Exergy-Optimum Coupling of Heat Recovery Ventilation Units with Heat Pumps in Sustainable Buildings

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ABSTRACT

This study shows that as a result of exergy destructsions in heat recovery ventilation units additional but avoidable carbon dioxide emissions take place due to the imbalance between the unit exergy of thermal power recovered and the unit exergy of fan power required for the additional pressure drop in the heat exchanger. Therefore, special attention needs to be paid in the design and control of heat recovery ventilation units in order to minimize such carbon dioxide emissions responsibility by a proper exergy-rational balance between heat recovered and power required. The potential improvements about the exergy rationality of the heat recovery ventilation units were investigated for several alternatives. These alternatives were: heat recovery ventilation-only (base case), coupling with an air-to-air heat pump in tandem or parallel to the heat recovery ventilation unit, and a heat pump-only case. In order to carry out such an investigation, a new exergy-optimum design and dynamic control model was developed. Under typical design conditions, this model showed that a heat pump in parallel configuration does not improve the exergy rationality unless its coefficient of performance is over 11, which is not practical with today's technology. Instead, passive solar and wind energy systems have been discussed and recommended. Results were also compared with condensing boiler, micro-cogeneration unit, fuel cell, and electric resistance heating cases. It has been shown that heat recovery ventilation with an air-to-air heat pump in tandem is the best in terms of the exergy-based coefficient of performance. Additional comparisons were made concerning avoidable and direct carbon dioxide emission responsibilities, global warming-potential and ozone-depleting potential, embodied energy, embodied exergy, and carbon dioxide recovery periods. A new composite index, which recognizes the direct relationship between the ozone layer depletion and the greenhouse gas emissions has also been introduced for comparing system alternatives in terms of their atmospheric footprint.

KEYWORDS


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INTRODUCTION

While buildings are getting more heavily insulated and tightened for energy conservation, indoor air quality (IAQ) concerns and demand for fresh-air ventilation is increasing, resulting in higher sensible heating and cooling loads. Therefore, there is a dilemma between energy conservation and indoor air quality. ASHRAE Standard 62.1 [1] provides precise requirements about fresh air changes per hour for several building typologies and indoor functions, which put constraints on energy conservation measures. In this respect, Debacker, Allacker, Spirinckx, Geerken, and Troyer analyzed the environmental footprint of ventilation in buildings, which is mandatory in Belgium since 1991 [2]. They considered natural supply air and exhaust, natural supply and mechanical exhaust, and mechanically-controlled heat recovery ventilation (HRV). They applied Pareto optimality in carrying out a life-cycle assessment analysis. Based on the First Law and environmental costs only, the HRV alternative offered the most preferable solution for the typical set of dwellings in Belgium. Ke and Yanming carried out a similar study in China [3]. They considered four climate zones in eight cities and investigated the applicability of HRV. Their metric of applicability was based on investment-specific cost and “energy” savings.

In simple terms, what an HRV unit does for energy savings and the economy is the sensible pre-heating of outdoor air in winter or sensible pre-cooling in summer with the exhaust air. The unit exergy of the thermal power gained from preheating or precooling, simply defined by the ideal Carnot cycle, is relatively small due to limited temperature rise or decrease in outdoor ventilating air. This exergy gain generally is not a match with the additional electrical exergy demand of the oversized or additional fans required for HRV operation in addition to the basic ventilation system requirements. Therefore, while an HRV unit may seem to be very efficient in terms of the First Law, often revealed by a high coefficient of performance (COP), it may prove to be inefficient according to the Second Law. In spite of this exergy-based inefficiency, the design and analysis of HRV systems are based only on the First Law of Thermodynamics. In order to better describe the performance and environmental footprint of an HRV unit, an alternative performance factor, namely, COPEX, which is simply the thermal power exergy gained divided by all power exergy inputs attributable to the heat exchanging process involved for HRV. COPEX, which is less than one is a way to express exergy destructions. Therefore, exergy destructions leading to avoidable CO$_2$ emissions may be minimized by letting COPEX value to approach one.

There is little research in the Second-Law aspect of HRV systems, while First-Law analyses are abundant. Likewise, Fouih, Stabat, Rivière, Hoang, and Archambault investigated the adequacy of the HRV system in low “energy” buildings [4]. They modeled an HRV unit using TRNSYS for dwellings in different climates of France and concluded that the adequacy of the HRV system depends on the building types, the heating loads, and the ventilation device characteristics. In their paper, Deymi-Dashteayaz and Valipour-Namanla [5] investigated the thermodynamics (First Law only) and thermodynamic feasibility of recovering waste heat from the computer racks in a data center using an air-source heat pump in Mashad, Iran and using it for space heating purposes. They reported that the system financially pays itself in 2.5 years and also improves the power usage effectiveness (PUE), energy reuse factor (ERF), and energy reuse effectiveness (ERE) of the data center [5]. Taha al-Zubaydi and Hong experimentally investigated counter-flow heat exchangers for energy recovery ventilation in cooling mode in buildings [6]. They determined that dimpled surfaces perform about 50% better than flat surfaces.

In terms of the First-Law again, HRV and exhaust-air heat pumps (EAHP) are quite effective in a district energy system, especially in cold climates by raising the return
temperatures to the district [7]. They compared the performance of a renovated building in a cold climate with HRV and three different EAHP connection configurations. The return temperature and energy use of the studied DH (District Heating) substations were modeled. The EAHP increased the weighted average return temperature of DH by 10°C or 15°C compared to HRV, depending on the connection scheme. The EAHP connection configurations had almost no effect on the seasonal COP of the heat pump, which was approximately 3.6 and corresponded to the measured best practice in the literature. Based on their simulations, they recommended the simplest EAHP connection scheme with the lowest DH return temperature. Cai, Mei, Liu, Zhao, and Wang have proposed to generate electric power through thermo-electric generators and then used it for refrigeration rather than keeping it as heat and utilize it as heat. They claimed that such a system has a large potential in the “energy”-efficient buildings [8]. They, however, ignored that heat has a relatively higher unit exergy than cold and thermo-electric cooling has low conversion efficiency. Therefore, it is better to utilize heat as heat.

Zhang, Fung, and Jhingan, S. have developed an Excel spreadsheet program in order to analyze and carry out a feasibility study of a residential energy recovery ventilator with a built-in energy economizer [9]. They pointed out that a built-in economizer makes the system much better in cost recovery. In their paper, Cheng, Shuli, and Ashish made a detailed review of the existing air-to-air heat and mass exchanger technologies for building applications [10]. They carried out an extensive investigation about the heat and mass exchanger-integrated, energy-efficient systems for buildings, ranging from passive to mechanical ventilation systems, defrosting methods, and dehumidification systems. Their review concluded that; heat exchangers result in insufficient airflow in passive buildings, HRV or ERV (Energy Recovery Ventilation) systems cause additional pressure drops, air leakage, and noise in the ducts, defrosting problems in cold climates, and finally, ERV systems require additional heat in dehumidification and regeneration phases. Building energy use is closely linked with CO₂ emissions, reported by many authors like Chenari, Dias, and Gáméno da Silva [11]. Their review showed that many factors must be taken into account for designing energy-efficient and healthy ventilation systems. They also concluded that utilizing hybrid ventilation with suitable control strategies leads to considerable energy savings, thus a reduction in CO₂ emissions. According to the 2014 report from the Intergovernmental Panel on Climate Change (IPCC), total anthropogenic GHG emissions have been continuing to increase over 1970, despite a growing number of climate change mitigation policies. Annual GHG emissions grew by almost 1.0 Gton of CO₂ by which 78% of emissions were from fossil fuel combustion. The built environment is responsible for about 40% of this portion [12]. In this respect, heat recovery systems in buildings have been represented as promising technologies by Cuce and Riffat, due to their capability of providing “considerable energy savings” in buildings [13]. Yet they ignored the presence of avoidable exergy destructions in their analysis.

According to another study, by applying an energy recovery system to building HVAC systems, roughly up to 66% and 59% of sensible and latent energy can be recovered [14]. In a detailed monitoring study of a UK dwelling, the efficiency of the installed mechanical ventilation with heat recovery was found to be over 80% [15]. However, all these figures are based on the quantity of energy defined by the First Law. These studies do not account for rating and evaluating methods, beyond providing a thorough account of different technological details, except thermal efficiency, and NTU values. Several other researchers have concentrated only on topics like fan noise, low First-Law efficiency, and leakage problems for de-centralized ventilation systems with a heat exchanger. According to these studies, decentralized ventilation is based on a single room or a small conditioned space, which has the potential of minimizing pressure losses due
to the short travel distance of the air, when compared with centralized ventilation [16]. Manz et al. [17] have tested and simulated the performance of various types of decentralized ventilation units for cold temperate climates. Recently, several studies about decentralized ventilation units with heat recovery have been carried out focusing on cold, temperate, warm, and humid climates. Smith and Svendsen have developed a short plastic rotary heat exchanger made of a polycarbonate honeycomb with small circular channels for single-room ventilation based on thermal design theory. Their experimental results demonstrated the potential of reducing heat recovery by slowing rotational speed, which is required to prevent frost accumulation [18]. The same authors investigated the effect of a non-hygroscopic rotary heat exchanger on a single-room about its relative humidity. They also studied the sensitivity to influential parameters, such as infiltration rate, heat recovery, and indoor temperature [19]. Coydon et al. have investigated several building facades, which are coupled with HRV systems and showed that counter-flow type heat exchangers recovered 64% to 70.0% of heat [20]. Again, all the research work cited above were based on the First Law.

