ANALYZING THE IMPACT OF THE PHIUS HRV/ERV PROTOCOL ON NORTH AMERICAN PASSIVE HOUSE CERTIFICATION

by

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Abstract

The stated heat recovery efficiency of HRV and ERV units in North American passive houses is dependent on the testing procedures and calculation methods established by several pertinent performance testing standards. This project highlights major differences between the applicable HRV/ERV standards for North American passive houses: the European Passive House Institute standard, the Canadian CSA-439-09 standard, and the American HVI-920 standard. It further examines the proposed PHIUS protocol which established η_{PHUIS}, a modified HRV/ERV heat recovery efficiency rating to more accurately reflect the North American climate. Simulations were performed to quantify its effect on the modelled annual heat demand for 31 certified passive houses. The results yielded two key findings. First, the margin of error for the new rating, η_{PHUIS}, relative to the existing rating, Ɛ, is a function of the regional climate given by the equation: y = 0.00001x + 0.0012. Locations with a colder climate have longer winters, thereby increasing the heating demand and intensifying the margin of error. Second, small to medium-sized houses with floor areas (<250m^2), which formed 90% of the sample study, have the largest impact on the margin of error up from 3.8% to 12% compared to large homes (>250 m^2) from 2.8% to 4.2%. The results validate the necessity for PHIUS' proposed η_{PHUIS} for North American HRV/ERVs.

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List of Abbreviations

PHI – Passive House Institute
PHIUS – Passive House Institute United States
CSA – Canadian Standards Association
HVI – Home Ventilating Institute
ERV – Energy Recovery Ventilator
HRV – Heat Recovery Ventilator
PHPP – Passive House Planning Package

1.0 Introduction

The energy efficiency of buildings has become both an environmental and economic imperative as a result of global climate change and the diminishing of conventional energy resources. The current trend of constructing low energy and passive houses is an example of reducing operating energy in the residential building sector. The concept of the passive house primarily focuses on a well-designed building envelope (walls, roof, windows, floors), which substantially limits the requirements for active mechanical systems to meet the home's heating and cooling demands. There are different strategies for building a passive house, however, a key reference is the German Passive House standard developed by the Passive House Institute (PHI) in 1996 (Passive House Institute, 2013). This pass or fail performance standard provides a set of applicable requirements and metrics for measuring energy efficiency, including a certification program which has been extensively used throughout Germany and Austria since 1990s (Feist, Peper, & Görg, 2001). Passive houses have been widely popular in Europe and the momentum has carried to North America, with thousands of buildings having been built globally to its specifications (Passive House Institute, 2015)

A key component in a passive house is the ventilation heat recovery system. Traditional single-family homes in North America rely on unintentional cracks and openings in the building envelope for ventilation. Unintentional ventilation heat loss through the building envelope is considerably greater than transmission heat loss, particularly in a well-insulated and airtight passive house (Juodis, 2006). Consequently, the reliance on the performance of mechanical ventilation units such as heat recovery ventilators (HRV) and energy recovery ventilators (ERV) is significant. With North America having relatively more extreme climates when compared to Europe, the efficiency of North American HRV/ERVs will therefore have a comparatively larger impact on a home's annual heat demand.

1.1 Objective and Scope of Research

There are three main criteria a building must meet to receive Passive House certification. They relate to space heating and cooling demands, building envelope airtightness, and total primary energy consumption. The heating and cooling energy demands can be achieved in part with highly efficient HRV/ERV units. The efficiency ratings of these units are measured and calculated as outlined in HRV/ERV performance testing standards, one of which is provided by PHI based on the central European climate (Passive House Institute, 2006a). In North America, the standards stipulating testing procedures for rating residential HRV/ERVs are the Canadian Standard Association's (CSA) C439 (Canadian Standards Association, 2009), Home Ventilating Institute's (HVI) 920 (Home Ventilating Institute, 2009), and the proposed Passive House Institute United States' (PHIUS) protocol (Wright, Klingenberg, & Pettit, 2015). The heat recovery efficiency of an HRV/ERV is determined by an array of factors, such as the insulation of the case, air tightness, heat exchanger technology, the supply and exhaust openings considered for energy balance, etc. However, the results for heat gains and heat losses (i.e. efficiency) for the same unit could be different if tested to varying standards, due to the differences in boundary conditions for calculation (Schild & Brunsell, 2003).

The objective of this project is to compare and quantify the effect of utilizing different certification standards for HRV/ERVs used in constructed North American passive houses. This paper will serve to answer two research questions:

1. What are the major differences in testing procedures and requirements between the pertinent performance testing standards for single-family HRV/ERVs used in passive houses?
2. How does the proposed PHIUS protocol impact the projected annual heat demand in certified passive houses across North America compared to current applicable standards?

Section 2.1 presents the concept of passive house in both the North American and European context. Section 2.2 provides a background on the use of HRV/ERVs in passive houses. Section 2.3 introduces the relevant standards which stipulate testing procedures and methods for calculating HRV/ERV performance. Section 2.4 scrutinizes the differences amongst the standards, thereby addressing the first research question. Section 3.0 analyzes how the differences affect an HRV/ERV's stated heat recovery efficiency. Section 4.0 examines the proposed method to reconcile the differences in varying standards so that an appropriate efficiency rating can be calculated. This new rating will be used as an input value to simulate the projected annual heat demands for each of the selected existing 31 projects. Finally, Section 5.1 quantifies and discusses the effect of the new rating on the projected annual heat demands, thereby addressing the second research question.

2.0 Background and Literature Review

The focus of this paper is to distinguish the discrepancies between major HRV/ERV performance certification standards and understand how they affect the calculations for annual heat demand in a North American passive house. It is important to acknowledge the differences in these standards are a consequence from being developed in different countries with distinct climates, manufacturer processes, and construction practices.

2.1 Passive Houses in North America and Europe

The concept of a passive house is prevalent throughout Europe, particularly in Germany due to Passive House Institute (PHI) and its certification program. Many neighbouring European countries have adopted the PHI standard, but fine-tuned the actual criteria to their climates, budgets, and cultural context. In Switzerland, the Minergie-P is the adaptation of the German passive house standard. Minergie-P eased the PHI heating demand limit by using a larger energy reference area (external face of exterior wall) rather than PHI's treated floor area (internal face of exterior wall) (Minergie, 2015). Similar leniency was found in Sweden, where they stipulated a heating peak-load-only limit of 15 W/m^2 compared to 10 W/m^2 in the PHI standard, with additional allowance for smaller buildings (Jacobson, 2013). In Brussels, Belgium, the requirement for primary energy demand became substantially more difficult to meet, changing PHI's 120 kWh/m^2a to 45 kWh/m^2a. However, renewable energy such as photovoltaic generation was permitted to meet this demand (Dockx, 2013).

By and large, European countries other than Germany where the PHI standard originated from, have modified the specific energy targets in order to suit their needs. Passive houses in North America should be no different; however, this is not the case. North American passive houses certified to date have been meeting the energy targets prescribed in the PHI standard with no adjustments. Although this standard makes sense in Germany, its applicability and practicality in North America are being disputed based on the presence of drastically different climates (Straube, 2009). For the central European climate in which it was developed for, the additional degree of building envelope investment required to meet the metrics was cost-competitive and in some instances cost-optimal (Wright, Klingenberg, & Pettit, 2015). The significant savings for choosing smaller mechanical systems as a result of a higher performance building offsets the extra initial investment. This leads to a more efficient building using less energy, resulting in long term economic savings from reduced operating energy consumption. However, this point of cost-optimality shifts in North America. Savings from replacing typical high-cost European baseline boilers and hydronic heating systems with smaller compact ventilation and heating units may not be achieved in North America where the market availability for high-capacity mechanical equipment is still quite high (Wright, Klingenberg, & Pettit, 2015). Furthermore, manufacturing companies for compact all-in-one units are relatively uncommon in North America, thus expensive oversea products negate potential savings. Another reason against using energy metrics derived from the European context is the difference in climatic conditions across the two continents. The correlation between degree-days and design temperatures is weak, and its relationship differs between North America and central Europe (Wright, Klingenberg, & Pettit, 2015). Degree-days and design temperatures affect annual energy demand and peak loads, respectively. Low annual energy demand results in cost savings from energy bills while low peak load reduces the size of mechanical equipment. Based on current North American passive houses built to the PHI standard, the common problem is they tend to overheat from excessive solar gains. Many of these homes have too much glazing, but are intentionally designed this way to reduce annual heat demand and meet the specified PHI limit. In other words, they were designed to meet certification limits in the winter, which is more challenging, even though it may not make sense in the summer with the extra heat gains. There are discussions arguing that the PHI standard is not economically-beneficial for cold climate North American housing, specifically in ASHARE Climate Zones 5 to 7 (Straube, 2009). Therefore, some of the recommendations provided by PHI may be impractical and disadvantageous to the typical North American homeowner.

In response to this, the Passive House Institute United States (PHIUS) formed a volunteer Technical Committee in 2011 and in collaboration with Building Science Corporation developed a standard adaptation in 2015 (Wright, Klingenberg, & Pettit, 2015). This new adaptation will implement the climate-specific standard in their PHIUS certification program, providing adjustments to the PHI criteria and making it suitable for North America's more extreme climates. The adjustments include modifications to HRV/ERV efficiency calculations found in current applicable performance testing standards. The committee will periodically update and revise the standard to reflect the ongoing changes in market prices, new materials, and climates.

2.2 Heat and Energy Recovery Ventilators in Passive Houses

The premise of the passive house is to provide significant reductions in heat losses to a point where internal and solar gains eliminate the need for a separate heating system and thus, heating can be supplied through the ventilation system. This is achieved primarily through excellent thermal insulation and airtightness of the building envelope. The exceptionally well-insulated envelopes will ensure surface temperatures of exterior walls and windows are close to ambient room air temperatures. This will eliminate the need for radiators to compensate for radiative asymmetries and cold air downdraught (Feist, Peper, & Görg, 2001). Furthermore, the added insulation will reduce the heating demand low enough where heating the ventilation air is adequate to meet space heating requirements.

Figure 1 illustrates the energy consumption by end-use in single family detached homes across the U.S. in 2009 (Energy Information Administration, 2009) and Canada in 2011 (Natural Resources Canada, 2011). In the U.S., approximately 42% of total energy consumption is used for heating, while cooling takes up only 8%. Similarly in Canada, approximately 65% of total energy is consumed for heating and only a fraction for cooling at 2%. It makes sense that energy consumption for heating is more in Canada because Canada is geographically farther north with colder climates than the U.S. It is evident from the statistics that space heating is the most significant portion of the total energy consumption in the North American residential sector. For this reason, this paper will focus on the discussion of energy demands for heating only.

