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# Design Solutions and Safety Considerations in Heat Pumps Using Propane (R290): A Review

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**Abstract**

The objective of this paper is to review modern design solutions applied to vapor-compression heat pumps using the environmentally friendly refrigerant propane (R290), with particular emphasis on refrigerant charge minimization and its impact on system energy performance, and to discuss the influence of compressor type, heat exchanger configuration, and the application of thermoelectric subcooling technology on the coefficient of performance and the seasonal coefficient of performance. Additionally, we examined the effects of supply voltage, water temperature, and climatic location on system efficiency. The reviewed results indicate significant potential for further optimization of heat pumps using R290 while simultaneously meeting safety and energy efficiency requirements.

**Keywords:** heat pump; R290 refrigerant; COP; SCOP

## 1. Introduction

Decarbonization processes and the transformation of the energy sector necessitate the implementation of new solutions in heating technology as part of a low-emission economy [1]. Heat pumps have become a key technology, serving as successors to solid fuel boilers. Their role is to ensure energy security while maintaining high efficiency and minimizing adverse environmental impacts [2]. Heat pump operation is based on renewable energy sources such as the ground or ambient air. Therefore, increasing the number of installed heat pumps has become one of the objectives of the REPowerEU plan, which is aimed at improving the share of renewable energy sources in the energy mix [3].

For a heat pump to perform effectively, it must be appropriately selected considering building standards, energy demand, and the climatic zone. In Europe, three climatic zones, namely warm, moderate, and cold, differing in heating season duration and design temperatures can be distinguished. When calculating the seasonal coefficient of performance, different design temperatures are assumed for each zone according to EN 14825 [4]:

- Warm climate: $T_{design,h}$ = 2 °C;
- Moderate climate: $T_{design,h}$ = −10 °C;
- Cold climate: $T_{design,h}$ = −22 °C.

In regions with mild climates, heat pumps can fully serve as both heating and cooling systems due to higher ambient temperatures [5]. Heat pumps operating in moderate and cold climates must demonstrate high efficiency at subzero temperatures and achieve higher supply temperatures of the heat distribution system. In retrofitted buildings equipped with high-temperature radiator systems, supply temperatures typically range from 75 to

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90 degrees Celsius. Achieving such temperatures under low-source-temperature conditions may not be possible without auxiliary peak heat sources [6]. Figure 1 illustrates representative outdoor temperature distributions for selected locations in Poland, highlighting the variability of climatic conditions and their impact on seasonal heat pump performance. These variations directly affect SCOP values and underscore the need for adaptable system designs that maintain efficiency across different operating conditions. The limited availability of high-temperature heat pumps constitutes a significant barrier to achieving established decarbonization targets.