There is quite a few exergy-based research about waste-heat recovery systems but they are concerned with the equipment components, without a holistic look for the exergy match between the supply and demand that needs to cover the wide range starting from the primary fuel input and ending at the final application through the waste heat recovery ventilation. Cuce, M. P., and Riffat, S. further provided a detailed account for the exergy analysis of such equipment [13]. Recently, an exergy analysis of a cross-flow heat exchanger was performed by Kotcioğlu et al. [21]. They have found that the heat exchanger efficiency decreases with the increasing air flow velocity. Yilmaz et al. [22] also presented an exergy-based performance assessment for heat exchangers. On the other hand, the exergy transfer effectiveness for heat exchangers has been described by Wu et al. [23]. These studies shed light on the exergy aspect of the waste heat recovery systems, yet they fall short of a holistic analysis of the overall performance and they exclude the relationship between exergy destructions and causing avoidable CO\textsubscript{2} emissions. Furthermore, exergy and CO\textsubscript{2} embodiments of the RHV construction material for LCA analyses were completely ignored. Despite certain shortcomings concerning the need for a holistic approach to the exergy performance of buildings in the literature, LowEX tool [24] that has been developed in the framework of Annex 37 by IEA ECBCS is an important step towards a better understanding of the importance of exergy analysis especially in low-exergy building applications, namely low-temperature space heating and high-temperature space cooling. However, this tool does not cover the avoidable CO\textsubscript{2} emissions due to exergy destructions, according to the Rational Exergy Management Model, which can be as large as direct emissions in magnitude. Furthermore, it does not cover yet embodied exergy destructions, which are especially important for nZEB and nZEXB equipment such that their payback periods are quite long.

The Need for an Exergy-Based Method with a Holistic View

With the ever-increasing awareness of the importance of utilizing the waste heat in air-conditioning and ventilation systems and green buildings, air-to-air heat exchangers are becoming a vital component of InZEB and nZEXB cases. According to the EU Directive 2010/31/EU on the energy performance of buildings (With the First Law of Thermodynamics), starting from the 31st December of 2020, all new buildings will be required to be nearly-zero energy buildings [25]. EU has not yet realized the importance of the Second Law in the quest for decarbonization. Yet, Tronchin and Fabbri have developed a new simplified method of evaluating the exergy of the energy consumed in buildings with the aim of finding a relationship between the HVAC loads of buildings including envelope heat transfer and the energy conversion plant and its sub-systems like
radiators [26]. They argued that the exergy analysis of energy consumption in heating and cooling of buildings could be a tool to evaluate an exergy tariff to promote low-exergy HVAC plants. They further argued that such an exergy-based building performance tool based on Annex 37 may be utilized to evaluate the relationship between energy/exergy consumption: Their new model evaluates the energy performance of buildings by establishing a direct relationship among exergy, temperature variations, and TOE. The originality of their model, which they name Exergy Performance of Buildings, emanates from the fact that their exergy-based model is structured on the energy-based EU Energy Performance in Buildings Directive. They compared the two approaches and concluded that exergy-based model provides a more comprehensive analysis technique such that it reveals exergy destructions, which are not acknowledged by an energy analysis. HRV is a sub-system in their model and therefore it may be instrumental also for the HRV performance.

Among many different types and means of recovering heat from the exhaust air, fixed-plate heat recovery is a simple yet widely used technology. The structure of fixed-plate heat exchangers is based on several thin plates arranged together in order to separate internal airflows. The airflow between these plates create an additional pressure drop, namely $\Delta P$ and thus lead to additional fan power demand in electrical form, namely $\Delta E$, that must be satisfied by either by fan oversizing or adding more fans. It must be reminded that $\Delta E$ only relates to the heat exchanger of HRV over the ventilation system without heat recovery.

Regarding the airflow arrangement, there are three types of fixed-plate exchangers: counter-flow, cross-flow, and parallel flow [27]. The typical efficiency of fixed-plate heat recovery is in the range of 50-80% [27]. In the existing building stock, however, there is less than 1% of heat recovery ventilation in Europe [28]. According to the same publication, in new buildings, mechanical extraction units (MEU) still dominate the market, while the share of HRV units is increasing rapidly.

Figure 1, which is a reproduction from the ASHRAE Handbook-HVAC Systems and Equipment, shows the basic model for a fixed-plate, air-to-air type HRV unit. In this figure, the stand-alone HRV model is isolated from its surroundings and from the additional/oversized fans and the power grid [29-a]. Even if such factors are included, they are limited to the quantity of energy recovered and spent by neglecting their different qualities. In fact, the unit exergy of thermal power gained or extracted (in cooling), $\varepsilon_H$ is substantially lower than the unit exergy of electric power, $\varepsilon_E$ which is virtually 1 W/W.

![Figure 1. Air Flows and DB Temperatures in a Stand-Alone HRV](Image)

There is an exergy imbalance between them. Furthermore, Figure 1 does not question the origin of power generation and the fuel used. Obviously, the fan power must be limited somehow for positive exergy gain from the exhaust air. The only ASHRAE source available, which deals with the limiting of fan power in air handling is the ASHRAE Standard 90.1 [29-b]. In Tables 6.5.6.1-1 and 6.5.3.1 of Standard 90.1 regarding the fan power limitations, the fan power is limited by a ratio of 1.2 BHP per
1000 cfm (1.896 kW/m$^3$s$^{-1}$) for flow rates less than 20,000 cfm (9.44 m$^3$s$^{-1}$) and for the given US climate zone of 7, B. To give an example, consider Table 1, which reproduces an edited version of a typical data set from the commercial literature [30]. This stand-alone HRV unit recovers 18.24 kW of heat, $Q$ at an airflow rate, $V$ of 0.83 m$^3$/s.

Table 1. Sample Data for a Commercial HRV Unit [30].

<table>
<thead>
<tr>
<th>Specifications</th>
<th>HRV Product Model Number</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
</tr>
<tr>
<td>Air Flow Rate, $V$ m$^3$/s</td>
<td>0.83</td>
</tr>
<tr>
<td>Thermal Efficiency, $\eta_I$</td>
<td>0.65</td>
</tr>
<tr>
<td>$T_i$, K</td>
<td>290.8</td>
</tr>
<tr>
<td>$Q$, kW</td>
<td>18.24</td>
</tr>
<tr>
<td>$E_{XH}$, kW= $Q$ x (1-$T_o$/T$_i$)</td>
<td>1.116</td>
</tr>
<tr>
<td>$\Delta E_{XE}$, $\Delta E$ kW (Kr ~ 1 W/W)</td>
<td>2 x 0.65 [kW]</td>
</tr>
<tr>
<td>First Law-COP, $Q/\Delta E$</td>
<td>14.03</td>
</tr>
<tr>
<td>Second-Law COPEX</td>
<td>0.613**</td>
</tr>
<tr>
<td>= $E_{XH}$ x (1-0.44$\eta_I$)/$\Delta E$</td>
<td></td>
</tr>
<tr>
<td>Approximate Weight, W kg</td>
<td>112</td>
</tr>
</tbody>
</table>

* For an outdoor air temperature of 0°C.
** With 44% in-house use of peak exergy

The First-Law efficiency, $\eta_I$ is 0.65. Outdoor air at 273 K is preheated to a temperature, $T_i$ of 290.8 K. It demands 2 x 0.65 kW dedicated fans for heat recovering purpose in the HRV unit itself, apart from the basic ventilation system. According to the traditional COP definition, this HRV unit has a COP value of 14.03 (18.24 kW/[2x0.65 kW]). This is a favorable value in terms of the First Law. If ASHRAE Standard 90.1, which permits a total fan power ratio of 1.896 kW/m$^3$s$^{-1}$ is applied to the first column of Table 1, 1.58 kW of fan power is permissible (1.896 kW/m$^3$s$^{-1}$ x 0.83 m$^3$/s) instead of 2 x 0.65 kW fan power, installed by the manufacturer. Accordingly, COP and COPEX values would reduce to 11.5 and 0.50, respectively. From the Second Law perspective, the total permitted fan power needs to be conservatively less than 0.83 kW instead of 1.58 kW. This is only about half of what ASHRAE Standard 90.1 permits. Obviously, ASHRAE Standard 90.1 does not refer to the Second Law at all and causes avoidable exergy destructions and thus more CO$_2$ emissions, because CO$_2$ emissions are proportional to exergy destructions (See Equation 21 in the following sections). In this example, the supply air to the indoor units needs to be temperature peaked by 6.2 K to raise it to a temperature of 299 K at the air terminal units in the indoor spaces, because the preheated air temperature, $T_i$ in Figure 1 is not high enough to satisfy the comfort heating loads and to maintain the DB comfort indoor air temperature, $T_i$. This means that additional unit exergy, $E_{XTP}$ of (1-290.8/299) W/W must be provided by an auxiliary air-heating system. On the return cycle of the air stream, part of this unit exergy, which is about 44% of the unit exergy (1-273/290.8) W/W of heat recovered by the HRV unit from the exhaust air may be treated in the form of exergy input, $E_{XA}$ in the quasi-closed loop of ventilation with an efficiency of $\eta_I$ from the exhaust air. Then the net exergy-based COP (COPEX) of the HRV unit with electrical and thermal inputs is (1-0.44$\eta_I$) x ($E_{XH}$ /$\Delta E$) is 0.613, a value which is quite less than one.

The most common five products with different capacities available in the international HVAC market were further compiled in Table 2. None of the manufacturers provide COP or COPEX values. The nearest value provided by a manufacturer was the specific fan power. The calculated values corresponding to these products with the above approach regarding COPEX values are always far from one, although COP values seem to be impressively high. It is obvious that without any reference to the COPEX term, design,
rating, and operation will not be rational and HRV units will keep being responsible for avoidable CO\textsubscript{2} emissions due to rather large exergy destructions. Therefore, an exergy-based holistic model is necessary, which also provides the answer to the question of where the electric power comes from and how it is generated.

Table 2. Typical Performance Data of Different World-Wide HRV Manufacturers.

<table>
<thead>
<tr>
<th>MANUFACTURER</th>
<th>Q</th>
<th>Em*</th>
<th>V</th>
<th>Fan Motor Power Provided by the MANUFACTURER</th>
<th>CALCULATED</th>
</tr>
</thead>
<tbody>
<tr>
<td>No.</td>
<td>Country</td>
<td>[kW]</td>
<td>[kW]</td>
<td>COP</td>
<td>COPEX</td>
</tr>
<tr>
<td>1</td>
<td>Japan</td>
<td>10.69</td>
<td>0.654</td>
<td>950</td>
<td>0.56</td>
</tr>
<tr>
<td>2</td>
<td>US</td>
<td>0.85</td>
<td>0.052</td>
<td>140</td>
<td>0.07</td>
</tr>
<tr>
<td>3</td>
<td>Japan</td>
<td>6.08</td>
<td>0.373</td>
<td>1000</td>
<td>0.475</td>
</tr>
<tr>
<td>4</td>
<td>UK</td>
<td>1.02</td>
<td>0.062</td>
<td>100</td>
<td>0.057</td>
</tr>
<tr>
<td>5</td>
<td>US</td>
<td>2</td>
<td>0.122</td>
<td>125</td>
<td>0.15</td>
</tr>
</tbody>
</table>

*Based on a regime of 273 K outdoor and 290.8 K supply temperature at HRV exit.
**With the assumptions of 40% recovered exergy from temperature peaking with 60% efficiency.