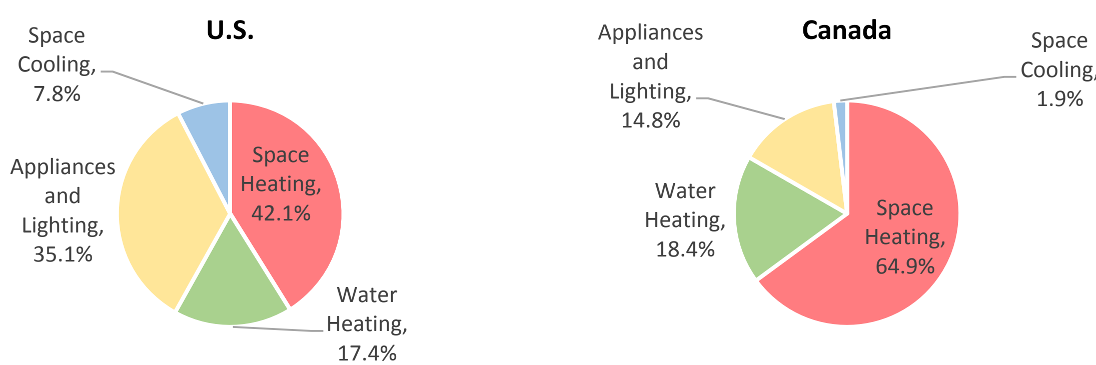

Figure 1. End-use consumption percentages in single detached homes, U.S. in 2009 versus Canada in 2011, (Energy Information Administration, 2009; Natural Resources Canada, 2011)

For a North American passive house to obtain Passive House certification, it must the meet all three compulsory energy consumption targets prescribed by the PHI standard as shown in Figure 2.

1. Certification criteria

Heating

Specific space heating demand ≤ 15 kWh/(m^{2}a)

or alternatively: heating load ≤ 10 W/m^2

Cooling[1] (including dehumidification[2])

Total cooling demand ≤ 15 kWh/(m^{2}a) + 0.3 W/(m^{2}aK) · DDH

or alternatively: cooling load ≤ 10 W/m^2

AND cooling demand ≤ 4 kWh/(m^{2}aK) · ϑ_e + 2 · 0.3 W/(m^{2}aK) · DDH – 75 kWh/(m^{2}a)

but not greater than: 45 kWh(m^{2}a) + 0.3 W/(m^{2}aK) · DDH

Primary energy

Specific primary energy demand for heating, cooling, hot water, auxiliary electricity, domestic and common area electricity ≤ 120 kWh/(m^{2}a)

Airtightness

Pressure test result, n_{50} ≤ 0.6 h^{-1}

[1] The criteria for cooling and dehumidification apply provisionally and may possibly have to be adapted with advances in knowledge. The requirements applicable for each building are calculated automatically in the PHPP ("Verification" Sheet).

ϑ_e: Annual mean outdoor temperature in °C

DDH: Dry degree hours (time integral of the difference between the dew-point temperature and the reference temperature of 13 °C throughout all periods during which this difference is positive)

[2] The partial requirement for dehumidification is described by the term '0.3 W/(m^{2}aK) · DDH'.

Figure 2. PHI certification criteria (Passive House Institute, 2013)

Specific space heating and total cooling demand is the energy consumption for all heating and cooling over a period of one year. As shown in Figure 3, the stipulated demand by PHI represents an 89 to 91% reduction in energy consumption when compared to Canadian averages in a single detached family dwelling from 2007 to 2011. This decrease shows the significant contrast between the conventional building stock in Canada and passive houses being certified around the world to the PHI metric. Heating and cooling peak load is the maximum heating or cooling capacity needed to maintain indoor design conditions, which is used to size furnaces and air conditioners. In North America, it is generally more cost effective to design passive houses to meet the space heating demand rather than the peak load. This is because it is expensive to add additional energy generation capacity which will only be used for a short amount of time. The International Energy Agency (IEA) report claim in some cases a reduction of up to 50% in the wholesale price of electricity can be achieved by reducing only 5% of the peak electrical demand (International Energy Agency, 2003). The only exception is found for homes located in northern cities with very long winter seasons, such as in Whitehorse and Yellowknife. There, meeting the peak load is easier because the extra energy generation capacity will be utilized frequently.

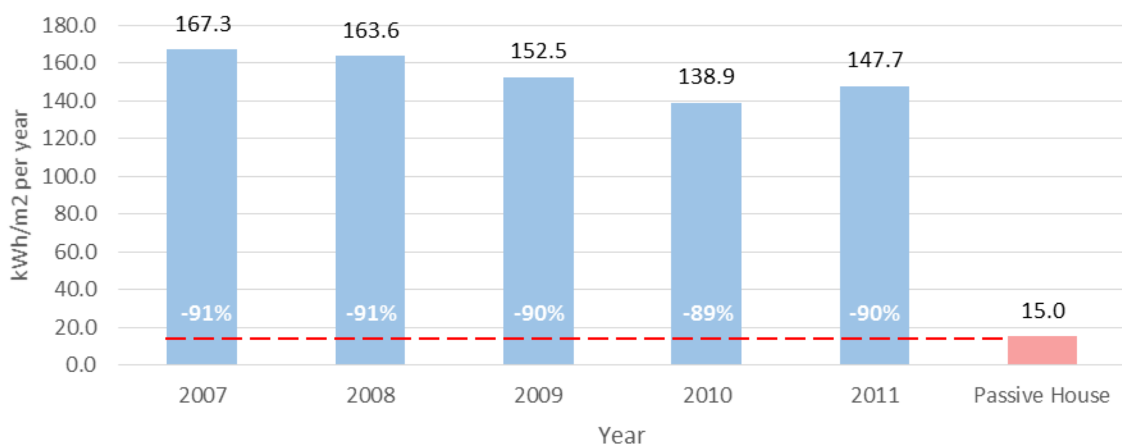

Figure 3. Energy demand for space heating and cooling in Canadian single detached homes

The space heating demand of 15 kWh/m^{2}a as defined by PHI is derived from a peak load of 10 W/m^2 in central European climates. This threshold for heating demand may fluctuate in different cities with different climatic conditions, but the peak load is climate-independent (Passive House Institute, 2006a). For example, a home in Stockholm with a peak load of 10 W/m^2 may use 20 kWh/m^{2}a whereas in Rome with the same peak load may use 10 kWh/m^{2}a. To qualify for Passive House certification, however, the building must satisfy either the 15 kWh/m^{2}a heating demand or 10W/m^2 peak load limit.

The German ventilation standard DIN 1946-6 stipulates 30 m^3/h as the minimum average rate of fresh air delivery per person required for good indoor air quality. If air has a specific heat capacity of 0.33 Wh/(m^{3}K) at 21^{o}C (ISO 7730) and can be raised by approximately 30 K (55^{o}C-21^{o}C) while avoiding odour emissions caused by pyrolysis of dust, then:

30 m^3/h/person x 0.33 Wh(m^{3}K) x 30 K = 300 W/person

This means the fresh air heater can supply 300 Watts per person and still deliver acceptable air quality. Assuming 30m^2 of living space per person, the maximum heating load at any given point must not exceed 10 Watts per square meter of living space (10 W/m^2). This is to make sure the required heat can be provided by the supply air. It is critical since the supply fresh air heater is the only heat source. Designing a building within this load, the mechanical system size will be minimized dramatically to realize economic savings. The fundamental concept is to minimize thermal loss and optimize thermal gain.

In order to meet the low space heating demand of $\leq$ 15 kWh/m^2a, it is highly recommended that high efficiency heat or energy recovery ventilators be employed. PHI further developed specific testing procedures for certifying HRV/ERVs to ensure the specific heating demand will be met. On the other hand, the North American standards CSA and HVI have their own testing procedures which certify HRV/ERV efficiencies for residential homes. Both HRV and ERV recovery devices can be used in the summer and winter. The main difference is that an HRV only recovers sensible (dry) energy while an ERV recovers both sensible (dry) and latent (moisture) energy. The decision to use one or the other depends on the regional climates, as there is no single solution for every situation. The Canadian Centre for Housing Technology conducted a field study to compare the performance of an ERV against an HRV, equipped in two identical side-by-side homes (National Research Council, 2012). They concluded that ERVs are effective in cold and dry climates, due to their ability to retain indoor humidity and prevent dryness. ERVs are also effective in warm and humid climates, as they provide better humidity control and reduce the electricity consumption of air conditioning dedicated dehumidifiers. Since the Canadian climate is predominately cold and dry, the discussion of ERVs in a Canadian passive house has more relevance compared to HRVs. In particular, ERVs with high efficiency heat exchangers are essential to meeting the annual heat demand of 15 kWh/m^2a.

Two common types of ERV heat exchangers for the single-family residential building application are the fixed plate and the rotary wheel. The fixed plate heat exchanger is typically constructed of thin plates stacked together with internal airstreams passing through where thermal energy is transferred from the outgoing airstream to the incoming airstream. Typical effectiveness of sensible heat transfer is 50-80% and the airflow can be arranged in counter-flow, cross-flow, and parallel flow (ASHRAE, 2009). The heat exchanger can be made from a humidity permeable material such as a polymer, which will transfer latent energy. A study has been conducted to measure the sensible and latent efficiencies of plate-type exchangers at typical interior heating and cooling conditions with varying exterior temperatures and humidities (Han, Choo , & Kwon, 2007). It was found that the heat exchanger artificially appeared to recover more sensible heat during winter conditions compared to the summer due to the heat gain from internal fans. Another study by simulation and experiment investigated a fixed plate heat exchanger made from plastic film in cross-flow configuration (Lu, Wang, Zhu, & Wang, 2010). It was found that the ERV had a sensible heat recovery effectiveness of 65-80%, depending on fan speed and pressure drop of 20Pa. Thus the effectiveness of heat recovery is not constant over variable airflow rates. More heat is recovered with lower airflow rates at lower fan speed settings.

On the other hand, a rotary wheel heat exchanger consists of a spinning wheel filled with an air permeable material such as polymer, aluminum, and synthetic fiber. The wheel spins between the two airstreams, transferring sensible energy. The speed of the rotor is typically slow at 3-15 rpm, with common heat recovery efficiencies above 80% (Mardiana-Idayu & Riffat, 2012). The rotor is powered by a third fan in addition to the two fans pushing air through opposing airstreams. A rotary wheel heat exchanger also recovers latent energy through the use of desiccants, typically made from silica gel or molecular sieve. The transfer of moisture occurs by the difference in partial vapour pressure across the two airstreams. In general, rotary wheels have higher heat recovery efficiencies than fixed plate configurations, however, fixed plates are generally more compact and require less maintenance due to a lack of moving parts.

2.3 Current Applicable HRV/ERV Performance Testing Standards

Specific to passive house applications, there are several major HRV/ERV performance testing standards under scrutiny in this project. The HRV/ERV performance testing standard published by PHI is used to declare “passive house suitable components” for specific use in the certification of passive houses. This standard is used widely throughout Germany and other parts of Europe. It provides a list of requirements and procedures for measuring the energy and acoustic performance of a heat/energy recovery device. In Canada, the CSA’s C439-09 standard is used for rating the performance of HRV/ERVs, but units certified under this standard may not necessarily be used for certification of passive houses (Canadian Standards Association, 2009). Similarly in the U.S., HVI’s Publication 920 is a standard based on the C439 standard (Home

Ventilating Institute, 2009). The American standard contains identical calculation methods to the Canadian standard for HRV/ERV performance, with the addition of a public product directory which lists all available units on the market for designers and homeowners to compare performance ratings. HVI also has a challenge component where one manufacturer may dispute against another manufacturer's performance rating and request the unit be re-tested by HVI. For passive house certifications in North America, units tested to any of the aforementioned standards can be used. However, if the chosen standard is any but the PHI standard, a percentage deduction will be applied to the overall rating or test result, consequently increasing the difficulty to meet passive house certification (Passive House Institute, 2013).

