Design Evaluation for a Finned-Tube CO₂ Gas Cooler in Residential Applications

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Abstract: Towards the introduction of environmentally friendlier refrigerants, CO₂ cycles have gained significant attention in cooling and air conditioning systems in recent years. In this context, a design procedure for an air finned-tube CO₂ gas cooler is developed. The analysis aims to evaluate the gas cooler design incorporated into a CO₂ air conditioning system for residential applications. Therefore, a simulation model of the gas cooler is developed and validated experimentally by comparing its overall heat transfer coefficient. Based on the model, the evaluation of different numbers of rows, lengths, and diameters of tubes, as well as different ambient temperatures, are conducted, identifying the most suitable design in terms of pressure losses and required heat exchange area for selected operational conditions. The comparison between the model and the experimental results showed a satisfactory convergence for fan frequencies from 50 to 80 Hz. The absolute average deviations of the overall heat transfer coefficient for fan frequencies from 60 to 80 Hz were approximately 10%. With respect to the gas cooler design, a compromise between the bundle area and the refrigerant pressure drop was necessary, resulting in a 2.11 m² bundle area and 0.23 bar refrigerant pressure drop. In addition, the analysis of the gas cooler’s performance in different ambient temperatures showed that the defined heat exchanger operates properly, compared to other potential gas cooler designs.

Keywords: supercritical carbon dioxide; experimental testing; finned-tube gas cooler

1. Introduction

The use of air conditioning systems is expanding rapidly around the world. An estimated amount of 700 million air conditioners will be operating in the world by 2030 [1]. This growing demand for air conditioning systems has enormous impacts on the environment.

Currently, a number of present regulations have been applied worldwide to control the use of harmful refrigerants [2–4]. The key implications of the use of conventional refrigerants include the depletion of the ozone layer and global warming. Based on the Montreal Protocol, a complete abolishment of chlorofluorocarbons (CFCs) was decided, due to their high ozone depletion potential (ODP) [5]. In addition, the phase out of hydrochlorofluorocarbon (HCFC) refrigerants was implemented [6]. On the other hand, the F-gas Regulation, first issued in the European Union in 2006, aimed to introduce measures for the reduction of fluorinated gases—hydrofluorocarbons (HFCs) and perfluorocarbons (PFCs)—in form of a phase-down, due to their high global warming potential (GWP) [7].
Instead, natural refrigerants are proposed as the substitute for the harmful refrigerants. Carbon dioxide (CO\textsubscript{2}) is a natural, low cost, non-flammable, non-toxic refrigerant. Subsequently, it has emerged as a credible natural refrigerant to replace HFCs and HCFCs. However, its unique critical point, high critical pressure of 73.8 bar, and low critical temperature of 30.98 °C, remarkably affects the performance of CO\textsubscript{2} refrigeration systems, as well as imposes special design and control challenges. When the ambient temperature is higher than the critical temperature of CO\textsubscript{2}, the system operates at supercritical conditions, and the heat rejection process occurs at a supercritical regime. In consequence, a phase change does not take place, and the heat exchanger in which this change of state occurs is called the gas cooler.

The impact of the gas cooler on the CO\textsubscript{2} refrigeration systems plays an important role, due to its high exergy loss. Therefore, it is considered vital to be further investigated and designed properly \[8\]. The finned-tube type for gas coolers is well established in the heating, ventilation, and air conditioning (HVAC) and refrigeration industries, due to its compactness and manufacturing flexibility. The design of the finned-tube heat exchangers affects considerably the overall heat transfer performance and system efficiency. Particularly, fin and tube thickness and the respective materials, spacing, and dimensions of the tubes and fins are crucial parameters of the design \[9\]. Fundamental studies about the heat transfer characteristics during the heat rejection process in tubes have been performed theoretically and experimentally by many researchers since Lorentzen and Pettersen \[10\] proposed the transcritical CO\textsubscript{2} cycle for mobile air conditioning systems.

