



Evaluation of surface-area requirements for air-cooled remote condensers

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ABSTRACT

Accurate condenser surface-area determination is a critical aspect of HVAC design. This is particularly challenging for air-cooled remote condensers operating with zeotropic refrigerants, where temperature glide and subcooling significantly influence the heat transfer. Existing methods offer limited transparency in engineering application or rely on oversimplified LMTD-based formulations, which represent these effects with insufficient physical rigour. This study addresses both shortcomings through a closed-form analytical framework using the LMTD method subphase-wise to dry air-cooled remote condensers. A geometry-independent temperature-potential utilisation factor (η_{cond}) is introduced, enabling explicit calculation of the required surface-area, the corrected logarithmic mean temperature difference, and the theoretical minimum surface. Notably, when the air-side capacity rate is the smaller of the two, η_{cond} is identical to the heat-exchanger effectiveness ϵ , enabling a single-step ϵ -NTU analysis of the entire condenser. Sensitivity analysis in a case study reveals that the subcooling region exhibits a fundamentally distinct parameter hierarchy, with the subcooling temperature dominant rather than the general inlet air temperature. Combined variation of the four theoretically adjustable parameters achieves surface-area reduction of up to 47.4%, whereas the most accessible engineering combination results in a 35.7% reduction. These findings necessitate the accurate LMTD formulation, as conventional LMTD simplifications introduce systematic errors of 56–59% overprediction and 48–57% underprediction. Validation against manufacturer data and literature sources confirms deviations below 1%. The presented transparent analytical framework supports engineering decisions for dry air-cooled condensers, including pre-design assessment, manufacturer selection verification, and fan-control strategy development.

1. Introduction

1.1. Literature review

Approximately 70–90% of building energy consumption corresponds to heating and cooling demand [1]. According to the International Institute of Refrigeration, refrigeration systems account for around 15% of global electricity consumption and contribute nearly 10% to greenhouse gas emissions [2]. Reducing these proportions requires, accurate determination of energy demand [3], based on proper load assessment [4] and consistent treatment of design parameters and their effects [5, 6]. The selection of appropriate energy generation systems is essential to satisfy the required demand [7,8]. Heat pumps substantially reduce CO₂ emissions [9], yet refrigerant selection becomes a central design consideration. Since a universally applicable refrigerant remains unavailable [10], research focuses on environmentally benign refrigerants (such as natural refrigerants and low-GWP zeotropic mixtures) [11].

Environmentally benign natural refrigerants (such as R744, R290, and R717) present specific safety risks during application [11], which motivates minimising the refrigerant charge [12]. For low-GWP zeotropic mixtures, the temperature glide during phase change presents a distinct engineering challenge that is often neglected in design calculations and receives only limited attention in the literature.

Beyond refrigerant selection, examination of individual system components represents another opportunity for system improvement. The condenser is a critical component of the refrigeration cycle, as it is responsible for heat removal from the refrigerant [13,14]. Heat rejection in condensers is typically achieved by either water cooling or air cooling. Water-cooled condensers (WCC) are implemented at higher capacity levels, where their higher heat-transfer intensity compensates for their complexity (including more potential failure points) [15]. However, such systems require considerable water consumption and impose significant thermal load on natural and artificial freshwater sources [16]. Due to these drawbacks, dry and wet (evaporative) air-cooled condensers (ACC)

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are increasingly applied, even at higher capacities [15].

ACCs reject waste heat into the atmosphere, reducing specific thermal pollution [16] and offering lower maintenance costs due to simpler design and the absence of water-treatment chemicals [16]. Nevertheless, ACCs require a large heat-transfer surface and significant fan-power input, and their thermal effectiveness is sensitive to wind effects [17, 18]. External air conditions (including dry-bulb and wet-bulb temperatures, dew-point conditions, and humidity) influence the thermal effectiveness of refrigeration systems [19,20]. High summer air temperatures reduce condenser heat rejection [21]. Yau et al. [22] report that a 1°C increase in outdoor dry-bulb temperature results in an approximately 2% reduction in COP. Practical experience indicates that above 35°C, the COP_R decreases from the 2.7–3.2 range to the 2.2–2.4 range [23]. At high external air temperature, various technical solutions are applied to maintain the required heat rejection.

The most common solution is raising the condensation temperature, which requires compressing the refrigerant to a higher-pressure level and thereby increases compressor power input (and thus the building's electrical energy demand) [21]. However, at external air temperatures above approximately 52 °C, the condensation pressure reaches a level at which system shutdown becomes necessary [23].

Another commonly applied approach involves increasing the overall heat-transfer coefficient (U) by utilising the latent heat of water (i.e., by using an evaporative ACC instead of a dry ACC). Fine water spraying enhances heat-transfer intensity at the condenser [24,25]. Wang et al. [26] experimentally analysed an air-conditioning system using an evaporative condenser. The COP increased by 6.1–18%, while compressor power input decreased by 14.3%. Hajidavalloo et al. [27] demonstrated that performance improvement increases with rising external air temperature. In the examined hot-climate conditions, energy consumption is reduced by up to 20%, while COP improves by approximately 50%. Nevertheless, improving heat-transfer intensity entails additional costs, as the more complex system configuration increases maintenance and repair requirements (including scaling and elevated legionella risk) [11]. Kostyák et al. [28,29] further highlighted that the effective applicability of evaporative systems in temperate climate zones is highly sensitive to external air temperature and humidity.

For these reasons, dry ACCs are researched and applied in residential [30] and industrial systems [31,32], and remain predominant in medium-capacity units. Nevertheless, researchers' efforts tend to investigate their partial displacement [33] or complete replacement by evaporative condensers [34]. However, in dry ACC development, further opportunities are identifiable. Chan et al. [35] establish a connection between the required condensation temperature and the number of intermittently operating condenser fans. Yu et al. [36] analyse variable-speed fan control in combination with condensation-temperature optimisation, and present COP improvements between 5.6% and 113.4%. Catrini et al. [37] demonstrate a near-linear increase in system energy efficiency with rising fan speed. Refrigerant subcooling provides significant effectiveness enhancement in dry ACCs [38]; however, this improvement exhibits pronounced refrigerant dependence, as shown by Pottker et al. [39] (18% COP increase for R1234yf and 9% for R134a).

An additional possibility for increasing transferred heat flow is the precise sizing of the heat-transfer surface. Its application requires careful consideration of the temperature conditions. This becomes especially important for zeotropic mixtures due to temperature glide and subcooling.

1.2. Approach, research gap, and scope

Condenser selection is a major consideration in cooling system design, since it directly affects energy efficiency and both investment and operational costs. A remote condenser configuration enables the spatial separation between the indoor unit and the condenser, providing flexible installation and effective heat rejection conditions. If the heat-rejection demands involve a relatively small heat-transfer surface-

area, a dry air-cooled remote condenser provides a simple and cost-effective option. Such cost-effective configurations become increasingly important as global cooling demand increases [40] and the challenge of cooling poverty intensifies [41].

ACC sizing methodologies vary considerably in accuracy, computational demand, and applicability. Methodology choice is critical where rapid and reliable decisions are required, namely in design workflows. These decisions arise during equipment selection and operational parameter setting, involving MSc-level engineers and largely based on limited information. Furthermore, scepticism is observed among these decision-makers regarding methodologies perceived as 'black-box' solutions, since the cumbersome verification of these calculations introduces a personal professional risk.

Considering these demands, several established scientific methods prove impractical. These are typically numerical simulation approaches (such as CFD- and TRNSYS-based methods [42–44]), which provide high computational accuracy and detailed distribution characteristics; however, they require substantial resources and extended calculation periods [45]. In contrast, methods based on limited input data require a precise thermodynamic and analytical model. Two such fundamental models are relevant at the engineering level: the LMTD and the ϵ -NTU descriptions, which are theoretically equivalent [46,47]. The sub-phase-wise application of the LMTD model provides a transparent treatment of temperature glide and subcooling. The required heat-transfer surface-area is determined implicitly by the ϵ -NTU formulation and explicitly by LMTD. Therefore, the present analysis adopts an LMTD-based method.

A review of previous research and analytical methodologies reveals four main research gaps.

1. At the level of rapid and reliable engineering decision-making [48], the effectiveness of black-box approaches remains limited [49] and results in excessive simplifications (which obscure critical effects).
2. The treatment of temperature glide in zeotropic refrigerants remains imprecise in both research and engineering practice [30].
3. The influence of subcooling receives only superficial and non-explicit analytical attention.
4. In the medium-capacity unit (5–50 kW), research remains limited in precisely determining the heat-transfer surface-area. Nevertheless, the decision in HVAC design practice between WCC and ACC is fundamentally determined by this parameter.

In response to these research gaps, this study develops a closed-form analytical framework for sizing dry air-cooled remote condensers. Departing from numerical simulation or black-box procedures, the formulation applies the LMTD method to each sub-process explicitly, enabling transparent treatment of temperature glide and subcooling. A key element of the model is the introduction of a geometry-independent condenser temperature-potential utilisation factor, defined to quantify the effective use of the available thermal driving force. This parameter enables explicit expressions for the corrected logarithmic mean temperature difference and for the required heat-transfer surface-area, and it analytically demonstrates the existence of a theoretical minimum surface. The analysis focuses on medium-capacity units (5–50 kW), where rapid and reliable engineering decisions are required, and where surface demand critically influences the choice between air- and water-cooled solutions. One motivation of this study is to support dry ACC selection in contexts where engineering convention already prefers wet cooling solutions; the presented framework therefore concentrates on specifically on the dry air-cooled configuration.

2. Theoretical background

In remote condensers (and in ACCs), the refrigerant rejects heat to the external air at approximately constant condensation pressure (p_c). The thermodynamic process comprises three subphases: desuperheating,

condensation, and subcooling, the last of which is absent in certain configurations.

For most refrigerants, condensation subphase is isobaric and isothermal; however, zeotropic mixtures (R4xx) remain isobaric only. These mixtures exhibit a temperature glide (ΔT_{glide} , at the condensation), defined as the temperature difference between the dew-point (T_{dew}) and the bubble-point (T_{bub}).

As heat is rejected, the cooling external air temperature increases from its inlet value ($T_{a,1}$) to intermediate levels ($T_{a,sc}$ and $T_{a,sh}$), and finally reaches its outlet value ($T_{a,2}$).

Fig. 1 depicts the heat-transfer process in four complementary representations [46].

Based on Fig. 1, heat-transfer equations are described for the overall process (Equation 1) as well as for each subphase (desuperheating: Equation 2, condensation: Equation 3, and subcooling: Equation 4). For the subphases, these expressions present, in sequence, the air-side, the refrigerant-side, and the conventional heat-transfer forms.

$$\dot{Q}_C = \dot{Q}_{sh} + \dot{Q}_{cond} + \dot{Q}_{sc} = c_{air} \cdot \dot{m}_{air} \cdot (T_{a,2} - T_{a,1}) = \dot{m}_{ref} \cdot (h_{sh} - h_{sc}) \quad (1)$$

$$\begin{aligned} \dot{Q}_{sh} &= c_{air} \cdot \dot{m}_{air} \cdot (T_{a,2} - T_{a,sh}) = \dot{m}_{ref} \cdot (h_{sh} - h_{dew}) \\ &= U \cdot A_{sh} \cdot \frac{T_{a,2} - T_{a,sh} - T_{sh} + T_{dew}}{\ln \frac{T_{dew} - T_{a,sh}}{T_{sh} - T_{a,2}}} \end{aligned} \quad (2)$$

$$\begin{aligned} \dot{Q}_{cond} &= c_{air} \cdot \dot{m}_{air} \cdot (T_{a,sh} - T_{a,sc}) = \dot{m}_{ref} \cdot (h_{dew} - h_{bub}) \\ &= U \cdot A_{cond} \cdot \frac{T_{a,sh} - T_{a,sc} - T_{dew} + T_{bub}}{\ln \frac{T_{bub} - T_{a,sc}}{T_{dew} - T_{a,sh}}} \end{aligned} \quad (3)$$

$$\begin{aligned} \dot{Q}_{sc} &= c_{air} \cdot \dot{m}_{air} \cdot (T_{a,sc} - T_{a,1}) = \dot{m}_{ref} \cdot (h_{bub} - h_{sc}) \\ &= U \cdot A_{sc} \cdot \frac{T_{a,sc} - T_{a,1} - T_{bub} + T_{sc}}{\ln \frac{T_{sc} - T_{a,1}}{T_{bub} - T_{a,sc}}} \end{aligned} \quad (4)$$

Where the ' $\dot{Q}_C$ ' (W) denotes the total heat released in the condenser, while ' $\dot{Q}_{sh}$ ', ' $\dot{Q}_{cond}$ ' and ' $\dot{Q}_{sc}$ ' (W) represent the heat released during desuperheating, condensation and subcooling, respectively. The mass flow rates of air and refrigerant are given by ' $\dot{m}_{air}$ ' and ' $\dot{m}_{ref}$ ' ($\text{kg} \cdot \text{s}^{-1}$). The inlet and outlet air temperatures are ' $T_{a,1}$ ' and ' $T_{a,2}$ ' (K), whereas ' $T_{a,sh}$ ' and ' $T_{a,sc}$ ' (K) denote the intermediate air temperatures after desuperheating and condensation. The ' c_{air} ' ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$) is the specific heat capacity of air. On the refrigerant side, T_i (K) denotes temperature and h_i ($\text{J} \cdot \text{kg}^{-1}$) denotes specific enthalpy at state ' i '. Thus ' T_{sh} ' and ' h_{sh} ' describe the superheated vapor entering the condenser; ' T_{dew} ', ' h_{dew} ' and ' T_{bub} ', ' h_{bub} ' correspond to the dew- and bubble-point states; while ' T_{sc} ', ' h_{sc} ' represent the subcooled liquid leaving the condenser. The overall heat-transfer coefficient is denoted by ' U ' ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$), and ' A_{sh} ', ' A_{cond} ' and ' A_{sc} ' (m^2) denote the heat-transfer surface-areas required for each subprocess (desuperheating, condensation and subcooling).