2.3.1 Key Terms and Approaches in Testing

The metrics used by the three agencies to measure HRV/ERV performance are different. PHI uses one metric to define the unit's overall heat recovery efficiency. CSA and HVI use one metric to define sensible, latent, or total heat recovery efficiency of the entire unit and another metric for the apparent effectiveness of the heat exchanger core. To obtain Passive House certification, the heat recovery efficiency value determined from the PHI standard is inputted into the calculation and energy modelling software PHPP. Along with other detailed information of the building, PHPP computes both the annual energy demand and peak load, yielding a pass or fail result in that category. For units tested to the standards provided by CSA and HVI, the value for apparent effectiveness is used instead with a deduction of 12% prior to being entered into the PHPP (Passive House Institute, 2013). The rationale behind this penalty is the difference in the level of rigour between the standards, with the PHI standard being more rigorous in different aspects (Passive House Institute, 2009). For example, it is the only standard of the three which emphasizes frost protection, acoustical properties, controllability, and electrical efficiency. Unlike CSA and HVI, it also stipulates a much higher minimum performance requirement, because its purpose is to certify highly efficient HRV/ERVs for use in passive house applications. The subsequent sections will discuss potential sources of this penalty and PHIUS's proposed method for calculating heat recovery efficiency in the North American climate, in an attempt to reconcile this 12% difference. The impact of PHIUS's proposed method will then be quantified in Section 5.1. Figure 4 illustrates the laboratory setup of the CSA HRV/ERV performance test.

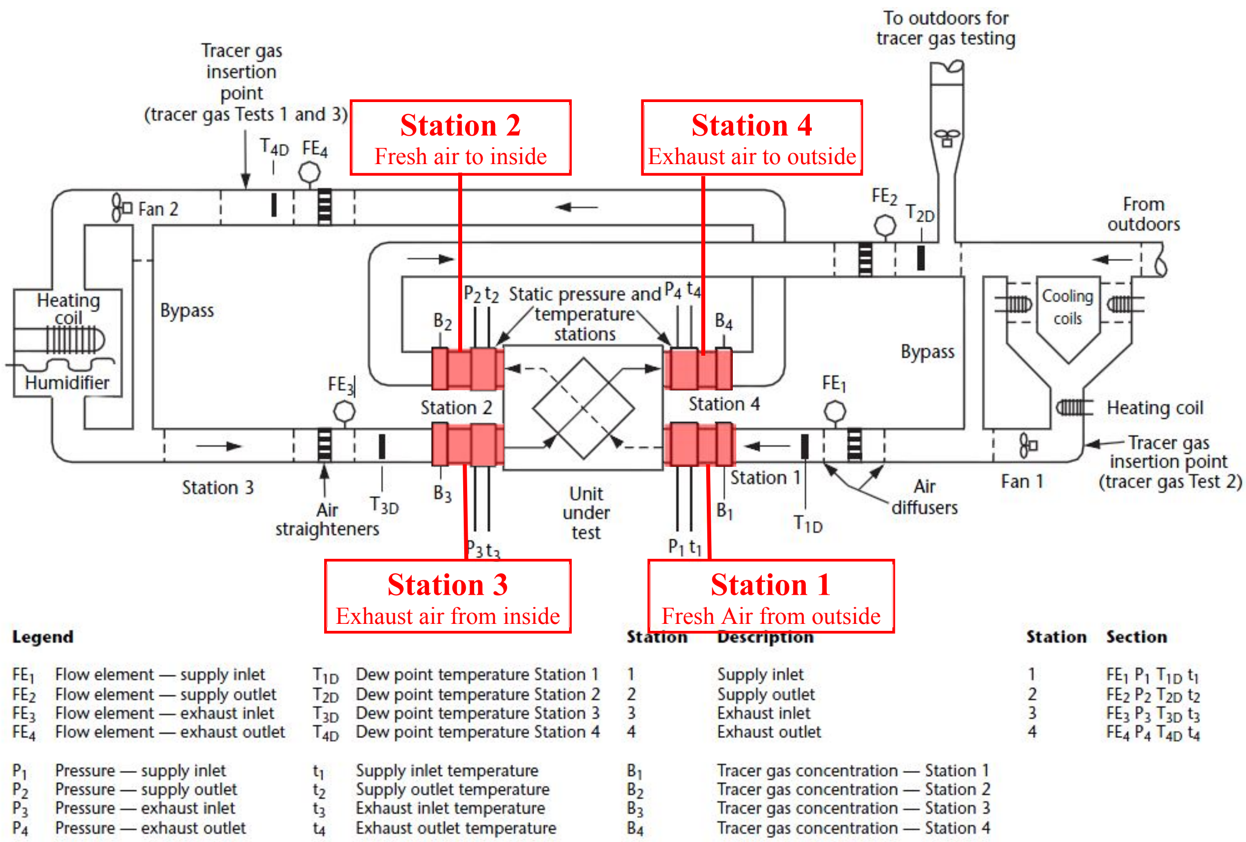

Figure 4. Schematic diagram of HRV/ERV laboratory testing setup (Reproduced from Canadian Standards Association, 2009)

The actual heat recovery efficiency of the HRV/ERV is determined by the amount of the energy transferred from the exhaust airstream to the supply airstream while deducting all energy consumed or lost in the process. However, the effectiveness of the heat exchanger in its ability to transfer sensible or latent heat without consideration for electrical consumption is also considered. In the CSA and HVI standard, this effectiveness is reported as *apparent effectiveness ($\mathcal{E}$)*. The equation for $\mathcal{E}$ may be useful in predicting the temperatures or relative humidity of the airstream at inlet or outlet positions when other temperatures or relative humidities are known. The following equation is given by CSA and HVI to determine apparent sensible, latent, or total heat effectiveness of HRV/ERV cores:

$$\boldsymbol{\varepsilon} = \frac{M_s \times (t_1 - t_2)}{M_{min} \times (t_1 - t_3)} \tag{1}$$

Where:
$\mathcal{E}$ = apparent sensible, latent, or total heat effectiveness
M_s = net mass flow rate of the supply air, kg/s

t = dry-bulb temperature, humidity ratio, or total enthalpy, respectively, at the locations 1, 2, or 3 as indicated in Figure 4, °C
M_{min} = M_s or M_e, whichever is less
where
M_e = mass flow rate of the exhaust air at Station 3, see Figure 4, kg/s

Equation 1 measures the unit's heat exchanger core effectiveness assuming ideal conditions where no air leakages within the unit take place. In reality, leaks or cracks in the unit lead to losses in air flow. Both CSA and HVI account for this by measuring the supply air contamination referred to as *exhaust air transfer ratio (R)*. The test is carried out using a tracer gas method.

The following equation is used for calculating R:

(a) If $\frac{B"_2}{B"_1} < 0.9$

$$R = 1 - \frac{B"_2}{B"_1} \tag{2}$$

(b) If $\frac{B"_2}{B"_1} \geq 0.9$:

$$R = 1 - \frac{B'_2}{B'_3} \tag{3}$$

Where:
R = exhaust air transfer ratio
B''_2 = concentration of tracer gas at station 2 (measured in the same units as B''_1), at location B_2, see Figure 4
B''_1 = concentration of tracer gas at station 1 (measured in the same units as B''_2), at location B_1, see Figure 4
B'_2 = concentration of tracer gas at station 2 (measured in the same units as B'_3), at location B_2, see Figure 4
B'_3 = concentration of tracer gas at station 2 (measured in the same units as B'_2), at location B_3, see Figure 4

CSA and HVI incorporates R into a separate equation to determine sensible and total heat recovery efficiencies, which better reflect HRV/ERV efficiency in the actual testing condition. This equation differs from ε because it considers the energy input attributed to fans, air leakages between streams, and air leakages between the unit and ambient air. Equations 4 and 5 are given to determine *sensible heat recovery (E_{SHR}) and total heat recovery efficiency* (E_{THR}), respectively:

$$E_{SHR} = \frac{\left[\sum_{i=1}^{n} M_{s,i} \times C_p \times \left(t_{5,i} - t_{1,i}\right) \times \Delta\theta\right] - Q_{SF} - Q_{SH} - Q_C - Q_D - Q_L}{\left[\sum_{i=1}^{n} M_{max,i} \times C_p \times \left(t_{3,i} - t_{1,i}\right) \times \Delta\theta\right] + Q_{EF} + Q_{EH}} \quad (4)$$

$$E_{THR} = \frac{\left[\sum_{i=1}^{n} M_{s,i} \times C_p \times \left(h_{5,i} - h_{1,i}\right) \times \Delta\theta\right] - Q_{SF} - Q_{SH} - Q_C - Q_D - Q_L}{\left[\sum_{i=1}^{n} M_{max,i} \times C_p \times \left(h_{3,i} - h_{1,i}\right) \times \Delta\theta\right] + Q_{EF} + Q_{EH}} \quad (5)$$

Where:
E_{SHR} = sensible heat recovery
E_{THR} = total heat recovery efficiency
i = ith time that data are recorded
M_s = mass flow rate of the supply air at Station 1, see Figure 4, kg/s
M_{max} = M_s or M_e, whichever is greater
Where
M_e = net mass flow rate of the exhaust air at Station 4, see Figure 4, kg/s
C_p = specific heat of the air, kJ/kgK
t_1, t_3 = dry-bulb temperatures for E_{SHR} at specified airstreams indicated in Figure 4, °C
h_1, h_3 = enthalpies for E_{THR} at specified airstreams indicated in Figure 4, °C
t_5 = net outdoor airflow temperature at Station 2, °C
h_5 = enthalpy of outdoor air at Station 2, °C
$\Delta\theta$ = time between flow measurements, s
Q_{SF} = energy attributed to supply fan motors, kJ
Q_{EF} = energy attributed to exhaust fan motors, kJ
Q_L = case leakage, kJ
Q_D = energy used for defrost, kJ
Q_C = casing heat transfer, kJ
Q_{EH} = energy used by heater in exhaust airstream, kJ
Q_{SH} = energy used by heater in supply airstream, kJ, and energy use attributed to compressor, kJ

PHI provides an equation for *heat recovery efficiency ($\eta_{HR,eff}$)*, which accounts for the electrical power from fans. This is similar to CSA and HVI's sensible heat recovery efficiency E_{SHR}, but the air leakages within the unit are tested separately following the Nordtest method (Passive House Institute, 2009). Equation 6 is given by PHI to calculate $\eta_{HR,eff}$, with subscripts rewritten to match equations provided by CSA and HVI.

$$\eta_{HR,eff} = \frac{(t_3 - t_4) + \frac{P_{el}}{M \times C_p}}{(t_3 - t_1)} \quad (6)$$

Where:
$\eta_{HR,eff}$ = sensible heat recovery efficiency
t_1, t_3, t_4 = dry-bulb temperatures at the locations indicated in Figure 4, °C

P_{el} = real electrical power, W
M = mass flow, kg/h
C_p = specific heat of the air, kJ/kgK

It is already evident that both Equations 4 and 6 are most representative of HRV/ERV heat recovery efficiency, which is useful for product comparisons. However, only Equations 1 and 6 are used as inputs in PHPP for calculating annual heat demand and peak loads. It should be noted that these equations also consider temperatures at different inlet and outlet positions. As mentioned earlier, different boundary conditions can lead to different results.