What is the Main Gap Between Current Study and Previous Studies

All previous studies covering -either energy analysis, or exergy analysis or both, do not include the embodied exergy in their LCA analysis. Furthermore, they do not address additional but avoidable CO\textsubscript{2} emissions, which are the result of exergy destructions. These unaddressed emissions may be equal or even higher than the direct CO\textsubscript{2} emissions. In addition, previous studies did not address the ozone depletion and global warming potentials that are associated with the HRV systems in conjunction with their power generation supply fuel and association with heat pumps, if coupled to them.

Implication Targets of the Study

Examples, which are given in Tables 1 and 2 show that a new exergy-based model is needed in order to fully understand the environmental performance and benefits of HRV units if there are any. The following implications are expected:

1. A method, capable of analyzing ‘Sustainable’ equipment by the Second Law,
2. A method, capable of estimating the actual CO\textsubscript{2}, ODP (Ozone Depleting Potential), and GWP (Global Warming Potential) footprint of heat recovering systems in buildings, including the origin of the energy input,
3. New design and rating metrics based on the Second Law,
4. Lifetime analysis method, which is a collection of the following payback periods:
   a) Embodied Exergy Payback,
   b) Embodied CO\textsubscript{2} Payback.
   c) Embodied Energy Payback,
   d) Investment Payback.

What Must be Novel

A new exergy-based evaluation model is expected to fill the gap in theory and practice by addressing the missing points in the literature like the analysis of the return of embodied CO\textsubscript{2}, energy, and exergy. These returns are far important than simple investment returns because they are all related to CO\textsubscript{2} emissions and global warming potential. Furthermore, the electric power source (Thermal power plants) and thermal recovery and conversion systems on-site like heat pumps with refrigerant leakages or even wind and solar systems and geothermal systems due to their associated exergy
destructions and embodied exergy, are responsible for ozone layer depletion and global warming. The model shall address and quantify these points, which are novel in heat recovery ventilation and in similar applications in the built environment. In this respect, the model shall permit to quantitatively analyze emissions and embodiment returns of combined systems like HRV and heat pump units, or HRV and small fuel cell units in residential green buildings. This will provide a complete account of the environmental footprint.

**DEVELOPMENT OF THE METHOD**

Firstly, the simple HRV diagram taken from the ASHRAE Handbook shown in Figure 1 must include the electrical exergy required by the oversized/added fan motors to move the air streams through the HRV unit itself in order to overcome the associated pressure losses. Figure 2 shows this first step of the model, which is the base case without a heat pump.

The total CO \(_2\) emissions responsibility, \(\Sigma CO_2\) of an HRV unit for a given thermal load, \(Q\), or any other similar system, which consumes electrical energy from the grid is a sum of the direct emissions responsibility -if power is delivered from a thermal plant- and avoidable CO \(_2\) emissions, namely \(\Delta CO_2\), according to exergy destroyments:

\[
\Sigma CO_2 = Q \left( \frac{c_{\text{mix}}}{COP_\eta_T} + \Delta CO_2 \right)
\]

(1)

The first term in Equation 1 represents the First-Law component of emissions and the second term represents the Second-Law component of emissions. The factor \(c_{\text{mix}}\) is the unit CO \(_2\) content of the primary fuel mix used in the energy sector that provides power to drive the fan, pumps, etc. of the on-site systems and equipment, like the fan motors of HRV units. The efficiency \(\eta_T\) covers the thermal plant, transmission, and distribution of power in the grid. Equation 1 at the same time represents the Ozone depletion potential and global warming potential because CO \(_2\) emissions are the prime cause of both. In addition, if mechanical compression systems with refrigerants like heat pumps are used further Ozone depletion takes place. The same also holds true for the attached cooling towers, while they release excess water vapor to the atmosphere, with the greenhouse effect.

**Base Case, HRV Only**

This model recognizes the presence of fans and their motors simply dedicated or attributed to the HRV boundary. It also recognizes the unit exergy of heat and power. According to this Model, exergy gain from the exhaust heat transfer, \(Q\) is given in Equation 2. This exergy gain raises the outdoor air temperature from \(T_o\) to \(T_1\).

\[
E_{XH1} = \left( 1 - \frac{T_o}{T_1} \right) Q
\]

(2)

The exergy gain in the HRV unit must be greater or at least equal to the exergy demand of fan motors on the outdoor and the exhaust side of the HRV unit:

\[
E_{XH1} \geq E_{XEO} + E_{XEHRV}
\]

(3)
The subscript 1 denotes HRV. If there is a coupled system like a heat pump it is denoted by the subscript 2. Thermal exergy gain in the HRV unit, namely $E_{XH1}$ is a linear function of the airflow rate, $V$ where fan power, thus the corresponding fan exergy demand is a power function of $V$ (See Equation 8). Therefore, there is a limit on the maximum airflow rate and the corresponding thermal exergy gain. The maximum flow rate allowed in many cases is less than the hourly fresh air requirement for maintaining the indoor air quality. Therefore, the remaining fresh air needs to be heated by another HVAC system, like a heat pump and the mix must be brought to the final supply air temperature, $T_f$. This requires the optimization of the outdoor airflow rate split between the HRV unit and another air heating system, like an air-to-air heat pump (HP), operated by grid power.

Payback Periods of HRV

An HRV unit, which is expected to save from total CO$_2$ emissions responsibility, the return of embodied exergy as well as investment, and energy spending must be accordingly analyzed. Table 3 compiles energy, exergy, and CO$_2$ emission embodiments for the major material types typically used in the manufacturing process of HRV units. Typically, steel, aluminum, and copper are used. Embodiments are expected to be recovered during the operation of the HRV. The commercial HRV model 1 as shown in Table 1 weighs 112 kg and has a material mix of about 50 kg Al, 45 kg steel, and 17 kg Copper. Therefore, the total embodied CO$_2$ for this model is 5343.4 kg CO$_2$. Other embodiments were also calculated by using Table 3:

a) Embodied Exergy, $E_{XEM}$: 15925 MJ (4423.6 kW-h),
b) Embodied Energy, $E_{EM}$: 11983 MJ (3328.6 kW-h),
c) Total embodied CO$_2$, $\Sigma CO_{2EM}$: 5343.4 kg CO$_2$,
d) Investment Cost, $I$: 1200 $ (Given, including installation costs).

If the HRV unit is operated for 3000 hours (moderate climate) in a heating season at an average thermal power of 5 kW, then the seasonal thermal energy savings will be 15000 kW-h. However, if the HRV fans require grid power, then the corresponding energy demand must be subtracted. With a seasonal average, $COP$ of 14.03, electrical energy spending is 1069 kW-h (15000 kW-h/14.03). It must be noted that the unit exergy of heat and the unit exergy of electricity are quite different and they should not be simply added or subtracted. However, just to demonstrate how the calculations are carried out in the industry, here they are subtracted. Therefore, this operation gives a net energy saving, $E_s$ of 13931 kW-h (According to the First-Law only).

a) Embodied Exergy Payback, $Y_X$

Because $COPEX$ is always less than 1 there is no finite payback period for embodied exergy. There is always net exergy destruction (From Table 1, $COPEX$ is 0.613).

Table 3. Embodied Energy, Exergy and CO$_2$ Emissions of Materials Used [30].
More clearly speaking, the exergy deficit is the difference between the exergy of thermal energy saved (24000 kW-h) and the exergy of the electrical energy used (1710.6): Exergy gain = 15000 x (1-273K/290.8K)-1069 (1) = -150.8 kW-h/heating season. The negative sign indicates exergy destruction. The unit exergy of electricity is 1 kW/kW.

b) Embodied Energy Payback, $Y_E$

\[ Y_E = \frac{E_{EM}}{E_s} = \frac{3328.6 \text{ kW-h}}{13931 \text{ kW-h}} = 0.24 \text{ heating season (1.5 months@16 h/day of operation)} \]  

(4)

c) Embodied CO\textsubscript{2} Payback, $Y_{CO2}$

\[ Y_{CO2} = \frac{\sum CO_{2EM}}{E_s \left( \frac{0.2}{0.85} - \frac{0.2}{COP \cdot 0.30} - 0.27[1 - COPEX] \right)} \]  

[if the denominator is > 0]  

(5)

In Equation 5, the term 0.2 kg CO\textsubscript{2}/kW-h is the unit CO\textsubscript{2} content, $c_i$ of natural-gas based on 1 kW-h of the lower heating value, used in a boiler with a seasonal-average First-Law efficiency of 0.85, which is assumed to be replaced by the HRV unit with a $COP$ value (14.03 from Table 1). The second term is the indirect CO\textsubscript{2} responsibility of the HRV unit, based on the use of grid power supplied from a natural-gas thermal plant. 0.30 is the power generation and transmission efficiency of the grid on average. The last term in Equation 5 represents the avoidable CO\textsubscript{2} emissions responsibility, $\Delta CO_2$ of the HRV unit (See Equation 25). Because the $COPEX$ value is less than one (0.613 from Table 1), the last negative term makes the denominator close to or less than zero, implying that the net CO\textsubscript{2} emissions savings are almost none. Therefore, the CO\textsubscript{2} embodiment is not practically recovered.

d) Embodied Investment Payback, $Y_c$

The simple investment payback, $Y_c$ is the function of investment cost, $I$, Cost of fuel, $C_f$, boiler efficiency, $\eta_B$, $COP$ of HRV, cost of electricity $C_E$, and the seasonal energy savings, $E_s$.

\[ Y_c = \frac{I}{E_s \left( \frac{C_f}{\eta_B} - \frac{C_E}{COP} \right)} \]  

(6)

If the average natural gas price, $C_f$ is about 0.035$/kW-h and $\eta_B$ is 0.85, average electricity price, $C_E$ is 0.1$/kW-h, $COP$ in heating is 14.03 (Table 1, Column 1), then from Equation 6, the investment payback period, $Y_c$ will be about 2.5 heating seasons.