Notably, the conventional heat-transfer form ($\dot{Q} = U \cdot A \cdot \text{LMTD}$) is admissible for each subphase individually, whereas the overall process resists this formulation and can only be described by the air- and refrigerant-side forms. However, by applying the air- and refrigerant-side forms of Eqs. (1)–(4), the air-side temperatures can be determined as a function of the inlet air temperature ($T_{a,1}$).

$$T_{a,2} = \frac{\dot{m}_{ref} \cdot (h_{sh} - h_{sc})}{c_{air} \cdot \dot{m}_{air}} + T_{a,1} \quad (5)$$

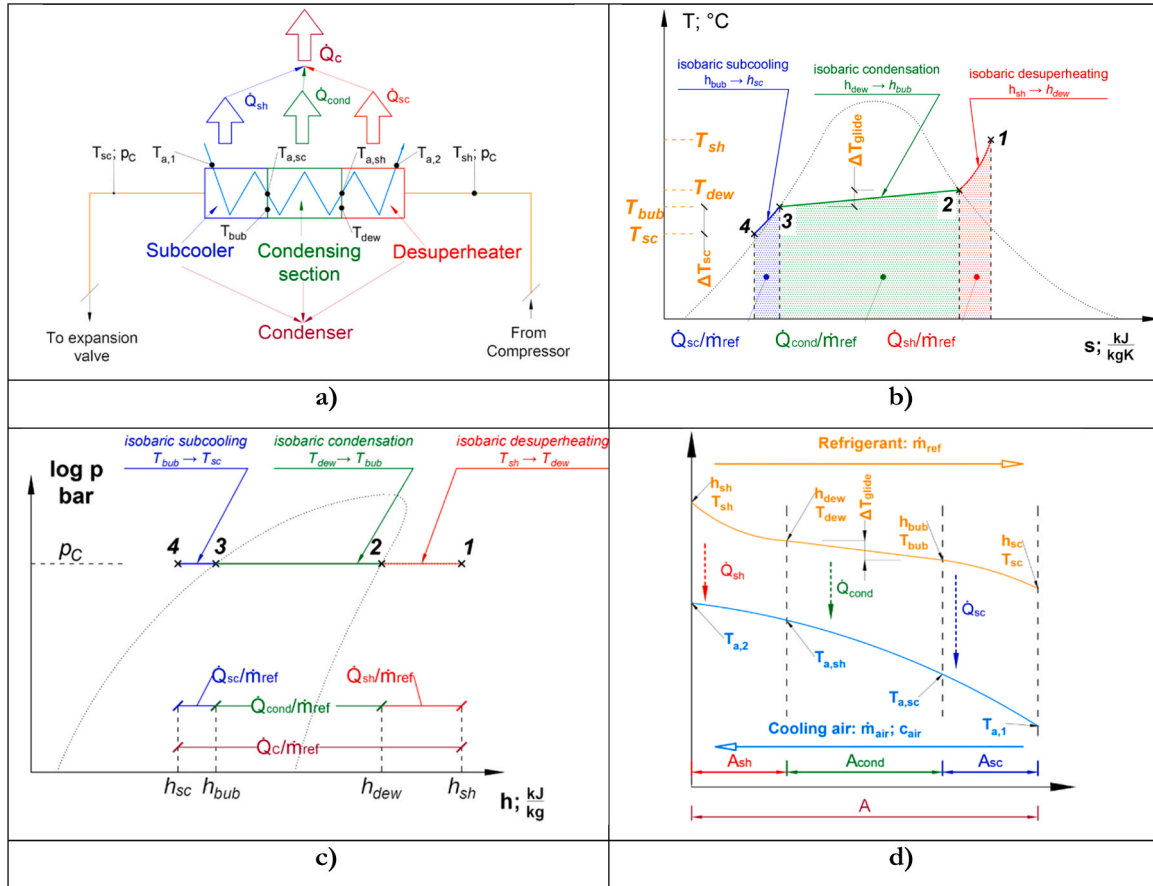


Fig. 1. Representation of the heat-transfer process in the condenser (distinguishing desuperheating and subcooling) for zeotropic refrigerants: a) Schematic indicating characteristic temperatures and heat flows; b) temperature–entropy (T – s) diagram, c) log pressure–enthalpy (log p – h) diagram, d) surface–temperature profile.

$$T_{a,sh} = \frac{\dot{m}_{ref} \bullet (h_{dew} - h_{sc})}{c_{air} \bullet \dot{m}_{air}} + T_{a,1} \quad (6)$$

$$T_{a,sc} = \frac{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air}} + T_{a,1} \quad (7)$$

The analytical model relies on a set of simplifying assumptions.

- The system operates under steady-state conditions, with transient and control dynamics excluded.
- Determination of the overall heat-transfer coefficient (U-value) requires the combined consideration of conductive and convective heat-transfer mechanisms. Incorporating temperature- and mass-flow-dependent convective coefficients introduces over-complication, which reduces model clarity. Therefore, this model follows industrial practice and treats U-value as a nominal parameter specified by the manufacturer. These U-values arise either from their Reynolds-number-based calculations or from their measurements at defined operating conditions.
- The nominal value of U remains approximately constant for minor temperature changes [46] and for restricted mass flow variations (within the turbulent regime) [50–52]; in contrast, evaporative cooling produces marked modifications (because of latent-heat effects) [24]. In this article, the U-value serves as a scaling parameter that proportionally adjusts the calculated surface-area without altering the relationships among the model variables.
- Condensation pressure (p_c) remains constant (refrigerant-side pressure losses neglected) and independent of the refrigerant mass flow rate ($\dot{m}_{ref}$).
- Temperature variation across the flow cross-section is negligible.
- The inlet external air temperature ($T_{a,1}$) remains constant in time.
- The absolute humidity of the cooling air stays unchanged over the entire process from $T_{a,1}$ to $T_{a,2}$. Therefore, Equations 1–4 already apply in their present form to dry ACCs, whereas evaporative ACCs still require refinement (e.g., accounting for evaporation and mass increase).

3. Results

3.1. Analytical derivation of an explicit condenser surface-area

One objective is the explicit determination of the condenser surface-area using the LMTD method applied to each subphase. To achieve this, the first step is to write the surface-area requirement for each subphase. For this, both the subprocess heat flows (the refrigerant-side forms of Equations 1–4) and the air-side temperatures (Equations 5–7) are substituted into Equations 2–4 (the conventional heat-transfer forms).

$$A_{sh} = \frac{c_{air} \bullet \dot{m}_{air}}{U} \bullet \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})}} \bullet \ln \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{a,1}) - \dot{Q}_C} \quad (8)$$

$$A_{cond} = \frac{c_{air} \bullet \dot{m}_{air}}{U} \bullet \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{bub})}{\dot{m}_{ref} \bullet (h_{dew} - h_{bub})}} \bullet \ln \left[\frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{bub} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})} \right] \quad (9)$$

$$A_{sc} = \frac{c_{air} \bullet \dot{m}_{air}}{U} \bullet \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{bub} - T_{sc})}{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}} \bullet \ln \frac{T_{sc} - T_{a,1}}{T_{bub} - T_{a,1} - \frac{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air}}} \quad (10)$$

By summing Equations 8–10 and rearranging the logarithmic expressions, the total condenser surface-area is obtained in Equation 11:

$$A = \frac{c_{air} \bullet \dot{m}_{air}}{U} \bullet \ln \left\{ \left[\frac{1}{\frac{T_{bub} - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sc} - T_{a,1})}} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{bub})}{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}} \right. \\ \bullet \left[\frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{bub} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{bub})}{\dot{m}_{ref} \bullet (h_{dew} - h_{bub})}} \\ \left. \bullet \left[\frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{a,1}) - \dot{Q}_C} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})}} \right\} \quad (11)$$

This expression is an explicit form and incorporates both temperature glide and subcooling. However, obtaining the form $\dot{Q} = U \cdot A \cdot \text{LMTD}$ still requires introducing an additional factor.

3.2. Condenser temperature-potential utilisation factor: definition and application to LMTD determination

As a first step, $\dot{Q}_C$ is obtained from Equation 11:

$$\dot{Q}_C = c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{a,1}) - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})}{\left[\exp \frac{A \bullet U}{c_{air} \bullet \dot{m}_{air}} \right] \left[1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})} \right]} \\ \bullet \left[\frac{1}{\frac{T_{bub} - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sc} - T_{a,1})}} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})}} \\ \bullet \left[\frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{bub} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{bub})}{\dot{m}_{ref} \bullet (h_{dew} - h_{bub})}} \quad (12)$$

The expression obtained from Equation 11 is a difference, which can be written as $\dot{Q}_C = c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{a,1}) \cdot (1-x)$. The $T_{sh} - T_{a,1}$ term represents the maximum temperature potential for heat-transfer, while $(1-x)$ acts as an efficiency-like factor. This efficiency provides the new factor required for the LMTD-based form:

$$\eta_{cond} = 1 - \left[\frac{1}{\frac{T_{bub} - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{\dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sc} - T_{a,1})}} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})}} \\ \bullet \frac{T_{dew} - T_{a,1} - \frac{\dot{m}_{ref} \bullet (h_{dew} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{a,1})}}{\left[\exp \frac{A \bullet U}{c_{air} \bullet \dot{m}_{air}} \right] \left[1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{sh} - T_{dew})}{\dot{m}_{ref} \bullet (h_{sh} - h_{dew})} \right]} \\ \bullet \left[\frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{bub} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{bub} - h_{sc})}{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{a,1}) - \dot{m}_{ref} \bullet (h_{dew} - h_{sc})} \right] \frac{1}{1 - \frac{c_{air} \bullet \dot{m}_{air} \bullet (T_{dew} - T_{bub})}{\dot{m}_{ref} \bullet (h_{dew} - h_{bub})}} \quad (13)$$

η_{cond} represents the temperature-potential utilisation factor at the condenser. This efficiency is unconventional because smaller values

indicate a lower thermal utilisation level and a smaller required surface-area. With this efficiency, Q_C is expressed as:

$$\dot{Q}_C = \eta_{cond} \cdot c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_{a,1}) \quad (14)$$

This form closely resembles the expression used in the ε -NTU method ($Q_{actual} = \varepsilon \cdot (c \cdot m)_{min} \cdot (t_{hot,inlet} - t_{cold,inlet})$). However, one important difference exists. The factor η_{cond} always requires $(c \cdot m)_{air}$, whereas ε (heat-exchanger effectiveness) is defined with $(c \cdot m)_{min}$. When $(c \cdot m)_{air} = (c \cdot m)_{min}$ (this condition occurs in almost all practical cases) η_{cond} becomes equal to ε . In this case, ε can be determined for heat exchangers with multiple sub-processes.

The total surface-area of a remote condenser can be determined using η_{cond} (substituting $c \cdot m_{air}$ from Equation 14 and $\dot{m}_{ref}$ from Eq. 1 into Equation 11, then rearranging the expression):

$$A = \frac{\dot{Q}_C}{U \cdot (T_{sh} - T_{a,1})} \cdot \ln \left[\left(\frac{T_{bub} - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{T_{sh} - T_{a,1}}{T_{sc} - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}} \right)^{\frac{1}{\frac{T_{bub} - T_{sc}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{bub} - h_{sc}} - \eta_{cond}}} \cdot \left(\frac{\frac{T_{dew} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}}{\frac{T_{bub} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_{dew} - T_{bub}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{dew} - h_{bub}} - \eta_{cond}}} \cdot \left(\frac{1 - \eta_{cond}}{\frac{T_{dew} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_{sh} - T_{dew}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{sh} - h_{dew}} - \eta_{cond}}} \right] \quad (15)$$

From this expression, the structure $Q_C = U \cdot A \cdot LMTD_C$ becomes apparent, where $LMTD_C$ is defined as:

$$LMTD_C = (T_{sh} - T_{a,1}) \cdot \ln \left[\left(\frac{T_{bub} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}} \right)^{\frac{1}{\eta_{cond} \cdot \frac{T_{dew} - T_{bub}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{dew} - h_{bub}}}} \cdot \left(\frac{\frac{T_{dew} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}}{\frac{T_{bub} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\eta_{cond} \cdot \frac{T_{sh} - T_{dew}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{sh} - h_{dew}}}} \cdot \left(\frac{1 - \eta_{cond}}{\frac{T_{dew} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{-1} \right] \quad (16)$$

The derived equations describe the most complex case, characterised by refrigerant temperature glide and subcooling. From an engineering perspective, three additional configurations are relevant: (1) glide without subcooling, (2) subcooling without glide, and (3) neither glide nor subcooling. In these special cases, if either $\Delta T_{sc} = T_{bub} - T_{sc}$ or $\Delta T_{glide} = T_{dew} - T_{bub}$ (in the denominators of Equations 13, 15, and 16) becomes

zero, the numerical result becomes singular. Resolving these zero-value problems requires either augmenting Equations 13, 15, and 16 with conditional constraints or deriving separate closed-form solutions for each case. To maintain engineering transparency, the latter approach is adopted. The related equations are listed in the Appendix.