2.4 Major Differences Between HRV/ERV Standards

The HRV/ERV standard by PHI is intended to be used together with the more comprehensive whole-house Passive House standard. It ensures that only highly efficient units are being employed to serve as an integral component of a high performance low energy building. On the other hand, the shared objective of the CSA and HVI standards is to provide an accurate methodology for testing and measuring HRV/ERV performance, thus allowing different units to be compared regardless of their level of efficiency. In light of this, a considerable degree of discrepancy exists between the North American and European standards. The proceeding subsections highlight several aspects of HRV/ERVs which are considered differently across the two continents.

2.4.1 Frost control

The process of heat recovery in a passive house HRV/ERV unit is illustrated as a schematic diagram in Figure 5. On the coldest winter days where the exterior air temperatures drop significantly, the low temperatures of the incoming air may cause the heat exchanger to freeze. The resulting build-up of frost will decrease the airflow rate and recovery efficiency of the unit, leading to performance deterioration (Kim, Choi, Ha, Kim, & Bang, 2010). In order to prevent damage to the unit, the incoming outdoor air needs to be preheated to a temperature warmer than -4°C prior to entering the heat exchanger (Feist, Schnieders, Dorer, & Haas, 2005). Electric coils may be used to preheat this air, however, the operation of the electric heater may be expensive.

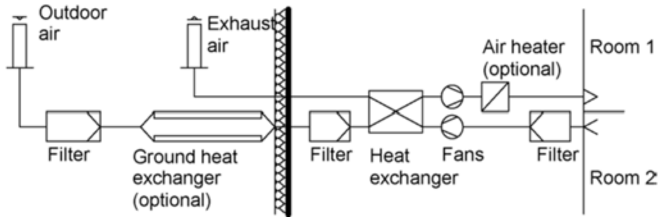

Figure 5. Schematic diagram of ventilation system and components in a single-family Passive House (Feist, 2005)

In European passive houses, the common method for frost control is to use ground-coupled heat exchangers, which are installed as earth tubes or brine-air heat exchangers (Ringer, 2011). Earth tubes are below grade pipes that draw outdoor air from a filtered air intake and preheat it using the moderate temperatures of the earth underground. The preheated air will be warm enough to go through the heat exchanger core without frost accumulation. Then it will be post-heated by optional electric coils before finally supplied into the house. Brine-air heat exchangers are similar to earth tubes except that it uses a liquid refrigerant to transfer heat instead of air through a thin pipe. If the temperature of the air exiting the heat exchanger falls below 5^{o}C, PHI requires that the HRV/ERV be automatically shut down to prevent damages to the electric coils.

In North American passive houses, preheating the incoming air is less common. Instead, HRV/ERVs usually use a temperature sensor to activate a fan-shut off defrost cycle whenever the outdoor air drops below a given threshold temperature (Peter Edwards Co., 2010). During this period the supply fan is shut off and the outgoing indoor air is automatically recirculated via bypass for a portion of every hour to warm up the heat exchanger core as shown in Figure 6. CSA and HVI accounts for the energy loss Q_D from the recirculated air in Equation 4. However, since the stale air is being recirculated instead of exhausted, the continuous stream of fresh air supply will be interrupted intermittently. This causes apparent impediment on the indoor air quality.

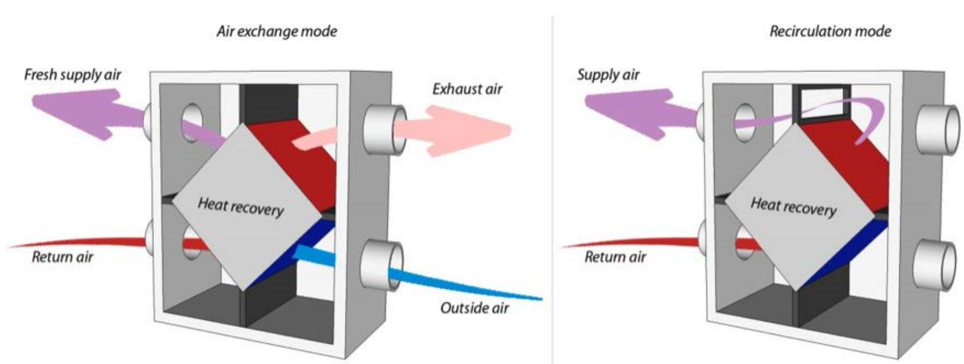

Figure 6. Recirculation mode (Cold Climate Housing Research Center, 2012)

The major difference between the two is that utilizing geothermal energy for frost control is considered free energy whereas recirculating stale air is not. Preheating cold air by ground-coupled heat exchangers is a form of frost prevention. Contrarily, stale air recirculation relies on warming up a core that already has frost deposits. During the defrost portion of the cycle, the potential for unbalanced flows can lead to depressurization of other household equipment, such as combustion appliances (Peter Edwards Co., 2010). Despite the simplicity for fan shut-off defrost without additional equipment, this method ultimately causes the unit to have poor efficiency due to the lack of heat recovery during the defrost cycle.

2.4.2 Pre and Post-Supply Air Heaters

It was discussed in the previous section that pre-heaters are used as a form of frost control. Post-heaters, on the other hand, are commonly used to raise the temperature of the supply air leaving the heat exchanger. Supply air heating is required to bring the temperature of the supply air leaving the heat exchanger to a minimum of 16^{o}C (Feist, 2006).

There are several ways to provide supplementary heating in a passive house: through a compact unit or supplementary air heater. Compact units are pre-packaged all-in-one units provide the heating, ventilation, domestic hot water, and cooling (if necessary). One of three heat generation methods are used in compact units: a small heat pump or micro-heat pump, condensing burner (using natural gas), or combustion unit for biomass fuel such as a fireplace, see Figures 7 and 8 (Passive House Institute, 2006b). A supplementary air heater can be used if the temperature of the supply air leaving the heat exchanger is not high enough to meet the heating load. Typically, it comprises of a hydronic system with hot water coils heated by direct solar thermal energy (rooftop hot water heater) or geothermal energy (ground heat exchanger). Compact ventilation and heating units have been designed specifically for use in passive houses and are popular in Europe. Unfortunately, compact units are uncommon in the North American market, thus the prices are kept relatively high. Also, geothermal energy is rarely chosen as the form of heating energy for passive houses in North America. In fact, air-source heat pumps are commonly used in Canada as a supplementary air heater to HRV/ERVs (Natural Resources Canada, 2014).

Figure 7. Fireplace as the heating system for ventilation air and domestic hot water (Passive House Institute, 2006b)

Figure 8 . Compact units with heat pump (Passive House Institute, 2006b)

2.4.3 Balanced versus Unbalanced systems

A balanced system is when the exhaust and supply airstreams have equal flow rates. Balanced airflows are important to avoid pressurizing and depressurizing the house. Pressurization occurs when the supply airflow is greater than the exhaust airflow. In the heating season, this may force heated moist air to the outdoors through the building envelope resulting in the loss of thermal energy and the potential of moisture-related damage in building components. Depressurization of the house occurs when the supply airflow is less than the exhaust airflow, pulling unheated outdoor air into the building interior where it will mix with the heated indoor air. Since this unheated outdoor air does not passed through the heat exchanger, it will reduce the enthalpy of the exhaust airstream, thereby reducing the performance of the HRV/ERV. Unbalanced airflows may cause humidity problems, wasted thermal energy, and a reduction in heat recovery performance. The ventilation losses are significantly more as a percentage of total heat loss in passive houses compared to less airtight houses with less efficient heat recovery (Feist, Schnieders, Dorer, & Haas, 2005).

HRV/ERVs with a single motor powering two fans use dampers to adjust and equalize the amount of airflow for each airstream. Units with separate motors for individual fans usually have a fan speed controller. In Europe, constant flow rate fans with electronically commutated motors (ECMs) are commonly. These fans automatically adjust its rotational frequency (rpm) to maintain a constant volume flow rate, ensuring the same amount of air in each airstream, thus creating a self-balancing unit. ECMs are found in North American furnace blowers, but as of 2010, they are not used in any HRV/ERV units (CMHC 2010). Only recently have ECMs been used in North American units, such as the UltimateAir RecoupAerator 200DX (Morosko, Personal Communication, 2015). PHI permits the unit to be either manually balanced or self-balanced. However, even if the unit is manually balanced upon installation, over time the pressure loss from ducts and the accumulation of dust resulting in filter clogging will cause the unit to be imbalanced, thus reflects the need for user controllability.

PHI permits a 10% airflow imbalance between the exhaust and supply airstreams. In the event of an imbalance of more than 10%, a correction will be applied to the temperature of the exhaust outlet at Station 4, which will penalize $\eta_{HR,eff}$. CSA and HVI also require the airflows to be balanced, but each must be within 3% from the four station average.

2.4.4 Air Leakage

The effects of unintentional airflows in ventilation units have been examined extensively. Both outdoor short circuit and indoor air leakages (inside the unit) will considerably impact the efficiencies of ventilation, heating load reduction and effectiveness of electrical energy use (Manz, et al., 2000). Outdoor short circuiting occurs when the exhaust airflows into supply air due to the positioning of exhaust and inlet grilles, velocity and direction of wind, and temperature difference between outdoor and exhaust air. For example, when the exhaust grill is placed near or directly below the inlet grill, the exhaust air may re-inter the fresh supply airstream, resulting in external air recirculation. Indoor air leakages occur when unintentional air penetrates the casing of the unit or there is cross leakage between the two airstreams. The airtightness of the unit and local pressure differences (determined by positions of fans) will control the direction and amount of airflow through the casing. To prevent air leakage and short-circuiting, appropriate unit construction in terms of insulation and installation of the units must be carefully considered.

PHI and CSA/HVI take different approaches in measuring indoor air leakages: both cross leakage and casing leakage. PHI references the Nordtest methods (NT VVS 022 and NT VVS 021) to measure internal and external air leakage while CSA use a tracer gas method. To determine cross leakage in the PHI test, the two sides of the exhaust airstream is sealed and applied both negative and positive-pressure, see Figure 9. The pressure in the supply airstream relative to the ambient air outside of the unit is brought to 0 Pa with an auxiliary fan. The input flow rate and discharge flow rates of the auxiliary fan will give a measurement of the size of cross leakage. Similarly, casing leakage in the PHI test is determined by measuring the airflow rate necessary to maintain a 0 Pa difference between the unit's interior and the ambient air.

HVI does not prescribe its own air leakage test, but rather references the tracer gas measurements performed in accordance with CSA. CSA stipulates three separate tests to determine the cross-leakage and casing leakage. For example, the first test measures cross leakage from exhaust to outdoor air and casing leakage from the exhaust air side. Inert tracer gas will be injected into a turbulent region before Station 3. The concentration of tracer gas found in the samples at Stations 2 and 4 will be used with Equations 2 and 3 to calculate the cross leakage, see Figure 10. If the amount of tracer gas found in the samples are less than what was injected initially, the remaining concentrations are contributed to casing leakage.

