (Fig. 1) through a graphical user interface. Operating conditions such as refrigerant and air mass flow are entered. The software then calculates the behaviour of the coil and displays the results in a table, chart or as 3D visualizations.

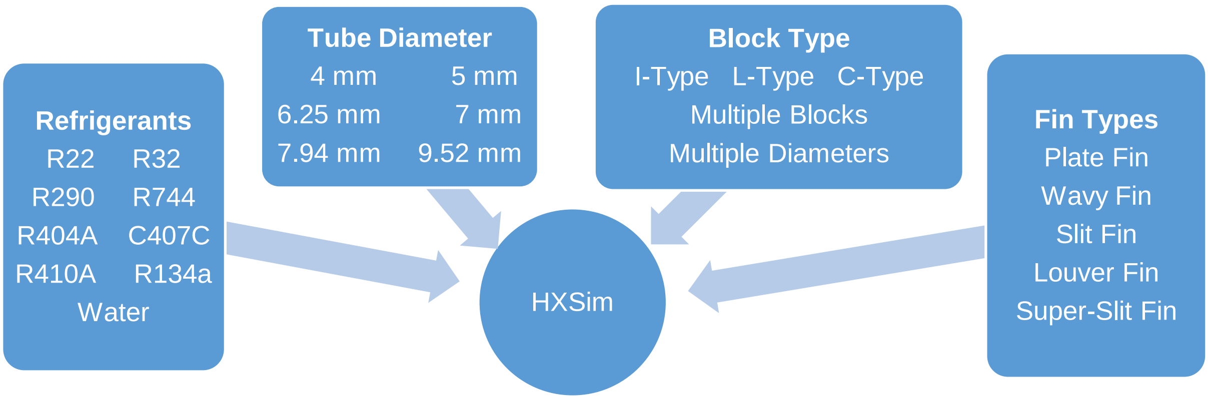

Figure 1: Tube diameter, block type, refrigerants, fin types and other parameters are input via menu choices within the HXSim software program.

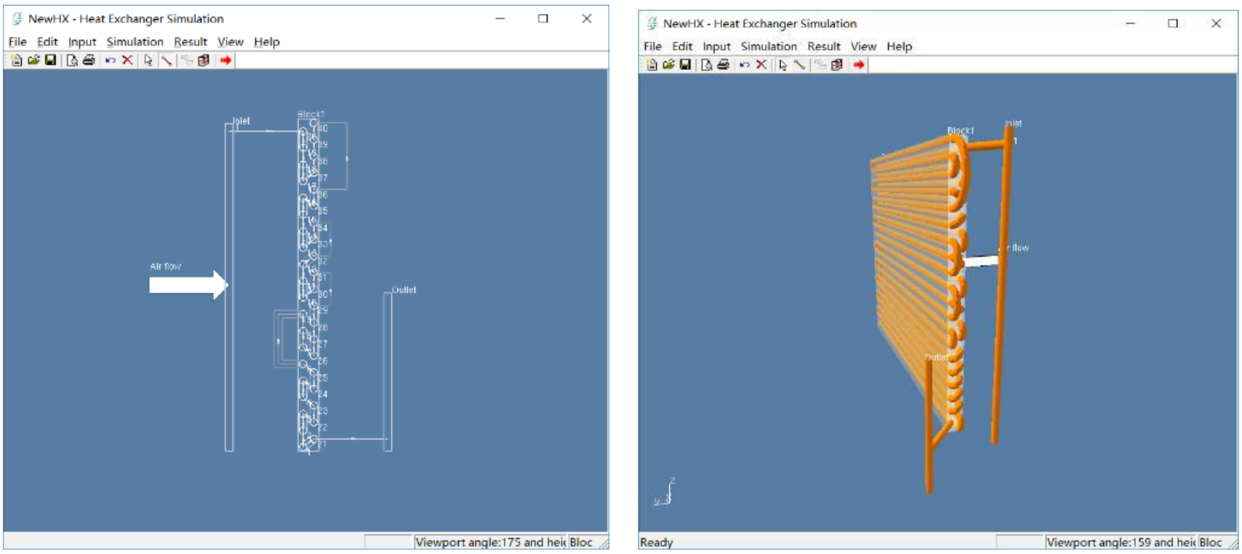

Figure 2: Tube circuitry is specified by linking tube ends together in a two-dimensional graphical interface. A three-dimensional visualization of the circuitry is then produced.