3.3. Condenser surface-area limits

Continuing the examination of η_{cond} applicability, Equations 12 and 13 indicate its permissible range to lie between 0 and 1. Adapting this observation to the surface-area expression (in Equation 15) extends the applicability of η_{cond} , as it determines the two limiting values of the condenser surface (corresponding to $\eta_{cond} = 0$ and $\eta_{cond} = 1$). At this lower limit, $A_{min} = \lim_{\eta_{cond} \rightarrow 0} A$, the temperature potential remains unutilised, and the resulting surface represents the lower bound of the condenser area, constituting a theoretical minimum:

$$A_{min} = \ln \left[\frac{T_{sh} - T_{a,1}}{T_{dew} - T_{a,1}} \cdot \left(\frac{T_{bub} - T_{a,1}}{T_{sc} - T_{a,1}} \right)^{\frac{T_{sh} - T_{dew}}{T_{bub} - T_{sc}} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{dew}}} \cdot \left(\frac{T_{dew} - T_{a,1}}{T_{bub} - T_{a,1}} \right)^{\frac{T_{sh} - T_{dew}}{T_{dew} - T_{bub}} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{dew}}} \right] \cdot \frac{\dot{m}_{ref}}{U} \cdot \frac{h_{sh} - h_{dew}}{T_{sh} - T_{dew}} \quad (17a)$$

$$A_{min} = \frac{\dot{m}_{ref}}{U} \cdot \frac{h_{sh} - h_{dew}}{T_{sh} - T_{dew}} \cdot \ln \left[\frac{T_{sh} - T_{a,1}}{T_{dew} - T_{a,1}} \cdot \left(\frac{T_{dew} - T_{a,1}}{T_{bub} - T_{a,1}} \right)^{\frac{T_{sh} - T_{dew}}{T_{dew} - T_{bub}} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{dew}}} \right] \quad (17b)$$

$$A_{min} = \frac{\dot{m}_{ref}}{U} \cdot \frac{h_{sh} - h_{dew}}{T_{sh} - T_{dew}} \cdot \ln \left[\frac{T_{sh} - T_{a,1}}{T_C - T_{a,1}} \cdot \left(\frac{T_C - T_{a,1}}{T_{sc} - T_{a,1}} \right)^{\frac{T_{sh} - T_C}{T_C - T_{sc}} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{dew}}} \right] \quad (17c)$$

$$A_{min} = \frac{\dot{m}_{ref}}{U} \cdot \frac{h_{sh} - h_{dew}}{T_{sh} - T_{dew}} \cdot \ln \left(\frac{T_{sh} - T_{a,1}}{T_C - T_{a,1}} \right) \quad (17d)$$

A transparent engineering description of this lower bound requires four separate expressions, depending on the presence of glide and subcooling (Equation 17a: glide with subcooling; Equation 17b: glide without subcooling; Equation 17c: subcooling without glide; Equation 17d: neither glide nor subcooling). In contrast, the upper limit reduces

to a single value: infinity, which is consistent with engineering experience. When $\eta_{\text{cond}} = 1$ (i.e., when the entire temperature potential is utilised), the required surface tends to infinity in all four configurations ($A_{\text{max}} = \lim_{\eta_{\text{cond}} \rightarrow 1} A = \infty$).

4. Validation

Before examining the practical relevance of the analytical framework, the model requires validation. The validation relies on three sources of operating data: the installed system, manufacturer-provided configurations, and literature-based results. Both the installed system and the manufacturer-provided configurations enable direct determination of U . In contrast, literature-based validation is more challenging because most publications ignore parameters required by the present model and apply simplified LMTD_C expressions. (Although a simplified LMTD_C is used and $U \cdot A$ is reported instead of A , correction of LMTD_C is feasible as defined by Equation 16.)

The installed configuration, which serves as the reference system in the subsequent investigations, operates at the ITSH facility in Debrecen. The investigated remote condenser is a CIAT EUROPA 2 C9103 HI 450 unit serving the heat-rejection demand of a CIAT DYNACIAT LGN 1200Z indoor liquid chiller (nominal cooling capacity: 307.8 kW). A representative operating condition is required for the analysis, one that remains close to the manufacturer's design specification while incorporating the effects of the as-built installation. Accordingly, the applied technical parameters represent a corrected design state based on the manufacturer's design data but refined through on-site measurements. These parameters are listed in Table 1.

The reference system has operated reliably since 2011. A review of replacement configurations is now appropriate, as the system is in its fifteenth year of service and employs a refrigerant with high global warming potential ($\text{GWP} = 1774$). For this reason, four manufacturer designs were requested for potential replacement, of which two suppliers provided sufficiently detailed data to apply the analytical model. These two suppliers provide three usable configurations for validation (and potential replacement). Manufacturer validation therefore utilizes these three configurations, each demonstrating suitability for the analytical framework.

Literature-based validation is generally limited by simplified LMTD expressions and incomplete operating data; however, two publications (Ref. [30] and Ref. [53]) provide sufficiently detailed information. Although both sources apply the same simplified LMTD expression, their presented operating data permit correction. Cuevas et al. [30] provide detailed operating conditions for twelve test cases, from which two representative operating states are selected: Test 1 (low-fan-speed range, medium condensation temperature, moderate subcooling) and Test 10 (high-fan-speed range, elevated condensation temperature, increased

subcooling). From the study by Morozyuk et al. [53], the "clean surface" operating condition is used.

Table 2 presents the manufacturer-based and literature-based data for validation.

The validation first uses the reference-system parameters (Table 1) and the manufacturer data (Table 2). Table 3 summarises the resulting deviation and the associated error analysis, including the absolute and relative differences as well as Mean Absolute Error (MAE), Root Mean Square Error (RMSE), Mean Absolute Percentage Error (MAPE), and the maximum absolute error.

Validation of the proposed analytical framework with the manufacturer data exhibits close agreement (about 1% deviation), reflecting either production-related rounding or the intrinsic uncertainty of the U -value. The error analysis confirms this precision, as the model achieves $\text{MAE} = 4.68 \text{ m}^2$, $\text{RMSE} = 5.28 \text{ m}^2$, $\text{MAPE} = 0.501\%$, and a maximum deviation of 0.859% in this validation. Furthermore, the values of η_{cond} (about 25% in all four cases) and the A_{min}/A ratio also remain close to those of the reference system. Although these observations are notable, the limited dataset permits only descriptive interpretation.

The literature dataset is also employed for validation. The independent U value is absent in Ref. [30] and Ref. [53]; therefore, the literature-based validation is restricted to the $U \cdot A$ term. Table 4 presents the deviation of the results determined by the analytical model, first relative to the published data and then to the values recalculated with the LMTD_C corrected by Equation 16.

Table 4 indicates that the $U \cdot A$ values derived from the analytical model deviate considerably from the published data (31.14%, 21.91% and 41.28%). However, adoption of the fully formulated LMTD_C (rather than the simplified expression) reduces these deviations to negligible levels (-0.02% , -0.03% and $+0.02\%$), consistent with the manufacturer-based validation. The aggregated error metrics also support this correction, as MAE decreases from $179.027 \text{ W} \cdot \text{K}^{-1}$ to $0.123 \text{ W} \cdot \text{K}^{-1}$, RMSE from $341.260 \text{ W} \cdot \text{K}^{-1}$ to $0.138 \text{ W} \cdot \text{K}^{-1}$, MAPE from 31.443% to 0.023%, and the maximum error from 41.28% to 0.03%.

These results indicate that the fully formulated LMTD_C provides accurate values, although distortions from simplified expressions require separate evaluation. As a first step, the effects of the proposed analytical framework are examined through a case study (based on the reference system).

5. Case study

5.1. Sensitivity of the condenser surface-area to initial parameters

The selection of the indoor unit fixes several major parameters relevant to condenser sizing, such as p_c , T_{sh} , or the refrigerant. According to Equation 12, four initial parameters ($T_{a,1}$, $\dot{m}_{\text{air}}$, U , and T_{sc}) remain adjustable in condenser design; therefore, the sensitivity of the required surface-area to these parameters is examined first individually and then in combination. These four parameters exhibit distinct degrees of operational flexibility. U and T_{sc} permit only marginal adjustment once the system enters service (material and construction decisions constrain the former, while the thermal design process constrains the latter). In contrast, both $T_{a,1}$ and $\dot{m}_{\text{air}}$ can vary considerably during operation. Direct influence over $T_{a,1}$ remains scarcely feasible, whereas $\dot{m}_{\text{air}}$ permits precise regulation through fan-speed control.

Fig. 2 presents the sensitivity of the condenser surface-area to the four adjustable initial parameters (using the data in Table 1). Percentage variation is applied to U and $\dot{m}_{\text{air}}$, whereas $T_{a,1}$ and T_{sc} vary in absolute increments and plotted on a separate (red) temperature axis to preserve graphical clarity.

Based on Fig. 2 and the maximum investigated variations (20% for U and $\dot{m}_{\text{air}}$, and 3.0°C for the temperatures), the sensitivity ranking is $T_{a,1}$ (-29.63%), U (-16.67%), $\dot{m}_{\text{air}}$ (-12.33%), and T_{sc} (-2.67%). Notably, a 3.0°C decrease in $T_{a,1}$ reduces the required surface-area by 29.63%,

Table 1
Technical data of the reference system at the corrected design condition.

Quantity	Symbol	Value
Heat released of the condenser	Q_C ; [kW]	407.0
Mass flow rate of the refrigerant	$\dot{m}_{\text{ref}}$; [$\text{kg} \cdot \text{s}^{-1}$]	1.941
Condensation pressure	p_C ; [bar]	19.74
Degree of superheating	$T_{\text{sh}} - T_{\text{dew}}$; [$^\circ\text{C}$]	35
Condensation temperature	$T_{\text{dew}}/T_{\text{bub}}$; [$^\circ\text{C}$]	49.72 / 45.04
Degree of subcooling	$\Delta T_{\text{sc}} = T_{\text{bub}} - T_{\text{sc}}$; [$^\circ\text{C}$]	5
Temperature of the inlet cooling air	$T_{a,1}$; [$^\circ\text{C}$]	35
Air volume flow rate of the fan	$\dot{V}_{\text{air}}$; [$\text{m}^3 \cdot \text{h}^{-1}$]	90,000
The overall heat-transfer coefficient	U_{nom} ; [$\text{W} \cdot \text{m}^{-2} \cdot \text{K}$]	31.23
Total heat-transfer surface-area of the condenser	A ; [m^2]	1656
Refrigerant		R407C

Table 2

Input data for the six validation cases, comprising three manufacturer-based configurations and three literature-based experimental datasets.

Symbol	Aermec	Trane Version 1	Trane Version 2	Reference [30] Test 1	Reference [30] Test 10	Reference [53]
Q_C ; [kW]	406.3	428.0	425.0	8.367	14.805	2.8
m_{ref} ; [kg·h ⁻¹]	7259.21	6781.80	6745.00	153.72	340.56	0.015
p_C ; [bar]	30.53	28.63	28.63	20.43	29.80	17.89
$T_{sh}-T_{dew}$; [°C]	35	35	35	57.86	35.83	
T_{dew} ; [°C]	49.86	50.6	50.6	44.95	61.71	38.20
T_{bub} ; [°C]	49.74	49.4	49.4	44.65	61.50	38.20
$\Delta T_{sc}=T_{bub}-T_{sc}$; [°C]	5	0	0	6	9.6	0
$T_{a,1}$; [°C]	35	35	35	32	32.2	24.6
V_{air} ; [m ³ ·h ⁻¹]	99,720	101,945	101,375	-	-	-
m_{air} ; [kg·s ⁻¹]	-	-	-	1.374	0.9	0.274
Refrigerant	R410A	R454B	R454B	R404A	R404A	R507A
U_{nom} ; [W·m ⁻² ·K ⁻¹]	57.32	44.86	44.57	-	-	-
A_{nom} ; [m ²]	663	900	907	-	-	-
$U \cdot A$; [W·K ⁻¹]	-	-	-	886	737	-
LMTD; [K]						7.52
P ; [kW]	86.9	100.02	100.02	5.629	10.012	1.031
Q_o ; [kW]	309.8	325.88	325.88	5.629	10.012	1.75
COP_R	3.57	3.26	3.26	1.454	1.672	1.70

Table 3

Validation results based on the installed reference system and three manufacturer configurations, including statistical error metrics.