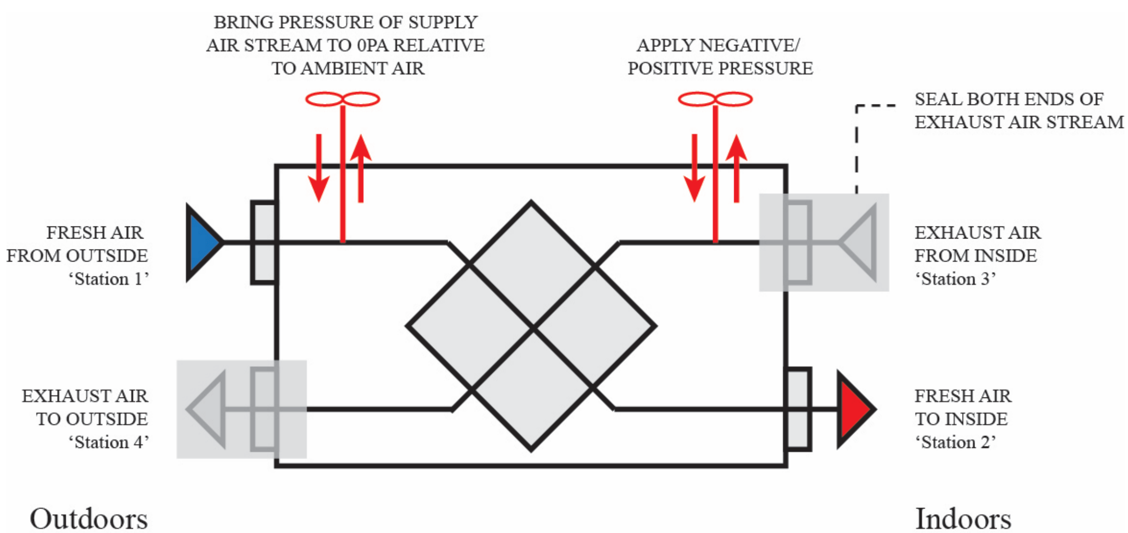

Figure 9. PHI test for measuring cross leakage

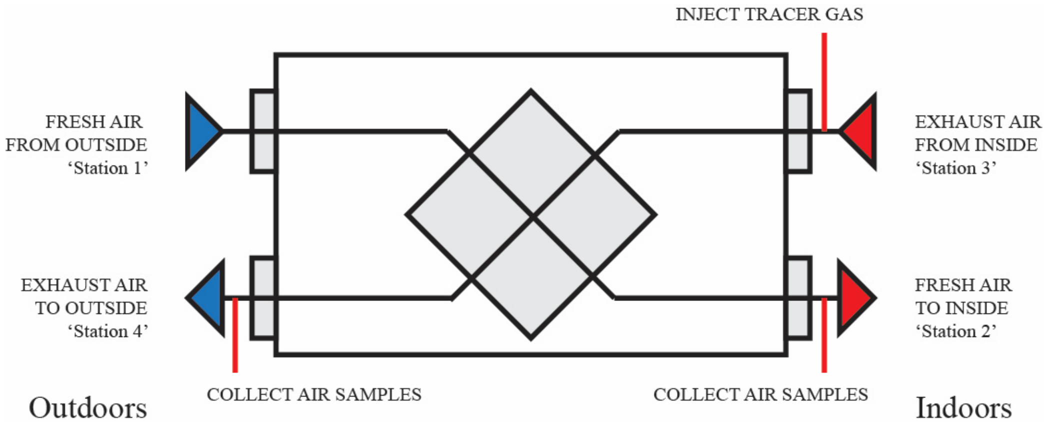

Figure 10. CSA/HVI test for both cross leakage and casing leakage

Both tests will subject the unit to negative and positive pressures. For PHI, a pressure differential in the range between 50 and 300 Pa will be conducted. Meanwhile the CSA test requires 50 and 100 Pa for fully ducted units, whereas non-ducted and partially ducted units require even less pressure differentials. PHI limits the amount of air leakage to less than 3% of the mid flow rate of the operational range. In the CSA test, the air leakage is measured and calculated as exhaust air contamination ratio R. This value affects the calculations for E_{SHR} and E_{THR}. In this sense, air leakage is considered, but there is no maximum limit for air leakage unlike the PHI test. The ambient room temperatures during the air leakage test are as follows:

Table 1. Ambient room conditions

CSA (C439)		PHI	
Temperature	15 to 35°C (59 to 95°F)	**Temperature**	20°C (68°F) +/- 1°C
Relative Humidity	20 to 60%	**Relative Humidity**	-

By limiting air leakages, the unit can perform at a higher efficiency and minimize cross contamination. However, cross leakage of the two airstreams causing indoor air quality concerns is further complicated for devices with recirculation defrost control, since cross contamination is at 100% during the defrost cycle when the exhaust air flows through the supply airstream.

2.4.5 Airflow Rates

PHI defines the operational range and flow rates for their tests. The operating range is defined as the range between the unit's highest and lowest fan speeds with an external pressure of 169 Pa and 49 Pa, respectively. The flow rates for the upper and lower limits of the operating range is determined by multiplying 1.3 with the highest speed, and multiplying 0.7 with the lowest speed, respectively, see Figure 11. The mid flow rate is the average between the highest and lowest speeds. These three flow rates (upper, mid, and lower) will be called upon during the different test measurements for acoustical efficiency, electrical efficiency, ventilation balance, airtightness, and heat recovery efficiency.

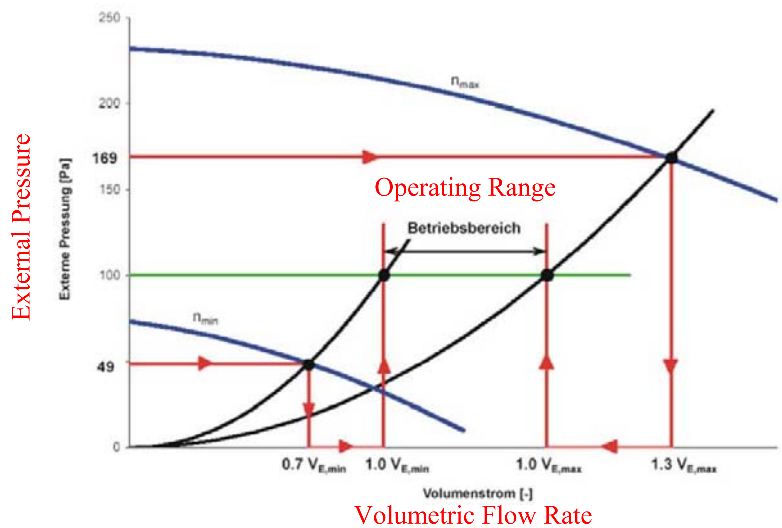

Figure 11. Graph showing operational range (PHI, 2009)

CSA and HVI do not specify flow rates and allow testing agencies to choose what flow rate to use when testing the unit. The agencies usually test the heating and cooling performance of the unit to the common high, medium, and low pre-sets defined by the manufacturer. The agencies will then list the different efficiencies for each of the tested flow rates, see Table 2. As discussed earlier in Section 2.2, heat recovery is not constant over variable airflow rates. This is why a separate heat recovery efficiency is listed for each net airflow.

Table 2. Example of HRV/ERV performance specification sheet (Bodycote Materials Testing Incorporated, 2005)

ENERGY PERFORMANCE									
	Supply Temperature		Net Airflow		Supply / Exhaust Flow Ratio	Average Power	Sensible Recovery Efficiency	Apparent Sensible Effectiveness	Net Moisture Transfer
	°C	°F	L/s	cfm		Watts			
HEAT-ING	0	32	30	64	0.98	49	83	96	0.69
	0	32	46	97	1.00	73	83	94	0.64
	0	32	95	201	1.02	260	81	93	0.55
COOL-ING	35	95	30	64	1.00	50	53**	Comments from testing agency:	
	35	95	63	133	1.02	121	44**		
*Description of Defrost:	Patented, climate dependant, controlled input heat.							Fan curve test was done at ERV maximum speed	

2.4.6 Electrical Efficiency

Electrical power consumption of the HRV/ERV includes all controls, fans, and external systems. Fan motors draw the most power in the unit, but the amount of power changes based on fan speeds. PHI requires the total electrical power consumption of an HRV/ERV to be less than 0.45 W/m^3h of transported supply airflow at the upper limit of the operational range. It also requires a maximum consumption of 1W in standby mode otherwise a complete disconnection from the electrical supply is necessary. On the other hand, CSA and HVI have no specific requirements for total electric power consumption while the device is running or in standby mode.

2.4.7 Fan Motor Waste Heat

Electronically commutated motors (ECMs) are commonly used in many residential HVAC systems. It consists of a brushless DC, three-phase motor with a permanent magnet rotor (GE ECM by Regal-Beloit, 2007). Two components that make up the ECM are: the fan drive and fan impeller. The fan drive is a microprocessor that converts AC power to DC power to operate the internal electronics. It then converts DC power to a three phase (3Ø) signal to power the fan impeller, while controlling the amount of frequency (rpm) and torque. The fan impeller is the permanent magnet synchronous rotor that propels the fan blades. Most ECMs have the fan drive integrated with the impeller, known as Integrated Motor Drivers (IMD) (Giacomini, Bianconi, Martino, & Palma, 2001).

Permanent split-capacitor motors (PSCs), on the other hand, are found in older HVAC equipment. They are less efficient than ECMs, use more electrical energy and produce more heat (Lukaszczyk, 2014). Unlike ECMs, the fan drive and fan impeller in PSC motors are not integrated together, but rather connected through long cables.

Generally there are two fans in an HRV/ERV to push air through the supply and exhaust airstreams. For ERVs with a rotary wheel, a third motor can be found to propel the wheel. An inefficient motor such as a PSC will generate more waste heat, raising the temperatures of the airstreams (Han, Choo , & Kwon, 2007). Depending on the location of the fan drives or fan impellers (if not an IMD), the additional waste heat may actually make the HRV/ERV appear more effective in recovering heat during the winter. In reality though, motor waste heat does not increase or reduce the efficiency of a unit to recover heat. In fact, it is ideal to choose efficient fan motors which limits the amount of heat it generates and electricity it uses.

PHI requires excellent fan efficiency (0.45 W/m3h), hence only ECMs can be used. However, since CSA and HVI have no limit on fan efficiency, PSCs with high waste heat may be used. CSA recognizes that the energy input attributed to fans (Q_{SF} and Q_{EF}) depends on the location of the fan motors, however, the location of the motor is dependant solely on the manufacturer's design.

2.4.8 Test Conditions

The test conditions including the precise range of temperatures and relative humidities are specified in each standard, as shown in Table 3. CSA requires an HRV/ERV be tested to the winter, summer, and low-temperature conditions. The outdoor test conditions for winter and summer season are 0°C/75% and 35°C/50%, respectively. An additional low temperature test condition of <0°C reflects the range of lower design temperatures in cold climate countries, such as Canada. For example, Toronto's winter design temperature is -17.2°C while Whitehorse is -35.3°C (ASHRAE, 2009). There are three testing conditions in the Canadian standard because there is a larger range of climate zones in North America as described by the International Climate Zone Definitions in ASHRAE-90.1-2007 (ASHRAE, 2007). In the PHI Standard, units only need to be tested to one condition, with a much smaller outdoor temperature range, from -15°C to 10°C. This reflects the fewer climate zones present in Europe.

Table 3. Test conditions for the different standards

	Outdoor		**Indoor**	
	Temp	**RH**	**Temp**	**RH**
Winter				
PHI	-15 to 10°C	-	20°C	-
CSA/ HVI	0°C	75%	22°C	40%
Summer				
PHI	-15 to 10°C	-	20°C	-
CSA/ HVI	35°C	50%	24°C	50%
Low Temp				
PHI	-15 to 10°C	-	20°C	-
CSA/ HVI	<0°C	-	22°C	40%

3.0 Analysis of Differences in HRV/ERV Standards Affecting Efficiency

The major differences outlined in Section 2.4 are not all-inclusive as PHI also requires testing of acoustical noise level, air filtration, electrical efficiency, etc. These items are not addressed within the CSA and HVI standards. However, Section 2.4 presented the key dissimilarities which contribute to the 12% test result penalty. A comparative summary of the items reviewed is shown in Table 4.

