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Simplified heat pump models for energy performance evaluation

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Abstract

Heat pumps are one of the most prominent technologies in enhancing energy efficiency in air conditioning systems. As one of the best technologies to use when the source is of a low-quality energy source, which is usually the case for renewable sources, they rapidly became popular as the demand for sustainable energy sources increased.

There has been a vast study on the working conditions of the heat pumps, more commonly to calculate the best set of temperatures at which the heat pump will work. However, in recent years, the problem of controlling the heat pump systems to achieve better performance was raised, to which numerous solutions are given. One of the most significant and widespread control strategies for heat pumps is using the variable refrigerant flow technology. This idea adds a variable-speed compressor to the typical heat pump cycle, which can adapt to the situation to avoid losses due to the on-off cycles.

The performance of variable refrigerant flow heat pumps, however, is not a straightforward objective to calculate because, most of the time, the heat pump will not work in the design condition, which will result in a system with an oversized heat exchanger. Therefore, most of the approaches to do these calculations are based on numerical and empirical formulas, usually developed by the manufacturer and unavailable to the public user.

This project aims to consider different methods to calculate the performance of a variable refrigerant heat pump in different working conditions. To this end, current methods developed for calculating part-load performance are considered; some modifications and novel ideas were presented, and the possibility of integration into the currently available strategies for calculating heat pump performance in full-load conditions was analyzed.

This project aimed to explore the intricacies of calculating heat pump performance to advance energy-efficient air conditioning systems. The research findings and insights obtained have the potential to enhance the design, operation, and control of variable refrigerant flow heat pumps. This, in turn, can lead to more effective utilization of renewable energy sources and facilitate the transition towards sustainable energy solutions.

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Nomenclature

a	Fitting coefficients, mechanical work	$[various]$
b	Fitting coefficients, condenser heat	$[various]$
c	Fitting coefficients, full-load performance	$[-]$
COP	Coefficient of performance	$[-]$
COP^*	Carnot coefficient of performance	$[-]$
EER	Energy efficiency ratio	$[-]$
EER^*	Carnot energy efficiency ratio	$[-]$
ff	Fluid correction factor	$[-]$
L	Supplied energy for compression	$[kj]$
$\dot{m}$	Mass flow rate	$[kg/s]$
n	Number of data	$[-]$
N	Rotary speed	$[rpm]$
P	Mechanical Power	$[kW]$
PFR	Performance factor ratio	$[-]$
PME	Percent mean error	$[\%]$
plr	Partial-load ratio	$[-]$
Q	Thermal energy	$[kj]$
$RMSE$	Root mean square error	$[\%]$
S	Entropy	$[kj/K]$
T	Temperature	$[K]$
U	Heat exchange coefficient	$[-]$
ΔT	Temperature difference	$[K]$
W	Mechanical energy	$[kj]$
y	Value of a variable	$[various]$
α	Fitting coefficients, part-load performance	$[various]$
γ	Irreversibility ratio	$[-]$
η	efficiency	$[-]$
ω	Rotary speed	$[1/s]$

Subscripts

A	Method A	L	Load side
B	Method B	M	Medium temperature
$comp$	Compressor	mod	Calculated from model
$cond$	Condenser	nom	Nominal
$cooling\ (c)$	Cooling mode	o	Outlet
dec	Declared from catalogue	pl	Part-load
fl	Full-load	plr	Part-load ratio
gen	generated	r	rotational
$heating\ (h)$	Heating mode	S	Source side
i	Inlet	X	Method X
j	The ordinal position of a value		

| Introduction

Heat pump technology has become vital to saving energy and to the environment. By swapping out expensive fossil fuels, lessening the need for costly energy solutions, and optimizing how thermal systems work, heat pumps offer a smart way to integrate renewable energy. They've improved efficiency, using eco-friendly refrigerants and working alongside renewable sources like solar. This makes them useful for homes, businesses, and industries, giving a greener option. This puts them at the center of the effort to make energy cleaner and more sustainable. They are not just for residential homes anymore – heat pumps have expanded to businesses and industries. Their flexibility to fit various energy needs makes them quite versatile.

All these advancements highlight heat pumps' role in shaping a cleaner energy future. As they get even more efficient and environmentally friendly, they will play a more significant part in our journey toward sustainability. [1] In their analysis of the heat pump prospective in the United Kingdom, Teng et al. [2] found that heat pumps hold the transformative potential to reshape the energy landscape, yielding substantial cost reductions pertaining to the assimilation of renewable energy while concurrently curbing carbon emissions. They play an instrumental role in domestic heating, enabling energy-efficient climate control, as well as in industrial processes that demand reliable and consistent thermal management. The efficacy of heat pumps in harnessing and modulating thermal energy stands as a cornerstone of the energy transition. The synergy between heat pump technology and renewable energy sources showcases a promising trajectory toward achieving heightened energy efficiency. Consequently, a reduced ecological footprint and enhanced energy security become attainable goals.

The simplicity of heat pumps made them the primary option for improving energy performance while dealing with heating and cooling. As a result, there has been a noticeable surge in both enthusiasm and motivation for advancing and delving into heat pump technology. This push for progress and exploration in the field has garnered considerable attention.

According to a report from the International Energy Agency [3], the scope of this surge is financially substantial. As of September 2022, approximately 550 billion USD had been earmarked to shield consumers from escalating energy costs. This financial commitment underscores the urgency and significance attributed to tackling the challenges posed by rising energy prices.

Simultaneously, numerous governments worldwide have initiated a strategic approach to propel energy efficiency. In particular, certain governments are encouraging the adoption of energy-efficient solutions like heat pumps. This endeavour has led to the implementation of targeted subsidies aimed at incentivizing the retrofitting of existing structures and the incorporation of heat pump systems. A noteworthy observation is that several European countries are at the forefront of this movement. These nations have adopted policies comprehensively addressing integrating heat pump technologies and energy efficiency measures.

The development of better heat pumps using new technologies and support from society and governments has caused a considerable increase in the heat pump market. A few things are driving this growth. New technologies are making heat pumps work even better, which makes them more attractive for homes and businesses. Governments are also passing rules and giving benefits to encourage the use of eco-friendly energy solutions like heat pumps. This combination of better technology and supportive policies creates a robust market environment.

The heat pump market's growth follows a steep curve, getting steeper each year. [4] The growing demand shows that people care about the environment and want to use energy wisely. More units are being sold each year, and the speed at which this increase is happening is getting faster and faster. In 2022, about 3 million heat pumps were sold in Europe. This was almost 40 % more than the number sold in 2021.

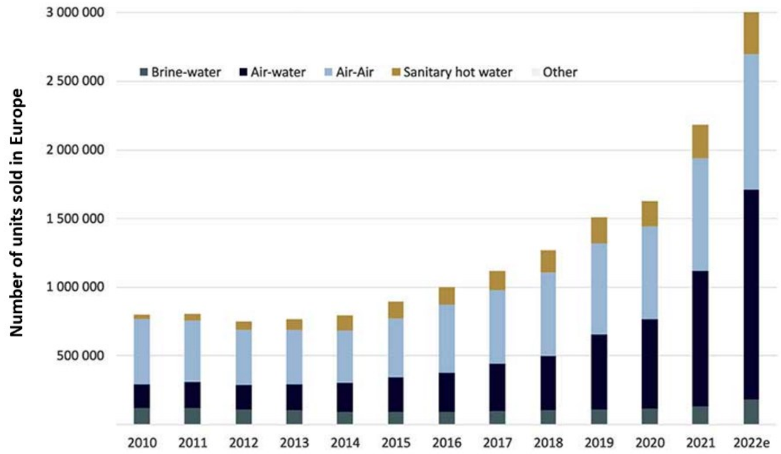

Figure 1 Trend of the market for heat pumps in Europe [4]

This growth is set to increase as Global Market Insight forecasts the European market for heat pumps to have a steady yearly growth of 18 percent until 2032 [5]. This growing market means that heat pumps will probably be the most common method for heating and cooling in Europe in the near future. Nevertheless, the studies and the knowledge of the technology are getting more specific and usually restricted. As a result, most of the studies analyzing heat pumps are focused on the effects and footprints of the heat pumps rather than their performance.

The improvements in how thermal generators work have significantly changed how the energy is being used. These changes mean that energy production is smoother and more efficient, especially when the generators are not working at full capacity. This progress is saving us money in the energy sector. A fascinating result of these changes [6] is the rise of flexible heat pumps. By combining the better performance of thermal generators with these modern heat pumps, we're finding more innovative ways to save money in the energy world.

Regarding backup power, these heat pumps effectively decrease the need for standby systems, reducing associated expenses. This adaptability also extends to renewable energy integration. By efficiently managing energy fluctuations, they enhance the stability of renewable sources and cut down the costs of additional balancing mechanisms. Multiple studies worldwide consistently highlight the noteworthy decrease in carbon emissions achieved through heat pump implementation. These findings emphasize the universality of this positive influence, showcasing heat pumps as versatile tools in our shared quest for sustainable energy solutions [7,8].

The comprehensive analysis of heat pumps mandates an integrated approach that amalgamates technical scrutiny with a profound comprehension of user dynamics. This symbiotic relationship between system functionality and user behaviour was underscored by the research conducted by Patteeuw et al. [9]. Their findings unequivocally demonstrated that user practices can notably influence heat pump efficiency, leading to potential performance deviations of up to 7 %.

The intricate interplay between heat pump operations and user engagement underscores the necessity of crafting models that seamlessly align with the diverse and evolving spectrum of user behaviours—optimizing operational efficiency hinges upon cultivating user-centric models in a landscape characterized by the continual advancement of heat pump technology. These models, designed with a malleable architecture, must adapt to diverse usage patterns and dynamically respond to evolving circumstances. Consequently, the trajectory of heat pump performance is not solely dictated by engineering intricacies but also by the nuanced interplay between user actions and system response.

The imperative to consider user behaviour profiles [10] when investigating heat pumps underscores the necessity for robust analytical approaches to assess the effectiveness of these systems in conjunction with other system components. Achieving this goal involves conceptualizing the heat pump as a simulation program, seamlessly integrated with diverse energy performance calculation methodologies. This holistic approach enables the comprehensive evaluation of the entire system's performance.

Developing a standardized model emerges as a pivotal requirement to realize this integration. This model must accurately forecast heat pump performance across operating conditions. Importantly, this predictive framework should find its foundation in fundamental physical principles, ensuring its reliability and fidelity. Simultaneously, the model must harmonize with the empirical data derived from real-world heat pump operations, thus bridging the gap between theoretical and practical domains.

As the model is designed, it must encapsulate the intricate interplay between the heat pump and its interconnected components, accounting for both the synergies and potential inefficiencies that can arise. The model can be fine-tuned to mimic real-world behaviours and outcomes by aligning with existing statistical information regarding operational heat pumps. This alignment bolsters the model's utility and increases its accuracy in predicting performance scenarios.

An objective is establishing a versatile framework that accommodates diverse user behaviour profiles and operational scenarios. This adaptability empowers the model to accurately forecast heat pump performance and energy consumption under varying conditions, thereby aiding in optimizing energy systems. A comprehensive assessment of the system's overall performance becomes feasible by employing a simulation program that seamlessly interfaces with other energy calculation methods. This holistic approach paves the way for more informed decision-making in designing, implementing, and managing energy-efficient systems centred around heat pumps.

In essence, the convergence of a standardized, physics-based model with empirical data and advanced simulation techniques offers a formidable toolset for understanding and enhancing the performance of heat pump-integrated energy systems. This holistic paradigm transcends the limitations of individual component analysis, enabling researchers and practitioners to comprehensively address the multifaceted challenges posed by real-world complexities and user-centric dynamics.

1 | Basic Concepts

A heat pump is a complex of mechanical components [11] that act as a thermodynamical system that can provide the desired transformation of energy when in need of converting the mechanical work to thermal energy. While being a complicated network of interconnected components, heat pump is based on a simple thermodynamic approach that is quite well-known. The basic cycle, which is sometimes also called "the reverse refrigeration cycle," uses compression as a means to enforce a non-spontaneous transfer of heat from a lower temperature to a higher one. This cycle can be analyzed with the first and second principles of the thermodynamics:

$$Q_h + Q_c + W = 0 \tag{1}$$

$$\frac{Q_h}{T_h} + \frac{Q_c}{T_c} + S_{gen} = 0 \tag{2}$$

In this equation, W is the energy required to compress the refrigerant; in the case of compression heat pump cycles, it is the mechanical power of the compressor. Subscripts h and c correspond to hot and cold sources, respectively; hence, Q_c and W are supplied to the unit, being positive andQ_h is provided by the unit to the hot ambient, which makes it negative.

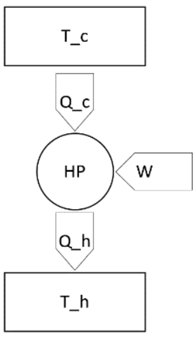

Figure 2 Thermal flows in a heat pump

S_{gen} is the entropy generated in the process due to reversibility, which is always positive. It can be seen, therefore that $|Q_c| > |Q_h|$, also, one can notice that if a heat pump works in both summer and winter conditions, the accumulation of heat in the external source during the summer period and extraction of heat during the winter period can sum up to a near zero value. This concept gives the heat pumps an advantage and opens up the possibility of using the ground as an external source.