Symbol	Reference system	Aermec	Trane Version 1	Trane Version 2
η_{cond} ; in %	25.10	25.14	25.28	25.52
A_{nom} ; in m ²	1656	663	900	907
A_{sh} ; in m ²	170.69	114.73	135.19	136.35
A_{cond} ; in m ²	1411.28	516.63	764.81	770.65
A_{sc} ; in m ²	74.02	31.64	0	0
A ; in m ²	1658.79	661.17	893.69	899.21
(Equation 15 and Eq. 20b)				
Deviation, in m ²	(-2.79)	(-1.83)	(-6.31)	(-7.79)
$(A - A_{nom})$				
Deviation, in %	(-0.169)	(-0.276)	(-0.701)	(-0.859)
$\left(\frac{A - A_{nom}}{A_{nom}}\right)$				
A_{min} ; in m ²	965.58	422.19	564.11	564.63
$\beta_1=A_{min} \cdot A^{(-1)}$; in %	58.31	63.68	62.68	62.25
MAE, in m ²	4.68			
$\left(MAE = \frac{1}{n} \cdot \sum A - A_{nom} \right)$				
RMSE, in m ²	5.28			
$\left(RMSE = \sqrt{\frac{1}{n} \cdot \sum (A - A_{nom})^2}\right)$				
MAPE, in %	0.501			
$\left(MAPE = \frac{1}{n} \cdot \sum \left \frac{A - A_{nom}}{A_{nom}}\right \cdot 100\right)$				
Maximum absolute error, in %	0.859			
$\max \left \frac{A - A_{nom}}{A_{nom}}\right \cdot 100$				

whereas an identical increase results in an 82.13% growth.

Fig. 3 decomposes the sensitivity analysis of Fig. 2 across the three distinct subphases (desuperheating, condensation, and subcooling) of refrigerant heat exchange.

For the required surface-area of desuperheating and condensation (Fig. 3), the sensitivity ranking is identical to that for the total condenser surface, differing only in the extent of the percentage changes. In contrast, the subcooling region exhibits a significant difference: the required surface-area is most sensitive to T_{sc} (−66.78%), followed by $T_{a,1}$ (−30.34%), U (−16.67%), and m_{air} (−0.48%). A notable point is that U affects all subprocesses identically, whereas T_{sc} and m_{air} are subphase-dependent—an aspect worth considering when designing condensers with a subcooling section.

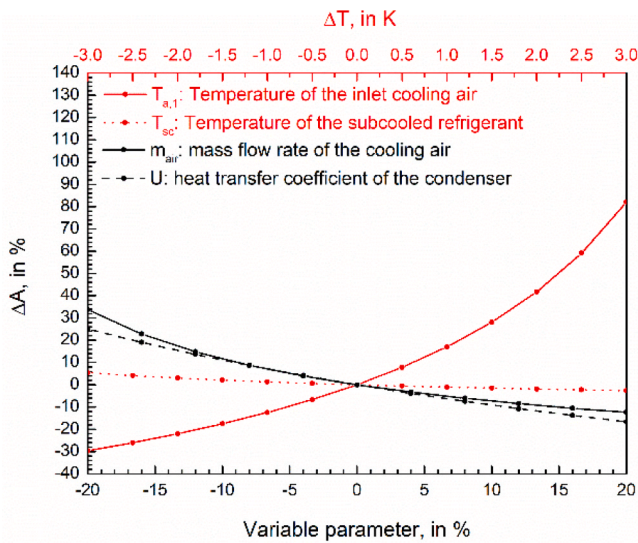
The continuation of the sensitivity analysis evaluates all six possible two-parameter combinations of the four adjustable parameters (m_{air} , $T_{a,1}$, U , and T_{sc}), as illustrated in Fig. 4. (The colour scale varies in 10% increments, where green shades indicate reduced surface demand and yellow-to-red tones denote increased demand relative to the reference system.)

Figs. 4a and 4e reveal two types of missing surface sections, each associated with the increase of $T_{a,1}$. An increase in $T_{a,1}$ reduces the minimum attainable T_{sc} (bounded by T_{bub} and $T_{a,1}$), and the combined increase in $T_{a,1}$ with a reduction in m_{air} prevents the removal of the prescribed heat flow Q_C . The smallest condenser surface arises from the combined modification of U and $T_{a,1}$ (Figs. 4d, 971.1 m², 58.64% of the reference value). A result of similar magnitude emerges from the

Table 4

Validation results based on three literature datasets, comparing published and corrected LMTDC values, including statistical error metrics.

Method	Symbol	Reference [30] Case 1	Reference [30] Case 10	Reference [53]
Value of LMTD _C using [30] (same as Equation 18)	η_{cond} , in %	8.52	24.95	14.42
	$U \cdot A$ (using η_{cond}), in $\text{W} \cdot \text{K}^{-1}$	610.11	575.52	218.63
	$U \cdot A_{\text{min}}$, in m^2	515.63	452.24	29.41
	$\beta_1 = (U \cdot A_{\text{min}}) \cdot (U \cdot A)^{-1}$, in %	84.51	78.58	13.45
	LMTD _C , in °C	9.4	20.1	7.52
	$U \cdot A$, in $\text{W} \cdot \text{K}^{-1}$	886	737	372.34
	Deviation, in $\text{W} \cdot \text{K}^{-1}$	275.89	161.48	153.71
	$(U \cdot A - U \cdot A_{\text{nom}})$			
	Deviation, in %	31.14	21.91	41.28
	$\left(\frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right)$			
	MAE, in $\text{W} \cdot \text{K}^{-1}$	197.027		
	$\left(\text{MAE} = \frac{1}{n} \cdot \sum U \cdot A - U \cdot A_{\text{nom}} \right)$			
	RMSE, in $\text{W} \cdot \text{K}^{-1}$	341.260		
	$\left(\text{RMSE} = \sqrt{\frac{1}{n} \cdot \sum (U \cdot A - U \cdot A_{\text{nom}})^2} \right)$			
	MAPE, in %	31.443		
	$\left(\text{MAPE} = \frac{1}{n} \cdot \sum \left \frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right \cdot 100 \right)$			
Value of LMTD _C using Equation 16	Maximum absolute error, in %	41.28		
	$\max \left \frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right \cdot 100$			
	LMTD _C , in °C	13.71	25.73	12.81
	$U \cdot A$, in $\text{W} \cdot \text{K}^{-1}$	609.97	575.33	218.41
	Deviation, in $\text{W} \cdot \text{K}^{-1}$	(-0.14)	(-0.19)	0.04
	$(U \cdot A - U \cdot A_{\text{nom}})$			
	Deviation, in %	(-0.02)	(-0.03)	0.02
	$\left(\frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right)$			
	MAE, in $\text{W} \cdot \text{K}^{-1}$	0.123		
	$\left(\text{MAE} = \frac{1}{n} \cdot \sum U \cdot A - U \cdot A_{\text{nom}} \right)$			
	RMSE, in $\text{W} \cdot \text{K}^{-1}$	0.138		
	$\left(\text{RMSE} = \sqrt{\frac{1}{n} \cdot \sum (U \cdot A - U \cdot A_{\text{nom}})^2} \right)$			
	MAPE, in %	0.023		
	$\left(\text{MAPE} = \frac{1}{n} \cdot \sum \left \frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right \cdot 100 \right)$			
	Maximum absolute error, in %	0.03		
	$\max \left \frac{U \cdot A - U \cdot A_{\text{nom}}}{U \cdot A_{\text{nom}}} \right \cdot 100$			

**Fig. 2.** Sensitivity of the required condenser surface-area to the four adjustable initial parameters, with $T_{a,1}$ and T_{sc} varied in absolute increments and U and m_{air} in percentage terms.

combined modification of m_{air} and $T_{a,1}$ (Figs. 4a, 1065.16 m^2 , 64.32% of the reference value), but this configuration supports a more practical implementation.

In the next step of the analysis, simultaneous variation of three adjustable parameters produces four possible combinations. To maintain clarity in the graphical representation, two parameters vary with identical magnitude in each combination. The corresponding results appear in Fig. 5.

Fig. 5 also exhibits missing surface sections, arising for the same reason as in the two-parameter analysis (from the increase in $T_{a,1}$). The surfaces become markedly steeper whenever both m_{air} and $T_{a,1}$ are among the varying parameters. Relative to the reference system, the largest surface reduction appears for the combined modification of $T_{a,1}$, m_{air} , and U (−46.40%, Fig. 5d), followed by U , $T_{a,1}$, and T_{sc} (−43.49%, Fig. 5c), then m_{air} , $T_{a,1}$, and T_{sc} (−36.89%, Fig. 5a), and finally m_{air} , U , and T_{sc} (−28.72%, Fig. 5b).

Finally, Fig. 6 presents the simultaneous modification of all four adjustable parameters. For graphical clarity, each parameter pair varies with identical magnitude (T_{sc} – $T_{a,1}$ and U – m_{air}).

Fig. 6 indicates that a surface reduction of up to 47.41% is attainable (U and m_{air} : +20%, T_{sc} : +3 °C, $T_{a,1}$: −3 °C). In contrast, simultaneous modification of all four parameters in the opposite direction (U and m_{air} : −20%, T_{sc} : −3 °C, $T_{a,1}$: +3 °C) results in a 277.92% increase in the required surface-area.

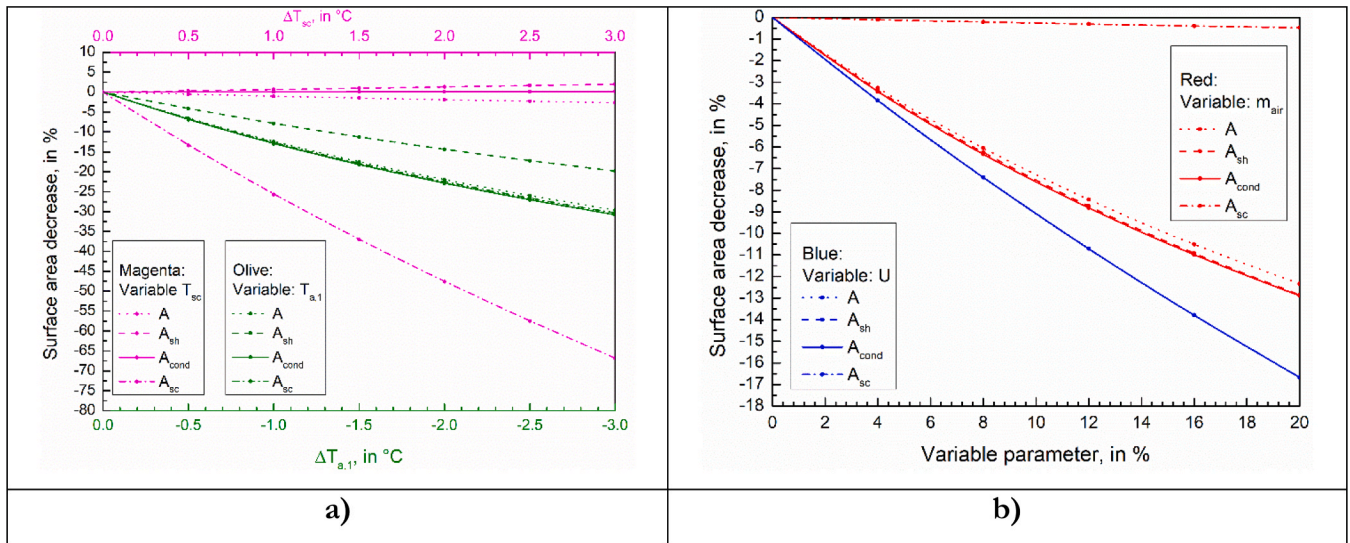


Fig. 3. Sensitivity of the condenser subphase surface-areas to the adjustable initial parameters, demonstrating distinct parameter hierarchies across regions: **a)** effect of subcooling and inlet-air temperature **b)** effect of air-side mass flow rate and overall heat-transfer coefficient.

According to the overall sensitivity analysis, increasing the number of simultaneously varied parameters reduces the required surface-area. (The minimum attainable values are: one parameter ($T_{a,1}$): 1165 m²; two parameters ($T_{a,1}$ and U): 971.10 m²; three parameters ($T_{a,1}$, U , and $\dot{m}_{air}$): 887.63 m²; all four: 870.94 m².) Across all parameter combinations (1, 2, 3, or 4), $T_{a,1}$ functions as the decisive parameter, as combinations including it result in smaller required surface-areas. Nevertheless, the distinction between theoretical sensitivity and practical engineering action requires explicit consideration. While the sensitivity analysis treats all four parameters as independent variables, real operational intervention reduces to a single adjustable quantity ($\dot{m}_{air}$). In operational terms, evaluating $\dot{m}_{air}$ in isolation, without accounting for the concurrent fluctuation of $T_{a,1}$, yields misleading estimates. This interdependence forms the basis of the fan-control strategy discussed in [Section 6.2](#).

5.2. Investigation of surface reduction using the condenser temperature-potential utilisation factor

The temperature-potential utilisation factor (η_{cond} , see [Section 3.2](#)) enables a more precise basis for assessing the condenser surface demand. The initial stage of this analysis considers the influence of all four adjustable parameters on η_{cond} , as illustrated in [Fig. 7](#).