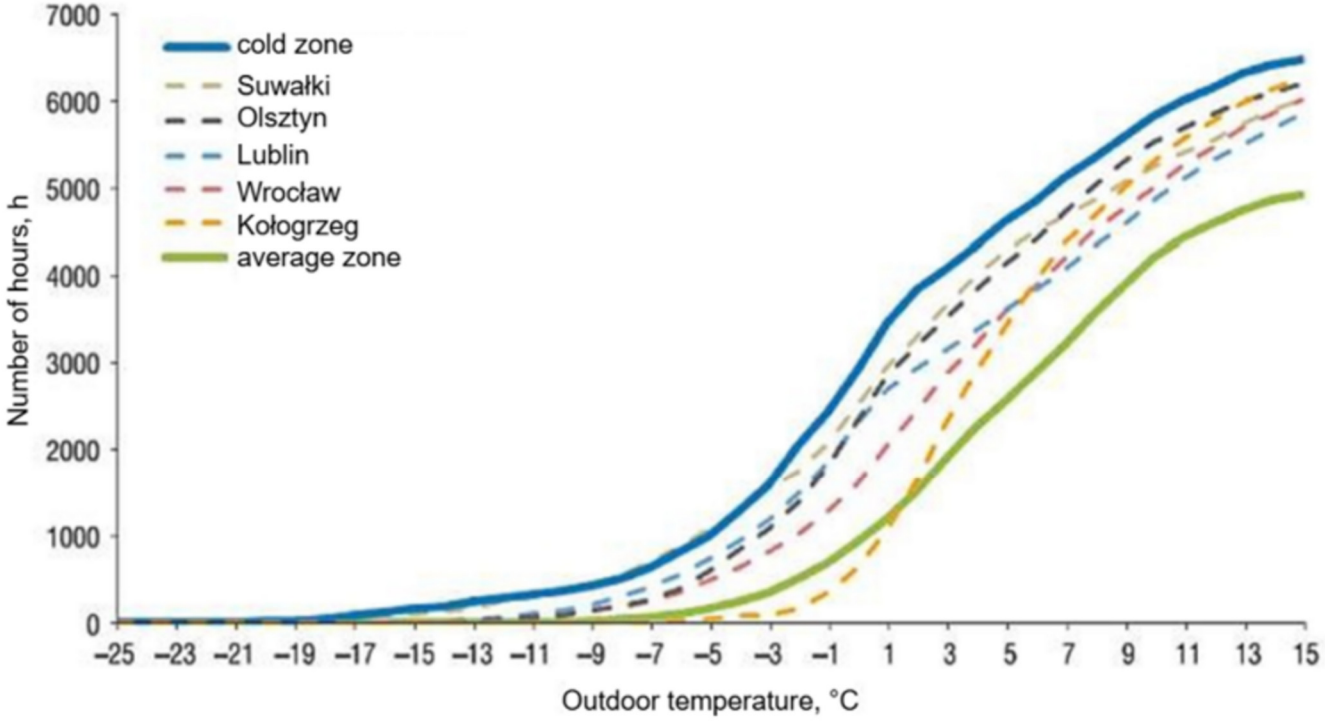



**Figure 1.** Comparison of climate curves for selected localities in Poland according to [7].

The refrigerant type significantly influences heat pump performance, as it mediates heat exchange between the heat source and sink [8]. In addition to thermodynamic, chemical, and physical properties, refrigerants must meet environmental criteria. Key selection parameters include zero ozone depletion potential, low global warming potential, and compliance with current European Union fluorinated gas regulations (Regulation (EU) No. 517/2014) [9]. Environmentally friendly refrigerants commonly used in heat pumps include propane (R290) and difluoromethane (R32), which are used as replacements for R410A [10].

Another major challenge for current refrigeration technologies is refrigerant charge minimization. This development direction addresses safety, system efficiency, the environmental impact, and operational costs [11]. Fluorinated refrigerants are being phased out due to their environmental impact, despite their favorable thermodynamic properties. The observable consequences of global warming necessitate the implementation of environmentally benign alternatives while maintaining adequate system performance [12].

The transition to low-global-warming-potential refrigerants has increased interest in research on hydrocarbon-based systems, particularly propane (R290). However, due to the flammability of this refrigerant (A3 safety class), significant design constraints arise, making system configuration, appropriate component selection, and, above all, refrigerant charge reduction key research and development priorities.

In this context, the present review focuses specifically on two interrelated research directions. The first is minimizing refrigerant charge, and the second is improving the efficiency of heat pumps operating in cold climates. These topics were selected because they directly address the main limitation of R290-based systems—the trade-off between safety and performance. Reducing the refrigerant charge enhances safety and compliance with current regulations but often leads to performance degradation. The application of thermoelectric subcooling (TESC) in heat pumps is a promising approach to mitigate this loss by increasing subcooling and, consequently, improving overall system efficiency.

This paper focuses on combining two concepts in the design of propane-based heat pump systems: refrigerant charge minimization and thermoelectric subcooling (TESC). Synthesizing these approaches is a novel direction. It may deliver satisfactory performance, especially in cold climates. The proposed solution helps maintain efficiency, ensure stable operation, and enable high supply temperatures, even at low ambient temperatures.

## 2. Theoretical Background

The operation of a heat pump is based on thermodynamic processes. In the evaporator, the refrigerant absorbs heat from the lower heat source, such as air, ground, or water, thereby increasing its temperature. The low-pressure vapor then enters the compressor, where its pressure, temperature, and enthalpy are increased. Subsequently, the refrigerant flows to the condenser, where it undergoes isobaric condensation and transfers heat to the upper heat source. The cooled liquid refrigerant then passes through the expansion valve, where it undergoes adiabatic expansion, resulting in a decrease in pressure and enthalpy [13]. The entire process is presented in the heating pump diagram in Figure 2.

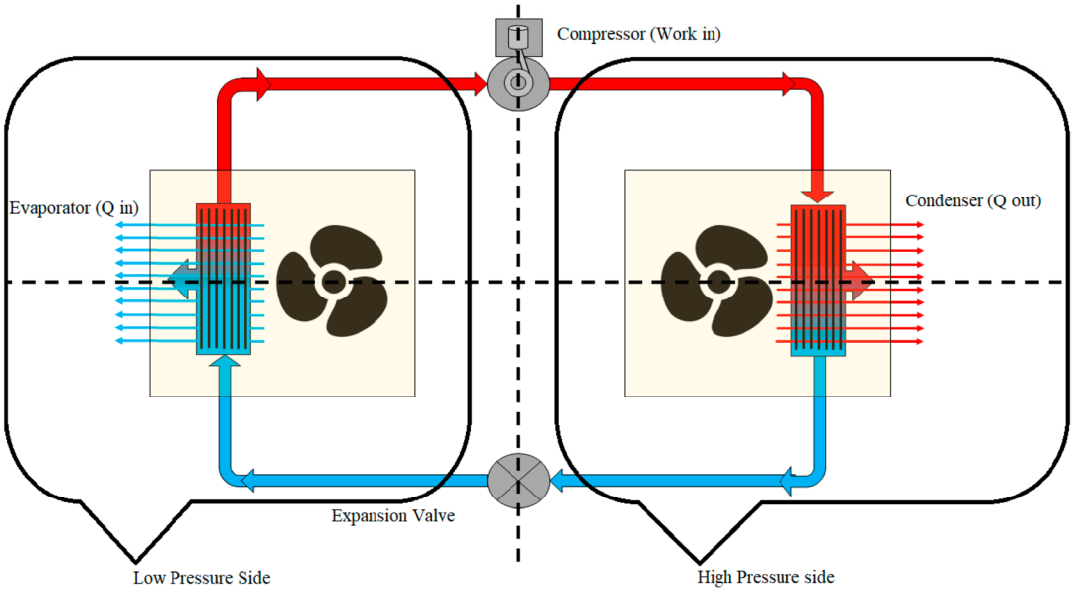



**Figure 2.** Basic elements of a heat pump system: compressor, evaporator, expansion valve, and condenser [13].

Heat pump performance is characterized by two coefficients: the coefficient of performance and the seasonal coefficient of performance. The coefficient of performance is defined as the ratio of delivered thermal power to compressor input power. Higher values indicate greater thermal energy output per unit of electrical energy consumed. To increase the coefficient of performance, system losses are minimized, the temperature difference between the heat source and sink is reduced, and high-efficiency compressors are employed.