Pitla et al. \[11\] conducted an investigation about heat transfer phenomena and pressure losses of CO\textsubscript{2} at supercritical conditions into a tube. They found that the majority of the deviations between the numerical and experimental values are within ± 20%, and a new heat transfer correlation was presented. Son and Park \[12\] carried out an experiment in order to investigate the gas cooling process of CO\textsubscript{2} in terms of heat transfer coefficient and pressure drop of the refrigerant. They described the variations of local heat transfer coefficient in the cooling process in the direction of the flow and proposed a more accurate heat transfer correlation. Zhang et al. \[13\] evaluated the performance of a printed circuit heat exchanger for cooling CO\textsubscript{2} with water. The analysis concluded that rapid variations in the thermodynamic properties of supercritical CO\textsubscript{2} increase entropy generation and therefore, to optimize the second law efficiency of the investigated heat exchanger, higher CO\textsubscript{2} mass flow rates should be used. Jadhav et al. \[14\] evaluated, using simulations, CO\textsubscript{2} gas coolers for air conditioning applications. For their investigation, a counter crossflow plain fin and staggered tube configurations were considered. According to the study, transverse tube spacing, gas cooler width, and air volumetric flow were the most influential parameters in the heat transfer mechanisms of the gas cooler.

Liu et al. \[15\] investigated experimentally the supercritical CO\textsubscript{2} characteristics in horizontal tubes with inner diameter of 4, 6, and 10.7 mm in terms of heat transfer phenomena and pressure losses. The authors concluded that the tube diameter significantly affects the heat transfer performance, and they proposed a new heat transfer correlation for the large diameter. Jiang et al. \[16\] investigated the convection heat transfer of CO\textsubscript{2} at supercritical pressures in a vertical small tube with inner diameter of 2.0 mm, experimentally and numerically. They studied the effects of various operational parameters and buoyancy on convection heat transfer in a small diameter. They concluded that when the CO\textsubscript{2} bulk temperatures are in the near-critical region, the local heat transfer coefficients vary significantly along the tube. Chai et al. \[17\] investigated, using simulations, the performance of finned-tube supercritical CO\textsubscript{2} gas coolers, combining a distributed modeling approach with the ε-NTU method. The results indicated that the performance of the gas coolers was enhanced by higher mass flow rates and lower tube diameters at the expense of higher pressure drops.

Other researchers have also investigated the performance of the air-cooled CO\textsubscript{2} gas coolers. Cheng et al. \[18\] presented an analysis of heat transfer and pressure drop experimental data and correlations for supercritical CO\textsubscript{2} cooling in macro- and micro-channels. Ge and Cropper \[19\] presented a detailed mathematical model for air-cooled finned-tube CO\textsubscript{2} gas coolers. They used a distributed method in order to obtain more accurate refrigerant thermophysical properties and local heat transfer
coefficients during cooling processes. The model was compared with published test results. The comparison showed that the approach temperature and the heat capacity are simultaneously improved with the increase of heat exchanger circuit numbers. Marcinichen et al. [20] conducted simulations to optimize the working fluid charge of the gas cooler. The optimal design of the study reduced \( \text{CO}_2 \) charge by 14%, compared to a reference design. Moreover, the analysis revealed the importance of the oil concentrations in the \( \text{CO}_2 \) pressure drop, which is up to 2.65 times higher for oil concentrations of up to 3%. Zilio et al. [21] experimentally evaluated two different gas coolers, one with continuous, and one with separated fins, and on two different circuit arrangements for a transcritical \( \text{CO}_2 \) cycle. Using a coil with fins, a heat flux improvement of up to 5.6% was identified, which corresponded to a coefficient of performance (COP) increase of up to 6.6% for a conventional \( \text{CO}_2 \) refrigeration cycle.

Gupta and Dasgupta [22] applied a similar modelling method to the one from Ge and Cropper, [19] in order to evaluate the performance of the heat exchanger being affected by the airflow velocity. Here, a higher gas cooler performance is achieved at a higher air flow velocity as it decreases the refrigerant’s approach temperature, and thus the heating capacity of the gas cooler is increased. Santosa et al. [23] built two \( \text{CO}_2 \) finned-tube gas coolers with different structural designs and controls, connected with a test rig of a \( \text{CO}_2 \) booster refrigeration system. They carried out experiments at different operating conditions while they developed models of the finned-tube \( \text{CO}_2 \) gas cooler. The analysis was conducted based on the distributed and lumped methods. They concluded that the heat exchanger design can affect the performance of both the component and the integrated system.

Although the heat transfer and pressure drop characteristics of the supercritical \( \text{CO}_2 \) in tubes have been investigated extensively using experimental and theoretical methods, research on the air-cooled finned-tube \( \text{CO}_2 \) gas coolers is still limited.