To quantify the performance [12] of a heat pump, it is convenient to use the ratio of desired output to the input. The input in a heat pump is the energy given to compress the refrigerant; in the case of a compression heat pump, it is the mechanical work (W). The desired output during the summer period will be Q_c while in the winter period Q_h is the energy that the heat pump is used for. Consequently, for the winter period coefficient of performance (COP) and for the summer period, the energy efficiency ratio (EER) are defined as performance indicators.

$$COP = Q_c/W \tag{3}$$

$$EER = Q_h/W \tag{4}$$

If the heat pump works without irreversibility, the performance factors depend only on the working temperature. In this scripture, these reversible performance factors will be called Carnot COP and Carnot EER, and will be shown as COP^* and EER^* respectively.

$$COP^* = \frac{1}{1 - \frac{T_c}{T_h}} \tag{5}$$

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$$EER^* = \frac{1}{\frac{T_h}{T_c} - 1} \tag{6}$$

In order to account for the irreversibilities, the ratio between the generated entropy and the change in the entropy as a result of exchanged heat is defined as the irreversibility ratio, and it is calculated with the following formulas [12]:

$$\gamma_{heating} = \frac{S_{gen}}{\frac{|Q_h|}{T_h}} = \frac{\frac{\Delta T_h}{T_h}}{1 + \frac{\Delta T_h}{T_h}} + \frac{\frac{U_c \Delta T_c}{T_c}}{\frac{U_h \Delta T_h}{T_h}} \times \frac{\frac{\Delta T_c}{T_c}}{1 - \frac{\Delta T_c}{T_c}} \tag{7}$$

$$\gamma_{cooling} = \frac{S_{gen}}{\frac{|Q_c|}{T_c}} = \frac{\frac{\Delta T_c}{T_c}}{1 - \frac{\Delta T_c}{T_c}} + \frac{\frac{U_h \Delta T_h}{T_h}}{\frac{U_c \Delta T_c}{T_c}} \times \frac{\frac{\Delta T_h}{T_h}}{1 + \frac{\Delta T_h}{T_h}} \tag{8}$$

In these relations, U and ΔT, respectively, refer to the heat exchange coefficient and temperature difference between the working fluid and the heat exchange medium for the corresponding subscript. This factor is related to the system's operation; therefore, it is associated with the different components of the heat pump, the operation condition, and many other possible contributing causes to the irreversibilities. In the equations 9 and 10, the performance indicators are shown to be related to the irreversibility ratio. This dependency on the operating condition renders the idea of having an exact prediction of the heat pump performance impossible.

$$COP = \frac{1}{1 - \frac{T_F}{T_C}\left(1 - \gamma_{heating}\right)} \tag{9}$$

$$EER = \frac{1}{\frac{T_C}{T_F}\left(1 + \gamma_{cooling}\right) - 1} \tag{10}$$

Another important indicator of performance for the thermodynamic cycles is the second principle efficiency, which essentially provides a measure to demonstrate the effect of irreversibilities on the performance of the system, and it is calculated as follows:

$$\eta_{2P} = \begin{cases} \frac{COP}{COP^*} & heating \\ \\ \frac{EER}{EER^*} & cooling \end{cases} \tag{11}$$

1.1 Main Components of a Heat Pump

The heat pump cycle consists of four primary components. As mentioned earlier, the main energy input is utilized to raise the pressure of the working fluid. In compression heat pumps, this function is carried out by a compressor. The compressor adjusts the pressure of the working fluid to compensate for the pressure drop that occurs between the evaporator and the condenser. This ensures that the working fluid maintains the appropriate temperature for the necessary phase change, which is essential for efficient heat exchange with external sources. This orchestrated process allows the heat pump cycle to effectively achieve its objective of transferring heat from a lower-temperature area to a higher-temperature one. This technology has widespread applications, including climate control and heating systems [12].

The essential task of reducing pressure to achieve the desired evaporator state is accomplished through the use of an irreversible expansion valve. While it's theoretically possible to optimize energy utilization by replacing the expansion valve with a turbine, practical constraints such as high costs and complex integration often make this approach less viable. In real-world applications, deciding to incorporate a turbine requires careful consideration due to the need for precise engineering, maintenance, and financial investment. As a result, practicality often favours sticking with expansion valves to strike a balance between operational efficiency and feasibility. This approach effectively balances practicality and performance, catering to various industries and sectors where precise temperature control and enhanced energy efficiency are crucial concerns [11].

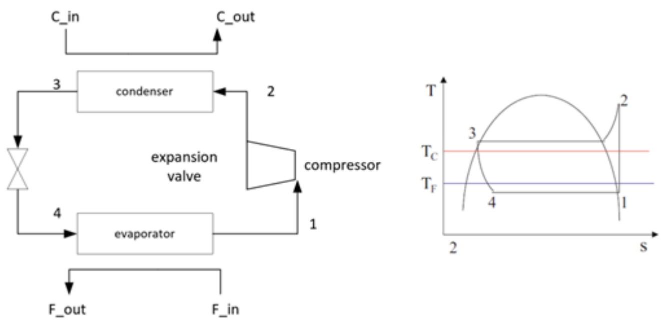

Figure 3 basic schema of heat pump cycle and the corresponding T-s diagram

The operation of the heat pump involves a meticulous exchange of thermal energy across two distinct interfaces, namely, an external heat source and an external heat sink. At the core of this intricate thermal exchange process lies the condenser unit, a pivotal component responsible for effecting the transformation of the gaseous outlet from the compressor into a condensed liquid state. This transformation is accompanied by a degree of subcooling to optimize thermal efficiency. Within this framework, the condenser plays a paramount role in furnishing the requisite heat to the system.

Conversely, the evaporator unit functions in a contrasting capacity, receiving a saturated mixture originating from the expansion valve. This mixture, characterized by its thermal properties, undergoes a heat exchange operation with the ambient surroundings. The outcome of this exchange manifests as the transition of the mixture into a vapour state, which may either exist in a saturated condition for instances involving wet evaporators or be subject to superheating in the context of dry evaporators. It is noteworthy that in cases where the evaporator's outlet exhibits a saturated phase, the implementation of a separator along the pathway to the compressor becomes essential to ensure the preservation of operational integrity.

In heat pump systems designed for both heating and cooling applications, a specialized component known as a 4-way valve plays a pivotal role in facilitating the transition between these two operational modes. This valve serves as a dynamic conduit for redirecting the compressed gas leaving the compressor, thereby enabling the heat exchangers to function either as condensers or evaporators, depending on the prevailing thermal requirements.

During the heating phase, the 4-way valve positions itself to direct the hot, compressed gas to the indoor heat exchanger. In this configuration, the indoor heat exchanger functions as the condenser, releasing heat into the indoor environment and effectively warming the designated space. Conversely, when the system shifts into cooling mode, the 4-way valve adjusts to route the compressed gas toward the outdoor heat exchanger. In this context, the outdoor heat exchanger takes on the role of the condenser, expelling heat to the external surroundings and producing a cooling effect within the indoor area [13].

1.1.1 Compressor

The component is to provide the pressure for the heat pump cycle to complete as required. [11] While two main types of compressors are axial (or reciprocating) and rotary, embracing technological progress in heating and cooling applications, the latter is usually used in heat pump applications. This is mainly because of their size, silence and smooth operation. They can be roughly categorized into compressors featuring a lone rotating axis, such as vane and scroll compressors, and those equipped with dual rotating axes. The term “rotary compressor” encompasses vane, lobe, screw, and scroll compressors. Vane and scroll compressors operate with a solitary shaft, constituting a single rotational axis. In contrast, the remaining types can possess either one or two axes. Compressors can also be divided based on their structure as follows:

- Open compressor: The driving motor and the compressor are independent air cooling entities linked through a mechanical connection.
- Hermetic compressor: Both the motor and the compressor reside within the same enclosure, and the motor's cooling is facilitated by the same fluid circulating within the compressor.
- Semi-hermetic compressor: This compressor is directly connected to the driving motor, contained within the same casing, yet the motor distinguishes a separate access point.

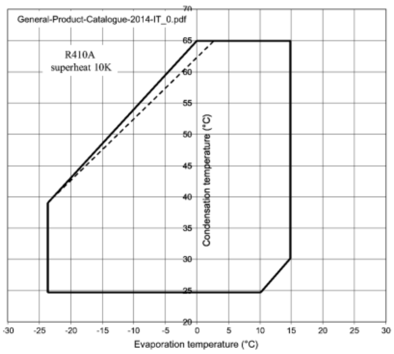

Figure 4 Operation range of a Copeland scroll compressor

For each specific refrigerant, every compressor is allocated a distinct section within the evaporation and condensation temperature plane (such as Figure 4). This demarcated area represents the operational scope of the compressor. Beyond this defined range, manufacturers do not assure the compressor's performance.

1.1.2 Heat Exchangers

At the core of heat pump systems are heat exchangers, which facilitate the transfer of heat between the working fluid and the surrounding environment. Heat exchangers are particularly important in the roles of evaporators and condensers.

The evaporator is a vital component in a heat pump system as it serves to absorb heat from the source medium, which could be air, water, or ground. During the evaporation process, the working fluid, typically a refrigerant, absorbs heat from the external environment and undergoes a phase change from a liquid to a vapour state.

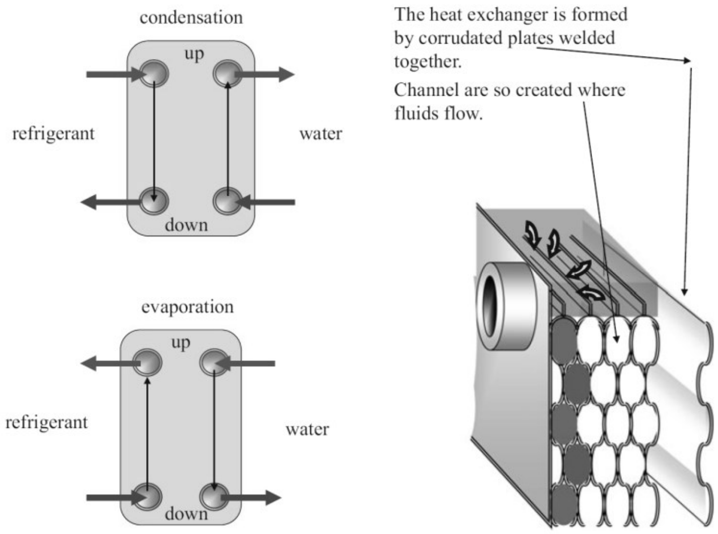

Figure 5 Scheme of a plate heat exchanger [11]

Designing efficient evaporators involves optimizing surface area, flow rates, and material properties. The increased surface area allows for more efficient heat transfer, while suitable flow rates ensure that the working fluid can absorb heat effectively without causing uneven distribution. The choice of materials is crucial, considering factors such as corrosion resistance and thermal conductivity to maintain system longevity and performance [14].

Condensers, another critical component of heat pumps, are responsible for releasing heat to the sink medium. In the condensation process, the refrigerant vapour releases heat, undergoing a phase change back to the liquid state. This heat transfer process is essential for maintaining the cycle of a heat pump system. The design of condensers, like that of evaporators, requires careful consideration of surface area, flow dynamics, and material selection.

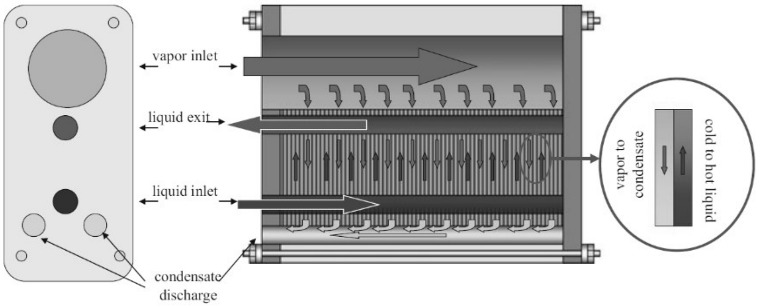

Figure 6 Scheme of a plate condenser [14]

There are various types of heat exchangers used in heat pump systems, including shell-and-tube, plate, and finned-tube heat exchangers. Each type has its advantages and disadvantages, impacting factors such as heat transfer efficiency, pressure drop, and compactness. The selection of the appropriate heat exchanger type depends on the specific application, space constraints, and overall system requirements [11].

Heat exchangers play a pivotal role in the operation of heat pump systems, serving as evaporators and condensers. They facilitate the transfer of heat between the working fluid and the surrounding environment, enabling energy-efficient heating and cooling solutions. Designing effective heat exchangers involves careful consideration of factors such as surface area, flow rates, and material properties. As society continues to emphasize energy conservation and environmental responsibility, the importance of heat exchangers in heat pump technology becomes increasingly evident.

Frost accumulation [15] on heat exchangers can significantly impact the efficiency and performance of heat pump systems. When operating in cold climates, heat exchangers are prone to frosting due to the moisture present in the surrounding air. Frost accumulation on heat exchangers leads to several detrimental effects that compromise the operation of heat pumps:

- Reduced Heat Transfer Efficiency: Frost acts as an insulating layer that hinders the efficient transfer of heat between the working fluid and the surrounding medium. This reduces the overall heat exchange rate, leading to decreased heating or cooling capacity of the heat pump system.
- Increased Energy Consumption: As frost builds up on the heat exchanger surfaces, the heat pump system requires more energy to overcome the insulating effect of the frost layer. This increased energy consumption not only affects the

operational costs but also undermines the energy-efficient nature of the heat pump.

- Shortened Equipment Lifespan: Frost formation can cause mechanical stress on heat exchanger components due to the expansion and contraction of the ice layer. This repetitive stress can lead to premature wear and tear, potentially shortening the lifespan of the equipment.