The minimum coefficient-of-performance values for specific heat pump types are defined in EN 14511 for designated operating points [14].

The seasonal coefficient of performance accounts for variable operating conditions over the heating season and depends on factors such as climate zone and heating season duration. Testing and calculation procedures are defined in EN 14825 [4].

Propane (R290) is characterized by a very low global warming potential of three, excellent thermodynamic properties, high thermal conductivity, and low viscosity. It belongs to safety group A3, indicating non-toxicity and high flammability. Therefore, systems using R290 must be hermetically sealed and designed with limited refrigerant charge to ensure operational safety [15]. Difluoromethane (R32) exhibits favorable thermodynamic properties, has a global warming potential of 675, and belongs to the mildly flammable A2L safety group [16]. Refrigerant charge reduction is essential to minimize the environmental impact and safety risks in the event of leakage.

## 3. Methodology and Results

In response to the current challenges, two scientific articles describing the design and operational performance of a 5–10 kW heat pump charged with propane were presented, with a particular emphasis on refrigerant charge minimization down to 150 g: "Low Charge Heat Pump Module—development and evaluation of charge-reduced designs using 150 g of R-290" (2020a) [17] and "Experimental Evaluation of a Charge-Reduced Heat Pump Module Using 150 g of Propane" (2020b) [18].

To achieve the targeted reduction in refrigerant charge, the volume of the heat exchangers was reduced by implementing asymmetric plates, the piping length was minimized, the pipe diameters were reduced, and the amount of oil in the system was decreased. The system was designed as a hermetically sealed unit in four different configurations, as presented in Table 1 below.

**Table 1.** Design variants of the heat pump (based on [17]).

| | **Variant 1 V1.0** | **Variant 2 V2.5** | **Variant 3 V2.6** | **Variant 4 V3.0** |
|---|---|---|---|---|
| Compressor type | Scroll | Rotary | Rotary | Rotary |
| Evaporator | 16-plate heat exchanger; long asymmetric | 16-plate heat exchanger; long asymmetric | 16-plate heat exchanger; long asymmetric | 28-plate heat exchanger; long symmetric |
| Condenser | 16-plate heat exchanger; long asymmetric | 16-plate heat exchanger; long asymmetric | 38-plate heat exchanger; short asymmetric | 46-plate heat exchanger; short asymmetric |

In symmetric heat exchangers, the inter-plate channels are identical, resulting in equal flow conditions of the working fluid on both sides; however, an increase in heat transfer is accompanied by a higher pressure drop. Different channel geometries on the hot and cold sides characterize asymmetric heat exchangers. The application of asymmetry improves pressure drop matching across varying mass flow rates and reduces the required heat exchanger surface area [19].

A numerical simulation of the heat pump operation was conducted for the B0/W35 operating point at compressor frequencies of 60 Hz and 120 Hz and for varying refrigerant charges. At 60 Hz, the system achieved a maximum heating capacity of approximately 5 kW and a maximum COP of 4.1 for configuration variant 4 with a refrigerant charge of roughly 300 g of R290. When operating at 120 Hz, the heating capacity nearly doubled for the same configuration variant, while the COP decreased to 3.5.

With a refrigerant charge of 150 g of propane, the system achieved heating capacities of 8.1 kW (variant 3, 120 Hz) and 4.1 kW (variant 3, 60 Hz), with corresponding COP values of 2.9 (variant 3, 120 Hz) and 3.4 (variant 3, 60 Hz). These results demonstrate the significant influence of both compressor frequency and refrigerant charge on system performance under identical boundary conditions.

Experimental investigations were carried out on the developed heat pump prototype. This study focused on measuring the coefficient of performance (COP), analyzing operational stability, assessing operational safety, and evaluating the impact of reduced refrigerant charge on thermal performance. The heat pump was tested in two configurations: System 1, equipped with a scroll compressor, and System 2, utilizing a rotary compressor. The results were obtained for compressor frequencies of 60 Hz and 120 Hz with varying propane charges at the B0/W35 operating point.

Figures 3–5 show a clear non-linear relationship between refrigerant charge and system performance. As the refrigerant charge increases from approximately 120 g to 170–200 g, both the coefficient of performance (COP, defined as the ratio of heating output to electrical input) and the heating capacity increase significantly, reaching their maximum within this range. Beyond 200 g, further increases in charge lead to a decrease in the COP, while heating capacity plateaus or slightly decreases.