New definitions about returns given above show that although HRV systems with fast economic returns are recommended for IAQ applications due to their First Law efficiency [1], they do not favorably payback in terms of exergy and CO\textsubscript{2} embodiments while more exergy is destroyed than recovered from the waste heat, unless the electrical power is supplied by on-site or near-site renewables. This disadvantage is further reinforced if the
net negative added value to the environment in terms of ozone depletion potential, ODP and global warming potential, GWP, which the HRV unit and the heat pump are directly and indirectly responsible are also considered with their interactions (See Figure 3 and Equation 26). In addition to the base case mentioned above, three case studies presented herein investigate whether the exergy rationality and environmental impact may be improved, namely by adding a heat pump in a parallel position or in a series position (downstream) of the HRV unit, respectively, and a stand-alone heat pump:

- Case 1: HRV coupled with a heat pump in parallel,
- Case 2: HRV coupled with a heat pump in series,
- Case 3: Heat pump only.

**Case 1. Heat Pump Parallel with HRV**

In order to design and optimize the split of the total flow rate of the outdoor air intake between the HRV and the HP units for maximum exergy-based performance, a new model was developed, which is shown in Figure 3.

This method identifies two parallel air flow ducts. One of them delivers outdoor air to the HRV unit. The second duct delivers the remaining outdoor airflow to the heat pump, HP. Hereby the outdoor air split ratio is represented by ratio $x$. If $x$ is zero, it means that HRV is not functioning (or absent). If $x$ is one then it means that the heat pump is not functioning (or absent). Airflow rates in the HRV unit are limited such that thermal exergy gain obtained, $E_{XH1}$ by heating the incoming outdoor air up to a temperature of $T_1$.

**Figure 3. Isolated Model for the Operational Diagram of the Method with a Parallel Heat Pump.** Note: Main HVAC fans that are served by the heat pumps are excluded.
must be higher than the electrical exergy demand of the two fans motors and the fan drive. This system designed and operating under optimum conditions is expected to also improve the ratio of energy consumption of the building to the energy consumption of the ventilation system, which may be similarly defined in terms of power usage effectiveness \((PUE)\) factor for data centers [5].

Across the HRV unit, there are two counter-air flows, namely the exhaust air at a flow rate of \(yV\) and the fresh outdoor airflow rate of \(xV\). Assuming that fans at both sides of the HRV unit are identical, then:

\[
Q \left(1 - \frac{T_o}{T_o + \Delta T}\right) = xV \rho C_p \Delta T \left(1 - \frac{T_o}{T_o + \Delta T}\right) \geq \left(E_{XEHRV} + E_{XEO}\right) = \frac{e \Delta P (x + y)V}{\eta_p \eta_{bm}} \quad (1 \text{ W/W}) \quad \text{(7)}
\]

Here, \(c\) in the last term represents the fan characteristic, \(\eta_F\) and \(\eta_{bm}\) are the fan and motor-belt efficiencies, respectively. They are assumed to be approximately constant with respect to their flow rates during operation. \(E_{XEHRV}\) and \(E_{XEO}\) represent the exergy demand of both fans, which operate on electricity. For standard air conditions at sea level and 15°C, it may be taken that \(\rho\) is 1.225 kg/m\(^3\) and \(C_p\) is (1.026 kJ/kg·K). Then, along with the following relationship between \(\Delta P\) and \(V\) for rectangular ducts:

\[
\Delta P = dV^m \quad (8)
\]

\[
\Delta T \left(1 - \frac{T_o}{T_o + \Delta T}\right) \geq \frac{e V^m (x + y)}{\eta_p \eta_{bm} x} \quad (9)
\]

The power \(m\) is 1.82 for the galvanized ducts. For other inner linings, sizes, and geometry, \(m\) changes between 1.8 to 1.9 [30]. According to Equation 7, the maximum flow rate between \(T_o\) and \(T_1\) (\(\Delta T\)) in the HRV unit is limited:

\[
\Delta T = \frac{(1 - x) T_{f} T_1}{x \left(\frac{x + y}{x} e\right)^{\frac{1}{n}}} \quad \text{(10)}
\]

\[
\Delta T = T_1 - T_o \quad , \quad T_f = x T_1 + (1 - x) T_2 \quad , \quad (11) \quad (12)
\]

then solving for \(T_i\{T_i > T_o\}\),

\[
T_i = \frac{T_f - (1 - x) T_2}{x} \quad , \quad (x > 0.40) \quad (13)
\]

The term \(x\) is subjected to the optimization algorithm (See Equations 17 and 19-a). The ratio \(y\) is separately controlled by optimized operating conditions of the HP for a given \(x\) ratio. \(T_i\) is a function of \(x\) for a required supply air temperature, \(T_f\). In order to solve \(T_i\), \(T_2\) is solved first for optimum exergy output in terms of the exergy-based \(COP_{HP}\), namely \(COP_{EX_{HP}}\) written only for the heat pump itself according to the ideal Carnot cycle:

\[
COP_{EX_{HP}} = COP_{HP} \left(1 - \frac{T_o}{T_2}\right) \quad \text{(14)}
\]

The \(COP_{HP}\) term may be linearly expressed in a given, narrow operating range [29]:
\[ \text{COP}_{HP} = q - r \left( T_2 - T_o \right) \]  \hspace{1cm} (15)

After Equations 12 and 13 are simultaneously solved and then the partial derivative with respect to \( T_2 \) is taken and equated to zero, the optimum \( T_2 \) for maximum \( \text{COPEX}_{HP} \) may be determined. Here, \( q \) and \( r \) factors are the linearization coefficients for the \( \text{COP}_{HP} \). If \( q \) is equal to 6 and \( r \) is 0.05K\(^{-1}\) for a given heat pump in a temperature range between 273 K and 300 K at an outdoor temperature, \( T_o \) of 283 K then the optimum \( T_2 \) is 337.7 K.

If the required supply air temperature, \( T_f \) is 310 K and \( x \) is 0.75, then, from Equation 13 \( T_i \) is 300.8 K. Therefore, the optimum temperature \( T_2 \) for the heat pump is out of range. In order to avoid this condition, a different type of model of the heat pump with different \( q \) and \( r \) values may be selected. This solution, however, is subject to the maximum allowable air flow rate, \( V \) that is limited by Equation 10. Now, knowing the optimum \( T_2 \) and knowing \( T_i \) in terms of \( T_2 \), for a known total fresh air flow rate requirement, \( V \), an optimization function, \( OF \) may be written in terms of \( x \) for maximum exergy gain from the exhaust air. If the electric motor of the additional/oversized fan is inside the HRV duct, the last term in Equation 17 represents the heat gain from the electric motor and its drive casing. This term adds exergy in winter (heating) but reduces exergy from the \( OF \) term in summer (cooling). If the electric motor and the casing are placed outside the ducts, this term drops.

\[ OF = \rho C_p x V \left( T_i - T_o \right) \left( 1 - \frac{T_o}{T_i} \right) + \rho C_p y V \left( T_2 - T_o \right) \left( 1 - \frac{T_o}{T_2} \right) - E_{EXHP} - \frac{cd \left( x + \left( 1 - a \right) x \right) V^{m+1}}{\eta_f \eta_{am}} \pm \]

\[ \{a = (1-y)/x\} \]  \hspace{1cm} (17)

The approximation given in Equation 18, which was derived from the information available in [30, 31], calculates \( Q_{FO} \). In this equation, \( H \) is a number less than one, which represents the net conversion ratio of the electrical power input, which is not converted to the shaft power to the heat transferred to the airflow in HRV.

\[ Q_{FO} = \left( H - \eta_f \eta_{am} \right) \left( \frac{c dx V^{m+1}}{\eta_f \eta_{am}} \right) \]  \hspace{1cm} (18)

In order to bring the supply temperature to the required design temperature, \( T_f \) at the exit of terminal units for proper satisfaction of the sensible indoor space heating load, the temperature may require to be peaked by an auxiliary heating system, which is arranged
in tandem to the HRV unit with or without the heat pump (See Figure 3). This auxiliary system might use primary fuel or power, and therefore additional ODP and GDP take place. \( E_{XTP} \) denotes the additional exergy spending for temperature peaking. In the quasi-closed outdoor/indoor system of the HRV unit, the exhaust air bears part of this thermal exergy and transfers it back to the outdoor fresh air drawn in with a thermal efficiency \( \eta_I \) across the heat exchangers of HRV and or HP. Because this thermal exergy is finite, as opposed to ambient (practically infinite) sources like air or water, and primary fuel or power input takes place during temperature peaking upstream in the indoor ducts, it may not be treated as ambient energy like the ground heat, ambient air, seawater, etc. Therefore, it must appear in the definition of COPEX, while traditional COP calculations often ignore this term (ambient sources). Referring to Equation 17, the overall COPEX of the system modeled in Figure 3, namely the HRV+ parallel heat pump Case 1, is given in Equation 19-a. The objective is to maximize \( \text{COPEX}_{HRV+HP} \).

\[
\text{COPEX}_{HRV+HP} = \frac{\rho C_p x V (T_1 - T_o) \left( 1 - \frac{T_o}{T_1} \right) + \rho C_p y V (T_2 - T_o) \left( 1 - \frac{T_o}{T_2} \right) \pm Q_{ex} - \frac{T_o}{T_1} \left( 1 - \frac{T_o}{T_1} \right)}{E_{XEH} + \frac{c_d (x + [1-a]x) V^{n+1}}{\eta_I \eta_{amb}} + \frac{E_{XEX}}{E_{OX}}}
\]

(19-a)

In Equation 19-a, \( E_{XTP} \) is the sum of the partial contributions of \( E_{XH1} \) and \( E_{XH2} \):

\[
E_{XTP} = \left( E_{XH1} + E_{XH2} \right)
\]

and,

\[
E_{XEH} = \rho C_p (1-a) V (T_2 - T_o) / (1 \ W/W)
\]

(20)

All terms in the objective function, \( OF \) and the \( \text{COPEX}_{HRV+HP} \) function given in Equations 17 and 19-a, respectively contain the split factor, \( x \), in such a manner that the objective function is a single function of this variable \( x \) with all other given and independently solved variables like \( T_o \) and \( T_2 \), respectively. Therefore, this model also provides an exergy-based control algorithm to maintain the maximum exergy rationality during operating under dynamic outdoor and indoor conditions. For the \( x < 1 \) condition, \( y \) is \((1-x)\). For \( x = 1 \) condition, \( y \) is either equal to zero (Case 1) or equal to \( x \), which is also equal to one when the heat pump is in series with the HRV unit (Case 2).

