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Economic Optimization of Rotary Heat Exchangers Using CO_2 Pricing Scenarios Based on Validated Fluid Dynamic and Thermodynamic-Based Simulation

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Abstract: Rotary heat exchangers have been widely used in paint shops, combustion power plants, and in heating, ventilation, and air conditioning systems in buildings. For these processes, many types of heat exchangers are available in the market: Tube-shell heat exchangers, plate heat exchangers, and rotary heat exchangers, among others. For the rotary heat exchangers, the problem is that there is no net present value method and lifecycle assessment method-based optimization found in the literature. In this work, we address this issue: An optimization is carried out with help of an empirically validated simulation model, a life-cycle assessment model, an economical assessment, and an optimization algorithm. The objective function of the optimization simultaneously considers economic and environmental aspects by using different CO_2 pricing. Different CO_2 pricing scenarios lead to different optimization results. The ambient air empty tube velocity $v_{a,\,2.1}$ optimum was found at 1.2 m/s, which corresponds to a specific mass flow m_{sp} of 5.4 kg/(m^2·h). For the wave angle β, the optimum was found in the range between 58° and 60°. For the wave height h^* the optimum values were found to be between 2.64 mm and 2.77 mm. Finally, for the rotary heat exchanger length l, the optimum was found to be between 220 mm and 236 mm. The optimization results show that there is still potential for technical improvements in the design and operation of rotary heat exchangers. In general terms, we recommend that the optimized rotary heat exchanger should cause less pressure drop while resulting in similar heat recovery efficiency. This is because the life cycle assessment shows that the use phase for rotary heat exchangers has the biggest impact on greenhouse gases, specifically by saving on Scope 2 emissions.

Keywords: rotary heat exchanger; thermal wheel; Kyoto wheel; optimization; CO_2 pricing

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1. Introduction

Nearly 100 years ago, in 1922, the Swedish engineer Frederick Ljungström invented the first rotary heat exchanger (RHX) made of steel for power plants [1]. This is a type of heat exchanger whose compact design works efficiently, and it was patented in 1930 [2]. Since then, researchers have continued to improve its design and operation based on expertise in thermodynamics, fluid dynamics, and structure engineering [1]. RHX became one of the most efficient types of heat exchangers for gas–gas streams and were later produced in aluminum in new applications for ventilation or paint shops [3–5]. According to Warren, "The story of this development work is a good example of how a basically simple idea can be developed and refined by coordinated efforts in countries around the world into the carefully engineered product that it is today" [1]. Until today, it has thus been used intensively in specific industrial processes [1] and in heating, ventilation, and air-conditioning (HVAC) in buildings [3] as a key element in heat recovery systems (HRS). Heat exchangers in general have been optimized over the years in order to improve their performance, especially tube-shell heat exchangers [6–8]. Other types of heat exchangers are also available in HVAC applications, such as regenerative heat exchangers and membrane heat exchangers [9]. Regenerative heat exchangers are the most efficient for HVAC purposes [4]. Regenerative heat exchangers, which is the classification in which RHXs belong, are challenging in terms of mathematical modeling [10]. Other authors have also tried to integrate economical or environmental aspects into the optimization of RHXs, but based on closed methods [11], which are limited in accuracy [12].

Nowadays, many manufacturers worldwide produce RHXs with different designs and efficiency levels. This efficiency is independently tested and certified by the organization Eurovent and their results are publicly available online. Thousands of different models made by various manufacturers from around the world have been certified [13].

The European Commission has passed laws governing ventilation systems over the last decade that increase HRS thermal efficiency above 67% starting in 2016 and above 73% as of 2018 for new buildings [14]. Thermal efficiency is defined by this law [14] as:

$$\eta = \frac{\vartheta_{2.2} - \vartheta_{2.1}}{\vartheta_{1.1} - \vartheta_{2.1}} \tag{1}$$

where η is the HRS thermal efficiency, and ϑ is the temperature for the corresponding stream in $^\circ$ C; $\vartheta_{2 \cdot 2}$ is the temperature of the supply air leaving the HRS and entering the room, $\vartheta_{2 \cdot 1}$ is temperature of the ambient air, and $\vartheta_{1 \cdot 1}$ is temperature of the process air, leaving the room and entering the HRS [14]. Schematics of the streams are depicted in Figure 1, where the type of HRS is an RHX and $\vartheta_{1.1} > \vartheta_{2.2} > \vartheta_{1.2} > \vartheta_{2.1}$ when the RHX is used as a preheater. When the RHX is used for cooling, then $\vartheta_{2.1} > \vartheta_{1.2} > \vartheta_{2.2} > \vartheta_{1.1}$ is used, but the definition of efficiency from Equation (1) is still valid. Kaup [15] advises that the European Union could further increase thermal efficiency η requirements, since it is assumed that the requirements in the enacted legislation will need to be reviewed.

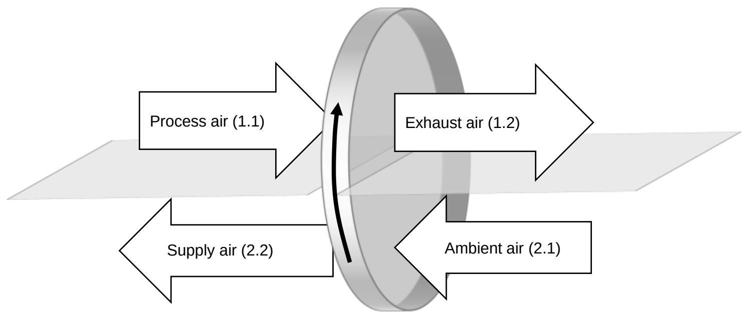

Figure 1. RHX working principle and stream names.

A further political influence is a price increase in taxes for emissions of greenhouse gases (GHG), which are measured in €/t CO_2 Equivalents and regularly referred to as "CO_2 pricing." As of now, in Germany, this only applies to industries and the energy sector, whereas in the future, it would also apply to buildings. This price increase will start with 25 €/t CO_2 Eq. in 2021 and will gradually increase until 2026, when the price should be between 55 and 65 €/t CO_2 Eq. [16]. Including buildings in CO_2 pricing creates both an opportunity and a challenge for further improving RHX design and operation.

For manufacturers, the challenge to produce competitive and affordable RHXs becomes more difficult because highly efficient heat exchangers frequently need more material [12] and are more expensive to manufacture. From the customer's point of view, many RHX models already on the market comply with the EU requirements [14]. However, since there is a myriad of RHX models and multiple possible operational points, it is not clear which is the best option to choose in light of the upcoming challenges. Furthermore, not all customers are completely aware of the conflict between the operation (heating and ventilation costs and energy monetary savings due to the HRS) and investment costs for their facilities. Therefore, customers might invest in a cheaper system in the construction stage of the project but then are forced to pay high heating costs when operating the facility. This is a short-sighted view. Hence, the entire product life cycle should be assessed in terms of economics and the environment, taking into account the thermodynamics and fluid mechanics aspects that have an impact on the RHX thermal efficiency.

The authors of this article have previously successfully developed a model for life cycle assessment (LCA) and cash flow analysis to calculate net present value (NPV) and internal rate of return (IRR) for RHXs [17]. This previous model shows the magnitude of the trade-off when optimizing either the economic or ecological aspects of RHXs. In that study, RHXs efficiently saved on GHGs and were economically less appealing (5% IRR) while RHXs were more economically appealing (70% IRR) and saved around 40% less GHGs than the aforementioned case [17]. This conflict arises from the fact that electricity is required to operate the RHX. Ventilators have to overcome the pressure drop produced by the RHX are therefore the largest energy consumers. Since heat and electricity have different prices and different GHG emissions per energy unit, basing a purchase on pure energetic optimization is short-sighted. Additionally, some RHXs that require more material to manufacture produce greater amounts of GHG emissions during the production process and also entail higher investment costs, but they are often more efficient during the use phase. This is an additional trade-off worth investigating.