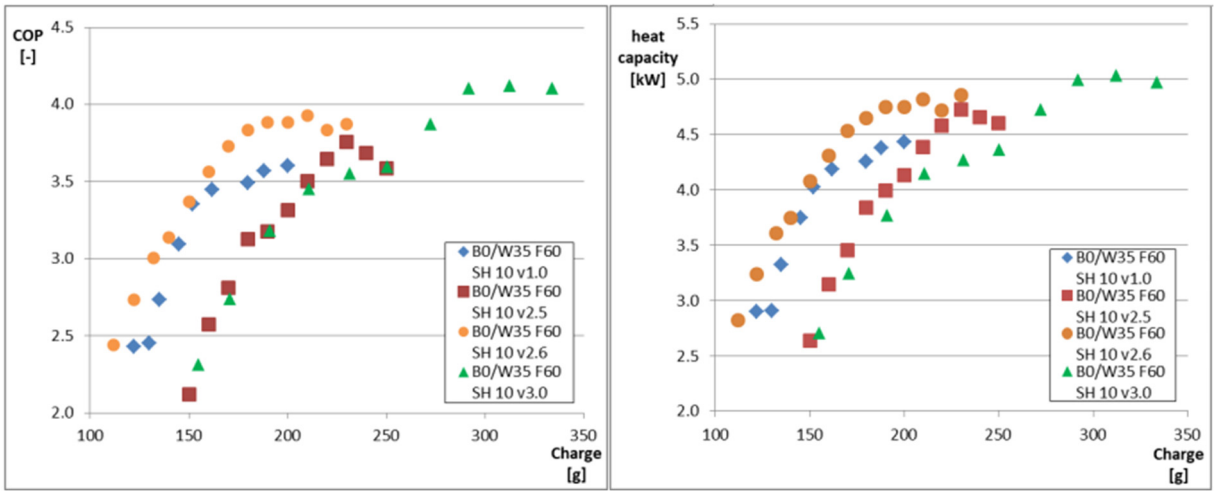



**Figure 3.** Test results for operating point B0/W35 at a frequency of 60 Hz [17].

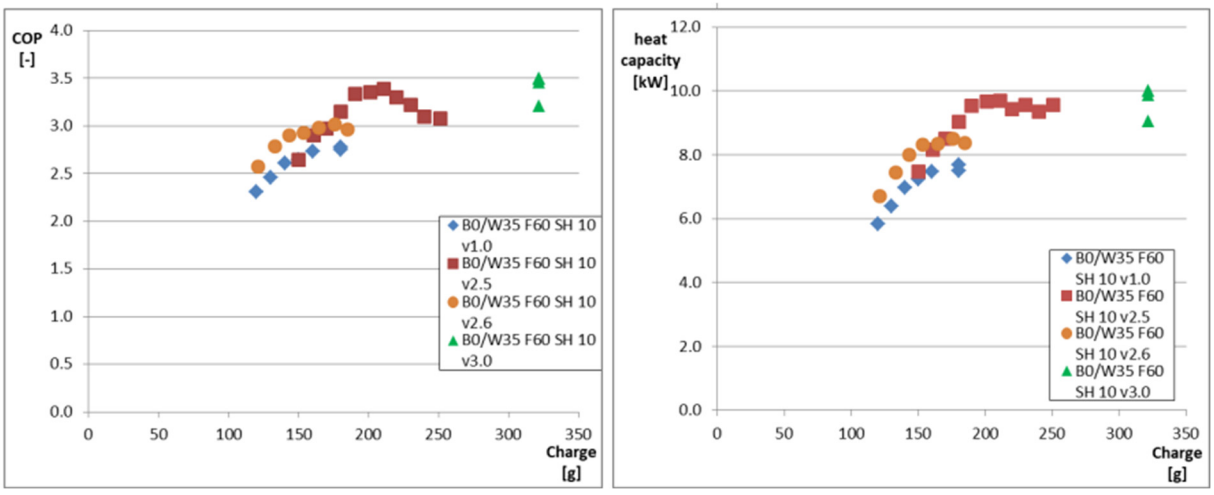



**Figure 4.** Test results for operating point B0/W35 at a frequency of 120 Hz [17].

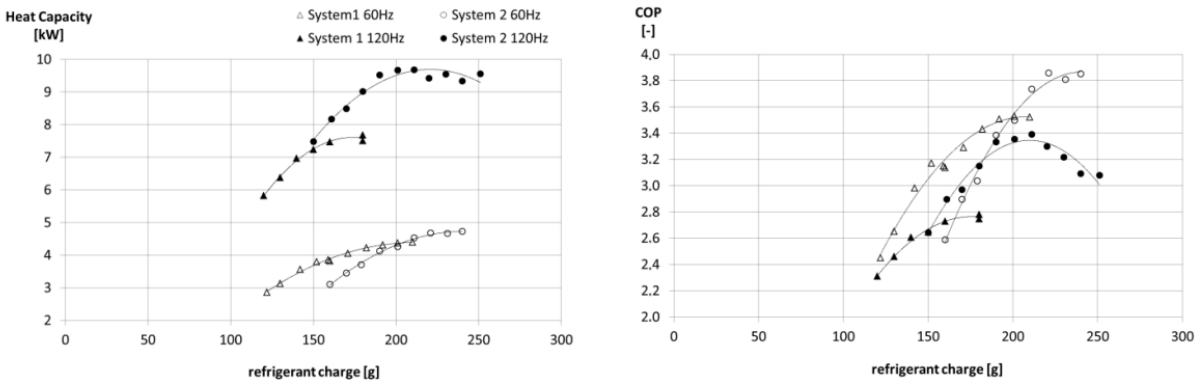



**Figure 5.** Operational test results: heating capacity and COP as a function of refrigerant charge for operating point B0/W35 [18].

This trend is further illustrated at low refrigerant charges (<150 g), where the system operates undercharged, resulting in reduced heat exchanger utilization and lower performance. Conversely, at excessive charge (>200 g), condensing pressure and compressor work increase, thereby reducing the COP despite relatively high heating capacity.

Taken together, these findings indicate that the optimal operating range is consistently 170–200 g across all configurations, ensuring stable operation and peak performance.

Figures 3 and 4 also demonstrate the influence of compressor frequency, which is the compressor's operating speed, measured in hertz (Hz). Increasing the frequency from 60 Hz to 120 Hz significantly enhances heating capacity; however, this is accompanied by a reduction in the COP due to higher electrical power consumption.