In this paper, mathematical calculations of the finned-tube \( \text{CO}_2 \) gas cooler are conducted, in order to establish a reliable design procedure. With focus on the heat transfer characteristics of the air side, the developed model is validated with an experimental setup using water as working fluid. Investigations of the effects of fan frequency, water inlet temperature, and water mass flow on the overall heat transfer coefficient are conducted, while deviations between the model and the test results are extracted according to the fan frequency. In addition, potential heat transfer correlations for the air- and refrigerant-side heat transfer coefficients have been studied. Finally, the model was applied to identify a reliable and efficient finned-tube \( \text{CO}_2 \) gas cooler design, as well as to evaluate its performance in different off-design conditions under varying ambient temperatures.

2. Materials and Methods

This study is part of a larger project of \( \text{CO}_2 \) air conditioning systems for residential applications and focuses on the gas cooler. A scheme of the considered \( \text{CO}_2 \) air conditioning system is depicted in Figure 1. Particularly, an efficient and reliable air finned-tube gas cooler was designed based on the boundary conditions, which are given in Table 1.

| Property                  | Value       |
|---------------------------|-------------|
| \( \text{CO}_2 \) inlet pressure (bar) | 93          |
| \( \text{CO}_2 \) temperature inlet/outlet (K) | 358.22/311.15 |
| \( \text{CO}_2 \) mass flow rate (kg s\(^{-1}\)) | 0.146       |
| Air temperature inlet/outlet (K) | 308.15/315.15 |
| Air mass flow rate (kg s\(^{-1}\)) | 3.601       |
| Heat duty (kW)            | 25.4        |
The **U**-value consists of three parts: air convection, refrigerant convection, and the conduction. Compared to the other two heat transfer contributions, the conduction plays a minor role. The heat transfer coefficients on the air- and refrigerant-side are crucial for the overall heat transfer coefficient, and the validation for them are considered necessary. In order to validate the calculations, the overall heat transfer coefficient of a defined air-cooled heat exchanger was investigated experimentally.

The experimental part was based on a test rig using water and employed a specific design of an air-cooled heat exchanger (HEX). The HEX type was a finned tube with a fan air cooling system. The **U**-value was investigated experimentally for the entire HEX for different conditions. The second part of the validation consisted of modelling the heat exchanger to simulate the finned tube of the HEX.

### 2.1. Theoretical Model

For the model calculations, a script was created in MATLAB R2019a [24], modelling the defined gas cooler. The model overall heat transfer coefficient of the HEX was calculated from Equation (1).

\[
U = \left[ \frac{1}{h_{\text{air}}} + \frac{A_o \times \ln \frac{d_o}{d_i}}{2 \times \pi \times L \times k} + \frac{A_o \times h_{\text{refri}}}{A_i} \right]^{-1}
\]  

where \(A_o\) and \(A_i\) represent the outer and inner surface of the tube, respectively.

\[
A_R = \frac{Q}{U \times \Delta T_{\text{LMTD}}}
\]

In order to design the **CO₂** gas cooler, a script in MATLAB R2019a [24] was developed. The simulation model calculated the overall heat transfer coefficient \(U\) of the gas cooler, based on the mass flows, inlet and outlet temperatures, and pressures of the medium. Subsequently, the required exchange area \(A_R\) was determined. Both parameters were based on the following equations:

\[
U = \left[ \frac{1}{h_{\text{air}}} + \frac{A_o \times \ln \frac{d_o}{d_i}}{2 \times \pi \times L \times k} + \frac{A_o \times h_{\text{refri}}}{A_i} \right]^{-1}
\]

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### 2.1.1. Air-Side Heat Transfer

In-line arrangement and circular finned tubes were assumed, in order to model the HEX. Based on the assumption of crossflow type, the air- and refrigerant-side heat transfer coefficients can be
calculated. The proposed correlation from VDI-Heat Atlas [25] was used in order to calculate the Nusselt number, using the following equation:

\[ Nu = C \times Re_d^{0.6} \left( \frac{A_o}{A_{to}} \right)^{-0.15} Pr^{1.7} \]  

(3)

with \( C = 0.22 \) for in-line arrangement. \( A_o/A_{to} \) is the ratio of the finned surface to the surface of the base tube, and for circular fins was calculated from the following equation:

\[ \frac{A_o}{A_{to}} = 1 + 2 \times \frac{H_f \times (H_f + d_o + t_f)}{s \times d_o} \]  

(4)

The Reynolds number was calculated by the equation:

\[ Re_d = \frac{\rho_{air} \times w_s \times d_o}{\mu} \]  