Therefore, the purpose of this work and the innovation of this article is to carry out an economic optimization of RHXs design parameters and operating conditions. Furthermore, the ecological aspect is taken into account by monetizing the GHGs. This optimization is based on a fluid dynamic and thermodynamics simulation while considering the economic and ecological aspects by assigning different scenarios for CO_2 pricing. The idea of these scenarios is to study how sensitive the design and operating parameters are to the CO_2 pricing.

It is worth noting that no previous work has been found within RHX literature that studies this economic or ecological optimization conflict. Unfortunately, scientific research on this topic is relatively scarce and further improvements proven by empirical evidence are restricted to individual cases and limited on the optimization scope to improving heat transfer efficiency [18–22]. This is probably due to the difficulty of simulating RHXs and the very specific technology, almost unknown outside its niche applications. Nevertheless, ventilation systems are gaining awareness in the general population, and the pressure on energy recovery and CO_2 savings is continuously increasing. Rotary heat exchangers can be part of the solutions on these topics.

2. Materials and Methods

RHXs are sometimes referred to as “thermal wheels,” “Kyoto wheels,” “heat recovery wheel,” “rotary air-to-air enthalpy wheel”, or “heat recovery wheel,” depending on the field of application, but the functioning principle is always the same. The working principle of RHX is based on the capacity of materials to store heat. Since heat is to be recovered from a specific gas stream (process air 1.1 in Figure 1), this stream flows through one sector of the RHX. At the same time, another gas stream (ambient air 2.1 in Figure 1) needs to be heated and flows in the opposite direction through the other sector of the RHX. The two streams are separated by sealings and a wall, which are integrated into a housing (not illustrated in Figure 1). Only the RHX, which is continuously rotating inside the housing, comes into contact with both streams. In this sense, it always absorbs heat in the one sector and releases heat in the other sector.

2.1. RHX Simulation: Design and Operational Parameters

RHX are manufactured by simultaneously coiling flat and a corrugated aluminum sheets as depicted in previous work [17]. The result is a “honeycomb” structure as shown in Figure 2. Operational parameters and design parameters influence the thermal efficiency η and the pressure drop of the RHX. Previous research conducted by the authors of this article [12] yielded a successfully developed simulation using the LINUS (Local Internal Nusselt Number for Sine Ducts) model for RHX, based on thermodynamic and fluid dynamic aspects that take these design and operational parameters into account. The simulation with the LINUS model was validated using empirical data from a pilot plant and data from the Eurovent database. This simulation makes it possible for a given set of design and operational parameters to predict the thermal efficiency η and pressure drop of the RHX within acceptable real-life limits. Furthermore, this simulation using the LINUS model delivers the most accurate predictions in thermal efficiency η found in literature [12]. This simulation is the basis for predicting heat energy savings and electricity costs. Following the same logic as in the previous work [12,17], the parameters for optimizing RHXs are:

- Wave height h^*: Distance between two metal sheets in which the waves are formed as shown in Figure 2.
- Wave angle β: Refers to the angle formed between the straight segment of the wave and the flat aluminum layer, also shown in Figure 2.
- RHX length l: Distance that the gas flows inside the RHX, as shown in Figure 3.
- Empty tube gas velocity v_a: The gas velocity measured just before or after the RHX as shown in Figure 3. Since gas velocity is a parameter that is temperature dependent, it changes the value from ambient air (2.1) to supply air (2.2) or from process air (1.1) to exhaust air (1.2). Therefore, in this work, it will only refer to the empty tube gas velocity of the ambient air (2.1) $v_{2.1}$. Moreover, since the RHX partially covers the flow area, the internal gas velocity is higher. This aspect has been already covered in previous work [12]. In this work, it is also assumed that no air flow takes place through the RHX seal clearances (no sealing leakages) and that the mass flow through both flow sides is identical (no infiltration). Hence, mass flow $\dot{m}$ is identical (no purge) to the four streams depicted in Figure 1. Furthermore, it is assumed that the flow to the RHX is evenly distributed, and as a consequence, the mass flow per surface area $\dot{m}_{sp}$ is also constant, which is equal to the product of the empty tube gas velocity v_a and density ρ of the corresponding stream.

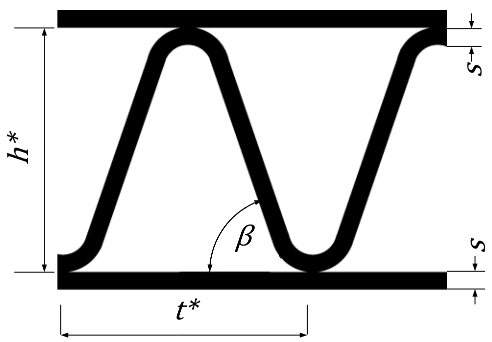

Figure 2. RHX cross-section.

Often in the literature, other parameters are considered to have an influence on heat exchangers thermal efficiency η. However, for the following reasons, they are not considered in this work on RHX optimization:

- Wavelength t^* shown in Figure 2: Since its value is directly dependent on wave height h^* and wave angle β, it is not necessary to be included. However, this is the method Eurovent uses to report their data [13].
- Material thickness s, shown in Figure 2: The developed simulation [12] demonstrated that RHXs with a smaller material thickness (s = 60 µm) have a higher thermal efficiency than thicker ones (s = 100–200 µm). This effect is consistent with the Eurovent data [13]. Additionally, thicker material requires more aluminum mass and therefore increases the RHX price and GHG emissions [17]. Consequently, there

is no economic or ecological conflict in this parameter. In addition, our cooperation industrial partner suggested the constructive challenges of material thinner than 60 µm be used. As a result, the material thickness s was set to 60 µm.

- Rotational speed n of the RHX shown in Figure 3: Higher rotation speed means higher thermal efficiency. This is consistent with simulation experimental data [12] and the literature [5]. Since the RHX drive train's power consumption is several orders of magnitude smaller than the ventilators' power consumption, rotational speed n does not significantly affect the system's electrical demand and therefore does not significantly affect the operation costs or the GHG emissions. Moreover, after a given rotational speed n, which is dependent on the RHX design and operational parameters [12], thermal efficiency is not further increased [5,23]. Therefore, rotational speed n was set to a constant value of 24 rpm for the purposes of this study.

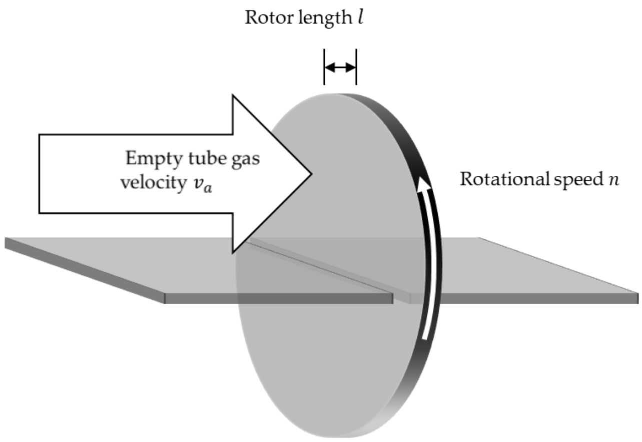

Figure 3. RHX length l and operational parameters.

2.2. Ecological and Economic Assessment Methods

The ecological evaluation is carried out using an LCA by implementing a cradle-to-grave approach. This means that the RHX life cycle is analyzed based on the raw material extraction through its end of life. Therefore, the LCA is divided into the following phases: Manufacturing (production), the use phase, and end of life (recycling). Previous work has already been done on this and the most important indicator is the GHG emissions (in CO_2 Eq.) when assessing the life cycle of an RHX [24]. The functional units (FU) chosen are 10,000 m^3/h and 100,000 m^3/h of ambient air (2.1 in Figure 1) that flows into the system at 10° C. Since temperature and volume flow act as the FU, these are the equivalent in mass flow $\dot{m}$ to 12,460 kg/h and 124,600 kg/h correspondingly. The reason for the FUs is due to transportation aspects: RHXs with diameters greater than 2900 mm are segmented into smaller pieces after being manufactured and rebuilt at their final location. Therefore, the smaller FUs focuses on the optimization of single-unit RHX and the larger FU focuses on the segmented ones. Since their pricing is calculated using different equations [24], these different equations would have to be used in these two scenarios.