Additionally, differences between system configurations are evident. System 1 (scroll compressor) shows a higher COP at a higher refrigerant charge, whereas System 2 (rotary compressor) shows more stable performance across a wider charge range but with slightly lower peak efficiency. These differences highlight the influence of compressor type on optimal system performance.

Figure 5 highlights the combined effect of refrigerant charge (the amount of refrigerant in the system, measured in grams) and compressor frequency (the speed at which the compressor operates, measured in hertz). It also shows the resulting performance differences between the analyzed system configurations. Increasing the compressor frequency from 60 Hz to 120 Hz significantly enhances the heating capacity (up to approximately 10 kW), but at the expense of a reduced coefficient of performance (COP), which is the ratio of heating output to electrical power input, due to higher electrical power consumption.

Building on this, the results also reveal distinct performance characteristics depending on the compressor type. System 1, which is equipped with a scroll compressor, achieves higher efficiency at elevated frequencies, likely due to improved volumetric efficiency at higher refrigerant mass flow rates. In contrast, System 2, which uses a rotary compressor, performs more favorably at lower frequencies, which may be attributed to lower mechanical losses.

For both systems, the optimal refrigerant charge is observed to be 170–200 g, confirming that performance is primarily governed by the combined effects of charge and operating conditions.

These results confirm that both refrigerant charge and compressor frequency must be jointly optimized to achieve the best trade-off between heating capacity and efficiency.

The highest heating capacity achieved was nearly 10 kW at a refrigerant charge of approximately 210 g of R290, as observed in System 2, which was operating at 120 Hz with a maximum COP of 3.8. At a refrigerant charge of 150 g of propane, System 1 achieved heating capacities of 3.9 kW at 60 Hz and 7.1 kW at 120 Hz. System 2 reached a heating

capacity of 7.5 kW when operating at 120 Hz. For a charge of 150 g of R290, a COP of nearly 3.2 was obtained for System 1 at 60 Hz.

The optimal refrigerant charge range for both system configurations was found to be 170–200 g. At higher operating frequencies, System 2, which was equipped with a rotary compressor, performed better, whereas at lower frequencies, System 1, which was also equipped with a rotary compressor, performed better. Conversely, the highest COP values were achieved at a refrigerant charge of approximately 200 g and at lower operating frequencies. Tests performed on the unit equipped with a rotary compressor at different lower heat source temperatures revealed that the heating capacity decreased from 5.9 kW to 3.9 kW as the source temperature dropped from 10 °C to −7 °C while maintaining a constant upper heat source temperature.

These results confirm that refrigerant charge and compressor operating frequency are key parameters governing the performance and stability of propane heat pump systems. Based on the analysis of Figures 3–5, a clear non-linear relationship between refrigerant charge and system performance is observed, with an optimal charge range of 170–200 g, where both the COP and the heating capacity reach their maximum values under stable operating conditions.

Deviations from this optimal range result in performance degradation. Undercharged conditions (<150 g) result in insufficient heat exchanger utilization and reduced efficiency, whereas an excessive charge (>200 g) increases condensing pressure and compressor work, leading to a decline in the COP despite relatively high heating capacity. Furthermore, increasing the compressor frequency from 60 Hz to 120 Hz significantly enhances heating capacity, but at the cost of a reduced COP due to higher electrical power consumption.

The results also demonstrate that system configuration plays a significant role in performance characteristics. The scroll compressor (System 1) performs more efficiently at higher operating frequencies, while the rotary compressor (System 2) shows better performance at lower frequencies, indicating different sensitivities to operating conditions.

A comparative summary of the influence of refrigerant charge on the COP, heating capacity, and operational stability is presented in Table 2, highlighting the identified optimal charge range and the associated performance trade-offs.

**Table 2.** Effect of the R290 refrigerant charge on the COP and system stability (based on [17,18]).

| Refrigerant Charge Range | Effect on COP | Effect on Heating Capacity | Stability | Interpretation | Operating Recommendation |
|---|---|---|---|---|---|
| <150 g | decrease | decrease | unstable | undercharged | avoid operation |
| 150–170 g | moderate | moderate | transitional | lower operating limit | suboptimal |
| 170–200 g | maximum | maximum | stable | optimum | recommended |
| >200 g | decrease | increase or stagnation | reduced stability | overcharged | avoid overcharging |

In the article published in 2022, titled Effect of Thermoelectric Subcooling on COP and Energy Consumption of a Propane Heat Pump (2020) [20], the impact of thermoelectric subcooling (TESC) on the coefficient of performance (COP), cooling capacity, and energy consumption of an air-to-water propane heat pump intended for single-family house heating was investigated. Marlow RC12-6-01LS (Marlow Industries (II-VI Marlow), Dallas, TX, USA) modules were selected for the study. The heat pump model was equipped with an internal heat exchanger (IHE), the schematic of the device is shown in Figure 6. The models of the vapor compression heat pump and the thermoelectric subcooling system

were implemented in MATLAB R2020b and integrated into a hybrid system. The TESC model employed the finite difference method to simulate the subcooling process.