[Fig. 7a](#) indicates that only $\dot{m}_{air}$ and $T_{a,1}$ influence the value of η_{cond} , in accordance with [Equation 11](#). (The apparent inconsistency with [Equation 13](#) is resolved by substituting [Equation 11](#) into [Equation 13](#).) The variation of η_{cond} follows the direction of the surface reduction; therefore, at a fixed $T_{a,1}$, a lower surface demand requires a reduced η_{cond} , realised through an increased $\dot{m}_{air}$. [Fig. 7b](#) presents the condenser surface demand (A) and $\dot{m}_{air}$, together with the four air-side temperature curves ($T_{a,1}$, $T_{a,sh}$, $T_{a,sc}$, $T_{a,2}$) defining the subphase boundaries, each expressed as a function of η_{cond} . The diagram offers visual confirmation of [Equation 17a](#) (indirectly also [Equations 17b, 17c and 17d](#)), demonstrating that the theoretical minimum surface is attained at $\eta_{cond} = 0$. This $\eta_{cond} = 0$ state features coincident air-side temperatures ($T_{a,1} = T_{a,sh} = T_{a,sc} = T_{a,2}$) and the associated infinite $\dot{m}_{air}$. This extreme condition highlights that A_{min} represents a theoretical limit, which becomes relevant in the practical interpretation (see [Section 6.2](#)).

A_{min} is obtained from [Equation 17a](#), where the indoor unit fixes several parameters and leaves only U , $T_{a,1}$, and T_{sc} adjustable. Accordingly, [Fig. 8](#) presents the resulting values of A_{min} as a function of $T_{a,1}$ and T_{sc} for a range of U values.

[Fig. 8a](#) indicates an exponential decrease in A_{min} at lower $T_{a,1}$ (hence the need to verify the manufacturer's cooling-air temperature), emphasising the importance of using the lowest attainable external air temperature. [Fig. 8b](#) demonstrates that T_{sc} affects A_{min} only marginally (except when it is within about 1 K of $T_{a,1}$).

This case study demonstrates that the analytical formulation provides a consistent basis for determining the condenser surface demand and for reducing it through the adjustment of four initial design parameters. Furthermore, the introduction of η_{cond} enables the identification of a theoretical lower bound for the required surface, which is, notably, non-zero. These findings establish the theoretical relevance of the analytical model, therefore practical applicability can be also examined. This case study (indirectly) applies the exact $LMTD_C$ value ([Equation 16](#)); however, the practical necessity of this precision remains questionable, and the deviations arising from simplifying the $LMTD_C$ value are therefore also assessed. Accordingly, the following section examines the effect of $LMTD_C$ simplification and the practical applicability of the proposed framework.

6. Discussion and practical application

6.1. Impact of simplified logarithmic mean temperature difference ($LMTD_C$) on the results

The analytical formulation presented in [Section 3](#) defines the $LMTD_C$ ([Equation 16](#)) with full subphase distinction, which at first glance appears unnecessarily overcomplicated. Therefore, this section compares the deviation in results between the fully formulated and the two simplified $LMTD_C$ approaches. The first simplified method is based on Stoecker's book [\[54\]](#) and represents the ACC as a condensation-only heat exchanger (this simplification is also applied by Cuevas [\[30\]](#) and Morozyuk [\[53\]](#)). The second method treats the ACC as a simple counterflow heat exchanger (an approximation suggested by practising HVAC engineers). [Fig. 9](#) presents the surface-temperature diagrams for these simplified methods.

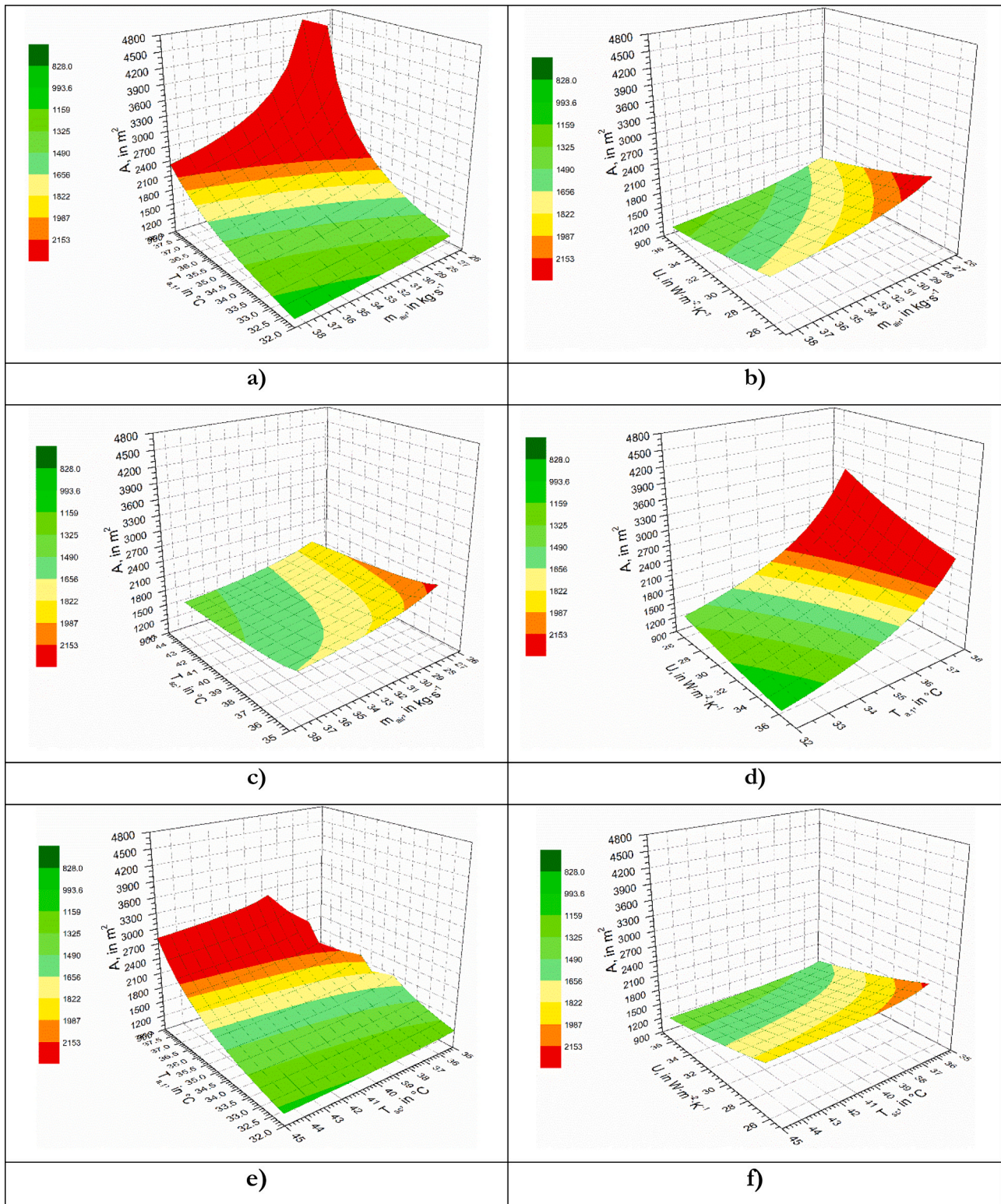


Fig. 4. Required condenser surface-area under simultaneous variation of two adjustable parameters, with colour scale varying in 10% increments: **a)** $T_{a,1}$; m_{air} **b)** U ; m_{air} **c)** T_{sc} ; m_{air} **d)** $T_{a,1}$; U **e)** $T_{a,1}$; T_{sc} **f)** U ; T_{sc} .

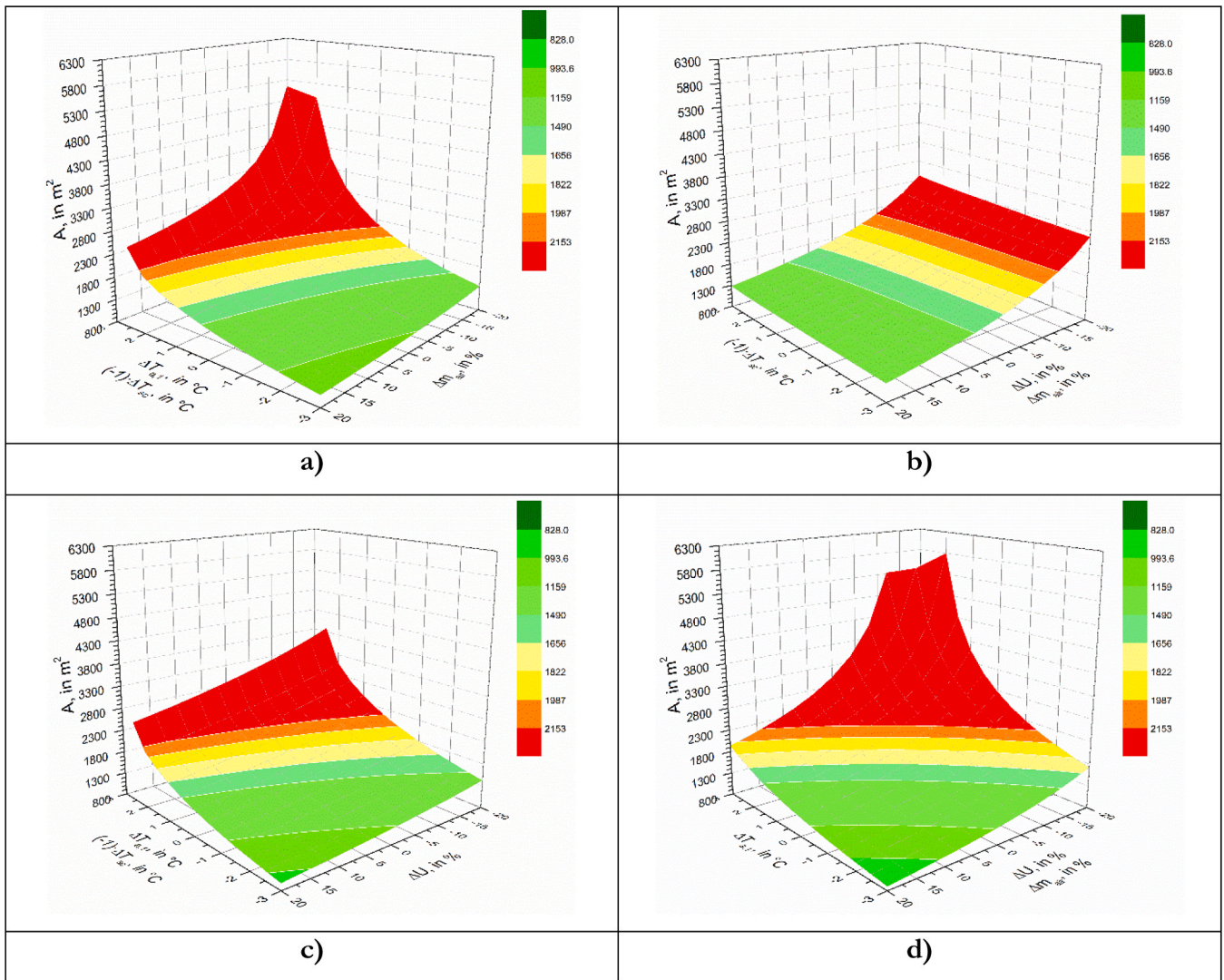


Fig. 5. Required condenser surface-area under simultaneous variation of three adjustable parameters, with colour scale varying in 10% increments: a) T_{sc} ; $T_{a,1}$; $\dot{m}_{air}$ b) T_{sc} ; U ; $\dot{m}_{air}$ c) T_{sc} ; $T_{a,1}$; U d) $T_{a,1}$; U ; $\dot{m}_{air}$.

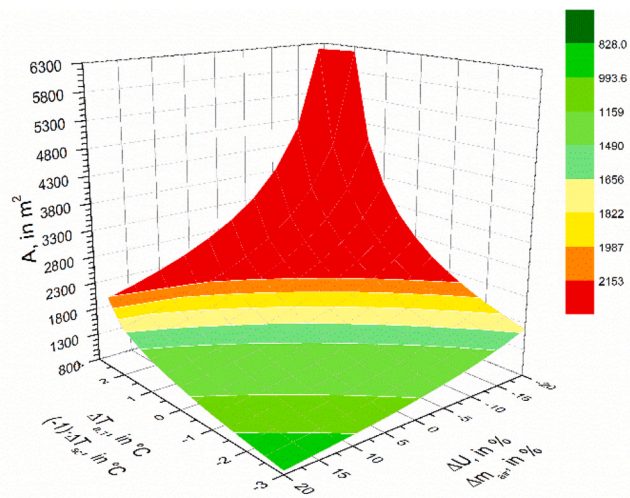


Fig. 6. Required condenser surface-area under simultaneous variation of all four adjustable parameters, with colour scale varying in 10% increments (the pairs $T_{sc}-T_{a,1}$ and $U-\dot{m}_{air}$ vary with identical magnitude).