(5)

where \( w_s \) is the velocity in the smallest cross-section and was calculated from the following equation:

\[ w_s = w_{inf} \times \frac{A_{inf}}{A_s} \]  

(6)

The air-side heat transfer coefficient was calculated from its definition, as the following equation shows.

\[ h_{air} = \frac{Nu \times k_{air}}{d_o} \]  

(7)

However, the air-side heat transfer coefficient was affected by the fins. The fins should be taken into consideration, thus the following equation was used [25]:

\[ h_{air,f} = h_{air} \times \left[ 1 - \left( 1 - \eta_f \right) \times \frac{A_f}{A_o} \right] \]  

(8)

The fin efficiency is defined as the ratio of the heat removed by the fin to the heat removed by the fin at wall temperature. The efficiency of the fin was calculated from the following equation:

\[ \eta_{fin} = \frac{\tanh X}{X} \]  

(9)

with [25]:

\[ X = \varphi \times \frac{d_o}{2} \times \sqrt{2 \times \frac{h_{air}}{k \times t_f}} \]  

(10)

and

\[ \varphi = \left( \frac{d_f}{d_o} - 1 \right) \left[ 1 + 0.35 \ln \left( \frac{d_f}{d_o} \right) \right] \]  

(11)

for circular fins [25]. So, the equation of the overall heat transfer coefficient used the updated air-side heat transfer coefficient as follows:

\[ U = \left[ \frac{1}{h_{air,f}} + \frac{A_o \times \ln \frac{d_f}{d_o}}{2 \times \pi \times L \times k} + \frac{A_o \times 1}{h_{effr}} \right]^{-1} \]  

(12)
2.1.2. Refrigerant-Side Heat Transfer

On the refrigerant-side, the Gnielinski correlation [26] was used to calculate the Nusselt number:

\[ Nu = \frac{f}{12.7} \left( \frac{Re_b - 1000}{Pr} \right) \left( \sqrt{\frac{1}{Pr^\frac{2}{3} - 1}} + 1.07 \right) \]  

which is valid in the range of \( 2300 < Re_b < 10^6 \).

In addition, the friction factor \( f \) was calculated by:

\[ f = (0.79 \ln(Re_b) - 1.64)^{-2} \]  

Here, the Reynolds number was defined as

\[ Re_b = \frac{G_{refri} \times d_i}{\mu} \]  

and \( G \) was defined as the mass velocity, and was calculated from the following equation:

\[ G_{refri} = \frac{m_{refri} / N_t}{\pi \times d_i^2 / 4} \]  

Finally, heat transfer coefficient at the refrigerant side was calculated from:

\[ h_{refri} = \frac{Nu \times k_{refri}}{d_o} \]  

Further investigation of potential heat transfer correlations was conducted. More specifically, comparisons between the correlations of Gnielinski [26] and Dittus–Boelter [27], and the correlations proposed by VDI-Heat Atlas [25] and by Schmidt [28] were made for the refrigerant- and air-side, respectively. The equations below illustrate the Dittus–Boelter’s [27] and Schmidt’s [28] correlations, respectively:

\[ Nu = 0.023 \times Re^{4/5} Pr^n \]  

where \( n = 0.3 \) for the fluid being cooled.

\[ Nu = C \times Re^{0.625} Pr^{1/3} \left( \frac{A_o}{A_{to}} \right)^{-0.375} \]  

where \( C = 0.3 \) for in-line arrangement.

2.2. Experimental Set Up

As it is referred, the \( U \)-value consists of three parts: the air convection, refrigerant convection, and the conduction. In the present case, the air-side heat transfer represents the main thermal resistance. Thus, the experimental set up aimed to identify a suitable heat transfer correlation with focus on the air side. In order to validate the calculations, especially the air-side heat transfer, a well-known working medium should be used for the experiments on the refrigerant-side. Here, water with well-known thermophysical properties and reliable heat transfer correlations at single-phase regime was selected as a working medium.

The test rig consisted of the heater, the gas cooler, the measurement equipment, and controls. To enable the information to be read and recorded, the instrumentations were connected to a data logging system. The test rig is shown in Figure 2 below.
The air-cooled HEX used for the experiments was the CU-713CX2 from Panasonic. The heater was from the Single® company, model STW 150/1-18-45-KS7. The K-type thermocouples and pressure transducers used were from OMEGA company with uncertainties of ±0.2 °C and ±1.5%, respectively, while the mass flow valve and meter with uncertainties of ±0.5% were manufactured by Bürkert.