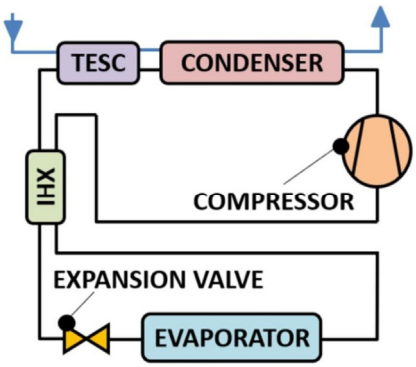



**Figure 6.** Schematic representation of the developed heat pump model with TESC [20].

The simulation study considered variable parameters, including ambient temperature ranging from −20 °C to 15 °C, water outlet temperatures of 40 °C and 55 °C, the number of TESC blocks (from 1 to 8; each block consisting of two modules), and the supply voltage of the thermoelectric modules (ranging from 0.5 V to 10 V DC). These parameters enabled the evaluation of electrical energy consumption, the degree of subcooling, the propane mass flow rate, heating capacity, compressor performance, and the COP. Additionally, an analysis of the advantages of the proposed system for space heating applications in different global locations was conducted.

The heat pump operating parameters were defined as follows:

- Temperature difference at the condenser (condensation): 5 K;
- Temperature difference at the evaporator: 7 K;
- Subcooling at the condenser: 3 K;
- Effectiveness of the internal heat exchanger (IHE): 50%;
- Useful superheating: 5 K;
- Non-useful superheating: 5 K.

Prior to the simulations, the parameters of the TESC blocks were defined, including the module type, the number of blocks, the supply voltage, and the working fluid parameters—namely, the inlet temperature and mass flow rate of propane and water. By coupling the propane heat pump model with the thermoelectric subcooling model, comprehensive simulation results were obtained. The number of TESC blocks was found to have a significant impact on overall system performance. As the number of blocks increased, the COP improved; for eight blocks, an increase of 7.64% was achieved at 40 °C, while 12.29% was achieved at 55 °C, compared to the system without TESC. The heat pump equipped with four TESC blocks also demonstrated satisfactory performance, with COP improvements of 6.03% and 9.54%, respectively. The degree of refrigerant subcooling at the condenser outlet varied with the number of TESC blocks (1–8) and ranged from 5.4 K to 19.07 K at a water temperature of 40 °C, and it varied from 6.46 K to 27.38 K at 55 °C. An increase in subcooling enhanced the TESC system's cooling capacity and improved heating performance. Regarding the heat pump system's cooling capacity, applying thermoelectric subcooling increased the cooling capacity per unit mass of propane; however, the propane mass flow rate decreased with increasing TESC block count, resulting in only minor changes in the overall cooling capacity.

By analyzing the COP values as a function of the number of applied TESC blocks and ambient temperature, as presented in Figures 7 and 8, it can be concluded that the COP increases with rising ambient temperature. Systems with more TESC blocks achieve higher

COP values. The application of TESC blocks in a heat pump system delivers measurable benefits, including an increased COP and improved overall system performance. Based on the simulation results, it was found that a higher supply voltage applied to the TESC system increases the heat pump's heating capacity.

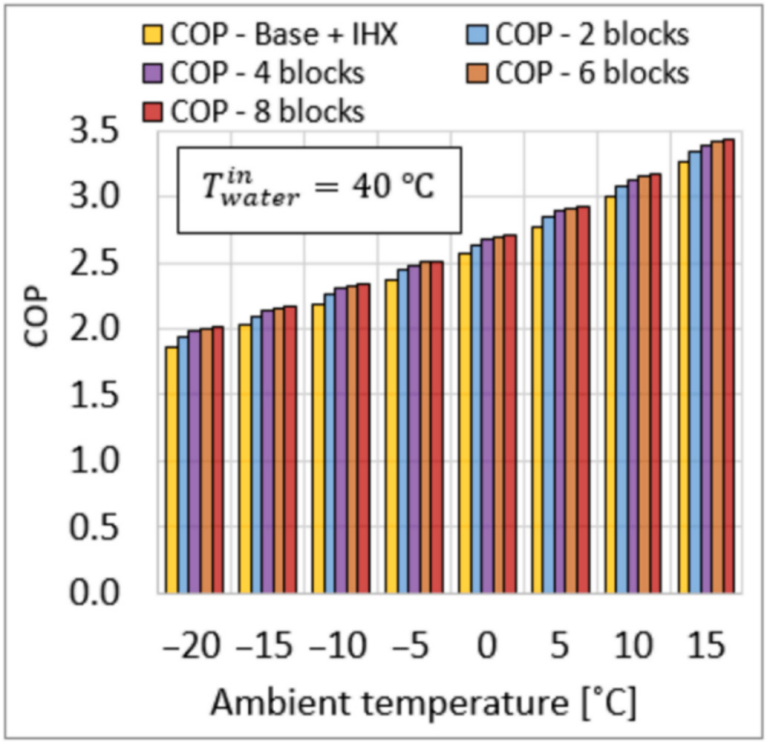



**Figure 7.** Results of studies showing the dependence of the COP on ambient temperature for water temperature at the inlet of 40 °C [20].