The software database includes thermophysical properties for eight refrigerants (R22, R410A, R404A, R407C, R134a, R32, R290 and R744) commonly used in vapor-compression systems as well as properties of water for hydronic coil design. It includes twelve different fin-types and tube diameters from 9.52 mm to 4 mm, including smooth and inner-grooved tubes. The tube circuitry is built by linking tube ends together using a graphical interface either in 2D or 3D (Fig. 2).

Empirical correlations for a wide variety of fin designs and internally enhanced tubes are built into the software. The designer can pull down menus, choose the feature to be varied, and select from an extensive list of types of fins and tubes. The appropriate correlations are then used in the simulations. The designer can capture the physical behaviour of intricately designed plate fins and round tubes with various diameters, wall thicknesses and microfin internal tube enhancements.

After more than a decade of continual improvement, this heat-exchanger simulation tool is now widely available in a single software program that runs on a laptop computer (ICA, 2021).

Two application design case studies are demonstrated in this paper, showing the capabilities and simplicity of HXSim software. First of all, a brief history of heat-exchanger simulation software is recounted.

2. THE EVOLUTION OF HXSIM DESIGN SOFTWARE

2.1. Core Computing Engine

The use of computational fluid dynamics (CFD) and finite element analysis (FEA) methods to analyse the airflow around the tubes and fins as well as computer simulations of refrigerant flow and temperatures inside the tubes continually improved as computing power increased. Ding et al. (2011) outline the core equations of tube-fin heat exchanger simulation software and provide references to earlier work

A parallel development in manufacturing was the successful fabrication of smaller diameter copper tubes. Early simulations focused on heat exchangers made from small diameter copper tubes driven by the desire to reduce materials usage in the high-volume manufacture of air conditioners. Initially, a limited number of tube correlations were available. Meanwhile, 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).

2.2. Correlations

More recently, ever more accurate, refrigerant-side correlations were determined for various diameters of copper tubes with and without internal tube enhancements, e.g., microfins, using many refrigerants and refrigerant blends. Separately, airside correlations were developed for various fin designs. These empirically determined correlations improved the accuracy of simulation software for an ever-growing combination of design parameters.

By way of examples, researchers from the University of Padova examined the behaviour of low-GWP refrigerants and refrigerant blends for flow boiling inside smooth (Longo et al., 2018a) and microfin (Mancin et al., 2018) tubes as well as condensation inside a smooth smaller-diameter copper tube (Longo et al., 2018b). Inoue et al. (2018) showed the combined effects of flow rates and microfin geometry on heat transfer coefficients and pressure drops for R1123/R32 flow in horizontal microfin tubes during condensation and evaporation. Sarpotdar et al. (2016a) developed correlations for small diameter (3 mm to 5 mm) louver-fin heat exchangers and also compared the performance of slit fins and louver fins (Sarpotdar et al., 2016b).

Various designs can be compared and evaluated in terms of multiple objectives, including reduced refrigerant charge, high efficiency, space constraints, or lower noise (low airside pressure drop). Cotton et al. (2019) reported on five different application case studies that used a multiple objective genetic algorithm (MOGA) to optimize heat exchanger designs. The MOGA approach still depends on knowledge-based design principles but the design process is aided and accelerated by reliable simulations and genetic algorithms.

3. SPLIT AC OUTDOOR UNIT

3.1. Structure of the Coils

The first case study is an I-Type tube-fin condenser coil for the outdoor unit (ODU) of a split air conditioner with a nominal cooling capacity of 3500 watts using R32 refrigerant. Both the original and the prototype ODU had identical block dimensions: 770 mm (length) by 36.4 mm (depth) by 505 mm (height).

In both cases, the fins were made from 0.105 mm thick aluminium plates. The fin pitch is dictated by the collar height made by the fin die. The fin pitch was tighter in the prototype design compared to the original design; the prototype uses slit fins with a 1.3 mm fin pitch whereas the original used wavy (corrugated) fins with a 1.4 mm pitch. These values are easily entered using pull down menus and popup windows.

Besides having smaller holes, the prototype fin has more holes and tighter hole spacings than the original. The original had two rows of twelve 7 mm diameter copper hairpin tubes; hence, 24 holes per column and 48 holes total; with a column-by-row spacing of 21 mm by 18.2 mm.