The experiments were carried out for different water mass flows, fan frequencies, and inlet water temperatures, as the following Table 2 shows. Figure 3a,b illustrate the air mass flow rate and fan power as function of the fan frequency.

### Table 2. Range and interval of the experimental variables.

| Range—Water Mass Flow Rate (L/min) | Range—Inlet Water Temperature (°C) | Range—Fan Frequency (Hz) | Range—Air Flow Velocity (m/s) |
|------------------------------------|-----------------------------------|--------------------------|--------------------------------|
| 5–9                                | 35–75                             | 50–80                    | 2–5                           |
| **Interval—Water Mass Flow Rate (L/min)** | **Interval—Inlet Water Temperature (°C)** | **Interval—Fan Frequency (Hz)** | 0.5 | 5 | 10 |

![Figure 2. Schematic of the gas cooler test rig.](image)

![Figure 3. (a) Air mass flow rate depending on fan frequency; (b) fan power as a function of fan frequency.](image)
Using Equations (20)–(24), the experimental overall heat transfer coefficient was calculated by:

\[ Q = m_w \times c_{pw} \times \Delta T_w \]  

(20)

\[ T_{air,o} = \frac{Q}{m_{air} \times c_{p_{air}}} + T_{air,i} \]  

(21)

\[ \Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\left(\frac{\Delta T_2}{\Delta T_1}\right)} \]  

(22)

\[ \Delta T_2 = T_{w,i} - T_{air,o} \text{ and } \Delta T_2 = T_{w,o} - T_{air,i} \]  

(23)

\[ U = \frac{Q}{A_R \times \Delta T_{LMTD}} \]  

(24)

2.3. Model Application

The validated model was finally applied to define an efficient and reliable gas cooler design, as well as to evaluate its performance for different off-design conditions. Within scope, the on-design analysis investigated different designs of heat exchangers, such as the number of rows and length of the tube. The target of the off-design analysis was to evaluate the operation of the gas cooler in different ambient temperatures.

2.3.1. On-Design

The developed model was utilized to design an air-finned CO₂ gas cooler. Therefore, three and four numbers of rows and finned tubes with inner diameters of 6.85, 16, and 22.5 mm were considered. In order to identify an efficient gas cooler design, a compromise between the bundle area and the refrigerant pressure drop of the gas cooler was aimed for. The model was provided with the inlet, outlet temperatures and pressures, medium mass flows, and the duty of the heat exchanger as input variables. In addition, the design properties of the tube were specified, so the total area of the tube was calculated. The air and refrigerant heat transfer coefficients, which were initially unknown, were assumed, and then the overall heat transfer coefficient was calculated from Equation (1). The required exchanged area was calculated from Equation (2), and so the required number of tubes was obtained. In the iteration process, the air and refrigerant heat transfer coefficients were updated using Equations (3)–(17). The iteration was continued until the relative tolerance of the overall heat transfer coefficient for two continuous iterations was equal to 0.001.

The pressure drop was calculated by the following equation:

\[ \Delta P = f \times \frac{C_{refri}^2}{2 \times \rho_{refri}} \times \frac{L}{d_i} \]  

(25)

\[ f = \left[1.82 \ln(Re_b - 1.64)\right]^{-2} \]  

(26)

which Filonenko [27] applies for \(10^4 \leq Re_b \leq 5 \times 10^6\).

In case of \(Re_b \leq 10^4\), Blasius [27] correlations was applied:

\[ f = \frac{0.316}{Re_b^{1/4}} \]  

(27)

The thermophysical properties of air and refrigerant like density, viscosity, specific heat capacity, and thermal conductivity were obtained from REFPROP version 10 database. The model used the mean temperature and pressure of the mediums in order to calculate the heat transfer coefficients.
2.3.2. Off-Design

The calculations were used to evaluate the overall performance of the gas cooler in different conditions. The defined gas cooler was investigated in different ambient temperatures. The boundary conditions and operational parameters for the off-design analysis were defined according to Table 3.

