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A Rational Methodology for the Design and Verification of High-Efficiency Refrigerators

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Abstract. Refrigerators based on vapor compression cycles are a major focus of energy optimization in an era of rising electricity costs and increasing environmental awareness. Typically, the design of these appliances relies on empirical guidelines, previous experience and physical prototyping. This work proposes a physics-driven methodology to design and verify high-efficiency refrigerators. The preliminary design procedure utilizes the logarithmic mean temperature difference method alongside well-established literature correlations for heat transfer. To verify the components and to compute the annual energy consumption, a dynamic model of the system is developed. This model is then used to conduct a parametric analysis and a simulation-based exploration of the parameter space. The aim of this exploration is to minimize the annual energy consumption. Preliminary results indicate that the physics-driven methodology not only reduces the annual energy consumption by almost 4%, but also optimizes the process by eliminating trial and error loops and, consequently, reducing the R&D costs.

1 Introduction and scope

In 2020, there were approximately 258 million refrigerators installed in Europe, with 92% of them operating in the residential sector [1]. Because these systems run continuously, they have a strong impact on the residential electricity consumption. Furthermore, strict regulations like Ecodesign and Energy Labeling [2] mandate the minimization of their annual energy consumption. The prescribed target is a 64% reduction in refrigerator energy consumption by 2030, compared to the 1992 baseline. Despite these demanding efficiency targets, the industrial design process still relies heavily on empirical methods and extensive experimental campaigns. This trial and error approach is time-consuming, implies high design costs and often leads to over-dimensioned components. Research in this field generally splits into two different approaches: high-fidelity numerical models and isolated optimization of individual components. Considering the first case, Hermes and Melo [3] proposed a first-principle dynamic simulation to assess the influence of several design parameters on the energy performance of frost-free refrigerators. Kumlutaş et al. [4] used computational fluid dynamics simulations to train an artificial neural network. The aim of their work was to investigate the design parameter values of refrigerators. Caglayan et al. [5] developed a dynamic model on the software Dymola to perform design optimization of refrigerators. At the same time, numerous isolated optimizations on single components have been discussed in the literature. Melo et al. [6] presented an experimental study on capillary

tubes to investigate the effect of their length, diameters and other relevant parameters. Shikalgar et al. [7] analyzed the behaviour of hot-wall condensers versus the one of a wire-tube one on the total heat flux. To conclude, design guidelines for each single component are available, but a rational global procedure is missing. This work addresses the gap by proposing a physics-based methodology for the design and verification of the main components of high-efficiency refrigerators. It is a unified and integrated step-by-step methodology that guides the designer from a preliminary design to a high-efficiency refrigerator.

2 Case study

The presented case study is a combined refrigerator unit, featuring a top-mounted refrigerator compartment and a bottom-mounted freezer compartment. To ensure completely independent thermal management, each compartment is served by a separate refrigeration circuit. Both configurations operate on a single-stage vapor-compression circuit, charged with R600a. Each refrigeration loop integrates four primary components configured as follows: the evaporator is a finned-tube ventilated heat exchanger located inside the refrigerated compartment; the condenser is a wire-on-tube heat exchanger operating under forced convection, positioned in the lower compartment of the refrigerator; the compressor is a hermetic compressor unit, also housed in the lower compartment. The compressor behavior is described by performance maps, provided by the manufacturer; finally a standard capillary tube is employed as the expansion device. The nominal operating conditions of the refrigerator are reported in Table 1. Their values are set according to IEC 62552 standard [8] for measuring refrigerator energy consumption in tropical regions. For the sake of brevity, only the refrigerator compartment is designed, since the proposed methodology is universal and can be generalized to both compartments. The refrigerator theoretical thermodynamic cycle is set by the designer.

3 Methodology

The flowchart in Figure 1 summarizes the main steps of the proposed methodology. The first step is to define the target conditions, i.e. the cooling requirements and the theoretical thermodynamic cycle. Based on those conditions, a preliminary design of the system is derived from some physical considerations. Next, a verification phase is performed using a dynamic model developed in the Matlab® environment. Finally, a simulation-based exploration of the design space is conducted using the dynamic model to find the system that has the lower energy consumption. In the following subsections, the methodology is explained in greater detail.

Table 1: Nominal operating conditions.