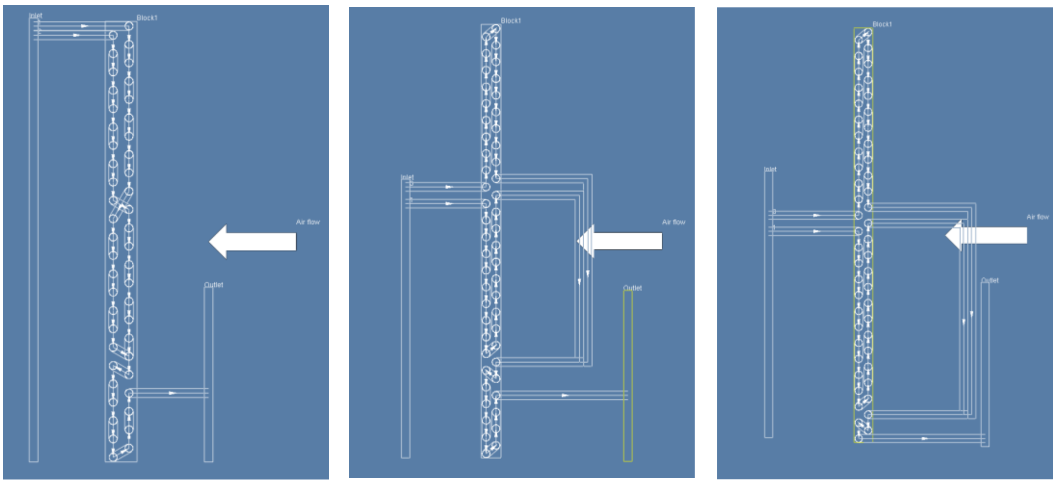

Figure 3. Circuitry (left to right) of the original design, first prototype and second prototype.

The prototype uses two rows of thirteen 5 mm copper hairpin tubes; hence, 26 holes per column and 52 holes total; with a column-by-row spacing of 19.5 mm by 11.6 mm. Again, these differences are a consequence of available fin dies and are easily entered in the simulation software graphical user interface.

Considerations of flow rates suggest that two circuits paths converging into one path is preferred. For a symmetrical flow circuit with thirteen hairpins in each row, two options are available: five hairpins in each row in each of two paths, converging into one path with three hairpins in each row (10-10 and 6); or six hairpins in each row in each two paths, converging into one path with one hairpin in each row (12-12 and 4). These two prototype configurations along with the original configurations are illustrated in Fig. 3.

3.2. Design Objectives

The original AC has an EER of 3.6 and a rated cooling capacity of 3500 W. On the refrigerant side, the discharge temperature and pressure are 2861 kPa and 68 °C, respectively; and the condensing temperature is 45.5 °C. On the air side, the air flow rate is 1800 $m^3 h^{-1}$. For an air conditioner cooling capacity of 3500 W, the condenser heat-exchanger capacity is 4470 W. Subcooling needs to be at least 5 °C below the air inlet temperature, which is 35 °C dry bulb and 24 °C wet bulb. Under these conditions, the mass flow rate is calculated to be 59 kg h^{-1}, according to the following equation.

$$m = \frac{Q}{h_{\mathrm{in}} - h_{\mathrm{out}}} \qquad \text{Eq. (1)}$$

Where m is the mass flow rate (kg s^{-1}); Q is the desired heat exchange capacity (W); and h_{in} and h_{out} are the specific enthalpies (kJ kg^{-1}) flowing in and out of the heat exchanger.

3.3. Performance Results and Graphical Visualization

COIL SIDE			
Fin Type	Slit	Utilized Tubes	52
Fin Material	Aluminum	Non Utilized Tubes	0
Fin Spacing [mm]	1.30	Circuits	3
Fin Thinkness [mm]	0.105	Tubes Per Circuit	17.33
Tube Type	Grooved	Coil Length [mm]	770.00
Tube Material	Copper	Coil Depth [mm]	23.20
Tube Dimension [mm]	5.00*0.21*0.14	Coil Height [mm]	507.00
Holes	26	Outer Area [m2]	13.166
Rows	2	Inner Area [m2]	0.576
Tube Vertical Space [mm]	19.50	Coil Face Area [m2]	0.39
Tube Horizontal Space [mm]	11.60	Inner Volume [L]	0.659
Header In [mm]	9.5	Header Out [mm]	9.5
AIR SIDE		**REFRIGERANT SIDE**	
Air Inlet DB. Temp. [°C]	35.0	Refrigerant	R32
Relative Humidity %	40.3	Discharge Superheat [°C]	22.50
Air Outlet DB. Temp. [°C]	43.3	Condenser Temp.[°C]	45.50
Relative Humidity %	25.8	Subcooling [°C]	4.97
Air Flow [m3/h]	1835.6	Mass Flow [kg/h]	62.0
Air Mass Flow [kg/h]	2357.5	Pressure Drop [kPa]	160.903
Frontal Velocity [m/s]	1.3	Outlet Pressure [kPa]	2666.846
Air Pressure Drop [Pa]	18.0	Ref. Charge [kg]	0.20
Atmospheric Pressure [kPa]	101.3	Ref. Side H.T.C. [W/m2*K]	3130.434
Air Side H.T.C. [W/m2*K]	183.796		
CAPACITY			
Total Capacity [kW]	4.794		

Figure 4: Simulation results for the first prototype using 5 mm tubes are output as a table in HXSim.