The economic assessment is based on the same cash flow analysis as a previous work [17]. This previous research has shown that an optimization with IRR as an objective function results in an RHX that does not comply with the minimum requirements of EU law. Furthermore, RHXs with higher IRR have lower GHG emissions savings and RHXs with higher GHG emissions savings have lower IRR. In addition, because of the IRR definition, lower investments are preferred. Therefore, in this work, the economic aspect is evaluated using the NPV and not the IRR. Regarding investment costs, the GHG emissions costs had been added with help of the CO_2 pricing. In the use phase, the electricity and gas the costs increase according to the GHG emissions and corresponding pricing. Additionally, regarding the end-of-life step, revenue is added that takes the GHG emissions savings from the recycling process into account.

RHXs typically have a 10- to 20-year service lifetime, depending on the application. Therefore, the more conservative value of 10 years is assumed. It is also assumed that the prices and emissions for gas and electricity are the ones currently charged on the German market, since it is very likely that the CO_2 pricing could be applied in Germany [16]. The conditions for the economic and ecological assessment are presented in Table 1 below.

Table 1. Economic and ecological assumptions.

Condition	Value
FU: Ambient air volume flow $\dot{V}_{2.1}$	10,000.0 m^3/h or 100,000.0 m^3/h
Ambient air mass flow $\dot{m}_{2.1}$	12,460.0 kg/h or 124,600.0 kg/h
Temperature ambient air $\vartheta_{2.1}$	10 °C
Temperature exhaust air $\vartheta_{1.2}$	20 °C
Rotational speed n	24 rpm
Product lifetime	10 years
Working hours per rotor lifetime	20,000 h
Gas price	0.06 €/(kWh)
Natural gas industrial burner efficiency	95% [25]
Natural gas emissions at furnace	0.2506 kg CO_2 Eq./(kWh) [25]
Electricity price	0.30 €/(kWh)
Electricity emissions	0.6476 kg CO_2 Eq./(kWh) [26]
RHX recyclable percentage	90% w/w
GHG emissions scope	Direct GHG emissions (Scope 1) and electricity (Scope 2)

2.3. Optimization Method

The C# simulation algorithm developed in previous work [12] is combined with an optimization algorithm. Previous experience [17] with commercial optimization software showed that the commercial algorithm presented difficulties in finding the best values in the vicinity of the optimum and required immense computation time to find many digits after the decimal point for the rotor length, for example. This accuracy is not required for practical purposes. Unfortunately, the commercial software did not allow limiting the accuracy of the optimization. Additionally, the commercial software is a "black box," meaning that the documentation does not reveal the methods used. Therefore, in this work, an algorithm is used based on multi-dimensional, unconstrained optimization with a univariate direct search method. The way this method works is depicted in Figure 4 and described as follows: An initial set of values for each dimension (wave height h^*, wave angle β, RHX length l, and ambient air empty tube gas velocity $v_{2.1}$) is selected by taking the mean value of the minimum and maximum values given (the start vector $\vec{x_0}$ is then created). Initially, the working vector $\vec{x_i}$ is created and equal to the starting vector $\vec{x_0}$. Then a specific dimension is selected (e.g., wave height h^*). For each value between the minimum and maximum values and with a step size equal to the accuracy a, the simulation is carried out and the NPV(x_i) is calculated. According to the maximum value of NPV(x_i) obtained, the corresponding x_i is selected for replacing the original value from the working vector $\vec{x_i}$ and this process is repeated for the following three dimensions. After every dimension has been evaluated, the working vector $\vec{x_i}$ is checked to determine whether it equals starting vector $\vec{x_0}$. If they are equal, the optimization is terminated. If they differ from each other, the process starts over from the point where the dimensions are selected and now starting vector $\vec{x_0}$ equals working vector $\vec{x_i}$.

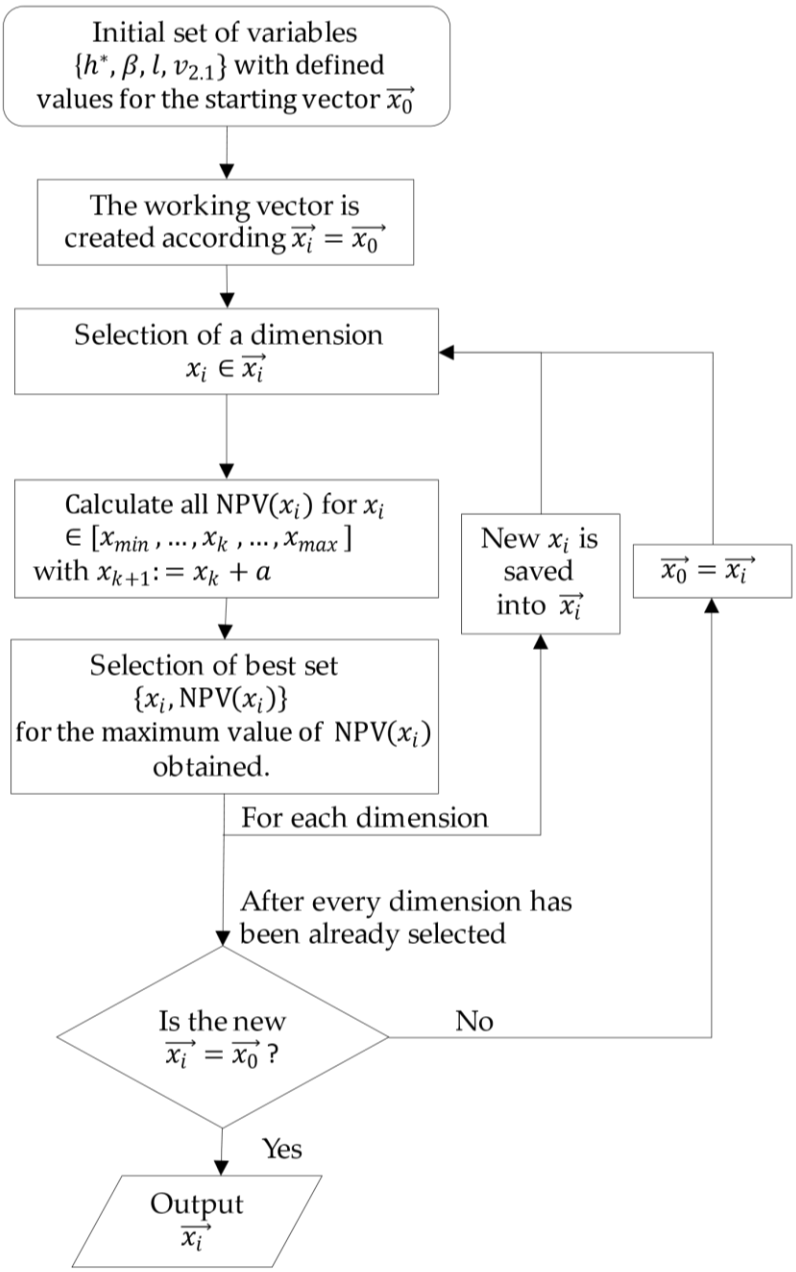

Figure 4. Optimization algorithm flow diagram.

It is documented in the literature [27] that the multi-dimensional unconstrained optimization with a univariate direct search method loses efficiency but not efficacy in finding the maximum in the vicinity of the optimum. To avoid this unnecessary and considerable computation time, the accuracy for each variable of the optimization was kept constant at a reasonable and practical value, depending on the variable itself. The accuracy used and the ranges are summarized in Table 2.

Table 2. Variables, accuracy, and ranges (including endpoints) for the optimization.