The value of $LMTD_C$ is determined by Equation 18 for the first simplified method (based on Fig. 9a) and by Equation 19 for the second one (based on Fig. 9b).

$$LMTD_{C,simple1} = \frac{T_{a,2} - T_{a,1}}{\ln \frac{T_C - T_{a,1}}{T_C - T_{a,2}}} = \frac{T_{a,2} - T_{a,1}}{\ln \frac{T_{dew}^{T_{a,1}} - T_{a,1}}{T_{dew}^{T_{a,2}} - T_{a,2}}} \quad (18)$$

$$LMTD_{C,simple2} = \frac{(T_{sc} - T_{a,2}) - (T_{sh} - T_{a,2})}{\ln \frac{T_{sc} - T_{a,2}}{T_{sh} - T_{a,2}}} \quad (19)$$

The case study and each of the proposals from the manufacturers provide the data required to evaluate the deviation of both $LMTD_C$ and A for the two simplified methods. Table 5 summarizes the initial data and the corresponding deviations for each simplified approach.

Table 5 indicates that Method 1 underpredicts $LMTD_C$ by 36.12–37.2%, which results in a 56.54–59.24% excess in surface-area. Although this approach is safer in engineering terms (i.e., it avoids an undersized heat-transfer surface), its application introduces a latent weakness. This weakness occurs when the refrigerant exhibits a substantial glide; the averaged condensation temperature occasionally falls below $T_{a,2}$ ($T_C \leq T_{a,2}$), which generates a non-positive logarithmic argument and invalidates the LMTD equation. (The reference system provides such an example.) In contrast, Method 2 overpredicts the value

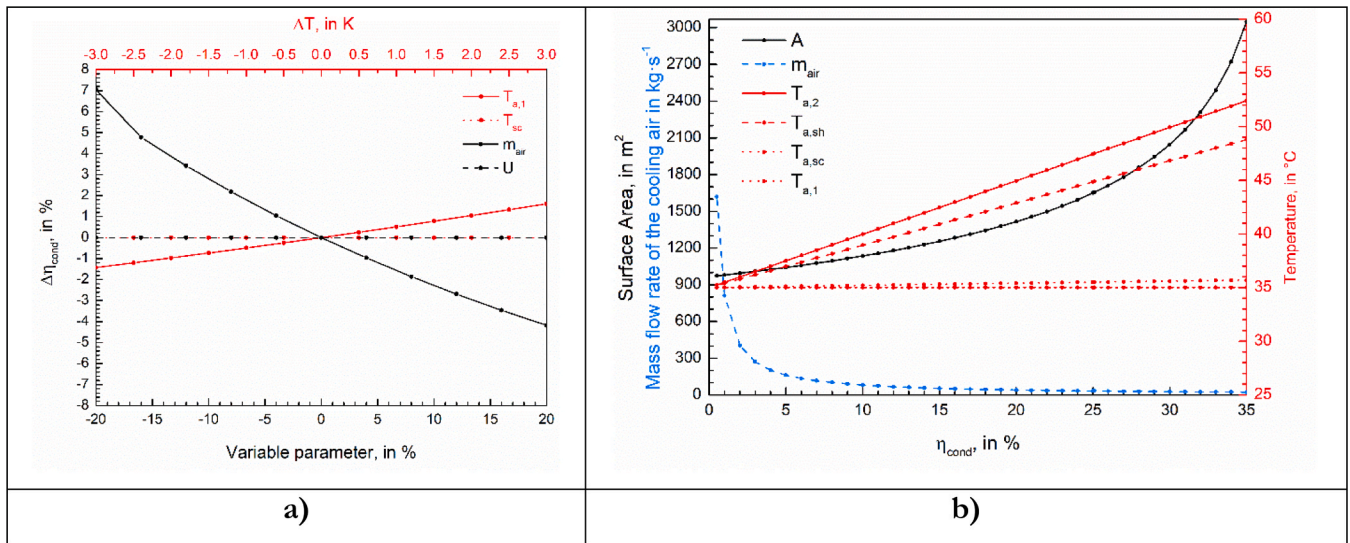


Fig. 7. a) Sensitivity of the condenser temperature-potential utilisation factor to the initial parameters b) Variation of air-side temperatures, condenser surface-area, and air-side mass flow rate as a function of condenser temperature-potential utilisation factor.

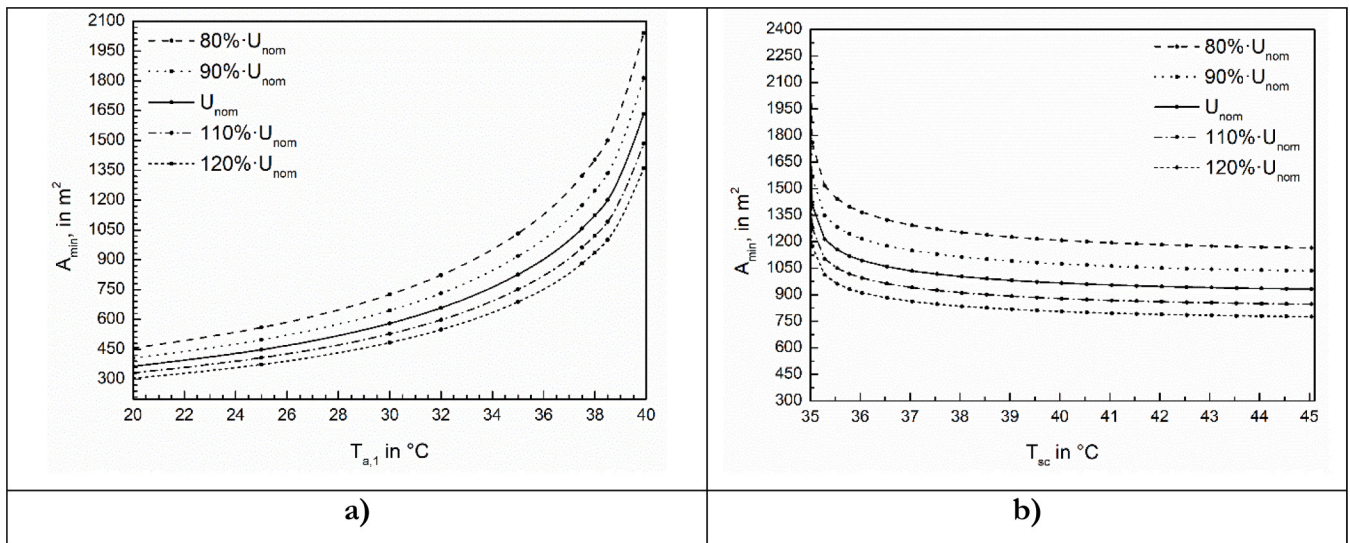


Fig. 8. Minimum required condenser surface-area as a function of a) inlet cooling air temperature b) subcooling temperature, for various overall heat-transfer coefficients based on Equation 17a.

of $LMTD_C$ by 92.2–130.6%, thereby reducing the surface-area demand by 48.0–56.6%. Comparison with the value of A_{min} in Table 3 reveals that this simplification can even yield a surface-area below A_{min} . Therefore, this approach is unsuitable for reliable engineering design, since it results in a critically undersized heat-transfer surface.

The magnitude of these deviations (overprediction up to 59% and underprediction up to 57%) confirms the necessity of applying the fully formulated $LMTD_C$ expression. Therefore, the following section can now examine the practical applicability of the proposed analytical framework.

6.2. Practical applicability and potential use cases

During the preparation of this study, several application areas emerged where the presented analytical framework exhibits potential relevance for practical HVAC design. Four principal directions of

application are identified with due consideration of practising engineers' views.

The **first application** arises at the initial stage of condenser design, before requesting the manufacturer's technical proposals. At this stage, the primary decision is whether heat rejection necessitates a water-cooled condenser or is sufficiently low for an air-cooled configuration. Since the surface-area is a key selection parameter, the minimum theoretical surface (A_{min}) can serve as a valuable consideration. The corresponding decision-support process is illustrated in Fig. 10.

A critical element of this process is the parameter β_1 , defined by the design engineer according to project-specific conditions. Its value ranges from 0 ($A=\infty$) to 1 ($A=A_{min}$) and varies with η_{cond} (generally, a larger η_{cond} determines a larger β_1 , reducing the estimated condenser surface area and favouring dry ACC feasibility). Based on the results in Tables 3 and 4, together with the two boundary conditions, β_1 is approximated by $1-\eta_{cond}^{0.7168}$ ($n = 9$, $R^2=0.959$), which can be refined with additional data

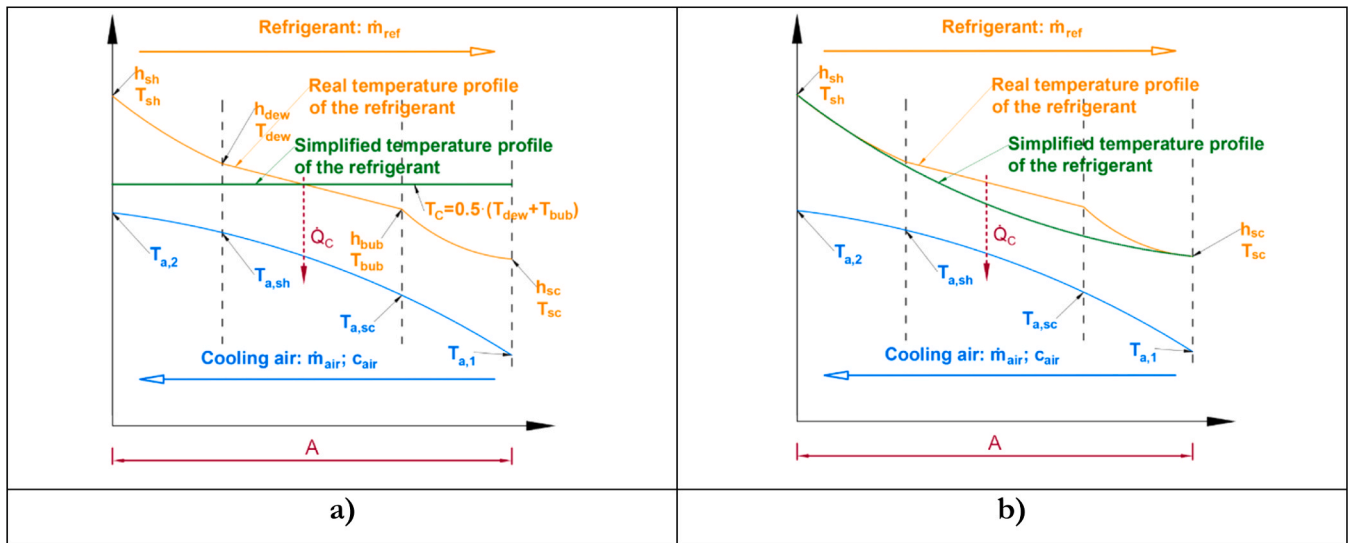


Fig. 9. Actual and two simplified surface-temperature profiles with visual deviations: **a)** condensation-only heat-exchanger representation; **b)** simple counterflow heat-exchanger representation.

Table 5

Effects of two LMTDC simplification approaches on LMTD_C value and required condenser surface evaluated across the installed reference system and three manufacturer configurations.

	Symbol	Reference system	Aermec	Trane Version 1	Trane Version 2
Initial data	$T_{a,1} / T_{a,2}$; in °C	35.0 / 47.5	35.0 / 47.5	35.0 / 47.7	35.0 / 47.8
	T_{dew} / T_{bub} ; in °C	49.72 / 45.04	49.86 / 49.74	50.6 / 49.4	50.6 / 49.4
	T_{sh} / T_{sc} or T_{bub} ; in °C	84.7 / 40.04	84.9 / 44.74	85.6 / T_{bub}	85.6 / T_{bub}
	Q_c ; in kW	407.0	406.3	428.0	425.0
	U ; in $W \cdot m^{-2} \cdot K^{-1}$	31.2	57.3	44.9	44.6
	A_{nom} ; in m ²	1656.0	663.0	900.0	907.0
	LMTD _C	7.87	10.69	10.60	10.51
Equation 16					
	LMTD _{C,simple1}	-	6.7	6.8	6.7
	Deviation at LMTD _C , in %	-	(-37.20)	(-36.12)	(-36.57)
	$A_{simple1}$, in m ²	-	1055.8	1408.8	1430.0
Method 1					
	Deviation at A , in %	-	59.24	56.54	57.66
	LMTD _{C,simple2}	16.09	20.55	24.28	24.25
	Deviation at LMTD _C , in %	104.5	92.2	129.1	130.6
Method 2					
	$A_{simple2}$, in m ²	809.8	345.0	392.9	393.3
	Deviation at A , in %	(-51.1)	(-48.0)	(-56.3)	(-56.6)

points. Subsequently, A_{min}/β_1 is compared with $A_{nom,max}$, defined as the project-relevant and largest nominal surface area in the air-cooled product-catalogues.

The **second application** concerns independent verification of manufacturer-provided sizing results. HVAC designers generally lack access to the manufacturers' sizing software, and this limitation creates dependence on calculations by manufacturer engineers (including potential input inaccuracies or selection bias). The analytical framework presented in this study reduces this exposure by enabling data verification. This decision-support process appears in Fig. 11.