**Case 2. Heat Pump in Series (Tandem) with HRV.**

In this case, the heat pump is placed downstream of the HRV unit in series such that the total airflow passes through both the HRV and the heat pump units. Therefore, \( x \) and \( y \) are equal to each other and mathematically speaking, both are equal to one. The downstream position is a better position for the heat pump, compared to an upstream position where colder air supply will reduce the COP of the heat pump. The same Equation 19-a applies with \( x = y = 1 \) in order to determine the maximum \( \text{COPEX} \) value for a given design or operating conditions. Furthermore, in Equations 17 and 19-a, \( T_f \) replaces \( T_2 \), \( T_1 \), which is determined from the HRV specifications, replaces \( T_o \) in Equation 16. Consequently, Equations 12 and 13 are not used.
Case 3. Heat Pump Only

The same Equation 17 applies with \( x = 0, y = 1 \), which means that there is not any HRV unit and the heat pump is exposed directly to the outdoor air at a temperature, \( T_o \), and has to deliver heat at the final supply temperature \( T_f \), because another auxiliary temperature-peaking system is not desirable from the cost and exergy points of view. This means that the heat pump has to operate at a lower \( \text{COP}_{HP} \), while a larger \( \Delta T \), which is equal to \( (T_f - T_o) \) exists. In case 3, \( T_2 \) directly replaces \( T_f \) at the absence of an auxiliary temperature-peaking heating system. See also Equations 14 and 15 for temperature replacements.

Avoidable CO\(_2\) Emissions Responsibility of HRV and the Heat Pump Due to Exergy Destruc-
tions

In the same Model, the additional (Avoidable) CO\(_2\) emissions responsibility, namely \( \Delta C O_2 \) due to exergy destructions is explained by the Rational Exergy Management Method (REMM) [32]. Figure 5 shows a sample exergy flow bar according to REMM for a single HRV unit that is driven by grid electricity.

![Exergy Flow Bar](image)

**Figure 5. Exergy Flow Bar for HRV Unit in Space Heating Mode (not to scale).**

This is a simple exergy flow bar, which covers only one application, one system, and one energy source. This bar is drawn starting from top showing the primary energy source temperature, \( T_f \) to the final application temperature in the HRV unit (indoor supply temperature at 300 K) and then to the environment reference temperature, \( T_{ref} \) (283 K) at the bottom. It is assumed that electric power is supplied through the grid where the electric power is generated in a natural-gas power plant. 2235 K is the adiabatic flame temperature of the natural gas. In this sample case, the exergy utilized (for the space heating demand), \( \varepsilon_{dem} \) starting from an outdoor temperature \( T_o \) of 283 K and ending at the outdoors is given by Equation 21, according to the ideal Carnot cycle.

\[
\varepsilon_{dem} = \left(1 - \frac{T_{ref}}{T_f}\right) = \left(1 - \frac{283 \text{ K}}{300 \text{ K}}\right) = 0.056 \text{ kW/kW}, \tag{21}
\]

while the original exergy supplied, \( \varepsilon_{sup} \) at the power plant is given by Equation 22,

\[
\varepsilon_{sup} = \left(1 - \frac{T_{ref}}{T_{sup}}\right) = \left(1 - \frac{283 \text{ K}}{2235 \text{ K}}\right) = 0.87 \text{ kW/kW}. \tag{22}
\]

According to REMM, if the major exergy destruction takes place at the upstream of the useful application (Heat Recovery), like in Figure 5, then \( \psi_R \), which is the Rational Exergy Management Efficiency is a simple ratio of \( \varepsilon_{dem} \) and \( \varepsilon_{sup} \) [33]. According to this definition, \( \psi_R \) is 0.064 for the above numerical example.
For 1 kW of exergy power supply:

\[ \psi_R = \frac{\varepsilon_{\text{dem}}}{\varepsilon_{\text{sup}}} \] (23)

Here \( \varepsilon_{\text{des}} \) is equated to 1-COPEX. Referring to Equation 23 and the identity in Equation 24, \( \psi_R \) seems to be equal to COPEX. However, because \( \psi_R \) is a measure of exergy rationality in terms of ideal Carnot cycle while COPEX is a measure of exergy efficiency in terms of various exergy destructions taking place in the system, this identity may not hold true for other more complex systems, while COPEX definition approaches to the Second-Law efficiency. If part of the exergy input to preheating is from the reclaimed heat that is a result of temperature peaking, \( E_{\text{XTP}} \), appears in the denominator of the COPEX term given in Equation 19-a. Therefore, for non-ambient exergy inputs, COPEX in Equation 24 must be corrected by a factor, \( w \). This factor is about 0.85 in building applications of HRV and heat pumps. If there is no temperature peaking, then \( w \) is one. This rule also applies to Equation 25 given below.

For a given unit exergy destruction, \( \varepsilon_{\text{des}} \) taking place in any energy conversion system or equipment like an HRV, its natural-gas equivalence that is necessary to spend in order to replace the said destroyed exergy will be \( \frac{\varepsilon_{\text{des}}}{0.87} \). Assuming that this exergy destruction is replaced in an equal amount of exergy generation in a non-condensing natural-gas boiler with an average reference thermal efficiency of 0.85, one may find the associated avoidable \( \text{CO}_2 \) emissions responsibility. In Equation 25, 0.2 is the unit \( \text{CO}_2 \) emission of natural gas per 1 kW-h of its lower heating value, \( c_i \). Then, Equation 25 simply relates the avoidable \( \text{CO}_2 \) emissions to the destroyed unit exergy per kW-h of heat. For the heat pump-alone case (Case 3), Figure 5 applies where \( \varepsilon_{\text{dem}} \) is replaced by \( \varepsilon_{\text{dem}} \times \text{COP} \). If COP is 5 then \( \psi_R \) is 5 x 0.064, which is 0.32.

\[ \Delta \text{CO}_2 = \left( \frac{\varepsilon_{\text{des}}}{0.87} \right) \left( \frac{0.2}{0.85} \right) = 0.27 \varepsilon_{\text{des}} = 0.27(1 - w \text{COPEX}) = 0.27(1 - \psi_R) \quad \{w=1\} \] (25)

In this case, \( \Delta \text{CO}_2 \) from Equation 25 is 0.18 kg \( \text{CO}_2 \)/kW-h of unit heat, \( Q = 1 \) kW-h, supplied assuming that no temperature peaking takes place (\( w=1 \)). Depending on the \( x \) value for an HRV and HP combination, the average \( \psi_R \) may be calculated by an algebraic weighted sum. If a fuel cell or micro-cogeneration unit replaces HRV and the HP, the exergy flow bar looks similar to the one given in Figure 6, because both systems generate electricity on the site from the natural gas fuel input. Fuel cell, however, has a lower heat output temperature, \( T_E \). In this case, \( \psi_R \) has a different definition [32]. If for example, \( T_E \) is 300 K for the micro-cogeneration unit and 350 K for a fuel cell then \( \psi_R \) values are 0.56 and 0.87, respectively.
Ozone Depletion and Global Warming Effects

Until now, the Ozone Depletion Potential, \( ODP \) and the Global Warming Potential, \( GWP \) have been separately treated regarding the impact of systems in the built environment, by ignoring the important relationship between the two. Therefore, a so-called zero-\( ODP \) refrigerant used in the compressor of a heat pump (For example, R227ea), which has a very high \( GWP \) value-about 2300 is recommended for reducing the Ozone depletion effect in the atmosphere. In fact, any increase in \( GWP \) increases \( ODP \).

While the air temperature in the lower atmosphere increases with an increase in greenhouse gases, air temperature in the stratosphere cools due to the blanketing effect of the greenhouse gases. This cooling triggers more Ozone depletion. Therefore, \( GWP \) has a definite relationship with \( ODP \). Conversely, any expansion in the ozone hole increases global warming. Although verified by observations this relationship was not mathematically expressed in a practical manner. In order to simply show the combined effect of a system like a heat pump with refrigerant leakages, a novel expression for the combined effect, namely Combined Ozone Depletion Index, \( ODI \) was developed:

\[
ODI = \frac{sGWP}{(1-ODP)} \times \left( \frac{ALT}{1} \right)^t
\]

This equation combines \( ODP \) and \( GWP \) of a given refrigerant by also referring to the atmospheric residence time, \( ALT \). The power (\( t \)) regarding the \( GWP \) term in Equation 27 includes the combined effect of water vapor released to the atmosphere due to fossil fuel combustion, from attached cooling towers or from increased evaporation from warmed-up seas, lakes, or rivers accepting reject heat from thermal power plants that provide electricity to air conditioners and heat pumps. Collection of condensates from condensing boilers do not help, because additional exergy destructions taking place during condensing the flue gas overweighs them in terms of avoidable \( CO_2 \) emissions responsibility of condensing boilers. Accordingly, there is neither any refrigerant nor heating and cooling equipment with compression or absorption cycle with actually non-zero \( ODI \) even if their reported \( ODP \) values are zero. For example, compare R744 and R227ea:

a- \( CO_2 \) (R744) values: \( ODP = 0, GWP = 1, ALT = 120 \) years,
b- R227ea (F gas) values: \( ODP = 0, GWP = 3500, ALT = 33 \) years.

For \( s = 0.1, t = 0.03 \) (including water vapor effect), and \( u = 0.01 \) values in hand, their \( ODI \) values are:

a- For \( CO_2 \) (R744) \( ODI \) is 0.115 and
b- For R227ea (F gas) \( ODI \) is 0.132.
These calculations ignore the additional effect of cooling towers, while they release moisture to the atmosphere with additional ODI. Therefore, cooling towers need to be eliminated or minimized by utilizing the reject heat. Dry cooling towers on the other hand use more electrical energy and they are again responsible for additional ODI, depending upon where the electricity comes from and how it is generated.