Variable	Unit	Accuracy	Min.	Max.
		a	x_{min}	x_{max}
Wave height $\boldsymbol{h}^*$	mm	0.01	1.00	4.00
Wave angle $\boldsymbol{\beta}$	°	0.1	30.0	60.0
RHX length $\boldsymbol{l}$:	mm	1	150	2000
Ambient air empty tube gas velocity $\boldsymbol{v_{2.1}}$	m/s	0.1	1.0	4.0
(Specific mass flow $\dot{m}_{sp}$)	kg/(m^2·h)		4.5	17.9

These ranges are roughly based on the ranges for which the nearly 1300 Eurovent aluminum RHXs have been certified [13]. These Eurovent ranges are presented in Table 3.

Table 3. Eurovent database minimum and maximum reported values.

Variable	Unit	Eurovent Minimum	Eurovent Maximum
Wave height $\boldsymbol{h}^*$	mm	1.35	3.80
Wave angle $\boldsymbol{\beta}$	°	30.0	60.0
RHX length $\boldsymbol{l}$:	mm	100	270
Ambient air **empty tube gas velocity** $v_{2.1}$	m/s	1.0	3.0
(Specific mass flow $\dot{m}_{sp}$)	kg/(m^2·h)	4.6	13.8

It should be noted that RHXs with an RHX length smaller than the commonly built 200 mm length provide lower thermal efficiency. Previous work [17] has shown that longer RHXs save more on GHG emissions, although they require more material. Therefore, when a price is set for CO_2 Eq. emissions, it is expected that the optimum RHX is longer than the usual 200 mm. Additionally, these RHXs that are smaller than 200 mm rarely reach the 73% thermal efficiency [13] required by law [14]. Still, for giving a possible range for improvement, RHX lengths l greater than or equal to 150 mm were considered.

Regarding the wave angle range, not only is the Eurovent database limited but so is the simulation model [12] because the models for the Nusselt number, which is used to calculate the heat transfer coefficient, is limited to maximum 60° for sine ducts [28].

The goal of this optimization process is to determine how the CO_2 pricing influences the optimal design and operational parameters of RHXs, while the NPV is objective function to be maximized. For reasons of simplification, the inflation rate is considered to be 0%. Based on political reasons [16], the CO_2 pricing defines the following scenarios:

- Scenario with 0 €/t CO_2 Eq.
- Scenario with 25 €/t CO_2 Eq.
- Scenario with 55 €/t CO_2 Eq.
- Scenario with 65 €/t CO_2 Eq.

Additionally, two optimization scenarios from previous work [17] have been updated under these conditions from Table 1 and presented for comparison reasons:

- Scenario of GHG emissions savings, where the objective function is to maximize CO_2 Eq. savings. In this case, a CO_2 pricing or inflation do not play a role.
- Scenario IRR, where the objective function is the maximization of IRR where CO_2 pricing is set to 0 €/t CO_2 Eq. and inflation is considered 0% annually. Please note that in the previous work, the IRR was used as an objective function instead of the NPV.

2.4. Comparing the Optimization Results to Eurovent Model

Finally, the results of the optimization step are compared in terms of GHG emissions and NPV to a Eurovent-certified RHX, model P_17-1100-WZV, whose thermal efficiency information is available online [13]. This RHX has been simulated successfully by the authors in the past [12]. This is an RHX with a high thermal efficiency (78–83%), so it is considered state of the art. The simulated RHX has the characteristics listed in Table 4.

Table 4. Eurovent model P_17-1100-WZV.

	Wave Height	Wave Length	RHX Length	Ambient Air Empty Tube Velocity	Specific Mass Flow	Rotational Speed	Material Thickness
	h^* mm	t^* °	l mm	$v_{2.1}$ m/s	$\dot{m}_{sp}$ kg/(m^2·h)	n rpm	s µm
Eurovent RHX	1.4	3.8	200	1.0, 2.0 and 3.0	4.6, 9.2 and 13.8	14	60

Since this Eurovent RHX is operated at three different ambient air empty tube velocities $v_{2.1}$, each of these operating conditions will be included in the comparison with the optimization results. For attaining the same FU, the RHX diameter is adjusted so the product of the cross-sectional area and the velocity is constant. Therefore, lower speeds require larger diameters and likewise larger velocity requires a RHX with a smaller diameter. The decisive question here is how much better the optimized RHXs are compared to the Eurovent RHX (for the different operating conditions) in terms of NPV and GHG emissions savings in the previous optimization scenarios.

3. Results

3.1. Optimization Results

Table 5 shows the results of the optimization where the objective function is the NPV with different CO_2 pricing and IRR optimized scenarios from literature [17] are also presented. Additionally, the values of the thermal efficiency η, IRR, GHG emissions savings, and RHX design and operational parameters are also shown for each scenario. The GHG or NPV savings are in comparison to the situation where no HRS is used.

Table 5. Optimization results NPV and IRR, CO_2 Eq. [17].

Optimization			Results							
Objective Function	**FU**	**CO_2 Pricing**	**NPV Savings**	**IRR**	**GHG Emissions Savings**	η	h^*	β	l	$v_{2.1}$
	m^3/h (Ambient air 2.1)	€/t CO_2 Eq.	t of €	%	t CO_2 Eq.	%	mm	°	mm	m/s
IRR [11]	10,000	0	21	71	107	60	2.78	60	150	2.7
GHG [11]	10,000	0	8	5	159	92	3.89	60	941	1.0
NPV	10,000	0	27	48	143	80	2.64	58.1	230	1.2
NPV	10,000	25	30	49	144	80	2.77	60.0	235	1.2
NPV	10,000	55	35	53	144	80	2.77	60.0	235	1.2
NPV	10,000	65	36	54	144	80	2.77	60.0	236	1.2
NPV	100,000	0	283	63	1424	79	2.74	60.0	220	1.2
NPV	100,000	25	318	67	1424	79	2.74	60.0	220	1.2
NPV	100,000	55	361	68	1440	80	2.74	60.0	236	1.2
NPV	100,000	65	376	69	1440	80	2.74	60.0	236	1.2

The first important observation from Table 5 is that the RHX optimized for IRR, due to its low thermal efficiency, does not comply with EU law. Where the RHX is optimized for GHG emissions savings, the RHX length l is extremely long, making it cumbersome for production, transportation, and installation. This increase in investment costs is not included in the model. Therefore, it is questionable whether these would really be the results for NPV and IRR. The actual results are definitely lower.

In the NPV optimized scenarios at an FU of 10,000 m^3/h AUL, the IRR and the GHG emissions values are between the IRR optimum and the GHG emissions optimum. Additionally, all NPV-optimized RHXs comply with the minimum thermal efficiency required by EU law. In the case where the CO_2 pricing is set to 0 €/t CO_2 Eq., the NPV savings are higher than those of the previous IRR and GHG emissions optimization scenarios. Although the higher the price, generally the more GHGs are saved, the saving effect by these CO_2 pricing is not considerable. One final aspect on the economic perspective is that the rotors optimized for the larger FU have approximately a 10% higher IRR. This is related to the investment pricing depending on the size, which is not linear, but after a certain size, the price increase becomes smaller.

A closer look into the RHX design and operational parameters reveals the following: The wave height h^* of the optimized RHX is relatively high. Rotors with these characteristics are not usually used in HVAC systems but rather are used in paint shop systems to reduce the risk of clogging and enable better cleaning. However, this is a very feasible value since RHXs up to a 3.8 mm wave height h^* exist according to the Eurovent database [13]. The wave angle of the optimized RHXs is independent of the CO_2 pricing or FU, and in almost every case, reaches the simulation limit set at 60°. Unfortunately, this limitation is based on the availability of heat transfer coefficients for higher wave angle β on which the simulation is based [12,28]. Nevertheless, there is a clear direction in terms of how large the wave angle should be, especially when comparing the greater range of values seen in the Eurovent data. The optimum RHX length l is between 220 and 236 mm, depending on the FU or CO_2 pricing. These are also feasible values according to the Eurovent data. Additionally, the optimal ambient air empty tube velocity $v_{2.1}$ was found to be 1.2 m/s, equivalent to a specific mass flow $\dot{m}_{sp}$ of 5.4 kg/(h·m^2). This value of air empty tube velocity $v_{2.1}$ is unaffected by the CO_2 pricing or FU.