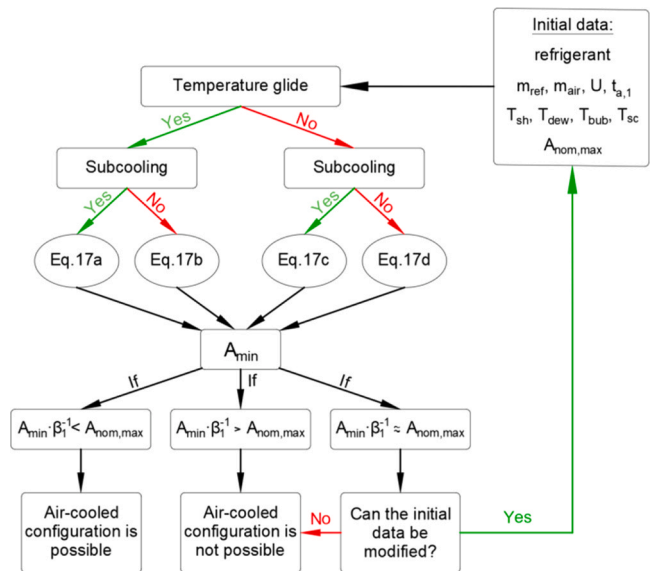


Fig. 10. Preliminary decision-support flowchart for condenser-type selection based on A_{min} .

The parameter β_2 applied in this process is defined by the design engineer according to project-specific conditions. Generally, a deviation of $\pm 5\%$ is acceptable (based on consultation with practising designers), although this value varies across countries and the project-specific conditions.

The **third application** is an alternative yet simple approach to fan control, derived from Equation 14. Assuming a constant value of the condenser temperature-potential utilisation factor ($\eta_{cond} = \text{constant}$), the cooling-air mass flow rate ($\dot{m}_{air}$) remains easily adjustable even when the external (inlet) air temperature ($T_{a,1}$) and the superheated refrigerant temperature (T_{sh}) vary. Thus, this approach supports energy-efficient fan operation in systems with variable condensation pressure (p_c) and exposure to variable external air temperature.

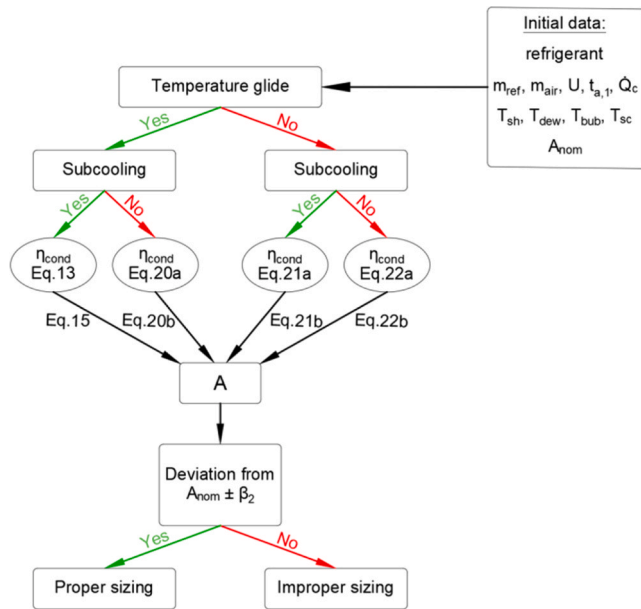


Fig. 11. Decision-support flowchart for independent verification of manufacturer sizing data based on η_{cond} .

The **fourth application** addresses early-stage operational diagnostics, where deviations from the design η_{cond} value indicate sub-optimal conditions. Elevated η_{cond} values typically reflect insufficient air flow (e.g., resulting from misconfigured fan control), abnormally high intake-air temperature at the condenser (e.g., caused by recirculation), or the compressor's inability to maintain the required condensation pressure. Conversely, reduced η_{cond} values indicate excessive air flow (e.g., here also resulting from misconfigured fan control) or unnecessarily high condensation pressure (e.g., caused by malfunction of the high-pressure safety switch). Thus, η_{cond} serves as an indicator for early detection of inefficient fan control and suboptimal operating states.

The four applications discussed above represent the principal directions; however, the method also supports further engineering decision-support tasks. This potential is demonstrated by two examples in this article. The first concerns the upgrade of existing remote condensers, where one of the central questions is whether pre-cooled inlet air or a higher-capacity fan unit constitutes the preferable solution. The presented method provides quantitative support for comparing associated investment, operational, and site-specific considerations. The second concerns the selection between equipment operating with different working fluids. The integrated consideration of conventional sizing parameters for remote condenser (e.g., air flow rate) and refrigerant properties (e.g., glide) enables rapid and reliable determination of expected performance and the required surface-area. For this reason, in the current transitional phase of refrigerant use, this method is particularly relevant.

7. Conclusion

7.1. Summary of main findings and practical implications

Sizing of remote condensers becomes increasingly critical due to the growing use of zeotropic refrigerants. Currently employed approaches (above all computationally demanding numerical methods and simplified LMTD-based formulations) often neglect temperature glide and subcooling. Hence, this study introduces a closed-form analytical

framework that resolves both shortcomings. The model determines the required surface-area explicitly and without iteration, by applying the LMTD method separately to the three heat-exchange subphases. The proposed model thereby provides physical rigour absent from simplified methods and preserves the transparency and low computational demand unattainable in numerical approaches. The proposed model separately resolves the sensitivity of the overall process and of the individual subphases. As demonstrated by the case study, the subcooling region exhibits a fundamentally distinct parameter hierarchy with T_{sc} dominant rather than general $T_{\text{a},1}$. This divergence indicates greater design flexibility for condenser surface-area reduction within the subcooling region than previously assumed. Simultaneous modification of all four adjustable parameters achieves the largest reduction in surface-area (47.41%). The more readily accessible combination of $T_{\text{a},1}$ and m_{air} alone produces a 35.68% reduction, which representing the principal engineering action available during post-commissioning intervention.

Defining the condenser temperature-potential utilisation factor (η_{cond}) represents the central result of this research. This factor provides a geometry-independent effectiveness indicator, and, when $(c \cdot m)_{\text{air}} = (c \cdot m)_{\text{min}}$, it yields the heat-exchanger effectiveness (ϵ) directly, thereby enabling single-step ϵ -NTU analysis of the entire condenser across all subphases. This indicator enables an alternative formulation of the required condenser surface-area (A), identifies a theoretical, non-zero lower bound for this surface (A_{min}), and supports the determination of the actual logarithmic mean temperature difference (LMTD_C). However, the practical necessity of this fully formulated LMTD_C is questionable, which motivates evaluation of the resulting values against two conventional simplifications. The condensation-only heat exchanger simplification increases A value by 56.5–59.2%, whereas the counterflow simplification reduces it by 48.0–56.6%. These significant deviations confirm the practical relevance of the fully formulated LMTD_C. Comparison with manufacturer data and previously published experimental datasets confirms the validity of the analytical framework, with deviations at the 1% level.

Grounding the evaluation in design considerations, this study identifies four principal applications of the analytical framework. The first two applications (each accompanied by a specific flowchart) relate to surface determination, while the remaining two are associated with the value of η_{cond} . The first application employs A_{min} to inform decisions between ACC and WCC configurations (in the early design phase), and the second provides independent verification of manufacturer sizing results. When η_{cond} remains approximately constant (third application), it allows for an alternative fan-control strategy. Substantial deviations of η_{cond} from its design value (fourth application) reveal operational irregularities, which establish it as a diagnostic indicator. These and further applications require attention to the model's assumptions and to the limitations of the current results, which delineate the scope for future research.

7.2. Limitations of the results and future research opportunities

The presented analytical framework provides a scientifically validated, engineer-oriented model for sizing and evaluating air-cooled condensers. Its primary relevance arises in systems employing zeotropic refrigerants and subcooling (with three additional configurations presented in the appendices); however, the model incorporates several inherent limitations. The following four principal limitations delineate the directions for future development and research:

- Since the presented model employs constant absolute humidity of the cooling air (see the model assumptions in Section 2), its direct applicability remains confined to dry air-cooled condensers.

Accordingly, future extension to wet-assisted dry and evaporative ACCs requires modification of the governing Equations 1–4.

- The model treats the overall heat-transfer coefficient (the U-value) as a nominal and scaling parameter, consistent with industrial practice. However, this simplification potentially restricts accuracy under conditions with pronounced humidity variation or substantial variation in air- and refrigerant-side mass flow and temperature. Therefore, the model incorporates an analytical expression for this dependency (via the U-value) in its future development.
- The model applies a steady state assumption. This confines the representation of start-up, shutdown, and transient dynamics (although HVAC systems predominantly operate within such steady state conditions). Thus, future development aims to incorporate non-steady-state dynamics into the model.
- The model applies a fixed temperature of the superheated refrigerant, thereby implying a constant discharge pressure. However, practical operation occasionally adjusts this discharge pressure in response to air-side variation. Therefore, future development aims to integrate the discharge-pressure variation into the model, thereby enabling accurate superheating determination required for effective condenser-surface optimisation.
- The model receives a comprehensive comparison against literature and manufacturer data; however, its validation remains incomplete within installed systems. This restricts exploration of further practical uses, although industrial responses reveal several potential applications. Therefore, future development integrates extended validation alongside the model refinement discussed above to enhance its practical relevance.

Nomenclatura

- T_{sh} (K) is the temperature of the superheated refrigerant.
- T_c (K) is the condensation temperature of the refrigerant.
- T_{dew} (K) is the dew-point temperature of the refrigerant.
- T_{bub} (K) is the bubble-point temperature of the refrigerant.
- T_{sc} (K) is the temperature of the subcooled refrigerant.
- $T_{a,1}$ (K) is the inlet temperature of the cooling air.
- $T_{a,sc}$ (K) is the cooling air temperature after condensation.
- $T_{a,sh}$ (K) is the cooling air temperature after desuperheating.
- $T_{a,2}$ (K) is the outlet temperature of the cooling air.
- ΔT_{sc} (K) is the temperature difference between the condensed and subcooled refrigerant.
- ΔT_{glide} (K) is the temperature glide of zeotropic refrigerant.
- $LMTD_c$ (K) is the logarithmic mean temperature difference for the entire condenser.
- $LMTD_{C,simple1}$ (K) is the simplified LMTD assuming semi-isothermal condenser.

Appendix

For conciseness, only the most complex case ($\Delta T_{glide} \neq 0$, $T_{bub} - T_{sc} \neq 0$) is included in the main text, while the three remaining special cases are presented in this Appendix. Firstly, Fig. 12 illustrates the corresponding pressure–enthalpy and surface–temperature diagrams for the processes of these cases.

- $LMTD_{C,simple2}$ (K) is the simplified LMTD assuming counterflow heat exchanger.
- h_{sh} ($J \cdot kg^{-1}$) is the specific enthalpy of the refrigerant entering the condenser.
- h_{dew} ($J \cdot kg^{-1}$) is the specific enthalpy of the refrigerant before condensation (at the dew-point).
- h_{bub} ($J \cdot kg^{-1}$) is the specific enthalpy of the refrigerant after condensation (at the dew-point).
- h_{sc} ($J \cdot kg^{-1}$) is the specific enthalpy of the refrigerant leaving the condenser.
- Q_c (W) is the total heat released in the condenser.
- Q_{sh} (W) is the heat released during desuperheating.
- Q_{cond} (W) is the heat released during condensation.
- Q_{sc} (W) is the heat released during subcooling.
- Q_0 (W) is the cooling capacity of the refrigeration system.
- P (W) is the electric power consumption of the compressor.
- A (m^2) is the total heat-transfer surface-area of the condenser.
- A_{cond} (m^2) is the heat-transfer surface-area required for condensation.
- A_{sh} (m^2) is the heat-transfer surface-area required for desuperheating.
- A_{sc} (m^2) is the heat-transfer surface-area required for subcooling.
- A_{min} (m^2) is the theoretical minimum heat-transfer surface-area of the condenser.
- A_{nom} (m^2) is the condenser surface-area in the air-cooled product-catalogues.
- $A_{nom,max}$ (m^2) is the project-relevant and largest value of A_{nom} .
- U ($W \cdot m^{-2} \cdot K^{-1}$) is the overall heat-transfer coefficient.
- U_{nom} (m^2) is the overall heat-transfer coefficient from the air-cooled product-catalogues.
- c_{air} ($J \cdot kg^{-1} \cdot K^{-1}$) is the specific heat capacity of cooling air.
- $\dot{m}_{ref}$ ($kg \cdot s^{-1}$) is the mass flow rate of the refrigerant.
- $\dot{m}_{air}$ ($kg \cdot s^{-1}$) is the mass flow rate of the cooling air.
- $\dot{V}_{air}$ ($m^3 \cdot s^{-1}$) is the volumetric flow rate of cooling air entering the condenser.
- η_{cond} (%) is the condenser temperature-potential utilisation factor.
- p_c (Pa) is the condensation pressure of the refrigerant.
- β_1 (–) is the safety factor for preliminary condenser sizing.
- β_2 (–) is the tolerance factor for manufacturer data verification.
- COP_R (–) is the coefficient of performance for cooling mode.
- $(c \cdot m)_{air}$ ($W \cdot K^{-1}$) is the capacity rate of air.
- $(c \cdot m)_{min}$ ($W \cdot K^{-1}$) is the smaller capacity rate from the two fluids.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