Running the simulation on the above specifications yields values of capacity, pressure drop and degrees of subcooling with output as illustrated in Fig. 4.

The original heat exchanger with 7 mm diameter tubes has a simulated capacity of 4500 W, which as expected is higher than the requirement of 4470 W. The two prototypes with 5 mm diameter tubes have even higher capacities of 4612 W and 4564 W. From these results, it can be concluded that the 5 mm heat exchanger is adequate as an outdoor condenser unit.

More importantly, the refrigerant charge is more than halved. It is reduced from 450 g for the 7 mm tubes to 200 g for the 5 mm tubes; and there are substantial savings in tube materials as well. The refrigerant pressure drop of the baseline is 20 kPa but 104 kPa and 128 kPa in the new designs. This a consequence of the smaller diameter tube. A moderate pressure drop in condensation has less effect on performance compared to a pressure drop in evaporation. This pressure drop may be acceptable, but it could also be lowered below 104 kPa by adding more branches to the condenser circuitry.

Refrigerant temperature can be calculated for many locations along the refrigerant path. Fig. 5 shows the 3D visualization results for temperature gradations from two perspectives. The slide of condensing temperature is within 3 °C; furthermore, the slides of condensing temperature in the two parallel paths are within 1.5 °C. The 5 mm tube heat exchanger has relatively good performance; hence, 5 mm tube heat exchangers are widely used in the outdoor units of air conditioners. The results show that the first prototype performs better than the second prototype. (Fig. 3 and Fig. 5).

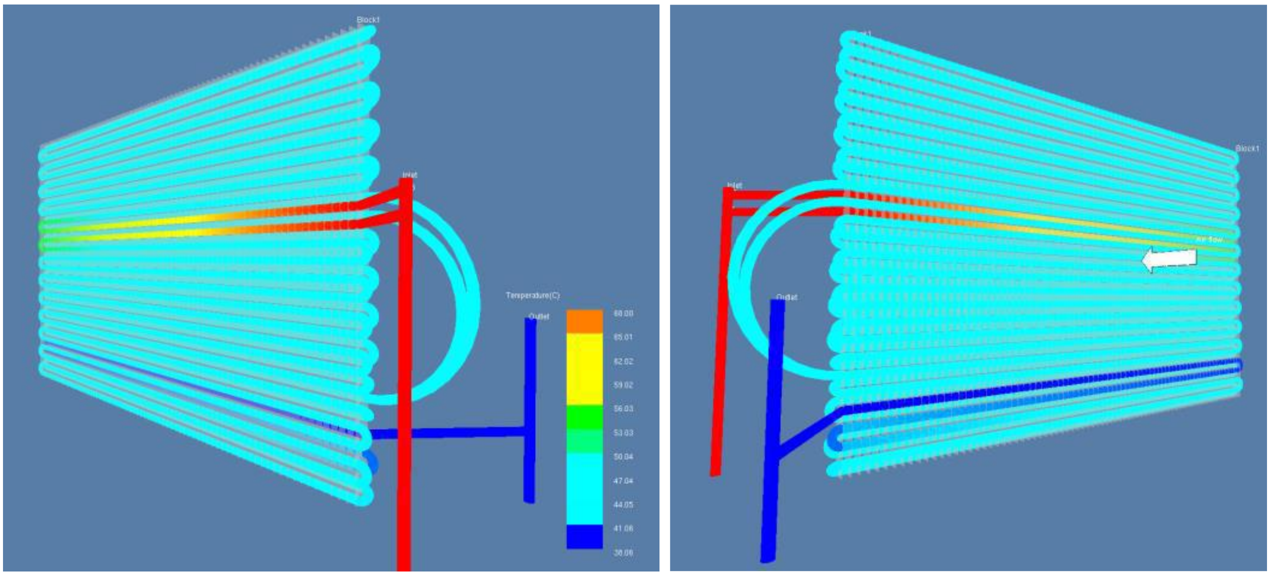

Figure 5: Simulation results can be output as three-dimensional graphic images.