3.2. Ecological Comparison of the Optimization to the Eurovent Model

Out of the nearly 1300 certified aluminum RHXs contained in the Eurovent database, there is no RHX that matches the optimization results from Table 5. These optimized RHXs provide (compared to common RHXs used in HVAC applications like the P_17-1100-WZV model) lower pressure drop and increased electricity and energy savings, resulting in a better NPV at the end of the lifetime. The optimized RHXs from Table 5 represent an increased investment and this could be a challenge for the RHX manufacturer. Still, the electricity and heating costs are much lower for the end customer to pay. In summary, the balance between increased investment and operation savings pays off for the customer with these optimized RHXs.

The optimized RHXs listed in Table 5 are compared hereafter to the Eurovent RHX presented in Table 4 operating at different ambient air empty tube velocities $v_{2.1}$ at both FUs. Please note that the GHG emissions can occur directly at the production site of the materials or use of the product (Scope 1) or take place in the power plant during electricity production (Scope 2). The savings presented in Table 4 are the differences between the "without HRS" scenario and the corresponding RHX, seen in blue in Figures 5 and 6.

The first noteworthy result is that the production and recycling steps of an RHX only marginally influence the overall GHG emissions of an HRS. Even without recycling, where GHG emissions are saved, the production is insignificant when compared to the GHG emissions during the use phase. In the use phase, the significant GHG emissions are from the additional electricity used by ventilators and heating. The optimized RHXs have low GHG emissions because of the low electricity consumption, although they are not as efficient on the heat savings as the Eurovent RHX at 1 m/s ambient air empty tube velocity $v_{2.1}$. The reason that it makes sense to reduce the electricity consumption at the expense of thermal efficiency is the GHG emissions of these energy sources as shown in Table 1. While natural gas has a GHG emissions of 0.2506 kg CO_2 Eq./kWh, electricity has GHG emissions of 0.6476 kg CO_2 Eq./kWh, 2.6 times higher. In addition, the price of gas is lower than electricity per energy unit. Therefore, from the environmental and economic point of view, it is still better to produce more heat using a gas furnace than to use more electricity to power ventilators and use a more thermal efficient RHX. This depends directly on the electricity sources, and this can change in the future if renewable electricity is used. Only in the case where the Eurovent RHX is operated at a low ambient air empty tube velocity ($v_{2.1}$ = 1 m/s) are the GHG emissions similar to the optimized RHXs. Finally, from the ecological point of view, there is not much difference between the different FUs. All the results are similar to having the lower FU multiplied by a factor of 10.

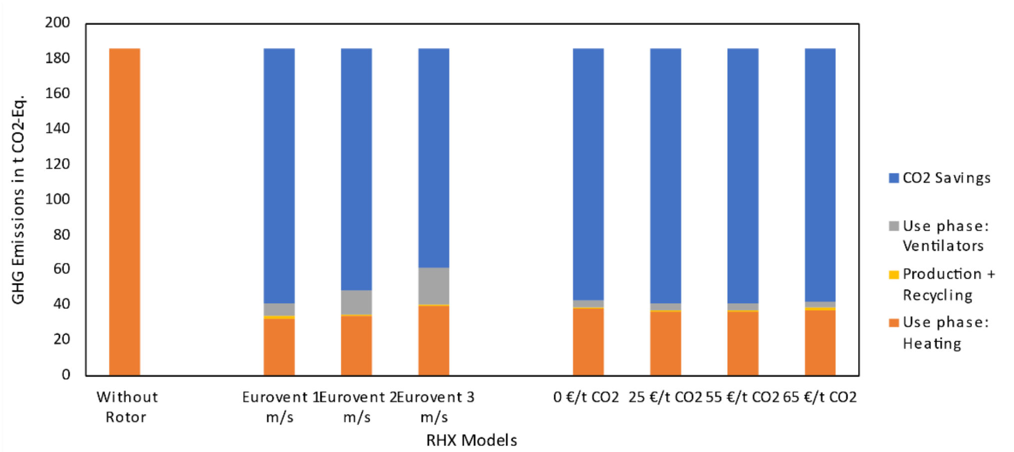

Figure 5. Total life cycle GHG emissions in CO_2 Eq. for different optimized RHXs and the Eurovent model at an FU of 10,000 m^3/h.

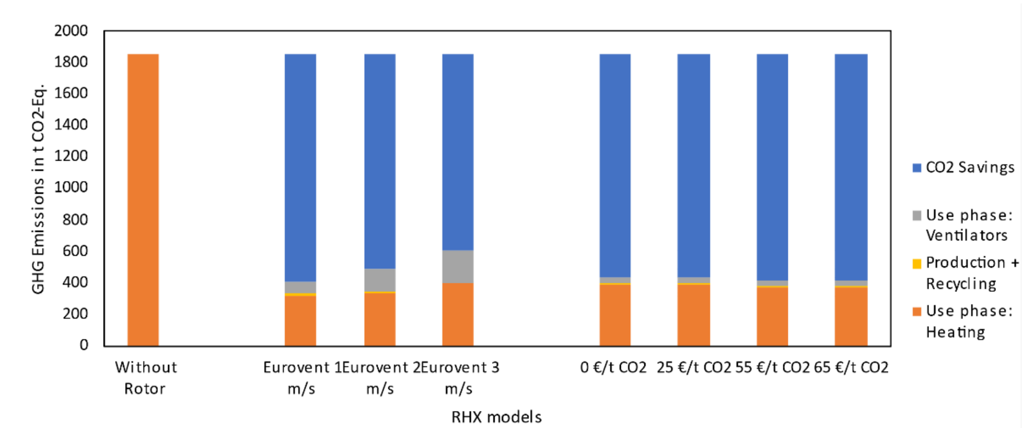

Figure 6. Total life cycle GHG emissions in CO_2 Eq. for different optimized RHXs and the Eurovent model at an FU of 100,000 m^3/h.

3.3. Economic Comparison of the Optimization to Eurovent Model

In this section, four different scenarios are presented for each CO_2 pricing. In each scenario, the Eurovent model for RHX is simulated at three different ambient air empty tube velocities $v_{2.1}$ for which Eurovent has made the information available at the two chosen FUs. With this information, the GHG emissions costs and NPV are calculated.

3.3.1. Scenario without CO_2 Pricing (0 €/t CO_2 Eq.)

Please note that the results for the NPV throughout the entire scenarios section have been normalized to ventilator electricity costs of the optimized RHX at 0 €/t CO_2 Eq. in order not to disclose sensitive information in cooperation with the industrial partner. The following are the results with no CO_2 pricing in Figures 7 and 8.

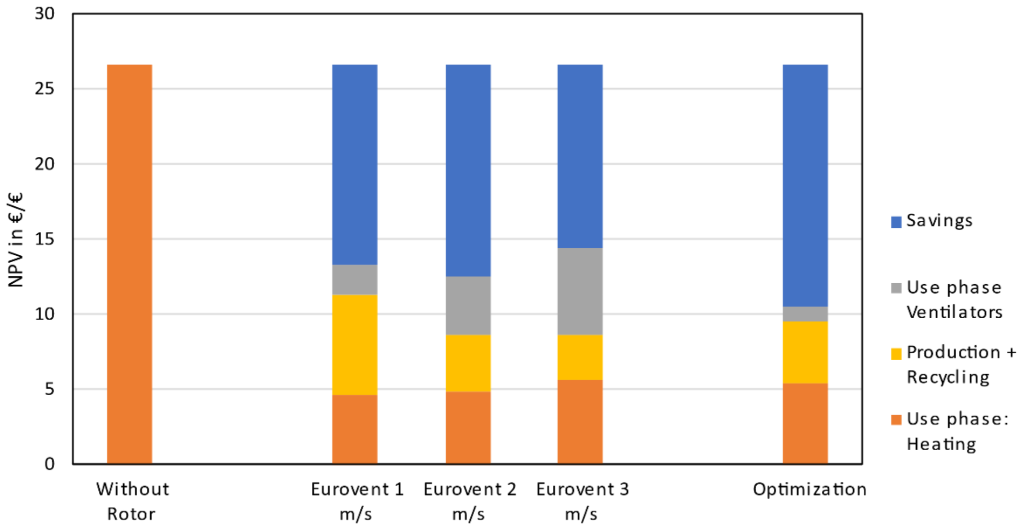

Figure 7. Scenario 0 €/t CO_2 Eq. with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 10,000 m^3/h.