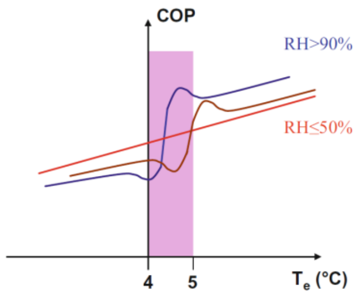

Figure 7 Effect of frosting on COP of heat pumps [15]

There are multiple strategies for solving the issue of frosting that occurs both outside and inside the tubes:

- Defrosting Cycles: Incorporating regular defrosting cycles into the heat pump system can prevent excessive frost buildup. There are two common methods for defrosting: air defrost, and reverse-cycle defrost [11]. In air defrost, the compressor is temporarily turned off while the evaporator’s fan continues working, melting the frost by the stream of air using the warm refrigerant to melt the frost. Reverse-cycle defrost involves reversing the cycle temporarily, making the condenser function as an evaporator to remove frost [15].

- Finned Surfaces: Designing heat exchangers with finned surfaces can help mitigate the effects of frosting. Fins increase the surface area available for heat transfer, which can discourage frost accumulation by promoting more efficient heat exchange. However, finned surfaces might require additional maintenance to prevent clogging.

- Hot Gas Defrosting: This method involves using a portion of the system's hot refrigerant gas to melt the frost. By diverting hot gas through the evaporator coils, the ice layer is gradually melted, and the heat exchanger is restored to its efficient operation.

- Electric Resistance Heating: Incorporating electric resistance heating elements within the heat exchanger can provide a localized heat source to melt the frost. While effective, this method increases energy consumption and should be used judiciously.

Frost accumulation on heat exchangers presents significant challenges to the efficiency and performance of heat pump systems. The effects of reduced heat transfer efficiency, increased energy consumption, and potential equipment damage underscore the need for effective frost mitigation strategies. Implementing techniques such as defrosting cycles, using finned surfaces, employing hot gas defrosting, and electric resistance heating can help maintain optimal heat exchanger performance in cold climates. Addressing the issue of frost accumulation is crucial to ensuring the continued effectiveness and sustainability of heat pump technology in various environmental conditions [15].

1.1.3 Expansion Valve

The expansion valve, which governs the refrigerant flow from high to low pressure, holds a crucial role in this cycle [11]. Various types of expansion valves are utilized in heat pump systems.

- Capillary Tubes: Capillary tubes are simple, cost-effective devices with narrow, lengthy tubes that restrict refrigerant flow based on capillary action. The refrigerant undergoes adiabatic expansion as it passes through the capillary, causing a drop in pressure. This pressure reduction facilitates the transition from liquid to the vapour state.

- Thermostatic Expansion Valves: Thermostatic Expansion Valves regulate refrigerant flow by maintaining a constant superheat at the evaporator outlet. A temperature-sensitive bulb senses the superheat and adjusts the valve's opening accordingly. This ensures that only superheated vapor reaches the compressor, preventing liquid floodback.

- Electronic Expansion Valves: Electronic Expansion Valves employ electronic controls to regulate refrigerant flow precisely. Sensors monitor various parameters such as evaporator and condenser pressures, temperatures, and compressor speed. These data are used to optimize valve opening, enhancing system efficiency.

- Fixed Orifice Valves: Fixed orifice valves, including fixed-size orifice and piston valves, offer simplicity in design but lack the adaptability of other valve types. They maintain a constant opening size, which may not be suitable for varying heat load conditions.

The choice of expansion valve depends on factors such as system size, load variations, and required precision. Thermostatic expansion valves are suitable for applications with varying heat loads, as they maintain a consistent superheat. Electronic expansion valves provide high precision and adaptability, making them ideal for complex systems, but they may be costlier. Capillary tubes are used in small, simple systems where precise control is less critical.

Expansion valves in heat pumps are vital components that significantly impact system efficiency and performance. Different types of valves offer distinct advantages and disadvantages based on application requirements. The appropriate selection of an expansion valve is crucial to ensure optimal heat pump operation. By understanding the equations, principles, and operational characteristics of various expansion valves, engineers and researchers can make informed decisions when designing and implementing heat pump systems.

1.2 Types of Heat Pumps

The heat pumps can be categorized based on the external sources with which they exchange heat [16]:

- Air-Source Heat Pumps (ASHPs): ASHPs extract heat from the outdoor air and transfer it indoors for space heating or cooling purposes. The performance of ASHPs is affected by outdoor temperature, making them less efficient in colder climates.
- Ground-Source Heat Pumps (GSHPs): GSHPs utilize the relatively constant temperature of the ground to provide heating and cooling. They achieve higher efficiencies compared to ASHPs due to the more stable temperature of the ground. GSHPs use vertical or horizontal ground loops to exchange heat with the earth.
- Water-Source Heat Pumps (WSHPs): WSHPs employ water bodies, such as lakes or rivers, as heat sources or sinks. They offer similar benefits to GSHPs but require access to water bodies.
- Air-to-Water Heat Pumps: Air-to-water heat pumps extract heat from the outdoor air and transfer it to a water-based heating system. They are commonly used for underfloor heating or radiators.
- Water-to-Air Heat Pumps: Water-to-air heat pumps extract heat from a water source, such as a well or lake, and transfer it to an air-based heating or cooling system. They are used in conjunction with forced air systems for both heating and cooling.

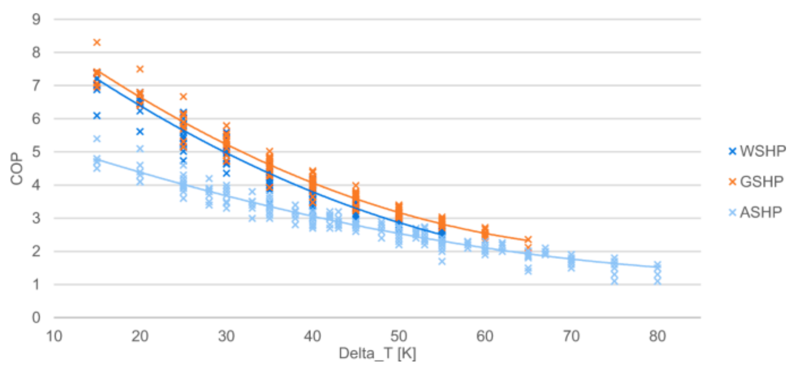

Figure 8 performance of different types of heat pumps with respect to the temperature difference between source and sink [16]

GSHPs exhibit the highest efficiency due to the stable ground temperature, while ASHPs are less efficient in colder regions. Air-to-water and water-to-air heat pumps have efficiencies similar to ASHPs, while WSHPs offer efficiency comparable to GSHPs. GSHPs tend to have higher upfront costs due to ground loop installation. ASHPs, air-to-water, and water-to-air heat pumps are more affordable. WSHPs may have moderate installation costs depending on water source availability. GSHPs and WSHPs have a smaller environmental footprint as they rely on renewable heat sources. ASHPs, air-to-water, and water-to-air heat pumps can still be more environmentally friendly than traditional methods. ASHPs, air-to-water, and water-to-air heat pumps are versatile and suitable for retrofitting. GSHPs and WSHPs are better suited for new constructions or major renovations due to their loop requirements.

The selection of a specific heat pump type depends on factors like climate, available resources, and budget constraints. The equations and figures provided in this essay enhance understanding, aiding informed decisions for sustainable heating and cooling solutions.

1.3 Heat Pump Operating Conditions

The correlation between a building's heating load and the external temperature can be represented by a linear relationship characterized by a negative slope. The capacity of a heat pump can be approximated using the graphs that correlate -linearly- the demand with the ambient temperature (as can be seen in figure 10).

It is worth reiterating that heat pumps commonly employ a volumetric compressor, leading to a dependency of their mass flow rate on their volumetric flow rate (which is tied to rotation speed) and fluid density. [17] This fluid density increases as the outdoor temperature rises due to the elevation of evaporation temperature and the associated temperature rise.

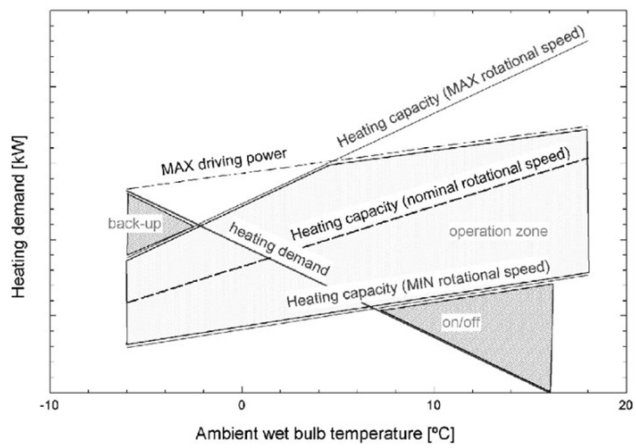

Figure 9 Heating demand and Capacity of a variable rotational speed heat pump in winter conditions [17]

Consequently, the supplied power by the heat pump escalates and exhibits an upward trajectory in relation to the outdoor temperature. Furthermore, the Coefficient of Performance (COP) experiences enhancement as the temperature of heat sources approaches one another.

During the summer season, the demand for cooling within a building amplifies in tandem with the external temperature and solar radiation. Concurrently, the temperatures of the heat sources diverge while maintaining a constant indoor temperature. This combination of factors results in a decline in cooling capacity and a reduction in the Energy Efficiency Ratio (EER). Consequently, an antithetical pattern is observed between the two graphs, one representing the required power and the other illustrating the supplied power.

The point of equilibrium, denoted as the balance point, emerges when the user's power requirement aligns with the heat pump's capacity. In the winter months, the heat pump's power output to the right of the balance point progressively surpasses the user's demand.

Examining the trend of outdoor temperatures at the desired location is imperative. The frequency of occurrence must be determined for each temperature value within the specified timeframe [18].

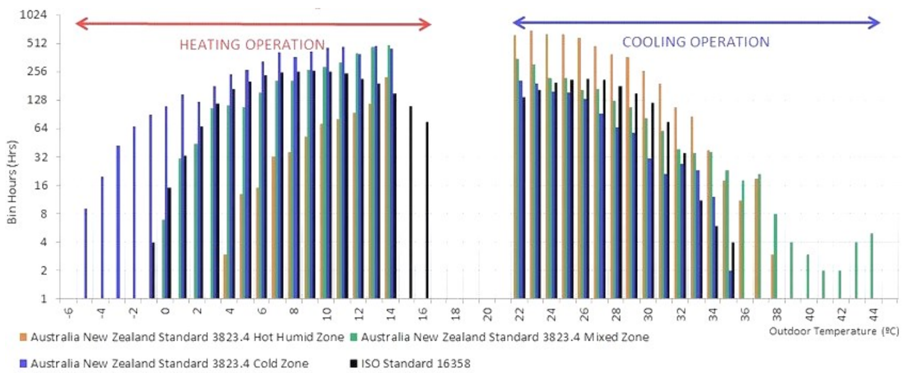

Figure 10 Distribution of bin temperatures for different climates

To comprehensively evaluate the efficiency of heat pumps, it's crucial to analyze the behaviour during partial-load operation. Manufacturers should supply various types of information [11], including:

- Heat pump type, indicating if it's reversible for both heating and cooling.
- Intended usage, such as only heating, heating and cooling, combined heating and domestic water, or only domestic water.
- Operation mode.
- Heat pump thermal sources.
- Moreover, manufacturers need to present the full-load performance data at different source temperatures (as per EN 14825 [19] and, for Italy, TS 11300 [20]). These standards outline the specific conditions.
- Additionally, the performance under a climatic part-load ratio (plr - the capacity of the heat pump divided by its full-load capacity) different from 1 should be provided. plr is expected to approach zero when the outdoor temperature reaches 16°C.

1.4 Control Methods for Heat Pump Operation

Heat pumps are equipped with control systems for multifaceted reasons, with efficiency optimization being paramount. The control system's ability to fine-tune variables such as compressor speed and refrigerant flow allows the heat pump to operate at its peak efficiency, leading to significant energy savings and cost reduction. Beyond efficiency, these systems play a pivotal role in temperature regulation, enabling the heat pump to maintain a consistent indoor climate by aligning its operations with set temperature parameters and prevailing environmental conditions. Load matching is another critical aspect facilitated by control systems, ensuring that the heat pump's output aligns precisely with the heating or cooling demands of the space, thus avoiding wasteful energy consumption resulting from oversized or undersized operation.

The compressor is a critical component in the functioning of a heat pump system. It raises the pressure and temperature of the refrigerant gas, allowing efficient heat transfer from lower temperature areas to higher temperature regions. [11] Consequently, regulating the compressor's operation is essential for maintaining the desired temperature conditions and optimizing the heat pump's performance.

The control systems keep a close watch on various factors like indoor and outdoor temperatures, the state of the refrigerant, and the need for heating or cooling. Using this information, the control system adjusts how the compressor operates. This could involve changing its speed, cycling patterns, or whether it's turned on or off. These adjustments are aimed at achieving the desired temperature output in the most efficient manner. This careful management of the compressor's actions not only helps save energy but also enhances comfort levels and overall heat pump efficiency. In the following pages, some of the control strategies are described.

1.4.1 On-off Method

The on-off control method is relatively simple and cost-effective to implement, making it a popular choice for controlling heat pumps in various settings. When the thermostat detects a deviation from the desired temperature, the heat pump switches on until the set temperature is reached. At this point, the heat pump turns off until the temperature deviates again. [21] This binary operation results in less frequent cycling compared to continuous operation, which can lead to reduced energy consumption, especially in moderate climates where the temperature fluctuations are within a narrow range.