4. CONDENSER COIL FOR A COOLING CABINET

4.1. Structure of the Coils

Table 1 lists the geometrical features of the two exchangers. The original condenser has 9.52 mm tubes in four rows with eight tubes per row; the prototype has 5 mm tubes in four rows with ten tubes per row. The original (with 32 tubes) has an inner volume is 0.433 L compared to only 0.142 L for the prototype (with 40 tubes).

The 9.52 mm fin die had (row by column) spacings of 21.65 mm by 25 mm; the 5 mm fin die had spacings of 19.05 mm by 16.5 mm. A wavy fin type was selected for both condensers, because this is an application for a dirty environment. The fin pitch in both cases was 3 mm, which again is suitable for long-term use with no maintenance.

Table 1. Heat Exchanger Geometry

Parameters	Original	Prototype
Tube diameter (mm)	9.52	5
Tube length (mm)	278	278
HX depth (mm)	86.6	66
HX height (mm)	200	190.5
Number of columns	4	4
Tubes per column	8	10
Row spacing (mm)	21.65	16.5
Column space (mm)	25	19.05
Fin pitch (mm)	3	3

4.2. Design Objectives

The purpose of the case study is to demonstrate refrigerant charge reduction using smaller tube diameter. The phase out of HFCs from light commercial refrigeration equipment is a success story that has resulted in the transformation of a major product category.

The present case study focuses on refrigerant charge reduction. Both heat exchangers of this case study use R290, which is a "natural refrigerant" with an ultralow GWP of 3. Holding the refrigerant type constant, simulations dramatically illustrate the charge reduction possible in switching from 9.52 mm (3/8 in. diameter) diameter copper tubes to 5 mm diameter copper tubes.

To meet the performance objective of the cabinet, a minimum capacity of 762W was required of the condenser. A calculation using Eq. 1 gives a mass flow rate of 18 kg h^{-1} based on the specific enthalpies of the refrigerant flowing into and out of the heat exchanger. It is determined that this flow rate can be met using a single path even for the 5 mm tube circuit. Flow circuitries are shown in Fig. 6.

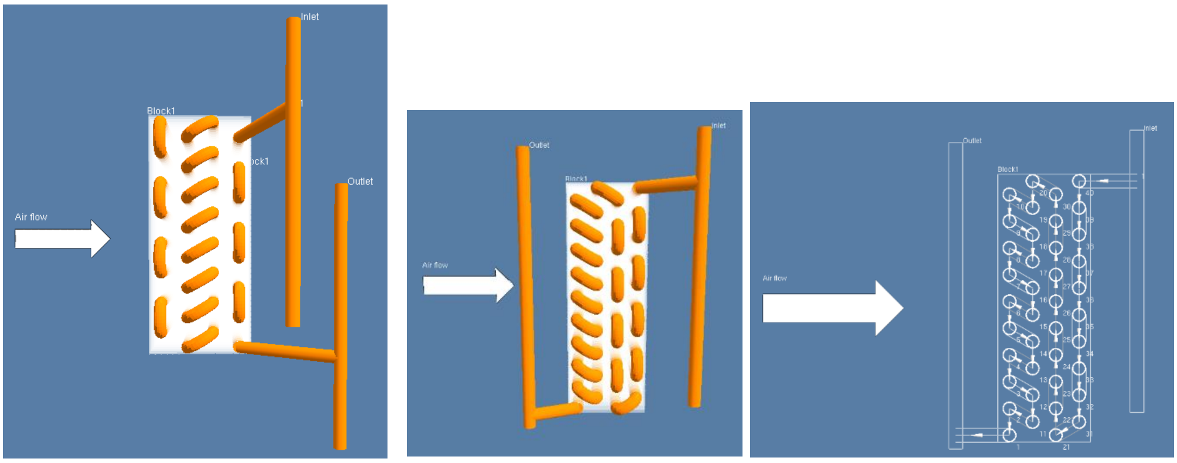

Figure 6: Original design (*left*: 32 tubes, 9.52 mm, single path) and prototype design (*middle and right*: 40 tubes, 5 mm, single path) for R290 condensers.

4.3. Performance Results

Table 2 lists key performance parameters obtained from the HXSim program. The results show that refrigerant charge is reduced from 135 grams to 100 grams while *increasing* the capacity from 762 W to 814 W. This "doing more with less" condition is possible because the refrigerant side heat transfer coefficient (HTC) is *nearly doubled* from 1130 W m^{-1} K^{-1} to 2016 W m^{-1} K^{-1}.