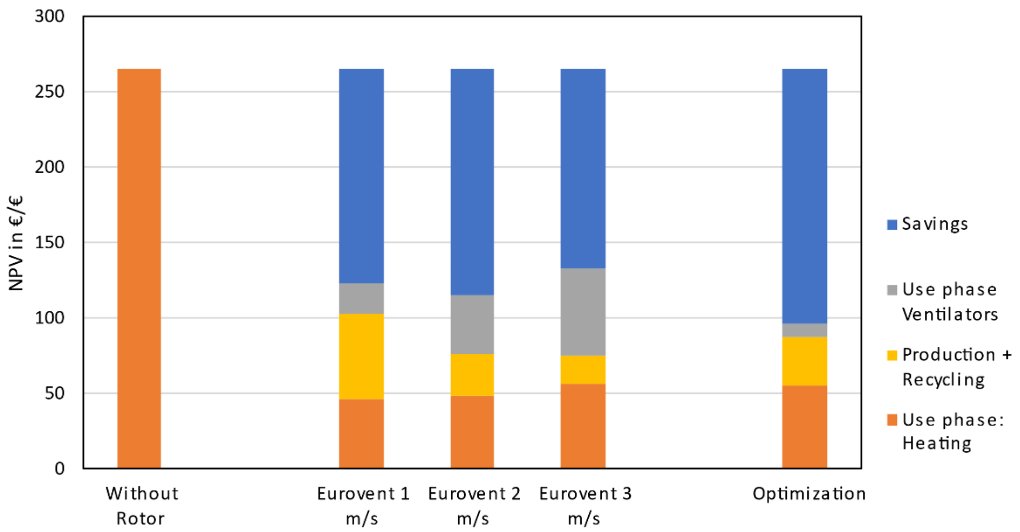

Figure 8. Scenario 0 €/t CO_2 Eq. with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 100,000 m^3/h.

In Figures 7 and 8, one can observe the different costs for electricity, heating, production, and recycling along the entire lifespan of different RHX models at the two examined FUs. Furthermore, the recycling step has a negligible overall effect economically. The heating costs alone are similar among the different conditions because the thermal efficiency is similar for all RHX models (between 79% and 83%). The biggest differences in these four conditions are the electricity and investment costs. This is because RHXs with a larger diameter (with lower empty tube gas velocity v, for the same amount of volume) cause a lower pressure drop (hence, lower electricity costs) but require larger investments for any given flow. Therefore, the investment and operating costs for electricity are in competition with one another. In addition, the investment cost for the optimized RHX is in the same order of magnitude as the Eurovent RHX, with an ambient air empty tube gas velocity $v_{2.1}$ of 2 m/s.

For the optimized RHX, the ventilators use less electricity than with the Eurovent RHX at different conditions. This is due to the higher wave height h^* of the optimized RHXs that results in lower pressure drop. Therefore, the ventilators have to compensate for a lower pressure drop and therefore require less electricity. This significant reduction in electricity costs makes the difference overall: The optimized RHX reduces the total costs from 16% to 28% compared to the Eurovent RHX. For the Eurovent RHX, by comparing the extreme operating conditions, one can observe that lower ambient air empty tube velocities $v_{2.1}$ have lower electricity costs than at 3 m/s. This is because the pressure drop depends directly on the square of the empty tube gas velocity v. On the other hand, to handle a given gas volumetric flow $\dot{V}$, the ambient air empty tube gas velocity $v_{2.1}$ is inversely proportional to the flow area of the RHX. The flow area is proportional to the square of the radius of the RHX. Therefore, for lower ambient-air, empty-tube gas velocities $v_{2.1}$, a larger flow area is needed, such that the diameter of the RHX is larger and consequently also the investment costs. To sum up, the investment and the electricity costs work against each other. Therefore, for the Eurovent RHX, the better solution for the empty tube gas velocities $v_{2.1}$ is the intermediate value of 2 m/s.

Please note also that the values for the FU at 100,000 m^3/h are roughly 10 times the values of the FU at 10,000 m^3/h for the economical (Figures 7 and 8) as well as the GHG emissions (Figures 5 and 6). This means that although the production process for larger RHXs might differ from smaller RHXs [24], the overall result is proportional to the FU. As a consequence, the same behavior is observed in the scenarios with CO_2 pricing. For purposes of readability, hereafter only the values of a FU of 10,000 m^3/h are presented in the text and the values with a FU of 100,000 m^3/h are in Appendix A.

3.3.2. Scenarios with CO_2 Pricing

In these scenarios, the CO_2 pricing substantially influences the heating and electricity costs because of their higher GHG emissions, as shown in Figure 5. The different scenarios are depicted in Figures 9–11.

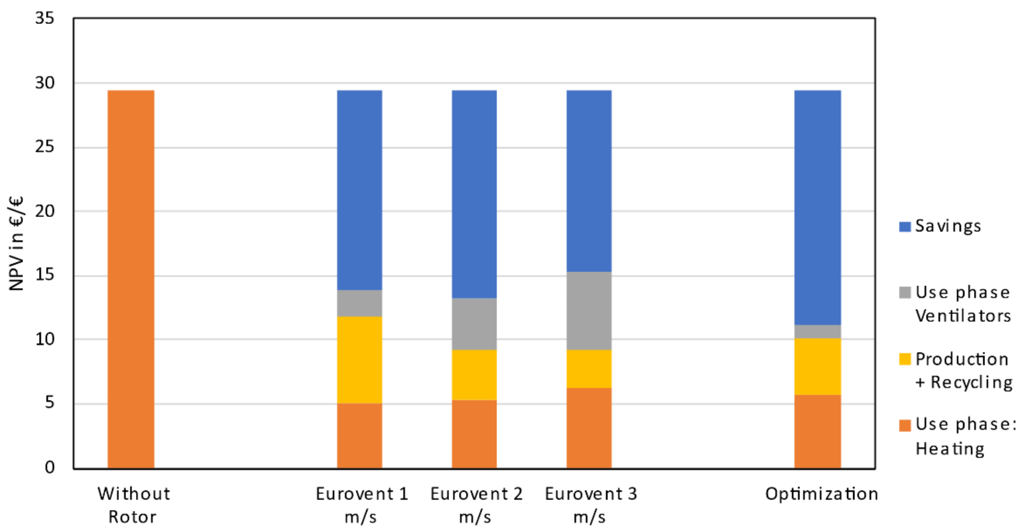

Figure 9. 25 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 10,000 m^3/h.