Quantity	**Symbol**	**Value**	**Unit**
Evaporating temperature	T_{e}	-5	°C
Condensing temperature	T_{c}	45	°C
Superheating	ΔT_{SH}	5	°C
Subcooling	ΔT_{SC}	3	°C
Refrigerator temperature	$T_{\mathrm{refrigerator}}$	5	°C
Freezer temperature	T_{freezer}	-18	°C
External temperature	T_{ext}	32	°C

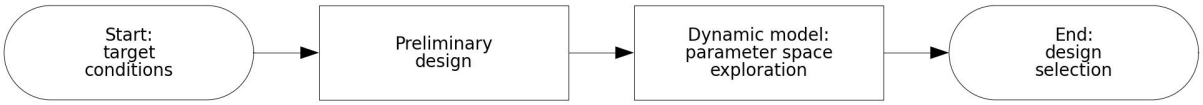

Figure 1: Flowchart representing the main steps of the proposed methodology.

3.1 Target conditions

The first step of the methodology consists of the definition of the operating conditions of the refrigerator. Specifically, two configurations are derived: the *normal cycle*, where the compressor runs at a minimum rotational speed, and a *pull-down cycle*, where the compressor works at a higher speed. The normal cycle occurs when the system must counterbalance the heat losses through the cabinet walls. Since the temperature difference between the compartment interior and the ambient temperature is set by the test conditions, the rate of heat transfer is derived from the geometrical and insulating properties of the compartment. This rate is the *minimum cooling requirement*. Conversely, the *maximum cooling requirement* is the thermal power required to offset the heat losses and to cool down a specific product load from its original temperature to the compartment set-point. Finally, the theoretical thermodynamic cycle is established by defining the condensation and evaporation temperatures, along with the specified superheating and subcooling. From these parameters, the refrigerant mass flow rate for both the operating conditions is computed.

3.2 System design

The second step of the methodology consists of the preliminary design of the individual components, as summarized in Table 2. First, the compressor is selected from a database of available commercial units. The compressor performance maps are known and are used to derive the coefficients of a 20-coefficient regression, which expresses the mass flow rate and the power consumption as a function of the evaporation temperature, condensation temperature and rotational speed. If the required mass flow rate is lower or higher than the design value, the minimum or maximum feasible mass flow rate of the selected compressor is considered for the design. In the preliminary sizing of the remaining components, these mass flow rates are applied to the design of the capillary tube and the heat exchangers, as will be shown in the following paragraphs. Indeed, each component is sized according to its most critical cooling requirements.

3.2.1 Capillary tube The capillary tube acts as the expansion device. It throttles the refrigerant to induce the required pressure drop between the condenser and the evaporator, according to the theoretical thermodynamic cycle. The mass flow rate is the minimum one, considering that the most critical operating condition is at the minimum cooling requirement. Given the internal diameter and the internal surface roughness of the tube, the length is derived by inverting the Darcy-Weisbach equation [9]. The Haaland [10] formulation of the friction factor has been used. Due to the progressive vaporization of the refrigerant, its thermodynamic properties vary significantly within the two-phase region and this requires a discretization into control volumes.

3.2.2 Heat exchangers After the selection of the compressor and the definition of the capillary tube length, the final stage of the preliminary design consists of the design of the heat exchangers. The heat exchangers design is governed by the maximum cooling requirements and is based on the logarithmic mean temperature difference (LMTD) method. An iterative procedure is needed since the overall heat transfer coefficient strictly depends on the heat transfer area. The bisection

algorithm is employed to solve the iterative loops due to its numerical robustness and simple formulation. The convergence stopping criterion is defined by a relative tolerance on the target variable lower than 10^{-4}.

In the evaporator, the boundary conditions regarding the temperatures and the mass flow rates are constrained by the system operating points defined in the first step § 3.2. The air-side inlet temperature is fixed by the internal-set point temperature, while the air-side outlet temperature acts as a design variable. On the refrigerant side, conversely, the temperatures inside the evaporator are directly derived from the thermodynamic cycle. Consequently, the objective of the evaporator design process is to determine the required heat transfer area to meet the design heat rate. The algorithm to design the evaporator is reported on the left side of Fig 2. In the first step, the effective heat transfer area is computed as a function of the total tube length. This length is initialized by the bisection algorithm within a physically admissible domain. This domain is bounded by a minimum technologically feasible tube length and the maximum available space within the compartment. Second, the heat transfer coefficients are computed using well-established literature correlations. Specifically, the air-side heat transfer coefficient is evaluated by the correlation proposed by Zukauskas [11] for a bank of tubes in cross flow. It should be noted that the correlation proposed by Wang et al. [12] could not be applied despite accounting for finned-tube heat exchangers. Indeed, the geometrical features of the problem were outside the range of applicability of this correlation. While Wang's correlation limits the maximum number of rows to 6, refrigerator evaporators usually feature a higher number of tubes in the flow direction. Meanwhile, the refrigerant two-phase heat transfer coefficient is computed using the Shah correlation [13] and the one-phase region heat transfer is computed via the Gnielinski correlation [14] for fluid flowing inside circular tubes. After the determination of the heat transfer area (A) as well as the overall heat transfer coefficient (U), the residual between the calculated UA and the design UA is computed. If the tolerance criterion is satisfied, the procedure stops, otherwise the tube length is updated via the bisection algorithm.