Table 3. Off-design boundary conditions.

| $T_{amb}$ (°C) | $T_{refri,i}$ (°C) | $T_{refri,o}$ (°C) | $P_{refri}$ (bar) | $m_{refri}$ (kg/h) | $m_{a}$ (kg/h) |
|---------------|-------------------|-------------------|------------------|-------------------|----------------|
| 20            | 78.3              | 20.02             | 93               | 184               | 93.15          |
| 21            | 79.3              | 21.03             | 93               | 184               | 92.79          |
| 22            | 80.3              | 22.03             | 93               | 184               | 92.29          |
| 23            | 81.2              | 23.04             | 93               | 184               | 91.80          |
| 24            | 82.2              | 24.05             | 93               | 184               | 91.30          |
| 25            | 83.2              | 25.07             | 93               | 184               | 90.60          |
| 26            | 84                | 26.12             | 93               | 208               | 101.67         |
| 27            | 84.7              | 27.2              | 93               | 233               | 112.59         |
| 28            | 88.6              | 28.31             | 93               | 257               | 125.66         |
| 29            | 85.9              | 29.48             | 93               | 282               | 132.37         |
| 30            | 86                | 30.71             | 93               | 306               | 141.14         |
| 31            | 86                | 32.01             | 93               | 330               | 148.68         |
| 32            | 85.8              | 33.52             | 93               | 379               | 165.14         |
| 33            | 85.5              | 35.09             | 93               | 428               | 179.17         |
| 34            | 85.3              | 36.61             | 93               | 476               | 190.51         |
| 35            | 85.1              | 38                | 93               | 525               | 199.50         |

Based on the off-design data, the overall heat transfer coefficient was calculated using Equations (3)–(17). In addition, investigation of different potential heat exchanger designs was conducted by comparing their overall heat transfer coefficient and the refrigerant pressure drop. The potential heat exchanger’s designs were based on the on-design analysis, and were chosen in terms of pressure losses and bundle area. The selected air-cooled HEXs were designed with identical finned-tube properties.

3. Results and Discussion

3.1. Validation of the Model

An experimental campaign was carried out in order to validate the mathematical calculations for the performance of an air-finned CO$_2$ gas cooler. The $U$-value from experimental results was compared with the $U$-value obtained from the model. The following results were obtained using the proposed correlation from VDI-Heat Atlas [25] and Gnielinski’s [26] correlation for the air-, and refrigerant-side, respectively. Figure 4a illustrates that most of the deviations were lower than 10%. Particularly, the average absolute deviation for 45 °C inlet water temperature and 50 Hz was 5%, while the maximum absolute deviation was 12%. The average absolute deviation for 50 Hz for different inlet water temperature was 11%. The calculations seem to approach the experimental results from 50 to 80 Hz. Figure 4b depicts that the absolute deviations for 80 Hz were even lower than 10%. Particularly, the absolute average deviations for 45 °C of 80 Hz was 5%, while the maximum absolute deviation was 9%. The absolute average deviation for 80 Hz of fan frequency for different water inlet temperatures was 6%.
The increase of the water inlet temperature affected in a similar way the overall heat transfer coefficient by 20% and 30% of 50 and 80 Hz, accordingly. The increase of the fan frequency showed an average increase of the overall heat transfer coefficient of 8%, 6%, and 6% in the ranges of 50–60, 60–70 and 70–80 Hz, accordingly. The increase of the water inlet temperature affected in a similar way the overall heat transfer coefficient of the heat exchanger Figure 5b. An average increase of 35% of the overall heat transfer coefficient from 35 to 75 °C of water inlet temperature for fan frequency of 60 Hz was revealed.

The investigation of different heat transfer correlations is considered necessary to identify the most suitable air-side heat transfer correlation. The comparison between the refrigerant-side heat transfer correlations shows that the deviations are approximately 1% for all the different fan frequencies, while the correlation by Gnielinski [26] calculates the heat transfer as lower than by that of Dittus–Boelter in Figure 6a. The comparison between the air-side heat transfer correlations shows that the deviations are highly affected by the fan frequency. Particularly, absolute average deviations for 50 and 80 Hz are 8% and 4%, accordingly, while the maximum absolute deviations are 10% and 5%. It is revealed that with the increase of the fan frequency, the deviations between the correlations become lower, as shown in Figure 6b.

Figure 6b.

Figure 4. Deviations between the model and experimental overall heat transfer coefficient for 45 °C water inlet temperature at (a) 50 Hz; and (b) 80 Hz fan frequency.