However, this simple method has its limitations. The on-off method may lead to noticeable temperature fluctuations within the controlled space. Since the heat pump is deactivated when the desired temperature is achieved, there might be slight variations in temperature before the heat pump cycles on again [11]. This can result in minor discomfort for occupants, especially in regions with extreme temperature changes. In climates with significant temperature variations, the on-off control method may not be as efficient. Frequent cycling during rapid temperature changes can offset the energy savings achieved during longer inactive periods [22].

If the heat pump experiences short cycling (rapid on-off cycles) due to frequent and small temperature deviations, it can lead to decreased efficiency and increased wear on the components, potentially undermining the benefits of the on-off control method. This method is most suitable for maintaining a relatively stable temperature range. It may not be the optimal choice for applications requiring precise temperature control, such as in industrial processes or scientific laboratories. The on-off method remains a valuable tool in the spectrum of heat pump control strategies, contributing to energy efficiency and occupant comfort when employed appropriately [23].

1.4.2 Step-wise Method

Step-wise compressor control involves adjusting the capacity of the compressor in predefined increments rather than continuously modulating its speed or operation. This control strategy is based on the principle that different thermal loads require varying levels of cooling or heating capacity. By adjusting the compressor's output in steps, the heat pump can closely match the required load without constant cycling, leading to reduced energy consumption and wear on the components [11].

This adjustment has to be done depending on the compressor type. As in a reciprocating compressor, it means bypassing the flow around some of the pistons [24]. As a result, some of the cranks, despite moving (and consuming mechanical work), do not produce any output and decrease the capacity of the compressor, hence the heat pump. In screw compressors, the control is operated using a servomotor linked to a slide which can control the flow rate into the compressor. In such compressors, there are usually four levels of operation, namely $10, 50, 75,$ and 100 percent of the full-load capacity. For the scroll compressors, this operation is done using a modulation algorithm that can move the scrolls with respect to each other, resulting in full or zero capacity [11]. An adjustment over the ratio of the time the compressor is working in full capacity over a full period can result in a modulated operation of the compressor based on the demand [11].

This method, while being costly, can result in a significant performance improvement in the heat pumps as it lowers the temperature fluctuations with respect to the on-off method. Also, there will be no need to switch the compressor on and off again, which means the wearing process will be extremely belated. Also, the determination of optimal steps is an engineering problem that can be challenging for the design of this system. [24]

1.4.3 Variable Refrigerant Flow

An inverter permits the alteration of the rotation speed of the compressor, enabling the adjustment of flow rate while keeping the compression ratio, volumetric efficiency, and isentropic efficiency constant. The inverter, employed with alternating current (a.c.) electric motors, transforms incoming grid a.c. Into direct current (d.c.) voltage. It subsequently produces electrical pulses as output, mimicking a.c. voltage with varying amplitude and frequency. The magnitude of this simulated voltage is controlled by adjusting the pulse amplitude through Pulse Width Modulation (PWM) at a consistent frequency. To modify the frequency of the simulated voltage, the frequency of the pulses is changed, thereby influencing the number of compressor rotations per second. Typically, maintaining a consistent magnetic flux is essential to prevent the iron core from becoming magnetically saturated. This saturation could lead to the rise of unwanted currents, resulting in overheating. In hermetic compressors, this overheating could extend to the refrigerant. To achieve this, a voltage linked to the frequency needs to be employed. As the frequency increases, the power also increases in a linear manner. The maximum attainable voltage corresponds to the voltage supplied by the electrical grid [11].

$$W_{comp} = a_0 \omega_r + a_1 \tag{12}$$

The utilization of an inverter imparts a noteworthy increase in both instantaneous and seasonal COP and EER. With an inverter in place, the heat pump can dynamically adjust its output to match the current heating or cooling demands. This adaptability ensures that the heat pump operates closer to its optimal efficiency. This enhancement translates to energy savings and reduced operating costs for the end user, a factor of increasing significance in today's energy-conscious world. [25]

An intriguing aspect of inverter-driven heat pumps is their capability to yield performance increases beyond their nominal values, even when operating at reduced flow rates. This phenomenon arises due to oversized heat transfer surfaces in comparison to the heat pump's design nominal conditions. This approach results in a reduction of temperature discrepancies among various heat exchangers and thermal sources within the system. By mitigating these temperature differences, the heat pump can maintain more consistent and efficient heat transfer, thus contributing to better performance.

The speed of the compressor is usually controlled with a variable-speed drive; the control system measures the inside temperature and changes the speed as an output. As a result, the heat exchange with the conditioned ambient can be modified. This means a smooth variation of the temperature and a low deviation from the reference point. It is important to remember that the only parameter that is controlled is the speed of the compressor, which will affect the mechanical output of the compressor, hence the heat exchange with the room. [26]

The control functions for this system affect the transient behaviour of the system. In this study, the model is considered to be a quasi-steady state and the transient behaviour is not accounted for. The variable-speed heat pumps provide better comfort and also, with their ability to operate at lower speeds for extended periods, experience less wear on components, leading to longer system lifespans and reduced maintenance costs. The abrupt starts and stops, which can generate noise in fixed-speed systems, are minimized in inverter heat pumps, contributing to a more serene indoor environment.

2 | Variable-speed Heat Pump Models

Given the significance of heat pumps in contemporary heating and cooling solutions, there arises a necessity for a simulation approach to effectively model these technological processes, particularly when addressing the configuration of appropriate systems. Despite the fundamental thermodynamic simplicity inherent to the technology, accurately forecasting the performance of a heat pump becomes exceedingly challenging when it operates beyond its standard conditions. Research efforts have thus been dedicated to formulating models that pertain to the operation of heat pumps under non-standard conditions, with a primary focus on systems employing on-off control mechanisms.

Fundamentally, the models developed thus far have been constructed either on theoretical principles, empirical data, or a combination of both. It's important to highlight that relying on empirical data yields more precise outcomes while demanding less computational resources. However, this approach necessitates access to data that might not be readily accessible beforehand. Conversely, the theoretical approach typically depends on certain parameters that are not easily obtainable.

An illustration of this can be seen in a study by Bordignon et al. [22], which focused on cascade on-off heat pumps. This research aimed to propose a TRNSYS model for the heat pump. The foundation of this study, which serves as the basis for the current thesis, rested on theoretical relationships between the components of the heat pump, including the utilization of the polynomial method. This particular method hinges on polynomial factors provided by the manufacturer, which vary for each individual system.

Another instance comes from the work of Browne and Bansal [27], who put forth an analytical transient modelling approach for vapour-compression chillers. This strategy relies on understanding system geometry factors such as pipe length and diameters, information that is typically not readily available to commercial users.

In recent periods, there has been a heightened focus on developing simulation techniques tailored to variable-speed heat pumps. Notably, there exist certain suggested models in this regard. For instance, the work of Kellang et al. [28] introduces a simulation model catering to solar-assisted heat pumps, while Del Col et al. [29] presents a model designed for ground-source heat pumps. Both of these models heavily depend on datasets provided by manufacturers.

Conversely, in a distinct approach, Liu et al. [30] employed a purely analytical technique within their investigation of multivariable control within vapour compression cycles. Their study sought to explore potential control strategies, thus targeting a manufacturer-based audience. Consequently, it relied on parameters like the length of the two-phase section in heat exchangers, which remain beyond the reach of typical commercial users. A comparable situation is evident in the analysis conducted by Xue and Lin [31], whose model aims to predict the performance of heat pumps using a set of equations, variables of which are not easily obtainable.

To address this issue, some research has been drawn upon insights from previous studies [22] and developed polynomial equations to approximate the power usage of the compressor. These equations hinge on the compressor's rotational speed and operating conditions, offering not only a more explicit understanding but also simplifying the computation of the heat pump's overall efficiency. Alongside assessing compressor power, the study also explored the correlation between heat pump capacity and temperature. Typically, this correlation is represented by a 9th-degree polynomial equation. Nevertheless, the intricate nature of such high-degree polynomials might impede their practical utility for ordinary users seeking real-world applications of the findings.

The first model that was developed for a variable-speed heat pump was proposed by researchers at Giorgia Tech; this model was based on comparing the COP of the variable-speed heat pump to the corresponding value for the fixed-speed heat pump at a certain "bin" temperature. [32]. This method, essential for various approaches in the field, plays a crucial role in evaluating the efficiency of variable-speed heat pumps, contributing to a more sustainable future. Among these approaches, one stands out for its significance and impact.

In order to establish universal methods for calculating heat pump performance, certain standards have been developed [19,33]. These standards revolve around the basis of the constancy of the 2nd principle of efficiency across different working conditions. Adhering to this principle, it is possible to apply an algorithm to determine COP and EER of the heat pump system. The performance factor is first calculated at the nominal condition, and then the 2nd principle efficiency is determined, enabling consistent utilization of this value throughout the model. When examining full-load conditions, researchers have not only relied on the established standards but have also presented a comprehensive study and simulation model for two different types of heat pumps, namely single-stage and cascade-cycle heat pumps. Additionally, another study by Bordignon et al. [34] compared various models capable of predicting heat pump full-load performance based on working temperature, utilizing available data from catalogues. These models can be adapted based on the amount of available information, such as the number of data points, and have demonstrated significant accuracy in predicting the system's heat extraction and coefficient of performance. The several proposed models are analyzed regarding the amount of available information and data points. Based on this study, some of the simplified models can be used to calculate the full-load performance of the system.

The process adapted in this study is combining the full-load models with part-load correction factors. First, the performance factor at full-load condition is predicted for different operating conditions, and then a correction factor for part-load condition is multiplied, hence arriving at a prediction for the performance of the heat pump at each possible operating condition.

2.1. Full-Load Operation

The analysis of the operation performance of a heat pump working in the full-load condition has been vastly studied, and several models have been proposed to predict it. The models which are introduced in this chapter use the available data to predict performance factors. A summary of these models can be found in tables 1-4.

Model	Proposed formula	Required variables	Minimum required number of datapoints
wh01	$COP = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 (T_{Lo} - T_{Si}) + c_4 (T_{Lo} - T_{Si})^2$	$T_{Si}, T_{Lo}, \dot{m}_S$	5
wh02	$COP = c_0 + c_1 (T_{Lo} - T_{Si}) + c_2 (T_{Lo} - T_{Si})^2$	T_{Si}, T_{Lo}	3
wh04	$COP = c_0 \exp(c_1 T_{So} + c_2 T_{Li}) + c_3 \frac{T_{So}}{T_{Li}} + c_4$	T_{So}, T_{Li}	5
wh05	$COP = c_0 \exp(c_1 T_{Si} + c_2 T_{Lo}) + c_3 \frac{T_{Si}}{T_{Lo}} + c_4$	T_{Si}, T_{Lo}	5
wh06	$COP = c_0 + c_1 T_{Si} + c_2 T_{Li} + c_3 (T_{Si} T_{Li})$	T_{Si}, T_{Li}	4
wh07	$COP = c_0 + c_1 T_{Si} + c_2 T_{Lo} + c_3 (T_{Si} T_{Lo})$	T_{Si}, T_{Lo}	4
wh08	$COP = \eta_{2P} COP^* (\Delta T_{min,c-h} = 10°C)$	η_{2P}	1
carnot	$COP = \eta_{2P} COP^*$	η_{2P}	1
wh09	$COP = c_0 + c_1 T_{Lo} + c_2 (T_{Lo} - T_{Si}) + c_3 (T_{Lo} - T_{Si})^2$	T_{Si}, T_{Lo}	4
wh11	$COP = c_0 + c_1 (T_{Li} - T_{Si}) + c_2 (T_{Li} - T_{Si})^2$	T_{Si}, T_{Li}	3

Table 1 Simplified models for full-load prediction of water-to-water heat pumps in heating mode

The simplest method for this measure is to assume the constancy of the second principle efficiency. A model that is also provisioned in the methods of calculations given in EN-14825, ISO-13256 [19,35] and ISO-13612 [33]. This assumption is because the second principle efficiency depends on the irreversibility factor, which in turn does not go through significant changes in low temperature variations.

In the calculation of COP^*, the temperatures are the entering secondary fluid temperature in the load and source side. However, if the load and source side temperatures are too close, the resulting COP will be hugely overestimated. Therefore a specific setpoint difference is considered (in this study 10°C), and if the difference of temperature exceeds it, then the lower temperature will be modified based on the setpoint difference. In this study, the nomenclature used in the paper by Bordignon et al. is adapted, and this method is named wh08, wc08, ah07 and ac05. In this nomination, the first letter denotes the type of heat pump (w for water-to-water, a for water-to-air), and the second letter distinguishes between heating (h) and cooling (c) modes. In this study, a model (denoted simply by "$carnot$") is analyzed that does not have the setpoint difference temperature, and as a result, the predicted performance factor can get to large numbers when the difference between source and sink temperature is low.