In contrast, in the condenser the air-outlet temperature is not known a priori, which implies that the air-mass flow rate is an unknown of the problem. Moreover, the geometry of the wire-

Table 2: Target conditions and method for each component.

Component	**Design cooling requirement [W]**	**Design target**	**Design output**	**Method**
Compressor	Minimum/ Maximum	Refrigerant flow rate	Compressor speed	Choice from a compressor list
Capillary tube	Minimum	Δp	Tube length	Darcy-Weisbach equation with Haaland friction factor
Forced convection evaporator	Maximum	UA	Tube length	Logarithmic mean temperature difference
Forced convection condenser	Maximum	Total tube length	Air flow rate	Logarithmic mean temperature difference

on-tube heat exchanger is fully defined. Another relevant difference between the evaporator and the condenser is that the first could be treated as a single control volume, while the second must be divided into three control volumes. This division into three control volumes aims to capture the distinct thermal behavior of the superheated vapor, the two-phase region and the subcooled liquid. To solve this mathematical problem, a specific double-nested loop procedure is implemented. The outer loop handles the air mass flow rate as the main iteration variable and is responsible for verifying the overall geometric constraint of the heat exchanger. For a given structural configuration, this loop passes the guessed air mass flow rate to the inner loop. The inner loop operates on a thermodynamic basis: utilizing the air mass flow rate defined by the outer loop, it calculates the required tube length for each of the three fluid regions. Specifically, for each zone, the algorithm computes the exact heat transfer area and thus the corresponding physical length necessary to exchange the specific thermal power prescribed by the thermodynamic cycle. Once the lengths of the superheated, two-phase, and subcooled sections are individually determined by the inner-loop, they are returned to the outer loop. Here, a convergence check is performed on the total dimensions. The sum of these three calculated lengths is compared against the fixed physical length of the condenser. If the residual exceeds the allowed tolerance, the outer loop updates the air mass flow rate via an iterative solver, and the process repeats. The algorithm reaches final convergence when the geometrical constraint is satisfied, meaning the sum of the three calculated zone lengths perfectly matches the total length of the condenser for the computed air flow rate. The workflow of the described double-nested loop is shown on the right side of Fig 2.

3.3 Dynamic model

Beyond the steady-state design of the main components of the refrigerator, a dedicated dynamic model is implemented to capture the transient behavior of the system. The model has been implemented in the Matlab environments, utilizing blocks from the Simscape thermal-fluids library [15]. The compressor is modeled with a custom-made block that implements the 20-coefficients of the regression that was calculated in the Section related to the system design § 3.2. In this way, based on the condensing and evaporating temperatures and the rotational speed given as an input by the control, the refrigerant flow rate and the power consumption are determined. After passing through the compressor, the refrigerant enters the condenser. This is modeled by a standard two-phase fluid to moist air Condenser - Evaporator block, which uses the Effectiveness-NTU method to evaluate the heat transfer through the walls. The subcooled refrigerant then enters the capillary tube which is a custom-made block composed of Pipe (2P) blocks. This discretization is necessary to better capture the thermal behavior of the refrigerant in the two-phase region. Finally, the refrigerant enters the evaporator, which is implemented using a "2P-MA" block identical in structure to the condenser. The air-side of the evaporator is connected to a volume chamber that mimics the refrigerator internal compartment, which interacts with the external environment through a series of thermal resistances and capacities. These latter components are fundamental to capture the transient thermal inertia of the system. Indeed, the compressor operates cyclically between two temperature thresholds dictated by the compartment thermostat. The presence of the thermal capacities allows the model to predict the start-up and shut-down periods of the compressor. In this way, a realistic annual energy consumption of the refrigerator is computed. It is worth mentioning that the thermal properties of the refrigerant are the result of a bi-linear extrapolation from lookup tables, whose values have been calculated using the CoolProp library [16]. These values are also used in the design of the components.