3.2. Sensitivity Analysis

The most satisfying results were obtained by using the correlations proposed by VDI-Heat Atlas [25] and by Gnielinski [26] for the air- and the refrigerant-side heat transfer, accordingly. Fundamental investigations regarding the dependence of the heat transfer characteristics (U-value) on different operational parameters and applied correlations were conducted. Particularly, the impact of the fan frequency, the water mass flow, and the water inlet temperature on the performance of the air-cooled HEX were investigated. The increase of both the water mass flow and the fan frequency enhanced the overall performance of the heat exchanger, as Figure 5a illustrates. The increase of the water mass flow rate had a strong impact on the overall heat transfer coefficient by 20% and 30% of 50 and 80 Hz, accordingly. The increase of the fan frequency showed an average increase of the overall heat transfer coefficient of 8%, 6%, and 6% in the ranges of 50–60, 60–70 and 70–80 Hz, accordingly. The increase of the water inlet temperature affected in a similar way the overall heat transfer coefficient of the heat exchanger Figure 5b. An average increase of 35% of the overall heat transfer coefficient from 35 to 75 °C of water inlet temperature for fan frequency of 60 Hz was revealed.
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The investigation of different heat transfer correlations is considered necessary to identify the most suitable air-side heat transfer correlation. The comparison between the refrigerant-side heat transfer correlations shows that the deviations are approximately 1% for all the different fan frequencies, while the correlation by Gnielinski [26] calculates the heat transfer as lower than by that of Dittus–Boelter in Figure 6a. The comparison between the air-side heat transfer correlations shows that the deviations are highly affected by the fan frequency. Particularly, absolute average deviations for 50 and 80 Hz are 8% and 4%, accordingly, while the maximum absolute deviations are 10% and 5%. It is revealed that with the increase of the fan frequency, the deviations between the correlations become lower, as shown in Figure 6b.

3.3. Design Procedure

The discussed model was applied. Thus, an on-design analysis was investigated in order to identify an efficient and reliable gas cooler in terms of pressure losses and required exchange area. The mathematical calculations were applied to on-design analysis using the correlations by Gnielinski [26] and those proposed by VDI-Heat Atlas [25] for the refrigerant- and air-side heat transfer characteristics, accordingly. The investigation of different number of rows (NR) and finned-tubes showed that the heat exchanger with four rows and the smallest diameter has a smaller bundle area (Figure 7).
Figure 7. Bundle area vs. tube length for 3 and 4 number of rows (a) outside diameter of 7.35 mm; and (b) outside diameter of 18.5 mm.

Taking the pressure drop into consideration, the results show that the tube with the outer diameter of 7.35 mm cannot be used, as the occurring pressure drop causes serious operation problems to the system, due to pressure drops exceeding 0.3 bar (Figure 8a). Instead, the heat exchanger designed with the tube with outside diameter of 18.5 mm can be used for lengths to 20 m (see Figure 8b). Figures 9 and 10 show the Reynolds number for air- and refrigerant-side, respectively.

Figure 8. Refrigerant pressure drop vs. length of the tube for 4 rows (a) outside diameter of 7.35 mm; and (b) outside diameter of 18.5 mm.
Based on the presented results, the final design of the air-finned CO₂ gas cooler is defined for a finned-tube (type-U), according to the specifications listed in Table 4.

**Table 4.** Gas cooler technical specifications.

| Gas Cooler Specifications          |       |       |
|-----------------------------------|-------|-------|
| Total length of the tube (m)      | 17    |       |
| Number of passes per tube (-)     | 14    |       |
| Number of passes (-)              | 43    |       |
| Length of each pass (m)           | 1.2   |       |
| Number of rows (-)                | 4     |       |
| Number of tubes per row (-)       | 3     |       |
| Bundle area (m²)                  | 2.11  |       |
| Refrigerant pressure drop (bar)   | 0.23  |       |

| Air Finned Tube Specifications    |       |       |
|-----------------------------------|-------|-------|
| Outside diameter (mm)             | 18.5  |       |
| Wall thickness (mm)               | 1.25  |       |
| Fin height (mm)                   | 10    |       |

### 3.4. Evaluation at Off-Design Conditions

The off-design analysis of the defined gas cooler was carried out for different ambient temperatures from 20 to 35 °C, and a comparison between potential gas coolers was made. The potential heat exchangers were chosen in terms of pressure losses and bundle area, while they had identical finned tube properties, except for the length of the tube. Particularly, the gas cooler with bundle area of 2.39 m² was characterized by its high pressure losses of 0.3 bar, while the gas cooler with bundle area of 2.81 m² showed relatively low pressure losses of 0.06 bar. The gas cooler with bundle area of 2.43 m²
was chosen, due to its combination of comparatively low pressure losses of 0.14 bar, as well as its low bundle area.