RESULTS AND DISCUSSION

It has been shown that the new method can be easily applied to the entire spectrum of any combination of HRV and HP units simply by varying the \( x \) and \( y \) values. Hence it is a versatile tool both for design, retrofit, and control of HRV units with or without heat pumps or any other auxiliary heating (cooling) units coupled to them, particularly in green buildings, where both the unit exergy demand of HVAC functions and unit exergy supply of renewable or waste energy resources are small. The need for such a sensitive balance in such a small exergy range requires an accurate, exergy-based optimum solution algorithm. A sample warehouse case with HRV and parallel HP combinations for a unit airflow rate of \( V = 1 \, \text{m}^3/\text{s} \) has been analyzed. Standard air conditions apply. The temperature of the supply air is not peaked. Table 4 provides the constant terms used. Equation 17 and Equation 19-a were used for a simple search of the optimum \( x \) and thus \( y \) values for the maximum \( OF \) value in an MS Excel worksheet in order to determine the optimum \( x \) and thus \( y \) values for maximum \( OF \). The results are shown in Figure 7. It is interesting to note that contrary to the general belief, heat pumps even with a \( COP \) value of 6 in heating mode do not contribute to sustainability and do not offer an optimum combination. \( OF \) values are always lower than the HRV-only case \((x = 1)\), when the Second Law comes into the picture: Figure 7 clearly shows that unless the \( COP \) is equal or more than 11 for this sample case, the HRV-only option \((x = 1)\) is always better. Even if such high \( COP \) values are possible at industrial scale or in nZEXB applications, such that the maximum \( OF \) \((5.8 \, \text{kW})\) in this case study takes place at the condition of \( x = 0.45\), \( COPEX \) is less than one as is true for all other options given in Table 5. It is obvious that any mechanical system running on grid electricity is not exergy rational.

Table 4. Constant Terms for the Case Study (in Equation 17 and 19-a).

<table>
<thead>
<tr>
<th>CONSTANT</th>
<th>( a )</th>
<th>( c )</th>
<th>( d )</th>
<th>( H )</th>
<th>( \eta_p \eta_{\text{bm}} )</th>
<th>( m )</th>
<th>( q )</th>
<th>( r )</th>
<th>( T_o )</th>
<th>( T_f )</th>
<th>( T_2 )</th>
<th>( E_{\text{XTP}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>VALUE</td>
<td>0.2</td>
<td>0.1</td>
<td>0.8</td>
<td>0.9</td>
<td>0.75</td>
<td>1.8</td>
<td>6</td>
<td>0.05</td>
<td>273</td>
<td>310</td>
<td>345</td>
<td>0</td>
</tr>
</tbody>
</table>

In Figure 7, \( x \) starts from 0.45, because \( T_f > T_o \) condition must be satisfied. If the curve is extended towards the \( x = 0 \) point of the graph, then the HP-only condition is reached. In this Case \( OF \) is the highest, which is 8.2, provided that \( COP_{\text{HP}} > 11 \) condition applies. If the heat pump is coupled in series, downstream with HRV, \( OF \) is at a single point on \( OF \) versus \( x \) diagram, because \( x \) and \( y \) are both set to 1. Here the \( OF \) term is 4.5.

Comparisons

In this study, the base case (HRV-only case), HRV with HP (parallel and series), HP-only, boiler, micro-cogeneration, fuel cell, and electric resistance heating with grid power were compared according to their direct \( CO_2 \) emissions and the avoidable \( CO_2 \) emissions, namely \( \Delta CO_2 \), and ODI responsibilities. The results are given in Table 5.
Figure 7. Change of OF value with x and y combinations for the Base Case and the Three Cases of HRV and HP Combinations.

The $\frac{\Delta CO_2}{CO_2}$ values listed in Table 5 for various systems and equipment remarkably show that avoidable $\Delta CO_2$ emissions, which are directly related to exergy destructions are as large as direct $CO_2$ emissions of equipment and systems. This might be a clue for the latest findings that sea levels will rise twice as much as previously anticipated. Because exergy is not measured but calculated, the -rather hidden to many- $\Delta CO_2$ emissions are avoidable by improving COPEX values and this puts exergy analysis into an important game maker role for avoiding global warming challenge, which is becoming a state of emergency lately. It may be argued that the major reason for low COPEX values in the reclamation of waste heat originates from the use of electric fans, pumps, and other ancillaries and they may be avoided by the use of heat pipes.

Table 5. Comparison with Condensing Boiler, Micro Cogeneration, and Fuel Cell.

<table>
<thead>
<tr>
<th>System Cases and Other Options</th>
<th>$\Psi_R$</th>
<th>$ODI$ (F Gas)</th>
<th>$\Psi_R$</th>
<th>$\Delta CO_2$</th>
<th>$\Sigma CO_2$</th>
<th>$\Delta CO_2$</th>
<th>COP</th>
<th>COPEX</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Case: HRV only</td>
<td>0.102</td>
<td>0.06$^1$</td>
<td>2.25</td>
<td>0.047</td>
<td>0.103</td>
<td>0.15</td>
<td>2.2</td>
<td>14.03</td>
</tr>
<tr>
<td>Case 1: HRV+Parallel HP</td>
<td>0.405</td>
<td>0.112$^2$</td>
<td>3.6</td>
<td>0.07</td>
<td>0.037</td>
<td>0.10</td>
<td>0.74</td>
<td>9.06$^3$</td>
</tr>
<tr>
<td>Case 2: HRV+Series HP</td>
<td>0.54</td>
<td>0.08</td>
<td>6.75</td>
<td>0.05</td>
<td>0.013</td>
<td>0.063</td>
<td>0.26</td>
<td>12.2</td>
</tr>
<tr>
<td>Case 3: HP only</td>
<td>0.51$^4$</td>
<td>0.178$^4$</td>
<td>3.79</td>
<td>0.06</td>
<td>0.088</td>
<td>0.148</td>
<td>1.47</td>
<td>11</td>
</tr>
<tr>
<td>Boiler only</td>
<td>0.135</td>
<td>0.27$^5$</td>
<td>0.5</td>
<td>0.25</td>
<td>0.254</td>
<td>0.504</td>
<td>1</td>
<td>0.8</td>
</tr>
<tr>
<td>Micro-Cogeneration</td>
<td>0.87</td>
<td>0.13$^6$</td>
<td>6.7</td>
<td>0.22</td>
<td>0.120</td>
<td>0.25</td>
<td>0.54</td>
<td>0.9</td>
</tr>
<tr>
<td>Fuel Cell</td>
<td>0.57</td>
<td>0.15</td>
<td>3.8</td>
<td>0.22</td>
<td>0.11</td>
<td>0.32</td>
<td>0.50</td>
<td>0.9</td>
</tr>
<tr>
<td>Electric Resistance</td>
<td>0.06</td>
<td>0.2</td>
<td>0.3</td>
<td>0.67</td>
<td>0.251</td>
<td>0.921</td>
<td>0.37</td>
<td>1</td>
</tr>
</tbody>
</table>

$^1$ With $\eta_T = 0.27$ and COP of HRV = 14; $^2$ Equal split of air preheating (in Figure 1); $^3$ With $\eta_T = 0.45$; $^4$ With COP = 5; $^5$ With $\eta_B = 0.85$; $^6$ Power output is prorated; $^7$ On grid power with $\eta_T = 0.30$ and energy supply mix with 0.4 kg CO$_2$/kW-h for power generation; $T_0 = 273$K for space heating.

Note: $\Delta CO_2 = 0.27(1-w \cdot COPEX)$. $\{0.85 \leq w \leq 1\}$

This argument may seem to be valid at a first glance but certain heat pipes, especially containing refrigerants, like R-134a [33] are claimed to have zero ODP, but in fact, they have non-zero ODI (see Equation 27). Therefore, while CO$_2$ emissions from power plants are avoided by reducing the use of grid power in heat recovery systems and equipment, ODI increases, which interrelates GWP to ODP. Therefore, the result is almost the same in terms of global warming. The use of water-ethanol, acetone, methanol mixtures with very low ODI may be other options but they have high smog-formation potential. Then the remaining solution for a sustained avoidance of large amounts of exergy destructions is new exchanger designs, extending even to morphing HRV units with 4 D printing.
technology, with embedded flexibility along with higher fan (including morphing fan blades) and higher motor efficiencies, supplied with renewable electricity on site.

The above discussion may hold true from the tiniest HRV unit in a small home to waste heat recovery from thermal power plants like heat recovery from stacks of coal-fired power plants. In such cases again the exergy recovered from the waste heat in the stack may be less than the exergy spent for the heat recovery mechanisms like pumps and additional stack fans if a careful design and exergy-based optimum control is not implemented. The boiler-only option has the lowest COPEX value. The highest COPEX value, which is 0.95 belongs to HRV+Series HP case, provided that $COP_{HP}$ is 11. Otherwise, i.e. for the condition of $COP_{HP} < 11$, there is no optimum solution and the singular solution goes to the $x = 1$ point (HRV only). The next better case is the parallel HP case and then the base case with COPEX = 0.613. The lowest COPEX is for electric resistance heating case that is running on grid electricity. This case has also the highest total CO$_2$ emissions rate.

The new composite index, namely $\psi_{R}/ODI$, which rates a system according to its exergy rationality per environmental footprint in terms of ODI is highest for HRV+Series HP case (Case 2). Therefore, it seems that the series coupling of the HP is a better alternative compared to parallel coupling with HRV. In turn, the HRV option has the lowest ODI value indexed to F gas. ODI originates from the GWP of the fossil fuels used in generating the electric power necessary for driving the oversized/added portion of the ventilating fans of the HRV unit.

A $COP_{HP}$ value greater or equal to 11 is only possible for industrial heat pumps, using Ammonia refrigerant. Figure 8 shows different cases of evaporator and condenser temperature cases. Obviously, in industrial scale, $COP_{HP}>11$ may be achieved at a small condenser and evaporator temperature differences (See Figure 9). According to Figure 9, the difference must be less than 20 K. This provides the condition that first the outdoor climate must be moderate and the building must be nZEXB type, which demands lower heating temperature, second, due to the industrial size of such heat pumps, district energy systems must replace individual heating and cooling systems. It is hard to achieve such high $COP$ values in smaller application sizes, like residential and small office buildings, unless the condenser/evaporator-temperature differences are very small.