Table 2. Key Heat Exchanger Performance Parameters

Parameters	Original	Prototype
Thermal capacity (W)	762	814
Inlet pressure (kPa)	1385	1385
Inlet temp. (°C)	105	105
Mass flow rate (g s^{-1})	3.7	3.7
Pressure drop (kPa)	2.3	40
HTC of Ref. (W m^{-1} K^{-1})	1130	2016
Subcooling (°C)	1.52	6.6
Weight of Ref. (g)	135	100

Meanwhile, refrigerant-side pressure drop is 2.3 kPa for the larger tubes compared to 40 kPa for the smaller tubes at the same mass flow (3.7 g/s). Although the pressure drop is higher for the smaller tubes, the absolute value is only a small fraction of the outlet pressures (1345 kPa for the smaller tubes and 1383 kPa for the larger tubes).

These simulations are easy to program and run in HXSim software. Refrigerant choices are not limited to R290. Similar reductions in refrigerant charge were obtained running the simulation with any of the available refrigerants.

5. CONCLUSIONS

Simulation tools are just what is needed in the fast-paced world of heat exchanger design today. Energy efficient, environmentally friendly ACR products are needed at every scale, including residential and light commercial as well as commercial and industrial scales. Furthermore, the development of affordable heat pumps for water and space heating are vital steps toward electrification and the use of carbon-free energy sources, while at the same time providing a multi-fold increase in the coefficients of performance compared to natural gas or electrical resistance heating. For these reasons, the International Copper Association in collaboration with Shanghai Jiao Tong University is offering the HXSim heat exchanger software to qualified heat exchanger designers (ICA, 2021).

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Comments from Reviewers: minor revisions

Reviewer 1:
The idea of the work is interesting and innovative, but the paper is a bit simple and "commercial". Some pictures are not useful for the reader and there are many typos.
NOTED. SEE RESPONSE TO REVIEWER 2.

THESE POINTS HAVE BEEN CORRECTED
The underscore dot "." should not be used in the measurement units, please change. DONE.
The Keywords list is really too long. SHORTENED.
Please, avoid "grams" and "kg/h" in the text. [CHANGED grams → g, AND kg/h → kg h^{-1}]
I personally don't like Figure 7 placed after the Conclusions. FIGURE 7 DELETED
In the Reference, the two works for Longo et al. 2018 are wrong, they report "Diani" instead "Mancin". CORRECTED. APOLOGIES. ERROR FROM PURDUE COVER PAGE HERE
https://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=2879&context=iracc

Reviewer 2:
The manuscript presents two case studies implemented on software created by the authors, called HXSim software. I certainly find it an interesting topic to the conference audience, but I think it is treated a little bit too commercially.
TONED DOWN PHRASES LIKE "FREE OF CHARGE" AND "USER FRIENDLY" AND WHOLE SCREEN SHOTS AND USER-FRIENDLY INTERFACE HAVE BEEN CROPPED FROM FIGURES

It is important to reduce the refrigerant charge amount, but it would be more interesting to give an example with a low-GWP refrigerant instead of R404A. Or, it would be even more interesting to compare different refrigerants and not just write that the simulation can be run with any of the available refrigerants. Not being a software usage tutorial, I ask the Authors to modify some images (figure 1, figure 4, figure 7) making them more readable. The text size is so small that it cannot be read. Also, I don't find it useful to present whole screen picture.

SECOND CASE STUDY NOW SIMULATES LOW GWP R290 INSTEAD OF R404A, SHOWING DROP IN REFRIGERANT CHARGE FROM SMALLER DIAMETER TUBES.
FIGURE 1 -- WHOLE SCREEN PICTURE OMITTED. OTHER IMAGE ENLARGED.
FIGURE 4 -- "WHOLE SCREEN" ELEMENTS CROPPED. TABLE ENLARGED/READABLE.
FIGURE 7 -- OMITTED

A note: the manuscript contains a lot of typos; I recommend reading it again.
WE HAVE PROOFREAD AGAIN AND CORRECTED TYPOS.

If you are required to submit the revised paper, please make sure to include a written response to the comments from reviewing committee in the additional page after the last page of your revised paper. In the final manuscript, the additional page will be deleted.
Please submit the revised paper from here.
URL: https://www.aicarr.org/Pages/Convegni/AreaRelatori/Send_paper_eng.aspx
Submission deadline: June 30, 2021 (CET)