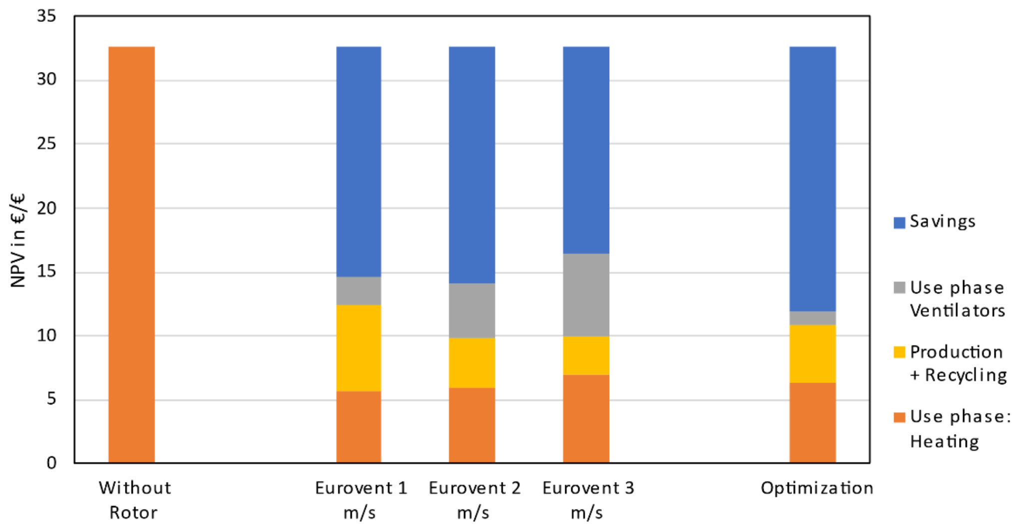

Figure 10. 55 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 10,000 m^3/h.

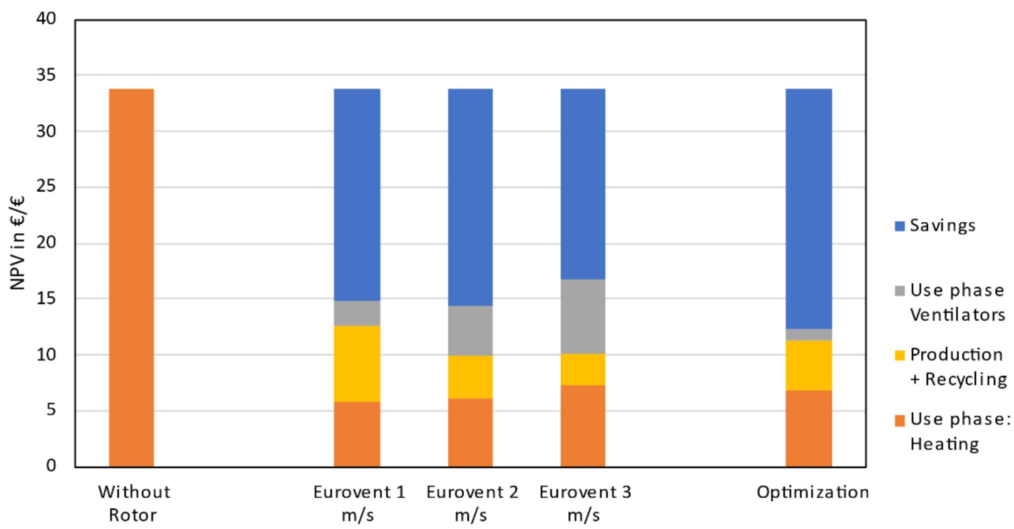

Figure 11. 65 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 10,000 m^3/h.

In these cases, the overall costs have been increased by the CO_2 pricing of the GHG emissions. Since the use phase has a greater impact on GHG emissions than production or recycling do, the effect of the use phase is amplified in the economic assessment when a CO_2 pricing is set, especially for the heating and electricity costs, as already mentioned. Regarding different scenarios where there is a CO_2 pricing as shown in Figures 9–11, the corresponding optimized RHX is the better option in terms of NPV for the same reasons as in the non-CO_2 pricing scenario (0 €/t CO_2 Eq. scenario). These optimized RHXs are the more economical solution in terms of NPV when considering investment and operating costs, even when taking different FUs into account.

4. Discussion

This work presents optimized RHXs in Table 5, which for customers, imply a better compromise between the investment and operation costs. However, this trade-off requires transparency on the plant manufacturer's part because he needs to communicate the overall economic and ecologic benefits to the end customer, which is how RHX manufacturers can produce more "expensive" RHXs during the investment phase. The short-term investment cost still gets amortized with the long-term heating and electricity savings. The NPV is higher in the new optimized RHXs compared to the RHX optimized in previous works. Looking at the IRR, the value of the optimized rotors is still close to 50% for the smaller FU and above 60% for the larger FU, which represents an increase in the amortization time as compared to the 77% for the IRR optimized rotor in our previous work. However, in any case, the period of amortization is within two years. Therefore, it is important when selecting an RHX to conduct an overall assessment of operating and investment costs. This could be a communication and business challenge for the plant manufacturer. The plant manufacturer should consider all of the costs involved, i.e., the investment as well as the operation costs for their customers. In addition, based on their experience, the industrial partner suggests that the ambient air empty tube velocity $v_{2.1}$ is often 3 m/s. This operating condition is economically detrimental for the end customer and has increased negative ecological impact.

This paper provides some transparency regarding the economic and environmental impact of currently used RHXs in comparison to the economically optimized RHXs. Interestingly, although many RHX configuration exist and have been certified by Eurovent, this work proves that RHXs can still be optimized under economic and environmental considerations.

One limitation of this work was to consider the GHG emissions exclusively in terms of the electricity grid mix and prices of services in Germany. Depending on the location and sources of electricity, this would result in a different optimum. For example, if electricity came from renewable sources, then it would make more sense to recover more heat and use more electricity. Then the optimum would get closer to the GHG optimum (Table 5). These are further circumstances that could be researched. Another limitation of this work is the angle range for wave angle β. Further studies could investigate whether steeper wave angles would further increase the heat transfer and would be economically and environmentally beneficial.

By comparing the energy efficiencies of the NPV-optimized RHXs to the state-of-the-art Eurovent RHX, one can observe that the thermal efficiency is approximately 80% in both cases. In the optimization cases, thermal efficiency improved slightly. The main bottleneck in bringing better technology to this market lies not in the technical challenges, but rather in the communication and decision-making process. As long as the decision-making process is short-sighted economically, i.e., focusing merely on the investment costs, the technical improvements cannot be brought to market.

5. Conclusions

The literature of LCA-based and NPV-based optimization for RHX is nonexistent. This is an area to develop in the future that would allow a better benchmarking with other types of heat exchangers. With this research work, a part of this gap was filled within the German context and within the limitations of the simulation models. This study could be repeated for different locations, with different electricity and heat sources, and the results would be different. A further research topic would be the inclusion of a simulation model with the same accuracy but over a wider range for the inclination angle (above 60°). For this study, they were out of the scope, but definitely interesting points for further research.

The NPV optimization results provides a compromise in terms of GHG emissions savings and IRR maximization between the other two optima from previous works, (where objective functions were GHG and IRR). In terms of NPV, the optimization of this work presents greater savings than the previous works from this research group. For each scenario (depending on CO_2 pricing or FU), there is an optimum presented. In terms of design and operating parameters: The ambient air empty tube velocity $v_{2.1}$ is not significantly sensitive to CO_2 pricing or FU, the optimum was found at 1.2 m/s, which corresponds to a specific mass flow $\dot{m}_{sp}$ of 5.4 kg/(m^2·h). Wave angle β is not significantly sensitive to CO_2 pricing or FU, but because the available models are limited to 60°, further research could concentrate efforts on developing Nusselt number models for sine ducts in the range between 60° and 90° to overcome this limitation. In this study, the best values were found between 58° and 60°. Wave height h^* and RHX length l are dependent on CO_2 pricing and FU, but on a relatively narrow range compared to the Eurovent database. The optimal values are between 2.64 mm and 2.77 mm for wave height h^* and between 220 mm and 236 mm for RHX length l. Independently of the CO_2 pricing and FU, in the complete lifetime of the product, the heating and electricity costs are greater than the investment. A lower investment on a RHX could be a more expensive system on overall for the complete lifecycle. The same applies to the GHG emissions. The main result of the optimization based on NPV is that the optimized RHX causes lower pressure drop, which significantly reduces electricity needs of the system while maintaining the thermal efficiency compared to the state-of-the-art RHX of the Eurovent database.

Author Contributions: Conceptualization: E.M., H.K., and N.T.; methodology: E.M. and N.T.; software: E.M. and N.T.; validation: E.M. and N.T. formal analysis: E.M.; investigation: E.M.; resources: N.T.; data curation: E.M.; writing—original draft preparation: E.M.; writing—review and editing: E.M., H.K., and N.T.; visualization: E.M.; supervision: N.T.; project administration: N.T.; funding acquisition: N.T. All authors have read and agreed to the published version of the manuscript.