A first simulation of the preliminary design was carried out and the annual consumption of

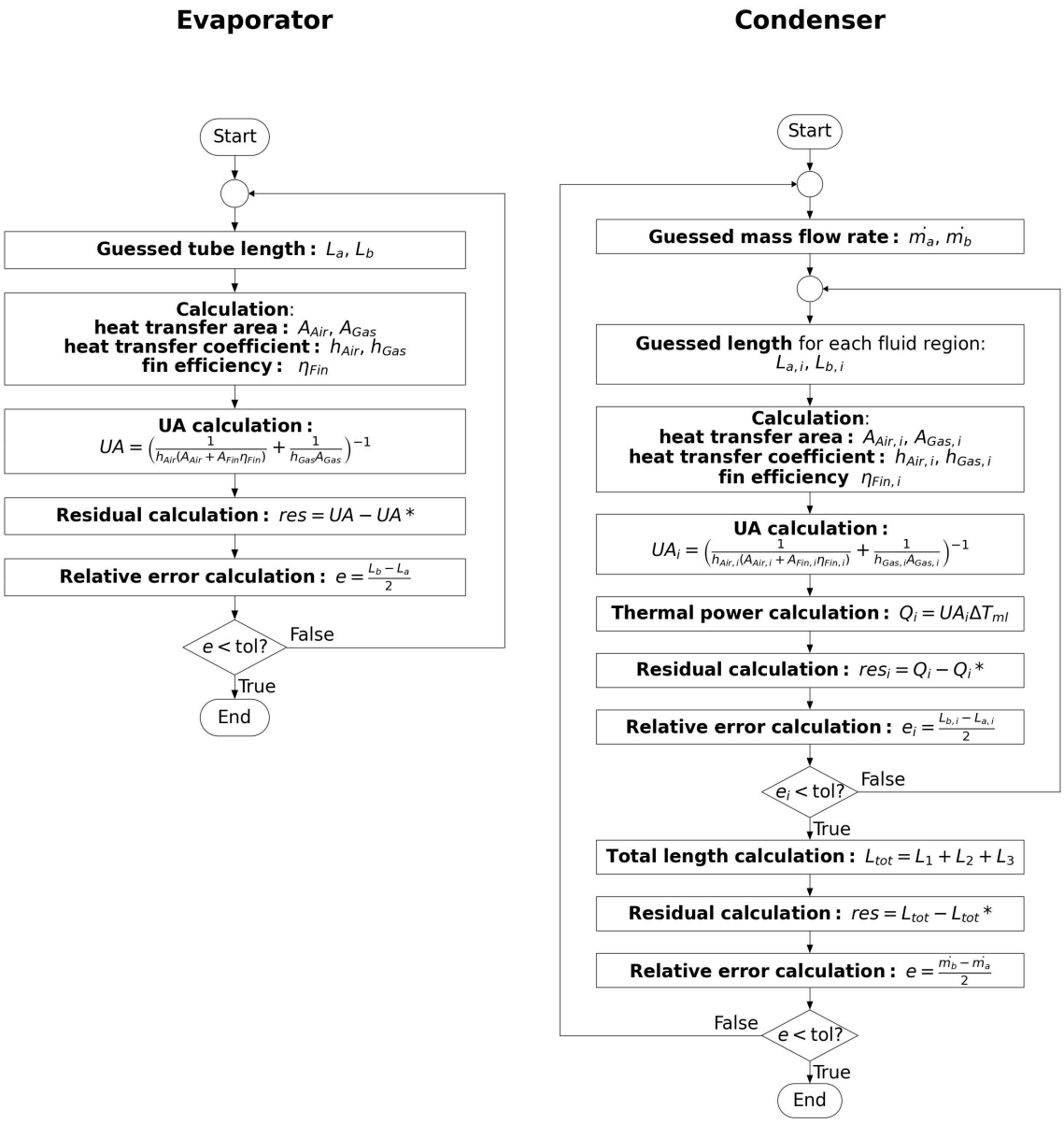

Figure 2: Flowchart representing the algorithms to design the heat exchangers.

the compressor was computed. Subsequently, a parameter space exploration was carried out. In fact, the dynamic model is an instrument to perform a parametric analysis to assess the influence of different parameters on the system performance. At the end of the exploration, the configuration with the minimal annual energy consumption of the compressor is selected. For the proposed case study, the influence of the compressor and of the capillary tube length is analyzed.