Table 4. Comparison of minimum requirements between applicable standards

Criteria	CSA/HVI Minimum Requirements	PHI Minimum Requirements
(Sensible) Heat Recovery Efficiency	>55%	≥75%
Cross leakage	-	≤3%
Minimum filtration	-	F7/G4
Minimum supply air temperature	-	≥16.5°C
Defrost required	No	Yes
Sound testing required	No	Yes
Low temperature emergency shutdown	No	Yes
Electrical efficiency	-	≤0.45 W/m^{3}hr
Temperature range for test conditions	35°C+ (<0 to 35°C+)	25°C (-15 to 10°C)
Airflow imbalance limit	<3% between the supply and exhaust airstreams	<10% from 4-station average measured, see Figure 4

Most of these differences suggest the performance test stipulated by PHI is more rigorous than that of CSA and HVI. For example, the common method of recirculating air in North America causes the HRV/ERV to have a lower efficiency since no heat is recovered during recirculation. The use of post and pre-electric heaters contributes to the electrical consumption of the whole system, whereas geothermal energy is considered energy-free. Efficient fans are not required in CSA/HVI, so even if the heat exchanger can transfer large amounts of energy effectively, the overall wattage of the entire system is not taken into account. PHI's air leakage test also subjects the HRV/ERV to higher pressure differentials.

3.1 Comparing Apparent Effectiveness (Ɛ) and Heat Recovery Efficiency ($\eta_{HR,eff}$)

Canadian Standard Association's (CSA) sensible heat recovery efficiency, E_{SHR}, corrects for several factors that add heat to the incoming supply airstream in a realistic situation. These factors include: heat generated from fans, heat transmission through the case, air leakages between the supply and exhaust airstreams, and defrost energy. Unlike E_{SHR}, the apparent effectiveness, Ɛ, assumes ideal conditions where none of aforementioned factors exist. Typically, Ɛ is higher than E_{SHR} for a given HRV/ERV, air flow, and temperature difference because it includes heat gained by the supply air from sources other than the exhaust air. It is common to use Ɛ to calculate the final delivered supply air temperature (t_2) into the room at a given flow rate for an HRV/ERV (Cold Climate Housing Research Center, 2012).

For PHPP energy modelling, the certified heat recovery efficiency rating $\eta_{HR,eff}$ from the PHI certificate is used. In the case of non-PHI certified units, the input value is the manufacturer's listed apparent effectiveness rating Ɛ, less 12%. This percentage deduction is not explicitly clarified by PHI, but can be attributed to the general inconsistencies between the applicable standards discussed above. To evaluate the applicability of this hypothesis, a study was carried out to examine the efficiency deviation when various HRV/ERVs are tested to the different standards (Stephens, 2013). The study subjected units from several manufacturers with different levels of insulation to the two testing methods as shown in Table 5, one prescribed by PHI and another prescribed by the European certification body TÜV, the Association of German Engineers (VDI), and the European Testing Center for Residential Ventilation Systems (TZWL). The former test method determines $\eta_{HR,eff}$, while the latter test method is equivalent to CSA's Ɛ. It was found the level of deviation lessens as it moves towards better insulated units. For poorly insulated units with higher heat losses, the difference between $\eta_{HR,t,eff}$ and Ɛ is significant. For well insulated units with lower heat losses, the difference becomes less noticeable.

Table 5. Efficiency deviations for units subjected to the two test methods (Reproduced from Stephens, 2013)

Manufacturer	Insulation Level	$\eta_{HR,eff}$	ε
		$\eta_{HR,eff} = \dfrac{(t_3 - t_4) + \dfrac{P_{el}}{M \times C_p}}{(t_3 - t_1)}$	$\varepsilon = \dfrac{(t_1 - t_2)}{(t_1 - t_3)}$
1	Low	69.9%	90.0%
2	Medium	59.2%	95.0%
3	High	93.0%	94.0%

Another similar test was conducted, but with 6 units. Out of the 6 units tested to both test methods, there was a deviation range between 16 to 31% with an average of 20% as shown in Figure 12. This study makes it difficult to justify PHI's 12% penalty on Ɛ since the difference between the two ratings is highly dependent on the quality and level of insulation for individual units.

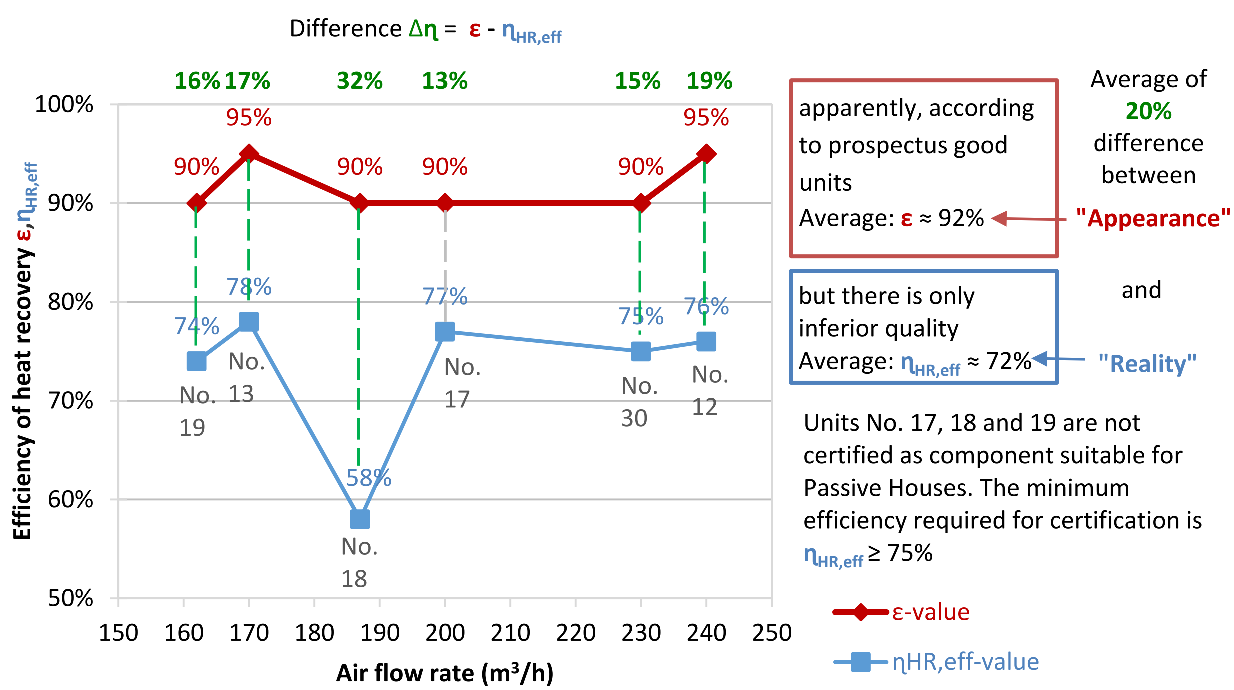

Figure 12. Comparing efficiency of heat recovery across various units using Ɛ and $\eta_{HR,eff}$ (Reproduced from Stephens, 2013)

3.2 Factors Contributing to an Increased Apparent Effectiveness (Ɛ)

The value for Ɛ is an inaccurate representation of HRV/ERV system efficiency. While the calculation for Ɛ considers temperatures at Stations 1, 2, and 3; the calculation for $\eta_{HR,eff}$ is based on temperatures at Stations 1, 3, and 4 as indicated in Figure 13. By accounting for the supply airstream going into the house (temperature at Station 2 or t_2), Ɛ may or may not include the additional motor waste heat generated by the fan drives, depending on where the manufacturer positions the fan drives- at the supply airstream, exhaust airstream, or both. The next Section examines how waste heat on a particular model will artificially raise the value for Ɛ.

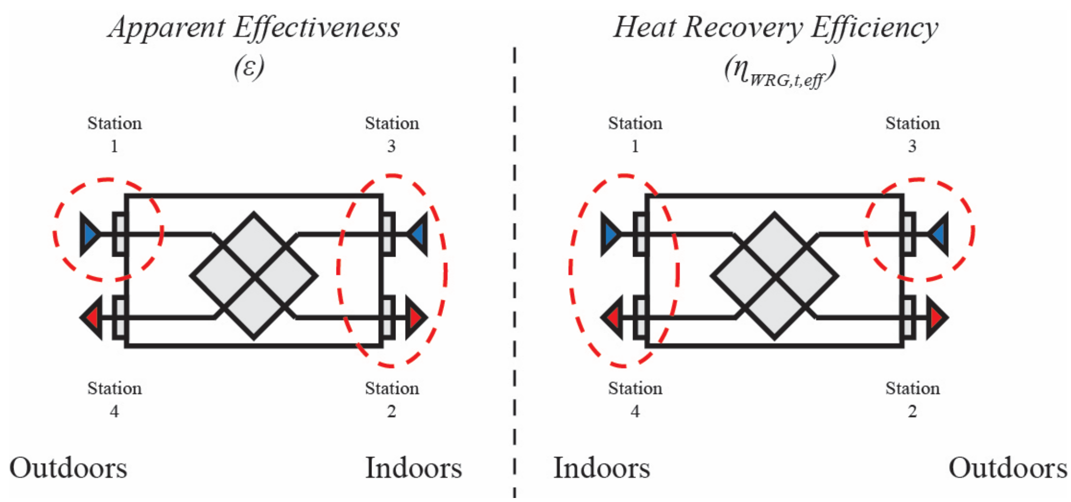

Figure 13. Different equations considering different temperatures

3.3 Impact of Motor Waste Heat on CSA/HVI Test Value

The most commonly used HRV/ERV model constructed in North American passive houses is the RecoupAerator 200DX ERV from UltimateAir Incorporated (PHIUS, 2015). The 200DX has three fan drives sending power to two fan rotors in each airstream and the moisture-transferring rotary wheel. All three fan drives are located in the exhaust airstream, upstream from the heat exchanger, see Figure 14 (UltimateAir, 2015).

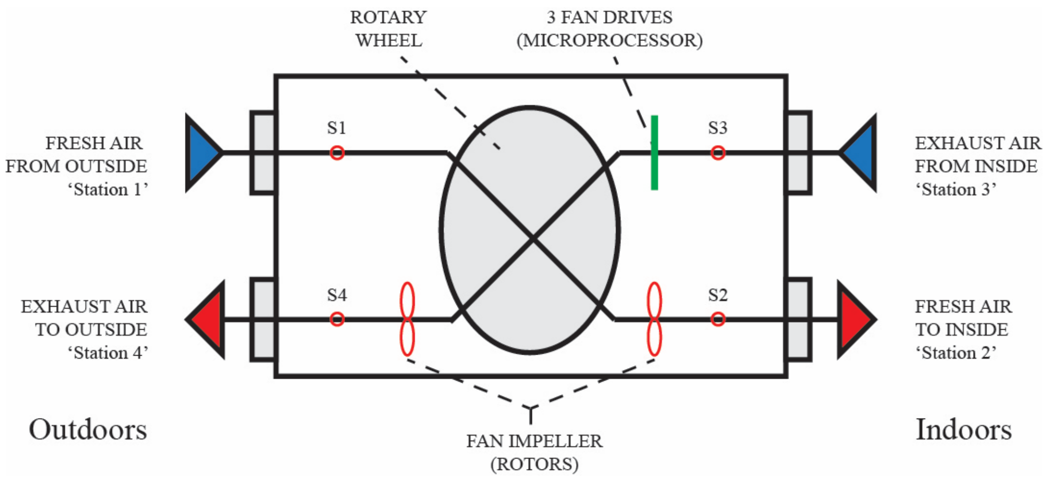

Figure 14. Schematic diagram of the RecoupAerator 200DX.