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Symbols List

Symbol	Unit	Description
a	-	Accuracy of the optimization
h^*	mm	Wave height
i	-	Dimension index
k	-	Index
l	mm	RHX length
$\dot{m}$	kg/h	Mass flow
n	rpm	Rotational speed
s	µm	Material thickness
t^*	mm	Wave length
$\vec{x}_0$	-	Starting vector for the optimization
$\vec{x}_1$	-	Working vector for the optimization
v	m/s	Empty tube gas velocity
$\dot{V}$	m^3/h	Volumetric flow

Greek symbols

β	°	Wave angle
η		Temperature efficiency
ϑ	°C	Temperature
ρ	kg/m^3	Density

Common subindices

a	Inflow
sp	Refers to the specific mass-flux per flow area of the RHX. Then in $kg/(m^2{\cdot}h)$
1.1	Process air
1.2	Exhaust air
2.1	Ambient air
2.2	Supply air

Appendix A

In this appendix, the supplemental data for a FU of 100,000 m^3/h for the scenarios with CO_2 pricing are presented.

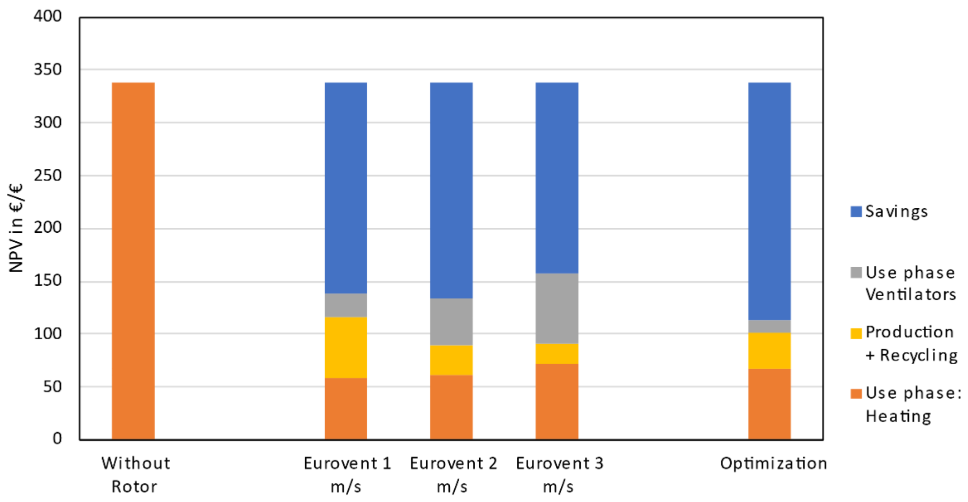

Figure A1. 65 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 100,000 m^3/h.

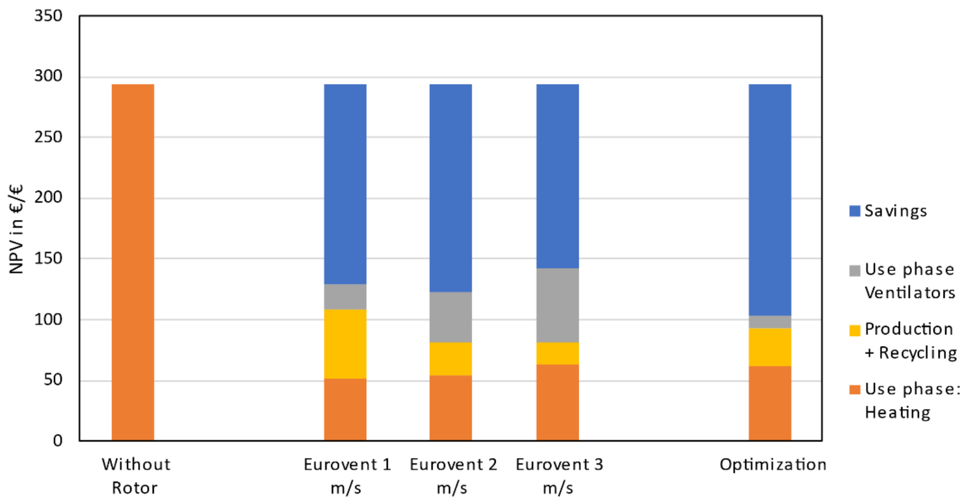

Figure A2. 25 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 100,000 m^3/h.

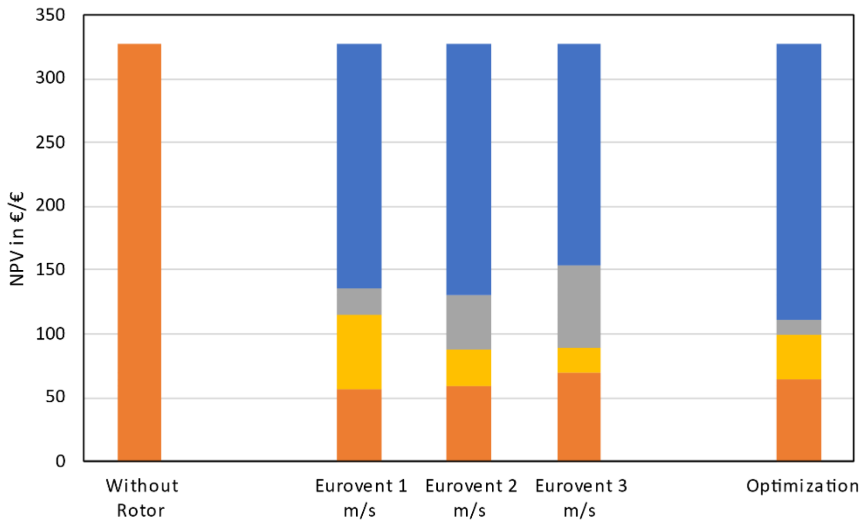

Figure A3. 55 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 100,000 m^3/h.

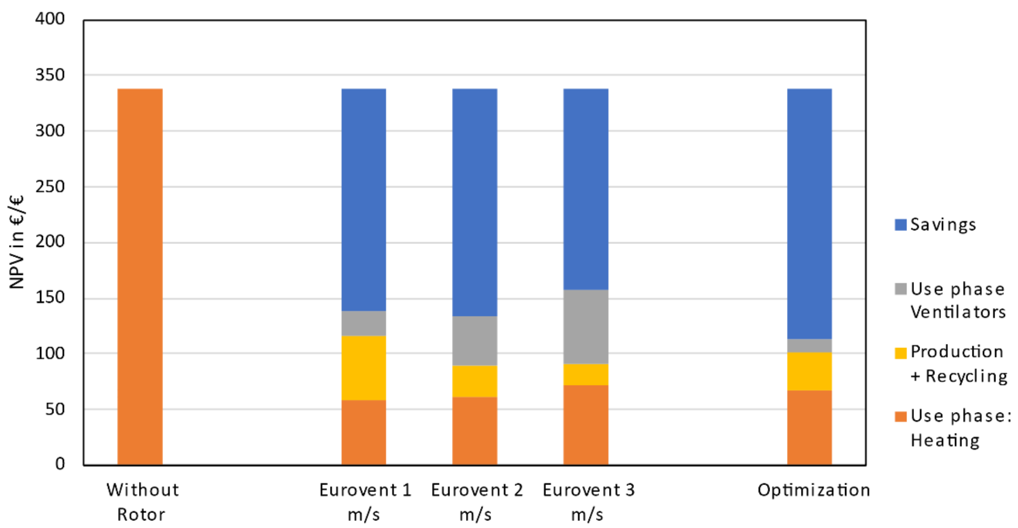

Figure A4. 65 €/t CO_2 Eq. scenario with optimized RHX and Eurovent RHX under different operating ambient air empty tube velocities at 100,000 m^3/h.

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