Model	Proposed formula	Required variables	Minimum required number of datapoints
wc01	$EER = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 (T_{Lo} - T_{Si}) + c_4 (T_{Lo} - T_{Si})^2$	$T_{Si}, T_{Lo}, \dot{m}_S$	5
wc02	$EER = c_0 + c_1 (T_{Lo} - T_{Si}) + c_2 (T_{Lo} - T_{Si})^2$	T_{Si}, T_{Lo}	3
wc03	$EER = c_0 \exp(c_1 T_{So} + c_2 T_{Li}) + c_3 \frac{T_{So}}{T_{Li}} + c_4$	T_{So}, T_{Li}	5
wc04	$EER = c_0 \exp(c_1 T_{Si} + c_2 T_{Lo}) + c_3 \frac{T_{Si}}{T_{Lo}} + c_4$	T_{Si}, T_{Lo}	5
wc05	$EER = c_0 \exp(c_1 T_{So} + c_2 T_{Lo}) + c_3 \frac{T_{So}}{T_{Lo}} + c_4$	T_{So}, T_{Lo}	5
wc06	$EER = c_0 + c_1 T_{Si} + c_2 T_{Li} + c_3 (T_{Si} T_{Li})$	T_{So}, T_{Li}	4
wc07	$EER = c_0 + c_1 T_{Si} + c_2 T_{Lo} + c_3 (T_{Si} T_{Lo})$	T_{Si}, T_{Lo}	4
wc08	$EER = \eta_{2P} EER^* \ (\Delta T_{min,c-h} = 10°C)$	η_{2P}	1
carnot	$EER = \eta_{2P} EER^*$	η_{2P}	1
wc09	$EER = c_0 + c_1 T_{Si} + c_2 T_{Si}^2$	T_{Si}	2

Table 2 Simplified models for full-load prediction of water-to-water heat pumps in cooling mode

The main advantage of this method is that it requires few datapoints and it bears accurate results at the vicinity of the nominal temperatures. However, the accuracy drops rapidly as the prediction gets further away from the nominal condition. This is the reason that some attempts studied [36] a correction factor for this method so that it can be accurate in larger intervals.

Model	Proposed formula	Required variables	Minimum required number of datapoints
ah01	$COP = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 (T_{Li} - T_{Si}) + c_4 (T_{Li} - T_{Si})^2$	$T_{Si}, T_{Li}, \dot{m}_S$	5
ah02	$COP = c_0 + c_1 (T_{Li} - T_{Si}) + c_2 (T_{Li} - T_{Si})^2$	T_{Si}, T_{Li}	3
ah03	$COP = c_0 \exp(c_1 T_{So} + c_2 T_{Li}) + c_3 \frac{T_{So}}{T_{Li}} + c_4$	T_{So}, T_{Li}	5
ah04	$COP = c_0 \exp(c_1 T_{Si} + c_2 T_{Li}) + c_3 \frac{T_{Si}}{T_{Li}} + c_4$	T_{Si}, T_{Li}	5
ah05	$COP = c_0 + c_1 T_{Si} + c_2 T_{Li} + c_3 (T_{Si} T_{Li})$	T_{Si}, T_{Li}	4
ah06	$COP = c_0 + c_1 T_{Si} + c_2 T_{Lo} + c_3 (T_{Si} T_{Lo})$	T_{Si}, T_{Lo}	4
ah07	$COP = \eta_{2P} COP^* \ (\Delta T_{min,c-h} = 10°C)$	η_{2P}	1
carnot	$COP = \eta_{2P} COP^*$	η_{2P}	1
ah08	$COP = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 (T_{Lo} - T_{Si}) + c_4 (T_{Lo} - T_{Si})^2$	$T_{Si}, T_{Lo}, \dot{m}_S$	5
ah09	$COP = c_0 + c_1 (T_{Lo} - T_{Si}) + c_2 (T_{Lo} - T_{Si})^2$	T_{Si}, T_{Lo}	3
ah10	$COP = c_0 \exp(c_1 T_{Si} + c_2 T_{Lo}) + c_3 \frac{T_{Si}}{T_{Lo}} + c_4$	T_{Si}, T_{Lo}	5
ah11	$COP = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 T_{Si}^2$	$T_{Si}, \dot{m}_S$	4

Table 3 Simplified models for full-load prediction of water-to-air heat pumps in heating mode

A rather complex model is found in TRNSYS [37] standard component library, which takes into account the flow rate of secondary source fluid. This model is utilizable when there are at least four data points available for the full-load condition. In this study, this model will be denoted with wh01, wc01, ah01, and ac01. For the heating mode of water-to-air heat pumps, the alternative is proposed to use outlet load temperature in the model ah08.

Another model does not consider the flow rate as an input, which can be regarded as ideal when dealing with some of the catalogues, especially those provided by southern European manufacturers. This relationship, however, sees its downside when the flow rate varies drastically among the datapoints. Therefore, it is not suitable to use when the demand has high-amplitude fluctuations in the same temperature difference. This model is denoted as wh02, wc02, ah09. The alternatives that use outlet temperature of the load side are proposed in ac02, ah02, and wh11. An alternative will add a term including the outlet load flow temperature, hence computing the one-degree effect of two temperatures as independent; this idea can be seen in model wh09.

Model	Proposed formula	Required variables	Minimum required number of datapoints
ac01	$EER = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 (T_{Li} - T_{Si}) + c_4 (T_{Li} - T_{Si})^2$	$T_{Si}, T_{Li}, \dot{m}_S$	5
ac02	$EER = c_0 + c_1 (T_{Li} - T_{Si}) + c_2 (T_{Li} - T_{Si})^2$	T_{Si}, T_{Li}	3
ac03	$EER = c_0 \exp(c_1 T_{Si} + c_2 T_{Li}) + c_3 \frac{T_{Si}}{T_{Li}} + c_4$	T_{Si}, T_{Li}	5
ac04	$EER = c_0 + c_1 T_{Si} + c_2 T_{Li} + c_3 (T_{Si} T_{Li})$	T_{Si}, T_{Li}	4
ac05	$EER = \eta_{2P} EER^* \; (\Delta T_{min,c-h} = 10°C)$	η_{2P}	1
carnot	$EER = \eta_{2P} EER^*$	η_{2P}	1
ac06	$EER = c_0 + c_1 T_{Si} + c_2 \dot{m}_S + c_3 T_{Si}^2$	$T_{Si}, \dot{m}_S$	4
ac07	$EER = c_0 + c_1 T_{Si} + c_2 T_{Si}^2$	T_{Si}	2

Table 4 Simplified models for full-load prediction of water-to-air heat pumps in cooling mode

The coefficients for this model are found using the generalized least square method. This method has been studied further and an analysis by Staffell et al. [38] provided a generic set of coefficients to be used instead of c_i for heating mode in the water-to-water heat pumps. For these generic coefficients, the result of the works in [34] shows a considerable deviation from the actual performance when dealing with the heat pump datasets provided for North American and Southern European markets.

AHRI standard also proposes models that consider only the source side temperature as the available data required for the sake of fitting. The models wc09, ah11 and ac07 are working with this assumption. Adding the source side flow rate into the account, models ah11 and ac06 exist to predict the water-to-air heat pumps in heating and cooling conditions.

The idea to implement an exponential term in the modelling was proposed in the work of Liu et al. [39], the coefficients of which are recommended to be calculated using a Generalized Reduced Gradient non-linear solver. This model (wh04, wc03) needs the inlet load side flow temperature, which can be unavailable in some data sources. An alternative will be the models wh05, wc04, and wc05, which require the output temperature of load side flow. The same idea of adding the exponential term is also applied to the water-to-air heat pump models, with the models ah03, ah04, ah10 and ac03.

A similar idea was to add a multiplication term, as proposed by [40], wh06 and wc06 are identical to their study, while as for the exponential models, the inclusion of inlet load temperature rendered the process of modelling impossible for some catalogues, so the alternatives, namely wh07 and wc07 are proposed to measure for this problem. This idea can also be applied to the water-to-air heat pumps within the models of ah05, ah06, and ac04.

2.2. Part-load Operation

As previously noted, simulating variable-speed heat pumps involves the concept of replicating their operation at full-load and subsequently forecasting their performance at partial-loads by applying a correction factor. To achieve this, it's essential to establish specific variables that pertain to the system's operation under partial-load conditions.

The first variable is then defined as the part-load ratio, it is the capacity demand divided by the full-load demand. $plr,$ then, by definition, can take the values between 0 and 1.

$$plr = \frac{actual\ capacity}{full\ capacity} \tag{13}$$

The second variable is the ratio of the performance factor to that of full-load capacity. The performance factor, PFR, is actually the correction factor that will be used to determine the part-load working performance of the system. This correction factor can be any positive number, as the part-load performance can sometimes surpass the performance at full-load conditions as the heat exchangers get oversized. However, this number will not practically be much larger than 1.

$$PFR_{heating} = \frac{COP_{pl}}{COP_{fl}} \tag{14}$$

$$PFR_{cooling} = \frac{EER_{pl}}{EER_{fl}} \tag{15}$$

2.2.1. Standard Methods

In the standards EN 13612 [33] and EN14825 [19], there are two methods provided for calculating PFR while taking into account the working conditions. The first method, referred to as Method A, a straightforward formula, is recommended for computing PFR. This formula assumes that PFR exhibits a rising trend up to a certain point when plr reaches 0.3 and then levels off at 1. While this assumption may not be entirely accurate, it is a conservative estimate, as in the real application of variable-speed heat pumps, when the operation is not in full-load, sometimes the operation results in higher extraction and rejection of heat. This method uses the following equations.

$$PFR_A = \begin{cases} 1 & if\ plr > 0.3 \\ \dfrac{plr(\frac{100}{25})}{0.9\, plr\left(\frac{100}{25}\right) + 1} ff_{plr} & if\ plr < 0.3 \end{cases} \tag{16}$$

$$ff_{plr} = \begin{cases} 1 & if\ internal\ secondary\ fluid\ is\ water \\ \left[1 - 0.25\left(1 - plr\left(\frac{100}{25}\right)\right)\right] & if\ internal\ secondary\ fluid\ is\ air \end{cases} \tag{17}$$

The second method provided by the standards (Method B) uses a set of measured datapoints to predict the performance of the heat pump. The temperatures used for the testing are proposed in the standard EN 14825 [19], and plr for testing is derived from a formula considering the working temperature and the design external air temperature, both for cooling and heating conditions. The test conditions are proposed based on the working mode (cooling, heating or other operations) and fluid type used in the heat pump. However, in the application of this method, it is possible to use any other set of data points as long as their plr covers a reasonable interval between 0 and 1.

After obtaining the test datapoints, a set of (plr, PFR) will be obtained, using which it is possible to interpolate the performance of the heat pump. Also, two trivial points ((0,0) and (1,1) for heating, (0,1) and (1,1) for cooling) are added to the test points. Therefore, considering a specific operating condition, with its respective plr, first the two test points which have larger and smaller plr are found, and then, their corresponding PFR is used to calculate the performance of the desired point linearly.

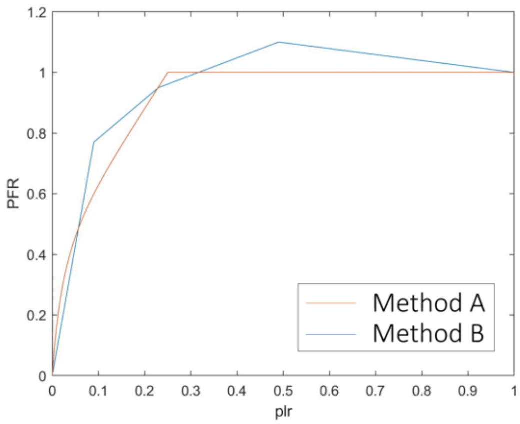

Figure 11 The standard methods to calculate PFR in heating mode

This method bears a significant accuracy; however, access to the test results and low uncertainty measurements that are required for doing the calculation is usually something that is not accessible to users. Therefore, it is not always possible to utilize it.

2.2.2. Proposed Method

Considering the physics ruling the principles of the heat pump, it is possible to frame a possible method for predicting the performance of variable-speed heat pumps. First of all, a critical measure in calculating the performance of any thermodynamic system is to study its energy input. As equation 12 shows, the mechanical energy moving the compressor has a linear relationship with its rotary speed; however, if the effect of temperature is taken into account, it will result in a more complex situation. In the context of simulation models for heat pumps, several proposed models have recognized the simultaneous effect of both the compressor's speed and the temperature on the compressor's power.

These models highlight the intricate relationship between these variables and their impact on the performance of the heat pump system. [41,42] Using an exhaustive and brute force simulation, it was realized that the best possible model was not considered in these suggested models. In order to conduct this simulation, a polynomial relationship between compressor power (as shown in equation 18), the working temperatures and the rotary speed was considered, and the best-defining models were obtained. This simulation was based on the EmersonClimate compressors data from the Copeland Select8 [43] software.