The fan drives use power to propel the impellers, generating waste heat in the process. This waste heat raises the temperature of the exhaust airstream, primarily increasing temperature t_2 at sensor S_2 as a result of heat recovery and secondarily increasing temperature at S_4. Since t_2 is directly proportional to ε (refer to Equation 1), this increases ε as well. Consider the following example. Jones (2001) suggests the waste heat raises the supply air temperature by 1°C. The effect on ε is shown in Tables 6 and 7.

Table 6. Actual winter test data for 200DX - Including waste heat from fan drives (Bodycote Materials Testing Incorporated, 2005)

Variables in Equation 1	(°C)
t_1 =	0
t_2 =	20.3
t_3 =	22
t_4 =	4.8
m_2 =	0.043 kg/s
m_3 =	0.042 kg/s
ε = 96.0%	

Note: ε was calculated using Equation 1

Table 7. Hypothetical winter test data for 200DX - Excluding waste heat from fan drives

Variables in Equation 1	(°C)
t_1 =	0
t_2 =	19.3 (-1°C)
t_3 =	22
t_4 =	4.8
m_2 =	0.043 kg/s
m_3 =	0.042 kg/s
ε = 91.3% (4.7% difference)	

Note: ε was calculated using Equation 1

Table 6 shows the measurements for the 200DX under CSA winter conditions from a testing laboratory (Bodycote Materials Testing Incorporated, 2005). Table 7 shows the hypothetical measurements under the assumption that no waste heat was generated, causing t_2 to be 1°C cooler. The difference of approximately 5% indicates the significant positive impact of fan motor waste heat on the final test result Ɛ. This testing advantage is accurate only for winter test condition. A testing disadvantage will occur for the summer conditions, see Tables 8 and 9.

Table 8. Actual summer test data for 200DX - Including waste heat from fan drives (Bodycote Materials Testing Incorporated, 2005)

Variables in Equation 1	(°C)
t_1 =	34.7
t_2 =	26.0
t_3 =	24.0
t_4 =	32.6
m_2 =	0.043 kg/s
m_3 =	0.041 kg/s
Ɛ = 84.4%	

Note: Ɛ was calculated using Equation 1

Table 9. Hypothetical summer test data for 200DX - Excluding waste heat from fan drives

Variables in Equation 1	(°C)
t_1 =	34.7
t_2 =	25.0 (-1°C)
t_3 =	24.0
t_4 =	32.6
m_2 =	0.043 kg/s
m_3 =	0.042 kg/s
Ɛ = 94.1% (9.7% difference)	

Note: Ɛ was calculated using Equation 1

Maintaining the assumption that the waste heat raises t_2 by 1°C, a significant negative impact on Ɛ is observed in the summer test. The system has to work harder to remove the extra heat added to the supply airstream. In this example, a unit with waste heat appears to operate approximately 10% less effectively. However, depending on the efficiency of specific fans used, the waste heat may raise t_2 to higher or lower temperatures (Jones, 2001; Navarro & Noyes, 2001). The intensified impact on Ɛ at a different temperature rise from 0.25°C to 1.5°C is illustrated in Figure 15.

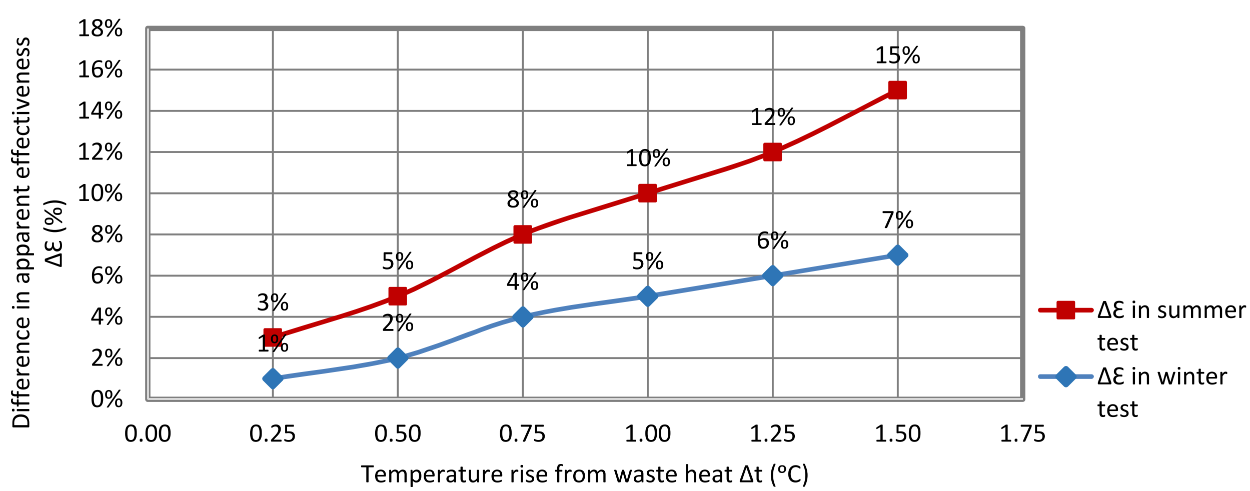

Figure 15. Relationship between temperature rise and apparent effectiveness

It is evident that as the temperature rise from waste heat increases, so does the difference between apparent effectiveness. The increase in apparent effectiveness in the summer test ranges from 3 to 15%, while in the winter test ranges from 1 to 7%. This contributes to the 12% penalty given by the PHI standard. Since North America is a predominantly cold-climate continent, the inflated winter test value makes it is easier to comply with the more challenging annual heating demand limit. Note that under the PHI test, $\eta_{HR,eff}$ is not affected by the fan motor waste heat in the 200DX since it does not account for t_2 in its equation.

In addition to the waste heat, there are other potential factors which may inflate ε for the winter test:

1. Exhaust air transfer ratio (R) – cross air leakage from the exhaust airstream to the supply airstream. The warmer exhaust air (~22^{o}C) may leak onto the cooler supply airstream (~20^{o}C), see Figure 16.
2. Casing air leakage and air infiltration - the warmer ambient air may leak into the cooler supply airstream, see Figure 17.
3. Heat transmission – the warmer ambient air may be transmitted through the casing material of the unit by conduction, see Figure 18.
4. Mass airflow imbalance during testing – if the mass flow rate of the exhaust airstream M_e is higher than the mass flow rate of the supply airstream M_s, then a pressure differential across the two airstreams will occur. This will be caused by the exhaust fans running at higher speeds. The higher pressure of the exhaust airstream will flow into the lower pressure of the supply airstream, see Figure 19.

Given all these factors, ε does not reflect the true energy recovery potential of the overall HRV/ERV system. Therefore, it should not be used for PHPP modelling.