3.4 Results and discussion

The current case study is designed according to the presented methodology. Since the proposed workflow is universal for the design of both refrigerators and freezers, only the results obtained for the refrigerated compartment are reported. Fig 3 shows the pressure - specific enthalpy

diagram of the theoretical cycle, the pull-down cycle and the normal operating cycle. The pull-down cycle is larger than the reference one, since the capillary tube is designed for the minimum cooling requirements. Instead, the normal operating cycle is stretched on the enthalpy axis, since the heat exchangers are over-dimensioned for this operating condition. In each case, the superheating and subcooling are guaranteed and the condenser temperature as well as the compressor outlet temperature are acceptable. Following the first preliminary design, a parametric analysis was conducted by testing different compressors. The unit used in the first evaluation was a commercial compressor with a 9 cm^3 displacement. Two smaller compressors, featuring displacements of 7 cm^3 and 5 cm^3, were then evaluated. Since smaller displacements inherently deliver a lower refrigerant mass flow rate at minimum speed, the capillary tube length was recalculated for each configuration to ensure proper matching. As reported in Table 3, the compressor annual energy consumption is 77.05 kWh/year for the largest unit, 75.87 kWh/year for the medium sized compressor and 75.96 kWh/year for the smallest one. These results indicate that upgrading to the medium-sized compressor reduces the annual energy consumption by 1.5%. If the capillary tube length is kept fixed at 3.3 meters, the medium compressor achieves an annual energy consumption of 74.14 kWh/year, thereby increasing the total energy savings to 3.8%.

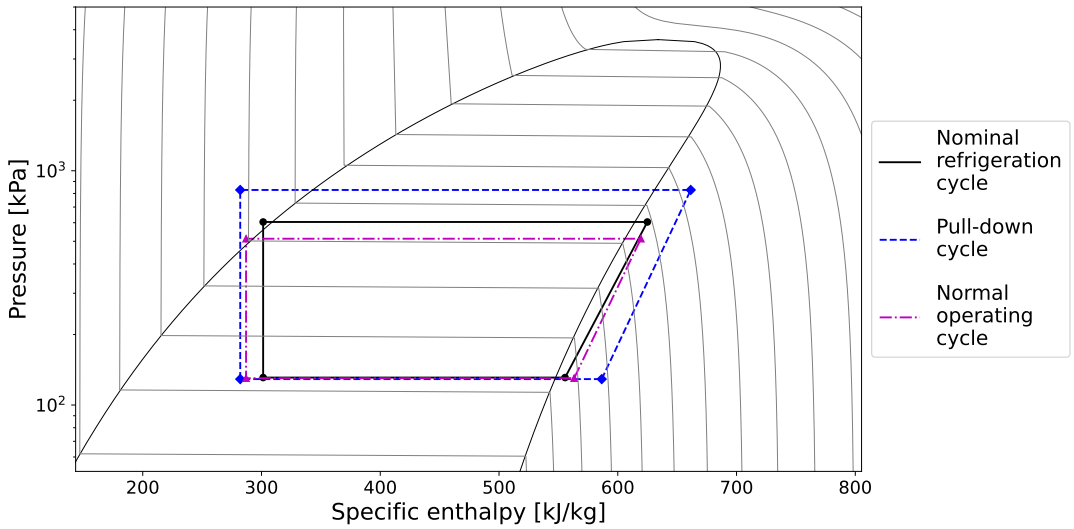

Figure 3: Pressure - specific enthalpy diagram of the thermodynamic cycle.

Table 3: Parametric analysis results: impact of compressor displacement and capillary tube length on the annual energy consumption.

Compressor displacement	**Compressor speed**	**Refrigerant mass flow rate**	**Capillary tube length**	**Annual energy consumption**
9 cm^3	950 rpm	1.35 kg/h	3.3 m	77.05 kWh/year
7 cm^3	950 rpm	1.03 kg/h	5.3 m	75.87 kWh/year
5 cm^3	950 rpm	0.71 kg/h	10.2 m	75.96 kWh/year
7 cm^3	950 rpm	1.03 kg/h	3.3 m	74.14 kWh/year

4 Conclusions

This work presents a physics-based methodology for the design and verification of high-efficiency refrigerators. It guides from a preliminary design to a parameter space exploration. Indeed, the proposed step-by-step procedure effectively bridges the gap between preliminary analytical sizing and dynamic system verification.

In the future, it would be possible to conduct a more comprehensive exploration of the parameter space by analyzing more parameters. Second, the framework could be used to couple the dynamic model with parametric optimization algorithms to automatically identify the minimal annual energy consumption. Finally, experimental validation campaigns on a physical prototype could be conducted to verify the transient capabilities and accuracy of the developed dynamic model.

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