This requirement for satisfying the condition $COP_{HP}>11$ may be derived by referring to Equation 15 by writing the temperature difference between the outdoor and supply temperatures. This derivation imposes a lower limit on the outdoor air temperature:

$$T_o \geq T_f - \left( \frac{q-11}{r} \right) \quad \{q > 11\} \quad (28)$$

A simultaneous solution of Equations 27 and 28 may further relate $E_{XP}$ to $T_o$. Thus, a variable-speed fan motor, which follows the outdoor temperature is essential. Equation 28 further indicates that a residential heat pump with high $q$ value and the low $r$-value is desirable. At the same time, in order to operate the heat pump at lower outdoor temperatures in colder climates, lower $T_f$ must be applied. This is only possible by decoupling the sensible heating (or cooling) loads and ventilation loads and dedicating sensible loads to radiant panel systems [34]. For example, if $T_f$ for ventilation only case is 295 K (22°C) in an nZEXB, $q$ is 12, and $r$ is 0.05K$^{-1}$, then from Equation 28, the outdoor temperature must be higher than 275 K (2°C). This is a serious restriction for the operation of the heat pump in HRV coupled applications, particularly during cold climates.
On the other hand, the electrical power exergy demand, $E_{XF}$ for an HRV unit for a given sensible load, $Q$ is limited and must be related to the $T_o$ limit given in Equation 28:

$$E_{XF} \leq Q \left(1 - \frac{T_o}{T_f} \right)$$  \hspace{1cm} (29)

Table 5 shows that for all known electro-mechanical systems used for heat recovery in buildings, which seem to be very profitable and environment friendly, $COPEX$ values, which reveal and evaluate their global sustainability aspects better than $COP$ are less than one, which indicate that all electromechanical systems naturally destroy exergy and the best alternative is to minimize exergy destructions by an exergy-based method, like the one presented in this paper.

However, Table 5 deals only with HRV-dedicated systems and equipment covered by the isolated model shown in Figure 3. The origin of the electrical power supply is not included. If a holistic insight about the performance and responsibilities of HRV units and their ancillaries in the built environment is required, an expanded model is possible if the domain is stretched upstream back to the origin of the electrical power generation, transmission, and transformation simply by introducing their total efficiency, $\eta_T$ to Equation 17, which is shown in Equation 30. This expansion affects the isolated model in Figure 3 at the HRV and heat pump power inputs. This simple introduction makes it possible to trace the responsibilities of HRV units back to the primary fuel input concerning the electrical power supply through the grid. See Figure 10. Terminal units in this Model are represented by the last term regarding $Q_{FO}$ in Equations 17 and 30.
The current average $\eta_T$ value in EU28 countries is 0.4 [36]. Therefore, the HRV Fan term (The third term) in Equation 17 increases by a factor of 2.5 (1/0.4). Obviously, the $OF$ values plotted in Figure 7 will decrease but the overall conclusions will remain the same. While Equation 1 already includes the term $\eta_T$, the holistic model may be applied to modify the $\Delta CO_2$ emission responsibility term, because there are additional exergy destructions taking place at the thermal power plants.

Two types of plants were identified namely a coal-fired (Anthracite) plant with economizers and a combined-cycle, natural gas plant. Respectively, adiabatic flame (exergy source) temperatures are 2453 K and 2343 K. Reference temperature is 273 K. Exit temperatures from power generation stage is 550 K for a coal-fired plant and 450 K for the combined cycle, natural gas plant.

$$OF = \rho C_p x V \left( T_1 - T_o \right) \left(1 - \frac{T_a}{T_i}\right) + \rho C_p y V \left( T_2 - T_o \right) \left(1 - \frac{T_a}{T_i}\right) - \frac{E_{xEHF} - \frac{cd \left(x + [1-a]x \right) V^{m+1}}{\eta_e \eta_m \eta_T}}{\eta_e \eta_m \eta_T} \pm$$

$$\pm q_{FO} \left|1 - \frac{T_o}{T_{av}}\right|.$$  \hspace{1cm} (30)

Figure 10. Expanded (Holistic) Model of Heat Recovery in a Building with Grid Power.

Figure 11 shows the destroyed unit exergy of these two types of power plants. Let $c_{ic}$ and $c_{iNG}$ are the capacity-weighted factors of unit $CO_2$ emissions of installed thermal power plants, namely coal-fired and natural-gas-fired, respectively. Then for the energy mix of a given country, represented by $c_{imix}$, that may be approximated from Equation 32, the second term of Equation 1 is modified in the following form:

$$\Delta CO_2 = 0.27 \left[ e_{desc} + \left( C_1 e_{desc} + C_2 e_{descNG} \right) \right]$$  \hspace{1cm} (31)

$$c_{imix} = C_1 \left( 0.6 c_{ic} \right) + C_2 \left( 0.2 c_{iNG} \right)$$  \hspace{1cm} (32)

Here 0.6 and 0.2 are the unit $CO_2$ emission rates of anthracite coal and natural gas, respectively, at adiabatic conditions in the air. The unit is kg $CO_2$/kW-h of fuel LHV.

$T_f = 2453 \text{ K}$

$T_E = 550 \text{ K}$

$T_{ref} = 273 \text{ K}$

$T_f = 2223 \text{ K}$

$T_E = 450 \text{ K}$

$T_{ref} = 273 \text{ K}$
Coal-Fired Power Plant

Combined-Cycle Natural Gas Power Plant

Figure 11. Exergy Destructsions in Two Types of Thermal Power Plants.

A better approach to eliminate the power supply-related disadvantages that have been revealed by the Holistic Model for the built environment is to move on towards passive houses. A renewable energy system example is the passive preheating of the outdoor air with a solar air heater system, typically installed on the roof of the sustainable building, which operates with the natural convection of colder outdoor air passing through the sandwiched duct beneath the PV panel. While colder air enters the PVT from the bottom, it rises and the indoor ventilation air is preheated. According to Figure 12, solar radiation intensity normal to the PVT surface, $I_n$ is 750 W/m$^2$. The Carnot cycle equivalent solar source temperature, $T_{sol}$, corresponding to $I_n$ is calculated from Equation 33 \[32\]. 1366 W/m$^2$ is the average value of the solar constant. Even when the Air-Air PVT operates without a fan (Natural Convection) it is responsible for avoidable CO$_2$ emissions.

$$\frac{I_n}{1366} = \left(1 - \frac{T_{ref}}{T_{sol}}\right)$$

In the summer period, additional equipment and exergy sinks are involved, which are shown in Figure 13. The PVT panel is cooled by air, which in turn, is cooled by the utility water. Utility water after being warmed by exchanging heat with the exit air from the PVT panel is further heated through a ground-source heat pump on demand and at the absence of the cooling load, in order to avoid the Legionella disease risk and then stored in a DHW tank. Part of the electric power generated by the PV cells drives the ground-source heat pump (GSHP), which satisfies the space cooling loads during the day time. The reject heat goes to the ground well. The air loop normally depends again on natural convection while a water pump is introduced for the ground loop. However, water pumps require less power than fans in transferring the same amount of heat \[30\]. In this arrangement, there is direct emissions responsibility at the power plant feeding the pumps through the grid.

$$\Delta CO_2 = 0.267 \varepsilon_{ex} = 0.267 \left(1 - \frac{290}{313}\right) = 0.02 \text{ kg CO}_2/\text{kW-h of heat}$$
Figure 12. Air-to-Air (PVT) Panel with Natural Circulation: Winter Mode.

Figure 13. Air-to-Air (PVT) Panel with Natural Circulation: Summer Mode.

Regarding a PVT system, the total output is not a simple addition of electricity and heat, because of their different unit exergy. Heat and cold also have different unit exergy. Depending on the ratio of power and heat generation, the rational exergy management efficiency, $\psi_R$ varies, which directly affects the added value of a PVT. Therefore the cost of a PVT must be levelized according to exergy in terms of $\psi_R$. In this study the Exergy-Levelized Cost, $ELC$ has been developed, which combines exergy rationality in terms of $\psi_R$, the selling price, $PC$, embodied costs, $EM$, panel area, $A$, and panel weight, $W$ [20]. $ELC$ serves for establishing a new comparative metric, which may also be used for rating the exergo-economic performance of a PVT system.

$$ELC = \frac{PC + EM}{\psi_R (A/W)}$$  \hspace{1cm} (34)

Furthermore, a second new metric, $TI$ evaluates any system in terms of its energy efficiency (First Law), energy rationality (Second Law) and ozone depletion index, $ODI$. This gives a total evaluation index.

$$TI = \eta R (1 - ODI)$$  \hspace{1cm} (35)

**CONCLUSION**

According to a recent study and an embodied exergy model about horizontal development versus vertical development in new settlements in the built environment by Kılıç, S. and Kılıç, B. [36], exergy is a game-changer. This also applies to the energy recovery in ventilation as shown in this paper. An HRV unit may seem very beneficial in economic and energy savings potential and even maybe touted to be environmentally benign. The same HRV unit actually proves to be exergy- irrational with existing technology, which depends upon grid electricity. First of all, more efficient electric motors, fans, and heat exchangers with less pressure drop across them need to be designed and new alloys and composites have to be used with fewer embodiments during their manufacture. However, these have diminishing returns in terms of the First Law.

Renewable energy sources with little or no electric power requirement must be utilized for effective solutions to make the $COPEX$ value approach or even exceed 1. Even if on-site solar electricity is used like the one shown in Figure 14, the Second Law asks the next question about what is the best rational way of using this electricity, either in an HRV unit or in public transport, and questions keep going on until the best allocation
scheme of renewable energy sources are set in a given district and set of buildings. Therefore, it is time to shift to the Second Law if global warming is to be actually reduced and CO$_2$ emissions are decoupled from the human-focused economic growth. In the meantime, the existing HRC units may be optimized such that COPEX approaches one.