| Gas Cooler Specifications |
|---------------------------|
| Total length of the tube (m) | 17 |
| Number of passes per tube (-) | 14 |
| Number of passes (-) | 43 |
| Length of each pass (m) | 1.2 |
| Number of rows (-) | 4 |
| Number of tubes per row (-) | 3 |
| Bundle area (m$^2$) | 2.11 |
| Refrigerant pressure drop (bar) | 0.23 |

| Air Finned Tube Specifications |
|-------------------------------|
| Outside diameter (mm) | 18.5 |
| Wall thickness (mm) | 1.25 |
| Fin height (mm) | 10 |

Figure 11a below shows that the defined gas cooler operates better than the other air-cooled HEX in different ambient temperatures. The defined gas cooler’s overall heat transfer coefficient was remarkably higher than the heat exchangers with the bundle area of 2.43 and 2.81 m$^2$. On the other hand, the $U$-value of the heat exchanger of 2.39 m$^2$ bundle area is near to the defined heat exchanger, ($A_B = 2.11$ m$^2$) but the refrigerant pressure drop is significantly higher (Figure 11b).

Figure 11. Comparison between potential heat exchangers in different ambient temperatures of (a) the overall heat transfer coefficient; and (b) the refrigerant pressure drop.

4. Conclusions

A design procedure for a CO$_2$ finned-tube gas cooler was developed and validated experimentally. The experimental focus was laid on the validation of the air-side heat transfer. Absolute deviations between the model and the experiments were extracted and prove the reliability of the selected heat transfer correlations. The developed simulation model was used to design an efficient gas cooler and
evaluate its performance in different ambient temperatures. Based on these results, the following conclusions are justified:

The deviations between the calculations and the experiments are highly affected by the fan frequency, the water mass flow, and the water inlet temperature. It was found that the absolute average deviations for 60–80 Hz are less than 10%. The increase in the fan frequency, the water mass flow, and the water inlet temperature caused an improvement of the overall heat transfer coefficient of the heat exchanger. The comparison between potential heat transfer correlations showed that the combination of the correlations proposed by VDI-Heat Atlas [25] and by Gnielinski [26] for the air- and refrigerant-side heat transfer coefficient, respectively, approached the experimental results better.

The heat exchanger with four rows has a smaller bundle area than with three rows, while the investigation of different tubes showed that the heat exchanger designed with the smallest diameter has the smallest bundle area and the highest refrigerant pressure drop. As a compromise between the bundle area and the refrigerant pressure drop, a gas cooler of 2.11 m$^2$ and refrigerant pressure drop of 0.23 bar was defined. Comparison between potential gas coolers was made, and the results show that the defined gas cooler operates more efficiently for different ambient conditions, compared to other potential heat exchangers.

Due to the pressure limitations of the equipment, the experimental validation of the model was conducted with water as working fluid. This procedure enables a reliable validation of the applied heat transfer correlation at the air side. However, the use of CO$_2$ as working fluid in the tubes would improve validation approach. Therefore, experiments with CO$_2$ are suggested for further work, next to the off-design analysis for the entire air conditioning system including the developed gas cooler model.

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**Nomenclature**

| Symbol | Description | Unit |
|--------|-------------|------|
| A      | Area        | m$^2$ |
| $\bar{c}_p$ | Mean specific heat capacity | J kg$^{-1}$ K$^{-1}$ |
| d      | Diameter    | m    |
| f      | Friction factor | - |
| G      | Mass velocity | kg m$^2$ s$^{-1}$ |
| h      | Heat transfer coefficient | W m$^{-2}$ K$^{-1}$ |
| H      | Height      | m    |
| k      | Thermal conductivity | W m$^{-1}$ K$^{-1}$ |
| L      | Length      | m    |
| m      | Mass flow rate | kg s$^{-1}$ |
| Nt     | Number of tubes | - |
| Nu     | Nusselt number | - |
| P      | Pressure    | bar  |
| Pr     | Prandtl number | - |
| Q      | Heat transfer rate | W |
| Re     | Reynolds number | - |
Abbreviations

CFCs Chlorofluorocarbons
CO₂ Carbon dioxide
COP Coefficient of performance
GWP Global warming potential
HCFCs Hydrochlorofluorocarbons
HEX Heat exchanger
HFCs Hydrofluorocarbons
HVAC Heating, ventilation and air conditioning
LMTD Logarithmic mean temperature difference
NR Number of rows
ODP Ozone depletion potential

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