$$\sum_{0\le x+y+z\le 2} a_{xyz}\omega_r^x T_S^y T_L^z \tag{18}$$

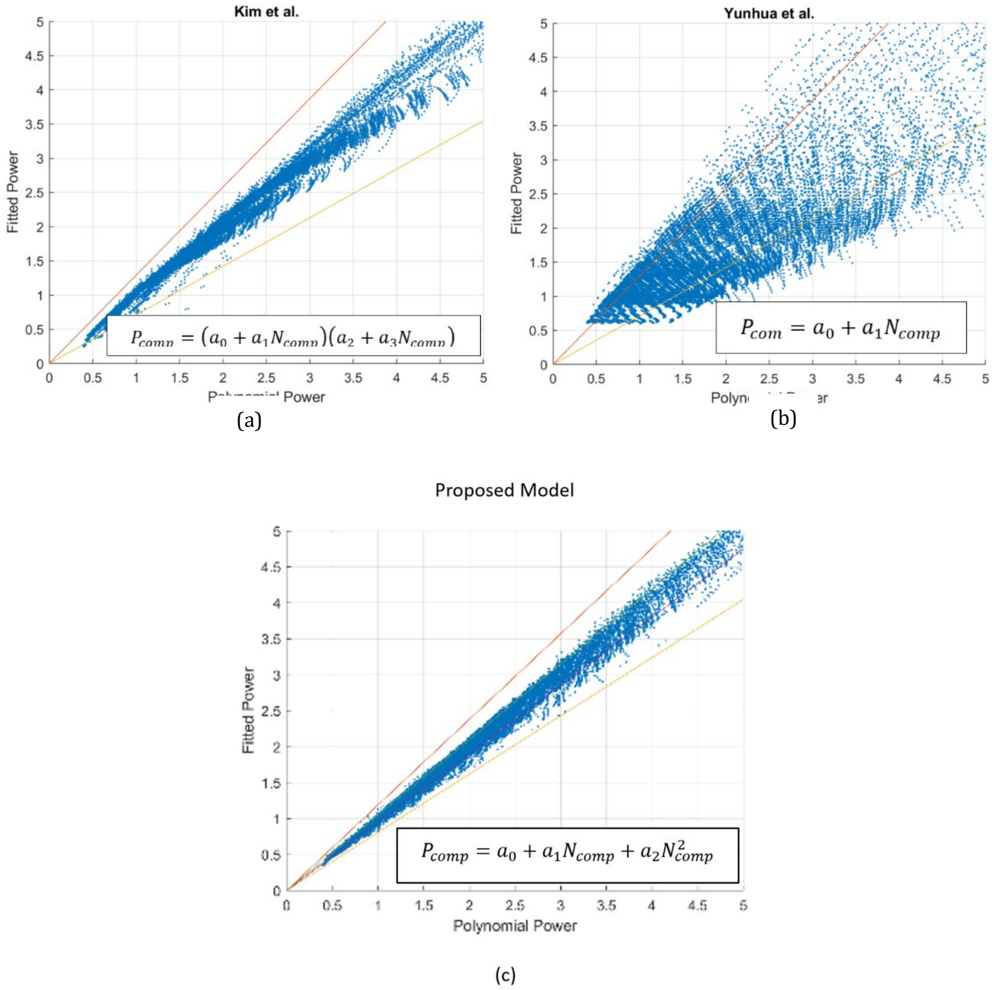

Figure 12 Accuracy of models for calculating compressor power (a) model proposed by Kim et al. (b) model proposed by Yunhus et al. (c) model used in this study

Considering the outcome of this simulation, it could be assumed that a better estimate of the power of the compressor can be obtained by such formula:

$$P_{comp} = a_0 + a_1\omega_r + a_2\omega_r^2 \tag{19}$$

Where a_1 and a_2 are themselves polynomial functions of T_L, T_S. This result can be used later when comparing the performance of the heat pump at part-load versus its performance at full-load.

The effect of changing the flow rate of the refrigerant is directly related to the heat capacity of the rejected heat of the heat pump. If a proportional correlation between fluid flow rate and the rotary speed of the compressor is considered, considering the formula $q = mc\Delta T$, it is derivable that:

$$Q_{cond} = b_1\omega_r \tag{20}$$

Again, in this formula, b_1 is a function of working temperatures. However, it can be seen that the main use of this formula is for comparing it with the full-load; therefore, the dependency on the temperature will not play a significant role.

Another important point to consider is the accessibility of the seasonal performance factors to the users (especially in the European markets). Seasonal performance factors ($SEER$ and $SCOP$) are the indicators of the performance of the system considering all the operating conditions that the machine experiences in its working period. Standard EN 14825 proposes a method to calculate these factors based on the method of bin temperatures.

Bin temperatures (intervals of one degree unit of temperature) are used to facilitate the calculation of a thermodynamic procedure over time as the temperature changes. Each bin temperature corresponds to a bin hour, which is the number of hours that the climate is estimated to experience the given temperature. To calculate the seasonal performance factor, the performance at each bin temperature is calculated, and then considering the bin hours, the accumulated performance variables are derived, from which the seasonal performance can be calculated.

In order to calculate the performance of the system, first, an intermediate approach is used to calculate PFR. This approach is in this paper denoted as X. After the calculation of PFR_X, the values resulting from the standard are used to correct PFR. Equations 19 and 20 are used to obtain the final formula. For the heating mode, it will be

$$plr = \frac{Q_{cond,pl}}{Q_{cond,fl}} = \frac{\omega_r}{\omega_{r,nom}}$$

$$\rightarrow PFR_{heating} = \frac{Q_{cond,pl}}{Q_{cond,fl}} . \frac{P_{comp,fullload}}{P_{comp,partload}}$$

$$= \frac{\omega_r}{\omega_{r,nom}} . \frac{a_0 + a_1\omega_{r,nom} + a_2\omega_{r,nom}^2}{a_0 + a_1\omega_r + a_2\omega_r^2}$$

$$= \frac{plr.\,\omega_{r,nom}}{\omega_{r,nom}} . \frac{a_0 + a_1\omega_{r,nom} + a_2\omega_{r,nom}^2}{a_0 + a_1 plr.\,\omega_{r,nom} + a_2 plr.\,\omega_{r,nom}{}_r^2}$$

$$= plr\,.\frac{a_0 + a_1\omega_{r,nom} + a_2\omega_{r,nom}^2}{a_0 + a_1 plr.\,\omega_{r,nom} + a_2 plr^2 \omega_{r,nom}{}_r^2}$$

$$= plr\,.\frac{a_0 + a_1\omega_{r,nom} + a_2\omega_{r,nom}^2}{a_0 + a_1 plr.\,\omega_{r,nom} + a_2 plr^2 \omega_{r,nom}{}_r^2}$$

As a result, and substituting ω_r With $plr.\,\omega_{r,nom}$

$$PFR_{X,heating} = \frac{plr}{\alpha_{0h} + \alpha_{1h} plr + \alpha_{2h} plr^2} \quad (21)$$

The same procedure can be conducted for cooling mode. (derivation is given in appendix B)

$$PFR_{X,cooling} = \frac{1 + \alpha_{1C} plr + \alpha_{2C} plr^2}{1 + \alpha_{0C} plr} \quad (22)$$

The first boundary condition for this equation is the fact that $PFR = 1$ when $plr = 1$, the second consideration for this equation is the plr_{max} in which the PFR reaches its maximum. In this study, the value of $\frac{T_{nom,cond} - T_{des,climate}}{T_{des,heating} - T_{des,climate}}$ is considered. Then, using the bin temperature methods, it is possible to calculate the $SCOP$ or $SEER$, thereby being able to have a third boundary condition for the equation and being able to calculate the $\alpha_0, \alpha_1, \alpha_2$.

Such an approach will result in a significant accuracy while dealing with high plr values, while the result tends to overestimate plr in the low part-load ratios. This issue is working in the opposite direction of method A in heating mode, so it can be resolved by making a weighted average between this method and the first method, which had the exact opposite problem.

$$PFR_{G,heating} = PFR_A(1 - plr^{1.35}) + PFR_{X,heating}(plr^{1.35}) \quad (23)$$

As there is no structured method like method A available for cooling operation, in the cooling process, the last step is not done and:

$$PFR_{G,cooling} = PFR_{X,cooling}$$

3 | Methodology

The process of proposing a model to predict the performance of a system consists of acknowledging the physics of the system and providing a method which then should be compared with real-world data.

In order to do so, the engineering software Matlab was used in this study. First, using the catalogue data, for each considered full-load model, the coefficients are calculated (based on the method of optimization recommended for the model). And then, using the part-load data, the part-load performance of the heat pump is predicted.

In the end, using the calibration analysis techniques, the predicted performance is compared with the available real-world data and the results are used to compare different methods of prediction of the heat pump performance.

3.1 Assumptions

The developed models are based on some assumptions. First, the performance of the heat pumps can be achieved using only the data given in the catalogues accessible to the users. As mentioned earlier in Chapter 2, there are already proposed models that can predict, with a relatively high accuracy, the performance of a variable-speed heat pump. However, such models rely on detailed information on the heat pump's components, which usually are not available to users. The geometric data, such as the length of the tubes and the structure of the heat pump, is not considered in the input. In the same way, there is no information about the thermodynamic cycle (such as pressure levels), which is only obtainable by measuring the refrigerant flow, using sensors which are, in turn, required for these models.

The other assumption is that in this study, the output performance factor is considered to be the same as the declared performance factor in the catalogues. This is not necessarily true, as with the change in climate conditions, such as humidity and atmospheric pressure, there will be a slight variation in the performance of the heat pump. Also, the effect of wearing is not considered, as there was no access to time series data for a heat pump in test conditions. Effects of electrical components and defrost in lowering the actual performance are considered not to be included in the declared value of performance factors in the catalogues ($COP\ and\ EER$) neither they are provided in the seasonal performance factors. In addition to the general assumptions, each of the models that are provided in the last chapter, have their own characteristics and simplification assumptions that are noted in their description in chapter 2.

3.2 Reference Heat Pumps and Adapted Models

In this study, the catalogues that were used to calibrate the models were for different sizing of two models of heat pump. One comes from the water-to-air Trilogy Series of ClimateMaster, while the other is the air-to-water Ferroli Omnia H/HN series. In the catalogue of the Ferroli heat pump, the available values are outlet water temperature (T_{Lo}) and inlet air temperature (T_{Si}) while for the ClimateMaster heat pump, both inlet and outlet water temperatures are given while still only having the inlet air temperature. Even though Ferroli is not a water-to-air heat pump, in this study, the models available in table 3 are adapted to predict its performance.

Considering the available data for each catalogue, for the heating mode, in full-load condition, it is possible to use $ah01, ah02, ah03, ah04, ah05, ah11$ for the ClimateMaster heat pump, the models $ah06, ah09, ah10$ for the Ferroli heat pump. The model $ah07$, which is based on the second principle efficiency, can be used for both.

3.3 Calibration Analysis

Calibration holds immense significance across a spectrum of disciplines, encompassing engineering, scientific research, and technological advancements. This pivotal procedure entails the meticulous comparison of measurements or data with established standards or references, a practice indispensable for guaranteeing the precision and dependability of results. Within the confines of this study, two main concepts are used: the Percent Mean Error (PME) and the Root Mean Square Error ($RMSE$). These metrics occupy a foundational position in evaluating the effectiveness of calibration processes, serving as indispensable instruments for enhancing the accuracy of measurements.

3.3.1 Percent Mean Error

Percent Mean Error is a metric used to quantify the average deviation between observed values and their corresponding reference values in calibration. PME is expressed as a percentage and provides insight into the systematic bias of a measurement system. It is calculated using the following formula:

$$PME = \frac{100}{n}\sum_{j=1}^{n}\left(\frac{y_{mod,j} - y_{dec,j}}{y_{dec,j}}\right) \tag{24}$$

PME helps in identifying whether a measurement system consistently overestimates or underestimates the true values. A PME close to zero indicates that the system is unbiased, while a positive or negative PME suggests a systematic overestimation or underestimation respectively. It is important to note that PME does not consider the magnitude of errors but focuses solely on their direction.

3.3.2 Root Mean Square Error

Root Mean Square Error is another critical metric used in calibration analysis. Unlike PME, $RMSE$ takes into account both the magnitude and direction of errors. It provides a measure of the overall accuracy of a measurement system. $RMSE$ is calculated using the following formula:

$$RMSE = 100\sqrt{\frac{1}{n}\sum_{j=1}^{n}\left(\frac{y_{mod,j} - y_{dec,j}}{y_{dec,j}}\right)^2} \tag{25}$$

4| Results

In this chapter, the results of the simulation will be demonstrated. Using tables and graphs, the accuracy of the calculations is compared, and attempts are made to find the reasons for the deviations from the actual data. The procedure for obtaining the results was discussed in the last chapter. After calculating the respective coefficients for each model, the calculated data were collected and compared with the data from the catalogues.

The errors were calculated, and the methods were compared based on their bias and precision. Using demonstrative methods, graphing the calculated values versus the real values, the performances at which the models do not show great accuracy were picked out, and hence, a deeper analysis was conducted on the reasons and the influential variables on the resulting error.

As mentioned in the third chapter, the analysis of the errors is conducted based on the calculation of percent mean error and root mean square error. The models were calculated with ten methods to calculate the full-load performance and three methods to provide the part-load correction in the heating mode. Also, for the cooling mode, eight models were used for the full-load, and two methods were considered to compute the final part-load EER. There were two different heat pumps under consideration, each with two different sizes.

4.1 Comparison of Models

Table 5 shows the PME and $RMSE$ of the values calculated using different methods for deriving the part-load COP of the ClimateMaster Trinity QE0930 heat pump. Table 6 shows the same value for the Ferroli OMNIA4. The same tables for other sizing of these heat pumps can be found in Appendix A.

From Tables 5 and 6, there are some ideas that can be interpreted. The first thing that can be seen is the significant difference between the errors in the first table and the error values in the second one.

	Full-load		Method A		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ah01$	4.00	0.28	14.64	−7.00	9.50	−2.25	11.32	−2.82
$ah02$	5.63	−1.14	15.39	−8.32	10.55	−3.63	11.3	−1.41
$ah03$	7.58	2.40	13.48	−5.03	8.34	−0.18	10.99	−4.96
$ah04$	6.41	−0.28	15.09	−7.50	10.30	−2.78	10.85	−4.47
$ah05$	7.72	−1.97	15.98	−9.08	11.50	−4.44	11.18	−6.14
$ah07$	6.82	0.79	13.18	−5.48	8.41	−0.90	9.10	−2.01
$carnot$	8.21	3.13	13.14	−3.48	9.12	0.32	11.66	−0.96
$ah11$	7.36	−2.08	15.94	−9.19	11.37	−4.55	13.66	−7.64

Table 5 Precision and accuracy comparison between different methods of COP calculation for ClimateMaster QE0930 -heating

Another interesting consideration is that there are some full-load models that show a weak accuracy compared with other full-load models; however, when integrated with partial-load methods to obtain the part-load COP, their accuracy improves. An example is the model $ah03$, which shows one of the worst $RMSE$s among the full-load models. However, when integrated with the part-load methods, it will become one of the most accurate methods. One reason to justify this observation is that the part-load methods tend to underestimate the result, and $ah03$ is one of the only full-load models that has a positive bias. Hence, the partial-load correction factor, with its negative bias, cancels out the positive bias of the full-load model. The same observation can be seen in the effect of implementing methods A and G with $ah06$. In the full-load condition, this model works way worse than $ah09$, but using methods A and G (which predict PFR with more underestimation) will make the final set of calculated COPs closer to the real data.