The new model may be transformed into user-friendly software in order to assist responsible designers and implement exergy-based control algorithms. Such a move will not just mimic existing tools, which are based on the First Law only [8] but far exceed them in optimization with true environmental contributions.

Apart from the above discussions, one needs to realize that the introduction of HRV technology reduces the size of the original HVAC system, like using ground-source or air-to-air chillers (heat pumps). The implication is the reduction of the installation cost of the original HVAC system versus the addition of the HRV unit in series or parallel. The same holds true for Embodied CO$_2$ emissions related to the material used for the heat pump and the HRV unit. An original A-A heat pump is downsized (Figure 14) but a new unit is introduced (HRV). The net embodied CO$_2$ at the beginning (operating time equal to zero) may be higher than the original. This initial increase, however, may be compensated at a time of $X_o$, because the COP of the HRV unit is higher. Thus the slope of the line is smaller) than the heat pump itself (See Table 5). For a ground-source application, the slope is smaller than the A-A case, because the COP is relatively higher. Yet the initial CO$_2$ embodiment requires additional groundwork and heat exchanging tube material etc. Therefore, the line is above the A-A case with a smaller slope. This, in general, renders a bigger positive impact of HRV with a bigger HRV, which brings the starting point $O$ is almost to the same point. But because the slope is not large enough this option may not return the CO$_2$ embodiment back. All these considerations show how in fact the problem is complicated even if a ‘simple’ HRV application is the question.

The importance of exergy in waste heat recovery is becoming an important also in the recent literature, due to the fact that the unit exergy of waste heat is comparably less than the unit exergy of electric power used for the heat exchanging process. Some Authors have started to consider exergy in their recent literature yet limited to basic thermodynamics and economy. For example, Ayachi et al. [37] determined the choice of system design and working fluids for an Organic Rankine cycle through a break down thermodynamic (Second-Law) analysis limited to system components. They have investigated two thermal sources, namely dry gas at 165 °C and moist gas at 150 °C. Their objective was to identify recovery solutions suitable for minimizing exergy destruction. In this respect, they analyzed several options including Organic Rankine and CO$_2$ transcritical cycles. They concluded that referring to both laws of thermodynamics, the optimum solution may be obtained through a set of suitable design steps, choosing a proper fluid and determining suitable operational such that pinch points are eliminated. Xu et al. [38] in their proposed hybrid ventilation air methane (VAM)-hybrid power generation system and a circulating fluidized bed, claimed that exergy and electricity (power) of the fuel (coal and VAM) are nearly equal to their embodied energies, and thus, the exergy efficiency of the system can be taken nearly equal to its energy efficiency. This claim shows that many authors take side steps to neglect the Second Law. Such research must be expanded to exergy rationality and must include embodied exergy and CO$_2$ emissions. In their study, Erguvan and MacPhee [39] carried out energy and exergy analyses for unsteady cross-flow overheated cylinders. They found that energy efficiency varies between 72% and 98%. The exergy efficiency for corresponding cases ranged between 40% and 64%. Their results suggest that exergy efficiency can be maximized especially in low-temperature applications like ERV systems by choosing specific pitch ratios for various Reynolds numbers. Similar studies available in the literature are helpful.
for equipment design from exergy point of view but do not address the overall and holistic rationality of exergy allocation.

As a final remark, we need to work inch-by-inch to re-wire exergy source and demand pathways to avoid the global warming emergency, starting from the tiniest equipment to much larger systems like power plants and metropolitan cities. The ultimate goal is to minimize avoidable CO₂ emissions by implementing new but simple methods, that do not appeal to big investments but a productive and positive state of mind.

Figure 14. Embodied CO₂ Considerations About Downsizing the HVAC Equipment Versus Introducing HRV.

NOMENCLATURE

A Solar panel irradiation area, [m²]
\(a\) Relationship between \(x\) and \(y\) (see Figure 3 and Equation 17)
\(a_{ch}\) Air Change per hour, [h⁻¹]
ALT Residence Time in the Atmosphere (Equation 27), years
\(C_1, C_2\) Ratio of the coal and natural gas usage, respectively in the installed national power generation capacity mix
\(c\) Constant for the fan characteristic (Equation 7)
\(C_f\) Unit cost of fuel, [$/kW-h]
cfm Cubic feet per minute, convert to m³s⁻¹ by multiplying by 0.000472
\(C_E\) Unit cost of electricity, [$/kW-h]
\(c_i\) Unit CO₂ emission rate, [kg CO₂/kWh] (Based on lower heating value)
\(c_{imix}\) Installed capacity-weighted unit CO₂ emissions rate of the fuel mix in national power generation system, [kg CO₂/kWh]
\(C_P\) Specific Heat, [kJ/kg·K]
CO₂ Direct CO₂ emissions, [kg CO₂/kW-h]
\(COP\) First-Law Coefficient of Performance, dimensionless
\(COP\text{EX}\) Second-Law Coefficient of Performance, dimensionless
d, e Constants in Equation 8 and 9, regarding additional pressure drop in the HRV unit
\(E\) Energy, [kW]
\(E_S\) Seasonal Energy Savings (Equation 6)
\(E_X\) Exergy, [kW]
\(ELC\) Exergy-Levelized Cost factor, [€-kg/m²]
\(EM\) Embodied cost of a solar panel [€]
\(E_{XA}\) Net Exergy Recovery on the Exhaust Side from Original Temperature-Peaking Process on the Pre-Heated Air Side of the HRV Unit, [kW]
\(E_{XF}\) Exergy Demand of Fan, [kW]
\(E_{XE\text{EO}}\) Electrical Power Exergy Demand of the Dedicated HRV Fan on the Fresh Outdoor Air Intake Side, [kW]
\(E_{XEHRV}\) Electrical Power Exergy Demand of the Dedicated HRV Fan on
the Exhaust Air Side, [kW]

\( E_{\text{EXTP}} \) Exergy Spent by Auxiliary Temperature-Peaking System, [kW]

\( E_{\text{EXH1}} \) Thermal Exergy Gained in the RHV Unit, [kW]

\( E_{\text{EXH2}} \) Thermal Exergy Gained in the HP Unit, [kW]

\( E_{\text{EXHRV}} \) Embodied Exergy of HRV unit, [MJ], [kW-h]

\( E_{\text{EXTP}} \) Exergy of the fuel or power input for temperature peaking of the Supply Air (If necessary), [kW]

\( FC \) Selling cost of a solar panel, [€]

\( GWP \) Global Warming Potential

\( H \) Net Conversion Ratio of the Electrical Power Input, which is not Converted to Shaft Power to the Heat Transferred to the Air Flow

\( I \) Investment cost, [$]

\( I_n \) Solar Radiation Intensity Normal to the PVT Surface, [W/m\(^2\)]

\( ODI \) Ozone Depletion Composite Index

\( ODP \) Ozone Depleting Potential

\( OF \) Objective Function (Equation 13), Net Exergy Gain of HRV+HP, [kW]

\( \Delta E \) Power Required to Compensate for the Pressure Drop in HRV Unit

\( \Delta P \) Pressure Drop, [Pa]

\( \Delta T \) Temperature Difference, [K]

\( Q \) Thermal Power, kW

\( q, r \) Linearized COP Factors for Heat Pumps (Equations 15 and 16)

\( s \) Constant in Equation 27

\( t \) Time, hour, year

\( T \) Temperature, [K]

\( TI \) Total Evaluation Index, dimensionless

\( T_1 \) Pre-Heated Supply Air Temperature at the Exit of HRV, [K]

\( T_2 \) Pre-Heated Supply Air Temperature at the Exit of HP, [K]

\( T_g \) Ground Source Temperature, K

\( T_i \) Indoor Air Temperature at the HRV Inlet on the Exhaust Side, [K]

\( T_o \) Outdoor Air Temperature at the Entrance of HRV, [K]

\( V \) Volumetric flow rate, [m\(^3\)/s]

\( W \) Weight, [kg]

\( w \) Correction factor for COPEX in Equation 25, 0.85 ≤ w ≤ 1.0

\( X \) Operating time, hour

\( x \) Outdoor Air Split Ratio Between HRV and HP

\( Y \) Simple payback period, number of heating or cooling seasons

\( y \) Indoor Air Split Ratio Between HRV and HP

**Greek Symbols**

\( \varepsilon \) Unit exergy, [kW/kW] or [W/W]

\( \rho \) Density, [kg/m\(^3\)]

\( \eta_I \) First-Law Efficiency, dimensionless

\( \eta_F \) Fan Efficiency

\( \eta_{bm} \) Belt-Motor Efficiency

\( \eta_B \) Boiler Efficiency, dimensionless

\( \eta_T \) Overall Efficiency of Power Generation and Transmission

\( \psi_R \) REMM Efficiency, dimensionless

\( \Sigma CO_2 \) Sum of direct CO\(_2\) and Avoidable CO\(_2\) Emissions, [kg CO\(_2\)/kW-h]

**Subscripts**

1 Any Variable Related to RHV

2 Any Variable Related to RHV
Boiler, Furnace, or Thermal Plant
Related to Investment Cost
Coal
Condenser
Demand
Destroyed (exergy)
Electric, Exit
Exhaust
Evaporator
Embodied
Electric motor or embodied
Energy
Exhaust Path of HRV
Heat Recovery Ventilation Related
Supply Air, or fuel (Source)
Fuel
Fan Heat Gain by the Incoming Outdoor Air in the HRV Unit
(in Winter)
Heat
Heat Pump
Inside, indoor
Fuel mix in the energy sector supplying the power grid.
Natural gas
Outside, outdoor (supply), Preheating Path of HRV
Break-even Point
Reference
Solar Energy Related
Supply
Power Transmission and Distribution
Temperature Peaking
Exergy
Power in Equations 8 and 9
Powers in Equation 27
Air to Air
Air Handling Unit
American Society of Heating, Refrigerating, and Air-Conditioning Engineers Inc.
Brake Horse Power
Dry-Bulb (Temperature)
District Heating
Domestic Hot Water
Exhaust Air Heat Pump
Energy Conservation in Buildings & Community Systems Programme
Energy Reuse Effectiveness
Energy Reuse Factor
Energy Recovery Ventilation (Sensible and Latent Heat)
European Union
Greenhouse gas
Ground-Source Heat Pump
Heat Exchanger
REFERENCES