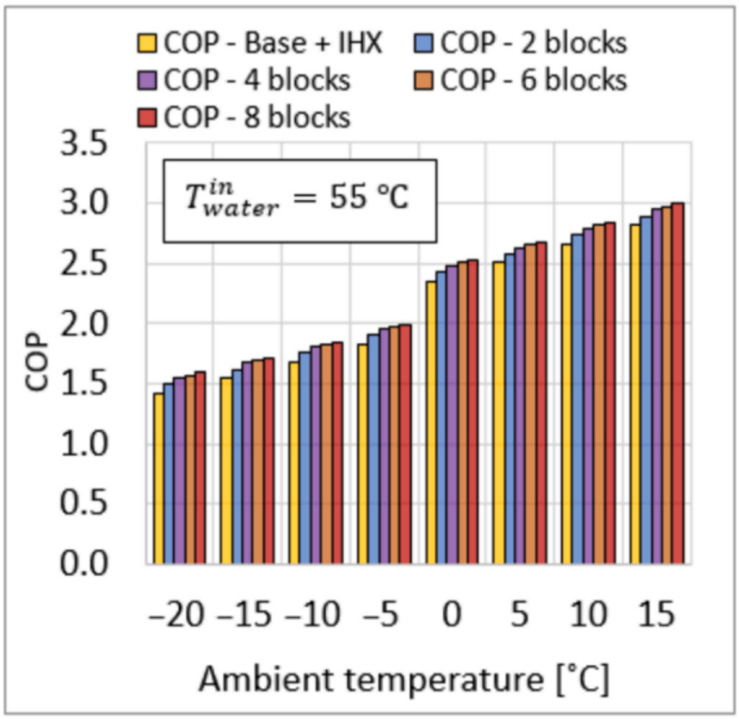



**Figure 8.** Results of studies showing the dependence of the COP on ambient temperature for water temperature at the inlet of 55 °C [20].

Additionally, a simulation of the system operation was performed for a single-story residential building with a floor area of 266.9 $m^2$ and a total internal volume of 1009 $m^3$; it was equipped with radiators and fan coil units and characterized by an overall heat transfer coefficient (UA) of 256.96 W/K. Based on these parameters, the space heating loads were determined to be 10.2 kW at an outdoor temperature of −20 °C and 1.28 kW at 15 °C. The analysis covered various climatic zones represented by selected cities worldwide:

- Helsinki (Finland);
- Minneapolis (USA);
- Harbin (China);
- Teruel (Spain);
- Saint Petersburg (Russia);
- Montreal (Canada).

The graphs in Figures 9 and 10 show that application of thermoelectric subcooling with an eight-block TESC configuration increased the seasonal coefficient of performance (SCOP) compared to the system without TESC. A greater number of thermoelectric subcooling blocks yielded greater improvements in the heat pump's seasonal performance. For a water inlet temperature of 40 °C and with the use of eight TESC blocks, the SCOP increase was recorded as follows:

- Helsinki (Finland): 5.82%;
- Minneapolis (USA): 6.21%;
- Harbin (China): 6.57%;
- Teruel (Spain): 5.45%;
- Saint Petersburg (Russia): 5.82%;
- Montreal (Canada): 6.09%.

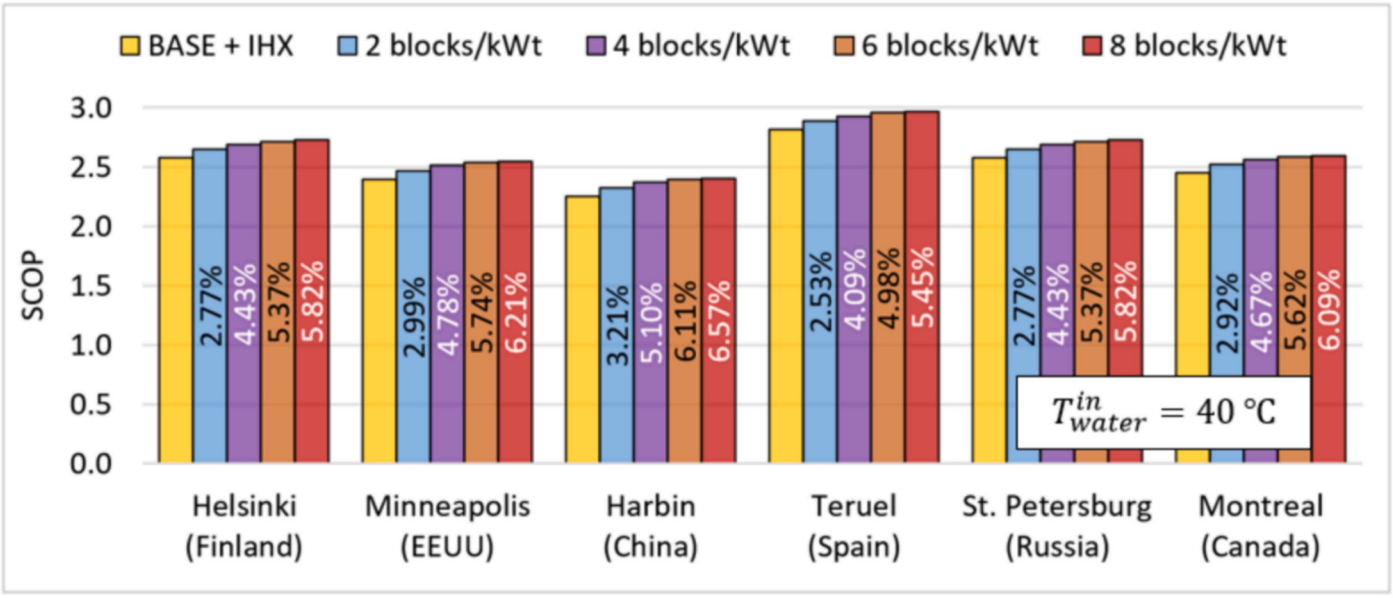



**Figure 9.** SCOP value for different locations and the number of TESC blocks used for an inlet water temperature of 40 °C [20].

For a water inlet temperature of 55 °C and the application of eight TESC blocks, the SCOP increase was as follows:

- Helsinki (Finland): 8.11%;
- Minneapolis (USA): 9.12%;
- Harbin (China): 9.98%;

- Teruel (Spain): 6.90%;
- Saint Petersburg (Russia): 8.10%;
- Montreal (Canada): 8.85%.