	Full-load		Method A		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ah06$	4.47	1.04	6.78	−5.30	4.91	−1.88	5.46	−0.95
$ah07$	6.39	−0.27	7.98	−6.70	7.46	−5.52	7.63	−7.39
$carnot$	7.15	2.16	7.48	−2.21	8.88	−2.12	8.91	−1.86
$ah09$	2.76	−0.74	6.42	−4.75	4.00	−1.59	5.32	−3.59
$ah10$	3.17	−0.30	6.20	−4.34	3.60	−1.16	4.18	−2.12

Table 6 Precision and accuracy comparison between different methods of COP calculation for Ferroli OMNIA4 - heating

The same format of tables can be obtained for the cooling mode. Table 7 shows PME and $RMSE$ values for the ClimateMaster Heat pump in the calculation of EER.

	Full-load		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ac01$	7.21	−1.59	8.76	−2.61	10.51	−1.97
$ac02$	4.85	−0.30	7.00	−1.43	9.12	−2.95
$ac03$	5.34	0.25	7.14	−1.33	7.69	−2.86
$ac04$	5.77	−0.82	7.75	−1.94	9.73	−3.46
$ac05$	7.22	−1.61	8.97	−2.33	12.23	−2.87
$carnot$	8.31	−0.97	10.13	−1.53	10.05	−1.78
$ac06$	7.39	0.97	7.28	−0.82	9.23	−0.65
$ac07$	4.72	−0.33	6.80	−1.18	8.93	−2.62

Table 7 Precision and accuracy comparison between different methods of EER calculation for ClimateMaster QE0930 - cooling

Considering the data available for the Ferroli heat pump, only $ac05$ was applicable, which did not bear different results from that of the climate master. It showed an $RMSE$ value of 7.17 percent for full-load operation while 8.43 and 11.91 percent for the part-load calculations of EER based on method B and method G, respectively. Also, in this table, the scale of the numbers is lower than that of Table 5. The reason for this difference will be discussed later.

An overall look at the tables will hint at a statement that was previously predictable. The more data is available, the more accurate it will be. The worst full-load models are those that operate with fewer variables and those that do not consider the mass flow rate. Also, the model based on the assumption of 2^{nd} principle efficiency to be constant does not follow the real data in a promising manner. Among these models, those that have more coefficients and need more data points to obtain, show the result with better precision.

Also, for the partial-load calculation, the same idea can be seen. Method G, which needs more input than method A, predicts the actual data with more accuracy, while method B, which needs actual test measurement data, bears significantly better results.

4.2 Models Accuracy

Looking at the data presented in the tables in the last section, it is crucial to verify the dependence of the error from the variables considered in the models and, consequently, the operating conditions. Figures 12, 13, and 14 show the actual heat pump performance (obtained from the catalogues), compared to the COP and EER that is calculated with the most accurate (lowest $RMSE$) among the proposed models. These methods resulted in maximum deviation from reality (error) of -8.06%, -3.24% and -2.99% respectively.

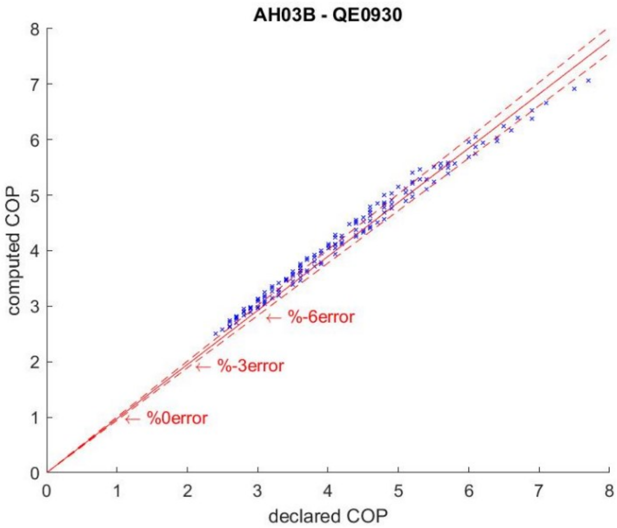

Figure 13 calculated and actual COP for the heating mode, based on ah03 full-load model and method B for ClimateMaster QE0930.

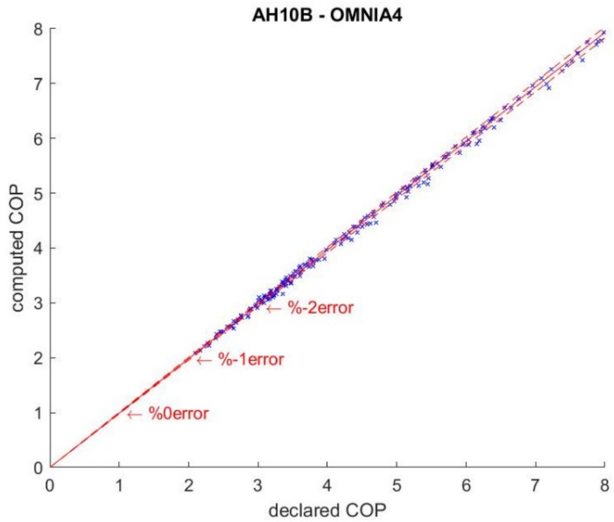

Figure 14 calculated and actual COP for the heating mode, based on ah06 full-load model, and method A for Ferroli OMNIA4.

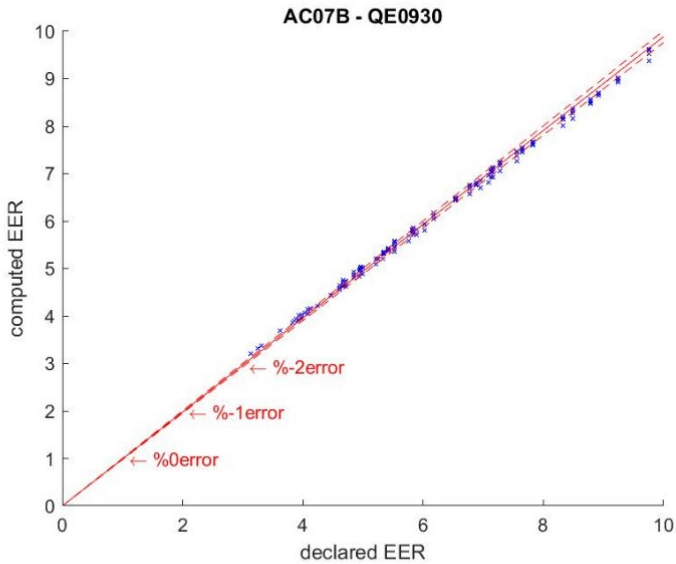

Figure 15 calculated the actual EER for the cooling mode, based on ac07 full-load model, and method B for ClimateMaster QE0930.

A look at figures 12,13 and 14 will show that there is a significant difference in accuracies obtained for the ClimateMaster QE0930 in heating mode and the other considered catalogues. The other observable issue is that the deviation from the declared COP or EER gets bigger as the declared value gets higher. This can be either a result of problems in the calculation of the full-load performance or an issue with the partial-load calculations. To better understand the reason behind this deviation, it is possible to see how the error changes with different variables. The error in relative terms is calculated using the equation 26.

$$Error = \frac{y_{mod,j} - y_{dec,j}}{y_{dec,j}} \tag{26}$$

If the error and deviation result from the problem in the full-load model in predicting the performance, it should be demonstratable using a graph that shows the trend of $Error$ as the full-load performance factor (COP_{fl} or EER_{fl}) changes. These graphs for the same heat pumps in some of the methods (the same as the last figures) are shown in figures 15,16 and 17.

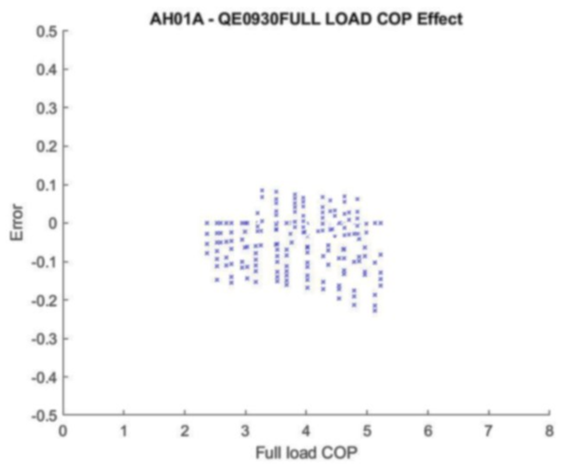

Figure 16 trend of Error as the full-load COP changes for the heating mode, based on ah01 full-load model, and method A for ClimateMaster QE0930

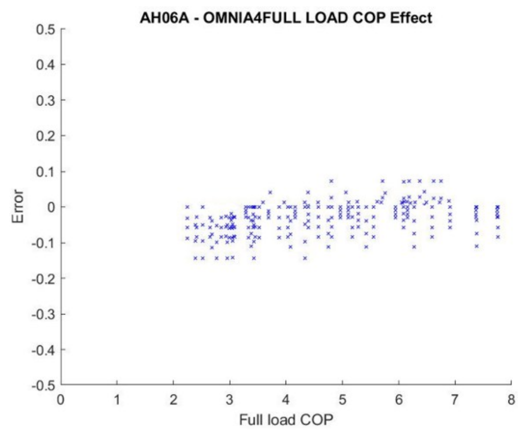

Figure 17 trend of Error as the full-load COP changes for the heating mode, based on ah06 full-load model, and method A for Ferroli OMNIA4

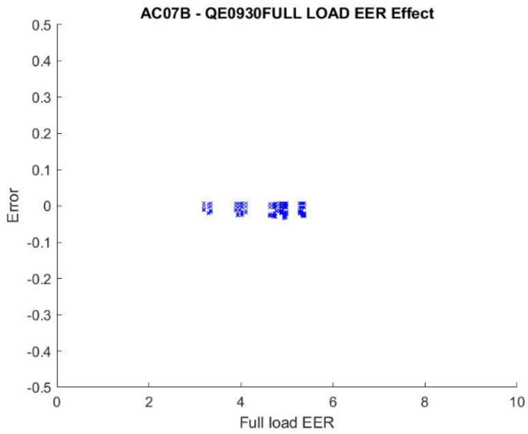

Figure 18 trend of Error as the full-load EER changes for the cooling mode, based on ac07 full-load model, and method B for ClimateMaster QE0930

From the previous figures, a significant trend in the Error value, while in different performance factor values, cannot be seen. This means that the reason for the deviation can be found in the calculations of the partial-load correction factor. In order to evaluate this guess, the trend of change in the error as PFR changes shall be considered. In the following figures, the mentioned diagrams are shown.

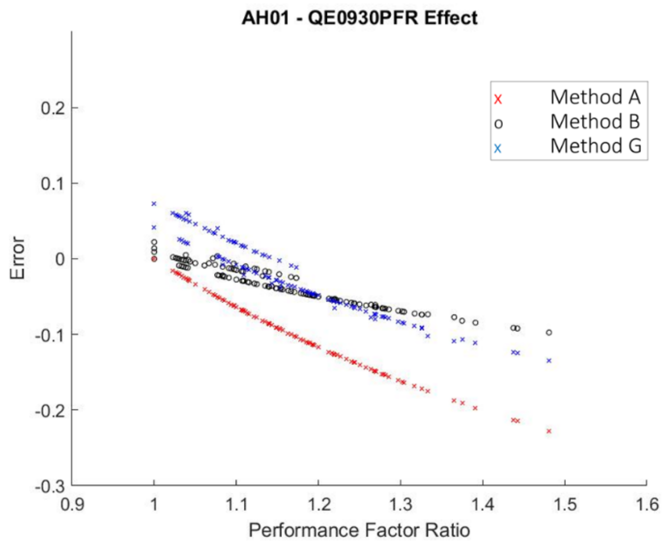

Figure 19 variation of Error with PFR for the heating mode, based on ah01 full-load model for ClimateMaster QE0930

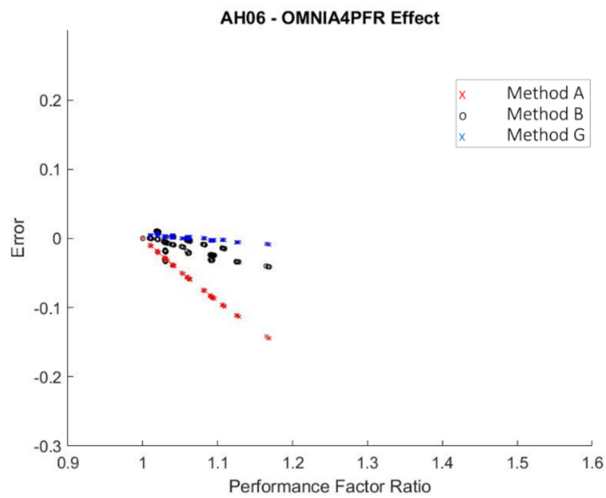

Figure 20 variation of Error with PFR for the heating mode, based on ah06 full-load model for Ferroli OMNIA4

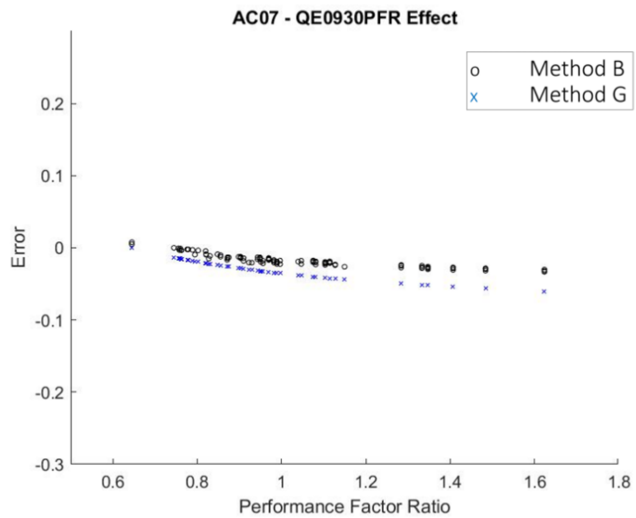

Figure 21 variation of Error with PFR for the cooling mode, based on ac07 full-load model for ClimateMaster QE0930

The last figures show an existing trend with the error and PFR as the models tend to predict lower amounts for the performance factor as PFR increases. One main reason for this observation is the fact that PFR is high when the heat exchangers are oversized, and therefore, the model needs the parameters from the geometry of the system to evaluate the performance of the system, which are not available in the simplified methods as mentioned in the assumptions in chapter 4.