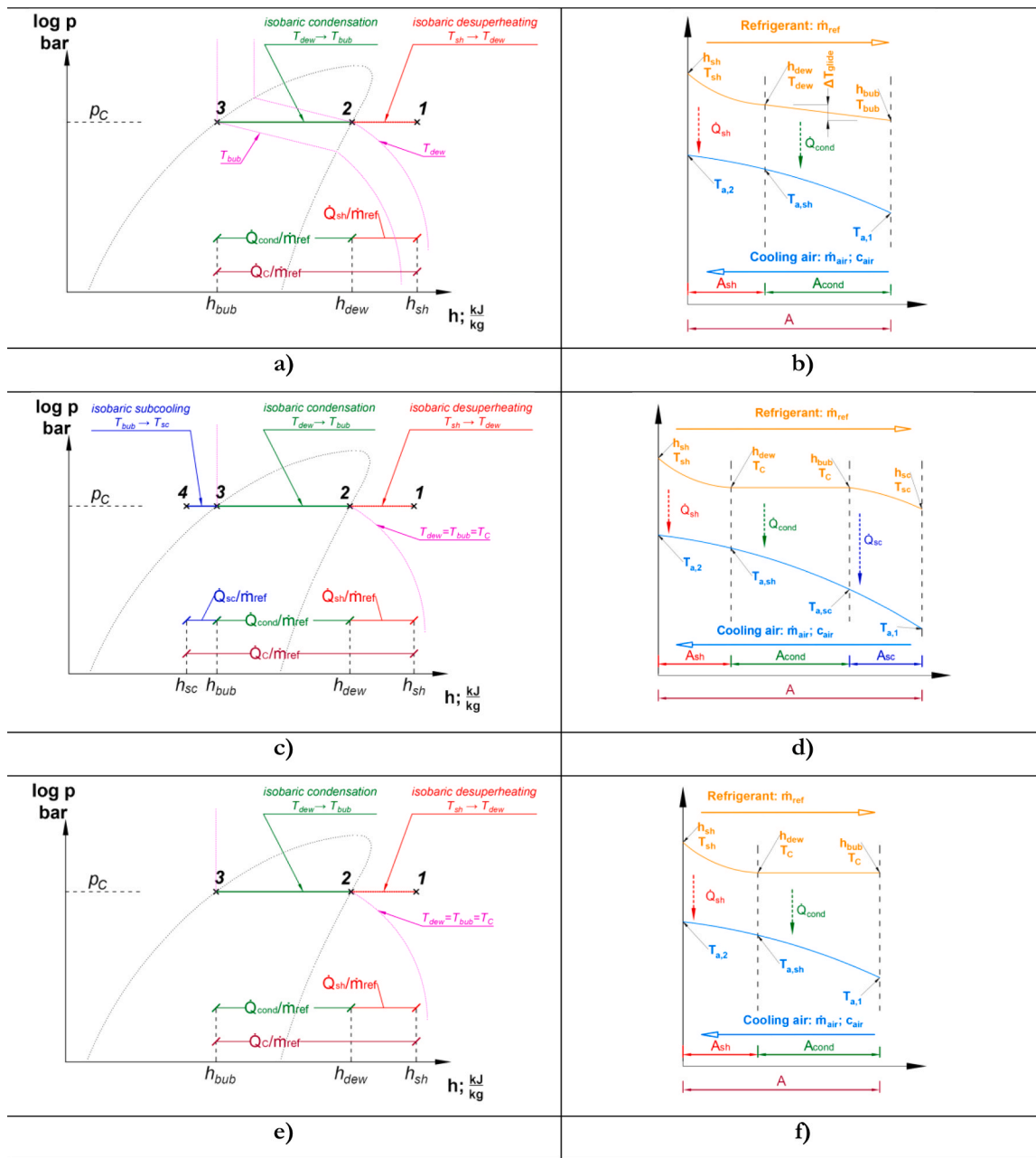


Fig. 12. Remote condenser specific cases: (a–b) glide only, (c–d) subcooling only, and (e–f) neither. Left: log p–h diagrams. Right: surface–temperature diagrams

For the first specific case ($\Delta T_{\text{glide}} \neq 0$, $T_{\text{bub}} - T_{\text{sc}} = 0$), the corresponding expressions for the condenser temperature-potential utilisation factor, surface-area, and LMTD are as follows:

$$\eta_{\text{cond}} = 1 - \frac{\frac{T_{\text{dew}} - T_{\text{a},1}}{T_{\text{sh}} - T_{\text{a},1}} - \frac{\dot{m}_{\text{ref}} \cdot (h_{\text{dew}} - h_{\text{bub}})}{c_{\text{air}} \cdot \dot{m}_{\text{air}} \cdot (T_{\text{sh}} - T_{\text{a},1})}}{\left[\exp \frac{A \cdot U}{c_{\text{air}} \cdot \dot{m}_{\text{air}}} \right] \left[1 - \frac{c_{\text{air}} \cdot \dot{m}_{\text{air}} \cdot (T_{\text{sh}} - T_{\text{dew}})}{\dot{m}_{\text{ref}} \cdot (h_{\text{sh}} - h_{\text{dew}})} \right]} \cdot \left[\frac{c_{\text{air}} \cdot \dot{m}_{\text{air}} \cdot (T_{\text{bub}} - T_{\text{a},1})}{c_{\text{air}} \cdot \dot{m}_{\text{air}} \cdot (T_{\text{dew}} - T_{\text{a},1}) - \dot{m}_{\text{ref}} \cdot (h_{\text{dew}} - h_{\text{bub}})} \right] \quad (20a)$$

$$A = \frac{\dot{Q}_C}{U \cdot (T_{\text{sh}} - T_{\text{a},1})} \cdot \ln \left[\left(\frac{1 - \eta_{\text{cond}}}{\frac{T_{\text{dew}} - T_{\text{a},1}}{T_{\text{sh}} - T_{\text{a},1}} - \eta_{\text{cond}} \cdot \frac{h_{\text{dew}} - h_{\text{bub}}}{h_{\text{sh}} - h_{\text{bub}}}} \right)^{\frac{1}{\frac{T_{\text{sh}} - T_{\text{dew}}}{T_{\text{sh}} - T_{\text{a},1}} \cdot \frac{h_{\text{sh}} - h_{\text{bub}}}{h_{\text{sh}} - h_{\text{dew}}} - \eta_{\text{cond}}}} \cdot \left(\frac{T_{\text{dew}} - T_{\text{a},1}}{T_{\text{bub}} - T_{\text{a},1}} - \frac{T_{\text{sh}} - T_{\text{a},1}}{T_{\text{bub}} - T_{\text{a},1}} \cdot \eta_{\text{cond}} \cdot \frac{h_{\text{dew}} - h_{\text{bub}}}{h_{\text{sh}} - h_{\text{bub}}} \right)^{\frac{T_{\text{dew}} - T_{\text{bub}}}{T_{\text{sh}} - T_{\text{a},1}} \cdot \frac{h_{\text{sh}} - h_{\text{bub}}}{h_{\text{dew}} - h_{\text{bub}}} - \eta_{\text{cond}}} \right] \quad (20b)$$

$$LMTD_C = (T_{sh} - T_{a,1}) \cdot \left\{ \ln \left[\left(\frac{1 - \eta_{cond}}{\frac{T_{dew} - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}}} \right)^{\frac{1}{\frac{T_{sh} - T_{dew}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{bub}}{h_{sh} - h_{dew}} - \eta_{cond}}} \cdot \left(\frac{T_{dew} - T_{a,1}}{T_{bub} - T_{a,1}} - \frac{T_{sh} - T_{a,1}}{T_{bub} - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}}} \right)^{\frac{1}{\frac{T_{dew} - T_{bub}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{bub}}{h_{dew} - h_{bub}} - \eta_{cond}}} \right] \right\}^{-1} \quad (20c)$$

For the second specific case ($\Delta T_{glide}=0$, $T_{bub}-T_{sc} \neq 0$), the refrigerants exhibit no temperature glide ($T_{dew}=T_{bub}=T_C$); the corresponding equations are as follows:

$$\eta_{cond} = 1 - \left[\frac{1}{\frac{T_C - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{\dot{m}_{ref} \cdot (h_{bub} - h_{sc})}{c_{air} \cdot \dot{m}_{air} \cdot (T_{sc} - T_{a,1})}} \right] \cdot \left[\frac{1 - \frac{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_C)}{\dot{m}_{ref} \cdot (h_{sh} - h_{dew})}}{1 - \frac{c_{air} \cdot \dot{m}_{air} \cdot (T_C - T_{sc})}{\dot{m}_{ref} \cdot (h_{bub} - h_{sc})}} \right] \cdot \left[\frac{c_{air} \cdot \dot{m}_{air} \cdot (T_C - T_{a,1}) - \dot{m}_{ref} \cdot (h_{bub} - h_{sc})}{c_{air} \cdot \dot{m}_{air} \cdot (T_C - T_{a,1}) - \dot{m}_{ref} \cdot (h_{dew} - h_{sc})} \right] \cdot \left[\frac{1 - \frac{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_C)}{\dot{m}_{ref} \cdot (h_{sh} - h_{dew})}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \frac{\dot{m}_{ref} \cdot (h_{dew} - h_{sc})}{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_{a,1})}} \right] \cdot \left[\exp \frac{A \cdot U}{c_{air} \cdot \dot{m}_{air}} \right] \cdot \left[\frac{1 - \frac{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_C)}{\dot{m}_{ref} \cdot (h_{sh} - h_{dew})}}{1} \right] \quad (21a)$$

$$A = \frac{\dot{Q}_C}{U \cdot (T_{sh} - T_{a,1})} \cdot \ln \left[\left(\frac{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{-\frac{1}{\eta_{cond}}} \cdot \left(\frac{T_C - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{T_{sh} - T_{a,1}}{T_{sc} - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_C - T_{sc}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{bub} - h_{sc}} - \eta_{cond}}} \cdot \left(\frac{1 - \eta_{cond}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_{sh} - T_C}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{sh} - h_{dew}} - \eta_{cond}}} \right] \quad (21b)$$

$$LMTD_C = (T_{sh} - T_{a,1}) \cdot \left\{ \ln \left[\left(\frac{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{-\frac{1}{\eta_{cond}}} \cdot \left(\frac{T_C - T_{a,1}}{T_{sc} - T_{a,1}} - \frac{T_{sh} - T_{a,1}}{T_{sc} - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{bub} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_C - T_{sc}}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{bub} - h_{sc}} - \eta_{cond}}} \cdot \left(\frac{1 - \eta_{cond}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{sc}}{h_{sh} - h_{sc}}} \right)^{\frac{1}{\frac{T_{sh} - T_C}{T_{sh} - T_{a,1}} \cdot \frac{h_{sh} - h_{sc}}{h_{sh} - h_{dew}} - \eta_{cond}}} \right] \right\}^{-1} \quad (21c)$$

For the third specific case ($\Delta T_{glide}=0$, $T_{bub}-T_{sc}=0$), where both glide (as in the second case, $T_{dew}=T_{bub}=T_C$) and subcooling are absent, the corresponding equations are as follows:

$$\eta_{cond} = 1 - \frac{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \frac{\dot{m}_{ref} \cdot (h_{dew} - h_{bub})}{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_{a,1})}}{\left\{ \left[1 - \frac{\dot{m}_{ref} \cdot (h_{dew} - h_{bub})}{c_{air} \cdot \dot{m}_{air} \cdot (T_C - T_{a,1})} \right] \cdot \exp \frac{A \cdot U}{c_{air} \cdot \dot{m}_{air}} \right\} \cdot \frac{1 - \frac{c_{air} \cdot \dot{m}_{air} \cdot (T_{sh} - T_C)}{\dot{m}_{ref} \cdot (h_{sh} - h_{dew})}}{1}} \quad (22a)$$

$$A = \frac{\dot{Q}_C}{(T_{sh} - T_{a,1}) \cdot U} \cdot \ln \left[\left(1 - \frac{T_{sh} - T_{a,1}}{T_C - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}} \right)^{-\frac{1}{\eta_{cond}}} \cdot \left[\frac{1 - \eta_{cond}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}}} \right]^{\frac{1}{\frac{h_{sh} - h_{bub}}{h_{sh} - h_{dew}} \cdot \frac{T_{sh} - T_C}{T_{sh} - T_{a,1}} - \eta_{cond}}} \right] \quad (22b)$$

$$LMTD_C = (T_{sh} - T_{a,1}) \cdot \left\{ \ln \left[\left(1 - \frac{T_{sh} - T_{a,1}}{T_C - T_{a,1}} \cdot \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}} \right)^{-\frac{1}{\eta_{cond}}} \cdot \left[\frac{1 - \eta_{cond}}{\frac{T_C - T_{a,1}}{T_{sh} - T_{a,1}} - \eta_{cond} \cdot \frac{h_{dew} - h_{bub}}{h_{sh} - h_{bub}}} \right]^{\frac{1}{\frac{h_{sh} - h_{bub}}{h_{sh} - h_{dew}} \cdot \frac{T_{sh} - T_C}{T_{sh} - T_{a,1}} - \eta_{cond}}} \right] \right\}^{-1} \quad (22c)$$

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