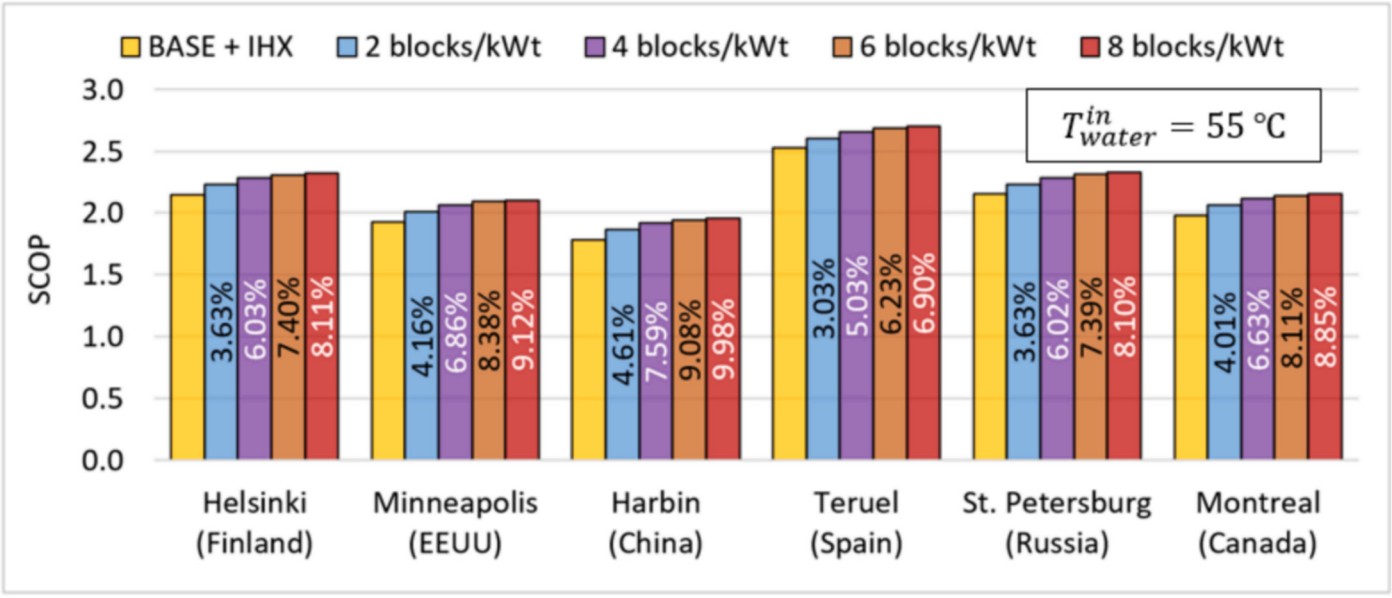



**Figure 10.** SCOP value for different locations and the number of TESC blocks used for an inlet water temperature of 55 °C [20].

## 4. Conclusions

HVAC systems play a significant role in decarbonization by replacing conventional heating sources while simultaneously reducing the sector's overall environmental impact.

Results from studies on heat pumps with reduced propane refrigerant charge indicate that it is possible to minimize refrigerant mass while maintaining satisfactory operational performance. The refrigerant charge has a substantial influence on both operational stability and system efficiency. The optimal refrigerant charge for stable operation was found to be 170–200 g. A charge of 150 g proved insufficient to ensure stable operation; however, it fulfilled safety requirements and minimized the potential environmental impact. The reviewed results indicate that performance degradation at a low refrigerant charge is primarily due to thermodynamic and flow-related limitations rather than the type of compressor.

A non-uniform refrigerant distribution in the heat exchangers and increased pressure losses led to higher electrical energy consumption, resulting in a reduced COP. To mitigate this effect, the refrigerant flow and distribution within the system should be optimized through improved control of the expansion valves. Nevertheless, the obtained results demonstrate that reducing the refrigerant charge in heat pumps can still yield satisfactory performance while also indicating promising directions for further technological development.

The integration of novel design solutions, such as thermoelectric subcooling blocks, into vapor-compression heat pump systems can significantly improve performance indicators, such as the COP and the SCOP. Optimizing operating parameters can be achieved by selecting an appropriate number of TESC blocks based on the specific system requirements. This approach highlights the potential for developing technologies tailored for operation in cold climates (e.g., Helsinki). However, TESC introduces additional electrical consumption and system complexity. Therefore, its effectiveness depends on achieving a net positive energy balance, which is more favorable at higher temperature lifts.

In conclusion, the conducted research confirms the validity of developing heat pump systems with minimized R290 refrigerant charge; however, particular attention must be paid to operational optimization to maintain high levels of efficiency, stability, and safety. The findings of this study constitute a valuable contribution to advancing heat pump technology, enhancing system performance, and potentially reducing the environmental impact.

The above studies demonstrate that it is possible to minimize the refrigerant charge while maintaining satisfactory system performance. Furthermore, integrating additional components into the heat pump system can enhance the COP, particularly under demanding conditions typical of cold-climate zones. When combined, charge-reduction strategies and system-level enhancements offer significant potential for improving overall heat pump performance; however, this also introduces increased system complexity. The reviewed research thus provides a meaningful contribution to the development of next-generation propane-based heat pump technologies for cold climates. Nevertheless, it should be noted that most available results are based on simulations, underscoring the need for long-term experimental validation. Furthermore, the trade-off between system complexity, energy consumption, and performance improvement requires careful evaluation in practical applications.

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