This is the main reason for finding meaningful difference between the $RMSE$ values of heating mode in the ClimateMaster product with other catalogues. Because in the aforementioned unit, the system reaches higher PFRs, the model fails to follow the actual performance data at more points in the operation range.

For these models, it is also possible to see the effect of temperature on the error; it is expected that the effect of temperature on the error should be the same as the effect of full-load performance because the partial-load correction factor is not strongly related to temperature.

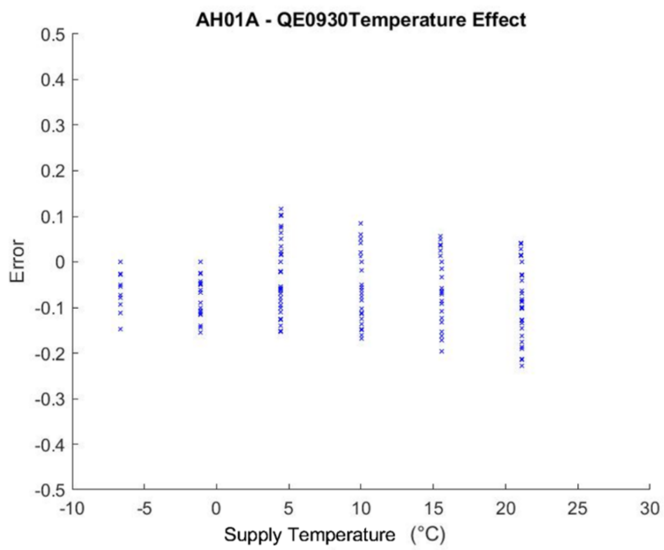

Figure 22 Variation of error for the AH01 model and method A with supply temperature in heating mode - QE0930

As can be seen in Figure 22, as expected, the temperature does not have a significant effect on the trend of error, and it looks like the errors in the calculation of partial-load COP are not dependent on the supply temperature. The same can be seen for other models and other datasets in the catalogue.

A similar analysis can be done on the effect of heating capacity (load) on the model's accuracy. This time, however, it is expected to see a trend as the capacity is related to plr, which has a meaningful contribution to the accuracy of the models. Looking at Figure 23, it is demonstrated that actually, this is in accordance with the observations and as the heating capacity gets higher (plr gets closer to 1) or it gets too low (plr reaches near zero), the error tends to be lower. However, when the unit is working in the loads, which results in plrs near 0.5, the error also gets higher.

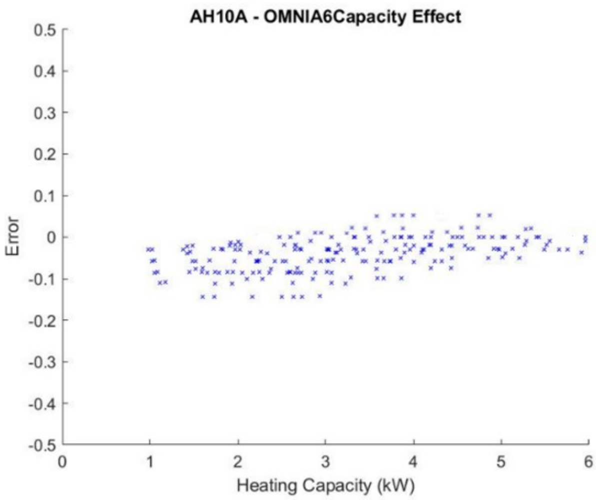

Figure 23 Variation of error for the AH01 model and method A with heating capacity - OMNIA 6

This trend can also be seen in the other models for different sizes and in cooling mode. Considering the effect of heating capacity and the fact that temperature difference is strongly related to capacity, the noises in the trend shown in figure 23 can be explained. Based on this justification, the noises should be cancelled when dealing with mass flow rate. In the ClimateMaster heat pump, for which there was access to the water mass flow rate, figure 24 was obtained. This figure demonstrates a more evident trend for the error. It is worth mentioning again that these graphs were reproduced for other sizing and also the cooling mode and the results were the same.

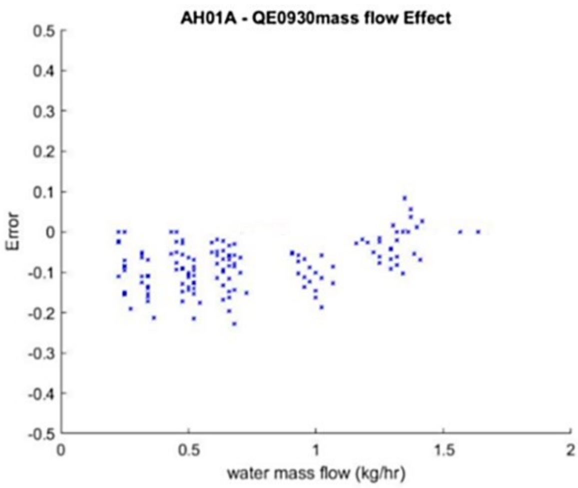

Figure 24 Variation of error for the AH01 model and method A with water flow rate - QE0930

| Conclusions

This thesis provided a method for predicting the performance of a heat pump that works with a variable-speed refrigerant control method. The main challenge in this work was to obtain an accurate algorithm to calculate the output of a heat pump using only catalogue data.

The scope of this work was to obtain a simplified method that can be used in creating a program utilizable in a simulation environment. This way, it will be possible to calculate the effect of using a variable-speed heat pump in integration with other parts of the thermal systems.

The first phase of the project was searching for the literature and finding the available or previously studied models. As mentioned in the second chapter, most of these models were not usable by a simple commercial user. However, splitting the problem into two different steps and solving each step separately helped achieve a more straightforward approach. It was decided to approach the solution by first using the available methods to calculate the full-load performance of the heat pumps and then using the standards and a new proposed method, calculating a correction factor that will help find the part-load performance of the unit. This approach assumed the correspondence of part-load performance to the full-load performance at the same working condition.

The proposed idea for the calculation of part-load performance uses the dataset of compressors to obtain a simple formula for the calculation of the mechanical power input and then, with a series of algebraic procedures, deriving a simplified formula that relates PFR with the partial-load ratio.

For the available catalogue data, and for both cooling and heating, different models for full-load calculations were used, and for the partial-load calculation, the methods A (just for heating) and B from the standard EN14825 were considered, and method G was proposed in this project.

As a result of the calculations, it was seen that in the full-load regime, for the water-to-air heat pump, the model $ah01$ with an $RMSE$ of 4 percent showed the best result in heating mode. This model considered the mass flow rate of the source flow in addition to the temperature. For the air-to-water heat pump, the best results were obtained by using the $ah09$ model, a simple quadratic model, which showed 2.76 percent $RMSE$. And for cooling mode in the water-to-air heat pump, the best model was $ac06$ with $RMSE$ of 4.72 percent, which is a quadratic model, with consideration for the mass flow rate.

The Carnot efficiency model ($ac05, ah07$) shows the least accuracy, and this is expected. Because this model assumes a constant 2nd principle efficiency, and it is independent of many variables that are actually influential in the performance of the unit. However, considering that it does not require much input data, it shows good accuracy with $RMSE$ values of around 7 percent in all the units.

However, adding in the calculation of the correction factor for the partial-load performance, it is always true that method B results in better accuracy than method G, which in turn is more accurate than method A. However, it should be noted that access to the information required in method B is not always guaranteed. The same issue can be found with method G, which needs less data. While method A is indifferent to the unit, and it uses only a simple function to obtain PFR.

Taking into account the partial-load performance, the best model to use in full-load is $ah03$, involving an exponential term, for the ClimateMaster water-to-air units in heating mode. It results in $RMSE$s of 13.48, 8.34, and 10.99 percent for methods A, B and G, respectively. For heating mode, in Ferroli OMNIA air-to-water unit, still, $ah09$ prevails with $RMSE$s of 6.42, 4.00 and 5.32 percent for methods A, B, G. These values for the part-load performance, especially the ones for method B and G are significantly low, which is promising. For the cooling condition in the ClimateMaster QE0930, $ac07$ is the best when coupled with method B with 6.80% $RMSE$, however with method G, $ac03$, which is an exponential model show the lowest $RMSE$ of 7.69 percent.

A visualization of the variation of error with different variables shows that the main error in the calculation of the part-load performance is due to the errors in PFR calculation. It was derived that as the PFR increases and gets more than 1, the error magnitude gets more prominent, and it will be more negative. This means the models cannot predict well the effect of oversize in the heat exchangers when the unit is with much higher performance than its full-load condition. It is possible to consider a study to analyze the possibility of using a correction factor for high PFR points. However, this is an almost impossible task. Because this correction factor would totally depend on the geometry of the heat pump, and not having access to the related data will make the efforts drastically more difficult.

The results of this thesis can be a step on which a program can be developed for the simulation of the variable-speed heat pumps, and its integration with other simulation environments can help better analyze the thermal systems, mainly used in heating and cooling applications for residential buildings.

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APPENDIX

A | Model Accuracy for Other Sizing of the Heat Pumps

	Full-load		Method A		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ah01$	3.92	0.30	13.80	−7.40	9.74	−2.45	11.69	−2.41
$ah02$	6.38	−0.98	14.05	−7.64	9.67	−3.18	10.93	−1.34
$ah03$	6.95	2.61	13.98	−5.38	7.20	−0.19	9.53	−5.10
$ah04$	6.38	−0.25	16.14	−8.47	10.37	−2.43	11.70	−4.13
$ah05$	6.81	−1.92	15.26	−9.64	12.58	−4.48	11.77	−6.29
$ah07$	6.66	0.87	11.63	5.82	7.01	−0.77	7.74	−1.75
$ah11$	7.49	−1.96	17.86	−10.55	10.77	−4.03	15.00	−7.85

Table A 1 Precision and accuracy comparison between different methods of COP calculation for ClimateMaster QE1860 -heating

	Full-load		Method A		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ah06$	4.20	0.93	7.26	−5.79	5.45	−1.71	6.22	−1.07
$ah07$	5.67	−0.24	8.54	−6.42	7.48	−5.03	7.98	−8.31
$ah09$	2.97	−0.63	7.17	−4.77	4.02	−1.45	5.76	−3.27
$ah10$	2.78	−0.30	6.54	−4.67	3.54	−1.12	4.21	−1.98

Table A 2 Precision and accuracy comparison between different methods of COP calculation for Ferroli OMNIA6 -heating

	Full-load		Method B		Method G	
	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]	$RMSE$ [%]	PME [%]
$ac01$	6.16	−1.55	9.67	−2.53	11.31	−2.00
$ac02$	4.36	−0.34	6.29	−1.29	10.45	−3.04
$ac03$	6.04	0.27	6.24	−1.21	7.77	−2.96
$ac04$	5.02	−0.81	7.42	−2.20	9.34	−3.22
$ac05$	8.15	−1.47	10.12	−2.66	10.42	−2.67
$ac06$	7.75	1.05	6.69	−0.73	8.22	−0.70
$ac07$	4.55	−0.35	6.72	−1.15	9.18	−2.67

Table A 3 Precision and accuracy comparison between different methods of EER calculation for ClimateMaster QE1860 - cooling

B | derivation of the formula for PFR in cooling

$$EER = \frac{Q_{cond} - W}{W} = \frac{Q_{cond}}{W} - 1$$

$$PFR = \frac{EER_{pl}}{EER_{fl}} = \frac{\frac{Q_{cond,pl}}{W_{pl}} - 1}{\frac{Q_{cond,fl}}{W_{fl}} - 1}$$

$$= \frac{b\omega - a_0 - a_1\omega - a_2\omega^2}{a_0 + a_1\omega + a_2\omega^2} \cdot \frac{a_0 + a_1\omega_{nom} + a_2\omega_{nom}^2}{b\omega_{nom} - a_0 - a_1\omega_{nom} - a_2\omega_{nom}^2}$$

As a further simplification, the work formula for the denominator is substituted with the linear model.

$$PFR = \frac{a_0 + \kappa_1 plr + \kappa_2 plr^2}{c_0 + c_1\omega} = \frac{1 + \alpha_1 plr + \alpha_2 plr^2}{\left(\frac{c_0}{a_0} + \alpha_0 plr\right)}$$

Knowing that c_0 and a_0 are both intercepts for the prediction of the same variable. They can be considered close to each other, hence:

$$PFR = \frac{1 + \alpha_{1C} plr + \alpha_{2C} plr^2}{1 + \alpha_{0C} plr}$$

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