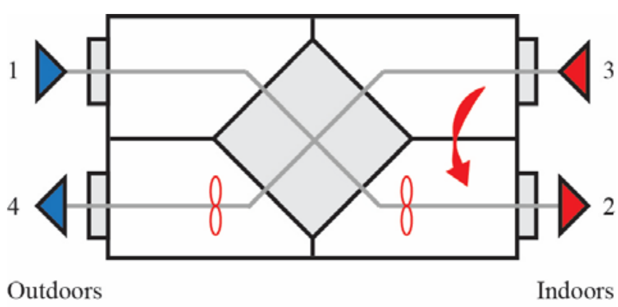

Figure 16. Exhaust air transfer

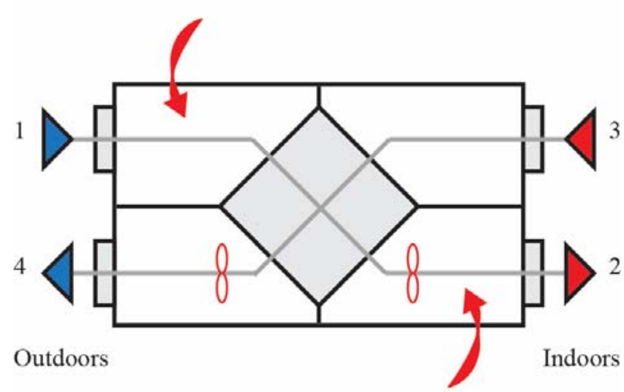

Figure 17. Casing air leakage and air infiltration

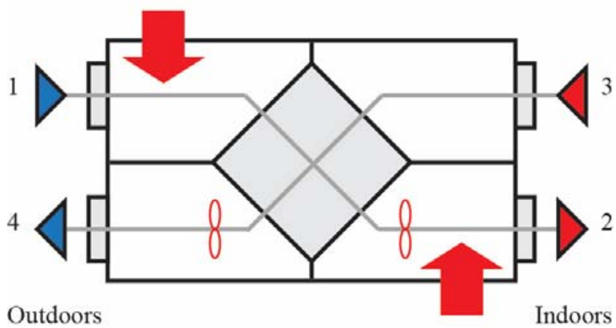

Figure 18. Case heat transmission

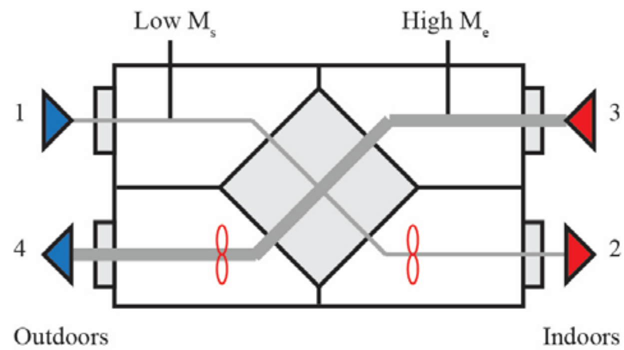

Figure 19. Mass airflow imbalance

4.0 PHIUS Proposal

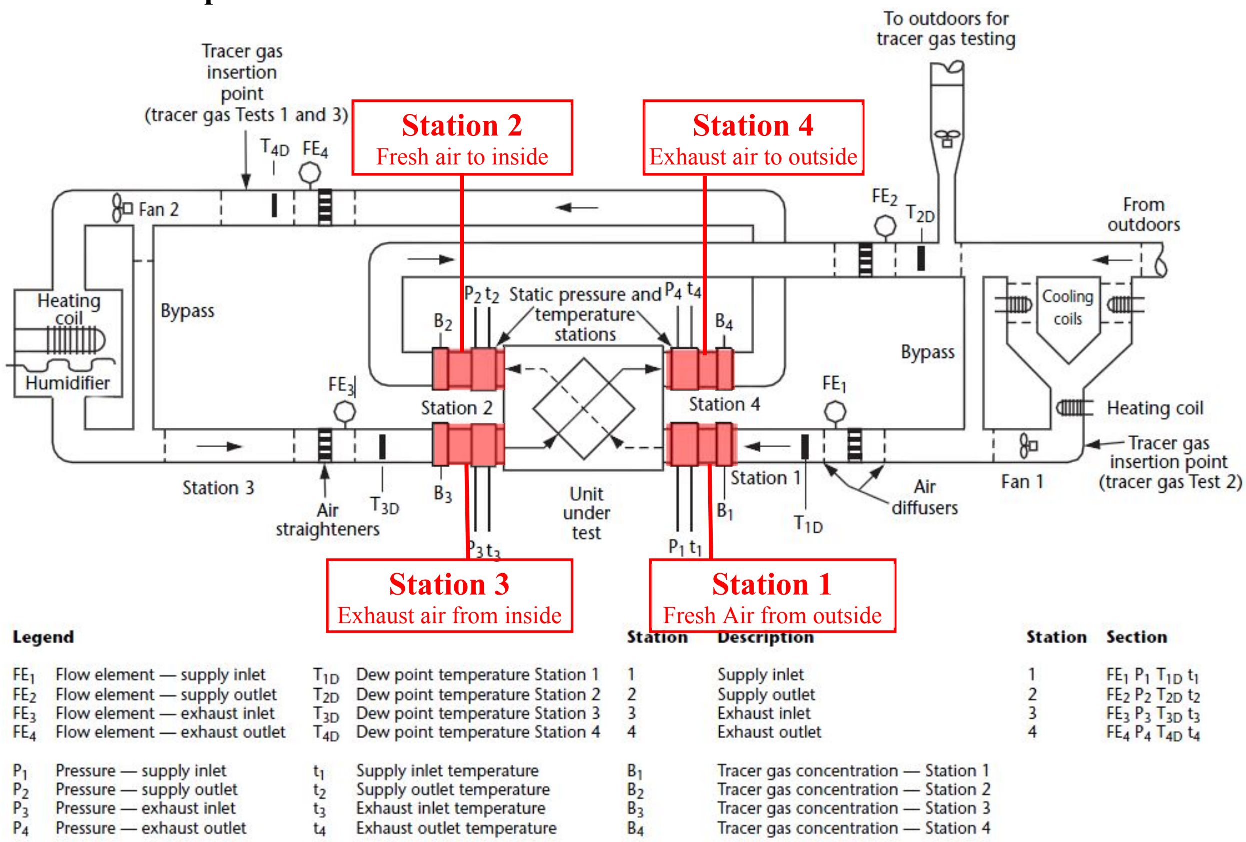

Figure 20. Schematic diagram of HRV/ERV laboratory testing setup (Reproduced from Canadian Standards Association, 2009)

To evaluate the true winter performance for a HRV/ERV system, the PHIUS Technical Committee proposes to use η_{PHUIS}, given by Equation 7. It is a modified version of CSA/ HVI's sensible heat recovery E_{SHR} from Equation 4 and it replaces Ɛ from Equation 1 to be used in PHPP simulations. This proposed calculation method serves to provide a more accurate representation of actual performance on site because it takes the stated efficiency of E_{SHR} provided by manufacturer and adds the total fan power measured on the field into the equation (Semmelhack, Personal Communication, 2015).

$$\eta_{PHIUS} = \frac{(\frac{E_{SHR}}{100} \times Maximum\ heat\ recovery) + Total\ fan\ power}{Maximum\ heat\ recovery}$$

$$\eta_{PHIUS} = \frac{(\frac{E_{SHR}}{100} \times ([M_s \times C_p \times (t_3 - t_1)] + P_{EF})) + P_T}{[M_s \times C_p \times (t_3 - t_1)] + P_{EF}} \quad (7)$$

Where:
η_{PHUIS} = sensible heat recovery efficiency proposed by PHIUS
M_s = mass flow rate of the supply air at Station 1, see Figure 4, kg/s
M_s = net mass flow rate of supply air
= M_2 x $(1-R)$
where
M_2 = mass flow rate of air measured at Station 2, see Figure 4, kg/s
R = exhaust air transfer ratio
C_p = specific heat of the air, kJ/kgK
t_1, t_3 = dry-bulb temperature at Station 1 and 3, respectively, see Figure 4, °C
P_{EF} = Electrical power of exhaust fan, W
P_T = Electrical power of both exhaust and supply fans, W
E_{SHR} = Sensible heat recovery efficiency from Equation 4

The rating of η_{PHUIS} is intended as a replacement test value for units tested under the CSA and HVI standards. It is to be used in PHPP energy modelling under winter conditions only. A replacement test value for summer conditions is being discussed, but has not been proposed thus far.

5.0 Methodology

Within the PHIUS project database, 55 certified North American passive houses using HRV/ERVs with either a PHI or CSA certificate had been accessed, as shown in Table 10. Approximately 56% or 31 of these houses are employed with the CSA-certified UltimateAir RecoupAerator 200DX. All 31 projects were selected for analysis. With the testing data provided by a third-party testing agency (Bodycote Materials Testing Incorporated, 2005), η_{PHIUS} was calculated using the proposed calculation method from Equation 7. Simulations were individually performed for each of the 31 existing PHPP files, with both ε less ~12% and with η_{PHIUS}. The results were compared to determine the impact of η_{PHIUS} on the modelled annual heat demand.

Table 10. Types of HRV/ERVs employed in certified North American Passive Houses

Performance Test	Unit Model	Number of units	Percentage of units
CSA	UltimateAir RecoupAerator 200DX	31	56.4%
PHI	Zehnder Comfo Air 200	6	10.9%
PHI	Zehnder Comfo Air 350	14	25.5%
PHI	Zehnder Comfo Air 550	4	7.3%
Total		**55**	**100.0%**

5.1 Simulation Results and Discussion

Based on the testing data for the UltimateAir RecoupAerator 200DX, the efficiency rating η_{PHIUS} of 89% was calculated using Equation 7, see Table 11. For the purposes of applying the new equation based on existing data, the exhaust fan power is assumed to be 50% of total measured fan power. The new η_{PHIUS} rating represents a 4 to 8% increase from $\eta_{HR,eff}$ = 81 to 85% (refer to column 2 in Table 12) previously used in existing PHPP files.

Table 11. η_{PHIUS} based on existing testing data

Data from third-party testing agency								**From Equation 6**
E_{SHR}	ε	M_s	C_p	$\Delta T = t_3 - t_1$	P_{EF}	P_T	$M_s \times C_p \times \Delta T + P_{EF}$	$\boldsymbol{\eta_{PHIUS}}$
Sensible Heat Recovery Efficiency (%)	Apparent Effectiveness (%)	Airflow (m^3/hr)	Heat Capacity of Air (Wh/m³K)	Temperature difference between Station 3 and 1 (°C)	Exhaust fan power (W)	Total Power (W)	Maximum heat recovery (W)	$\boldsymbol{\eta_{PHIUS}}$ **(%)**
83	96	108.36	0.33	22	25.5	51	811.58	**89**

The calculated η_{PHIUS} was used to perform 31 different PHPP simulations for houses in the US and Canada. A summary of the simulation results are presented in Table 12. Columns 1 to 7 represent values extracted from existing PHPP files, while columns 8 to 10 are values calculated in PHPP with η_{PHIUS} = 89%.

Table 12. PHPP simulation results

1	2	3	4	5	6	7	8	9	10
Project	**$\eta_{HR,eff}$ = ε – (~12) (%)**	**HS[1]**	**HDD[2]**	**RVV[3] (m^3)**	**TFA[4] (m2)**	**Q_H[5] (kWh/m^2a)**	**$Q_{H\text{-}PHIUS}$[6] (kWh/m^2a)**	**ΔQ_H = Q_H - $Q_{H\text{-}PHIUS}$ (kWh/m^2a)**	**Δq = $\Delta Q_H/Q_H$ (%)**
House A, UT	85	E	5700	645	258	14.60	13.99	0.61	4.2
House B, OR	83	HP	5000	364	146	12.87	12.05	0.82	6.4
House C, NC	83	HP	4600	479	192	13.41	12.81	0.60	4.5
House D, ME	83	HP	7875	266	106	12.00	10.74	1.26	10.5
House E, KS	83	HP	5500	401	160	13.00	12.15	0.85	6.6
House F, MA	83	HP	6000	314	126	14.20	13.00	1.20	8.5
House G, IL	83	HP	6685	254	102	14.98	13.19	1.80	12.0
House H, VA	84	HP	4000	398	159	11.10	10.57	0.54	4.8
House I, KY	83	HP	4434	254	102	14.98	14.23	0.76	5.1
House J, VA	83	x	4718	407	163	14.80	14.23	0.57	3.8
House K, OH	83	x	6143	406	162	14.10	13.06	1.04	7.4
House L, NC	83	GS HP	3761	920	368	11.17	10.85	0.32	2.8
House M, KY	83	HP	5248	225	90	14.98	14.16	0.82	5.5
House N, NS[7]	83	EB	7160	387	155	14.16	13.53	0.63	4.5
House O, PA	83	HP	5765	404	162	13.79	13.12	0.66	4.8
House P, NC	83	HP	3478	434	173	14.61	13.97	0.63	4.3
House Q, KY	83	HP	4432	254	102	14.67	13.88	0.79	5.4
House R, VA	83	HP	4739	581	232	14.32	13.53	0.79	5.5
House S, NY	83	E	7346	294	149	13.72	12.62	1.10	8.0
House T, NC	83	HP	2597	1171	469	12.97	12.56	0.41	3.2
House U, VA	83	HP	5130	316	126	14.83	13.94	0.88	6.0
House V, KY	83	HP	4434	254	102	14.10	13.34	0.76	5.4
House W, VA	83	HP	4685	570	228	9.50	8.90	0.60	6.3
House X, AZ	83	x	2511	508	203	11.01	10.54	0.47	4.3
House Y, WA	81	HP	6300	399	160	23.85	21.77	2.08	8.7
House Z, VA	83	x	4615	326	131	9.97	9.31	0.66	6.6
House A1, VA	83	HP	4257	316	126	14.35	13.63	0.73	5.1
House B1, NY	83	E	6273	295	118	10.79	9.78	1.01	9.4

House C1, NY	83	E	6273	294	118	13.22	12.11	1.10	8.4
House D1, NY	83	E	6273	295	118	10.85	9.87	0.98	9.0
House E1, NY	83	E	6273	294	118	11.42	10.41	1.01	8.8

[1]Heating system, where: E = electric resistance, HP = air-to-air heat pumps, GSHP = ground source heat pump, EB = electrical baseboards, and x = unknown. [2]Heating degree-days. [3]Room ventilation volume. [4]Treated floor area. [5]Annual heat demand calculating with $\eta_{HR,eff}$. [6]Annual heat demand calculating with η_{PHIUS}. [7]Province of Nova Scotia in Canada, all others are States located in the U.S.

The values in column 7 (Q_H) are existing modelled annual heat demands used to determine whether the particular house satisfies PHI's space heating requirement of 15 kWh/m^{2}a. The values in column 8 ($Q_{H-PHIUS}$) are new modelled annual heat demands based on η_{PHIUS} = 89%. The difference between the two is represented by column 9 (ΔQ_H). The sensitivity of energy savings to the calculation method is represented by column 10 (Δq). Δq denotes the margin of error between the existing and newly modelled annual heat demands based on the two methods of calculating heat recovery efficiency, and is given following equation:

$$\Delta q = \frac{\Delta Q_H - Q_{H-PHIUS}}{Q_H} \quad (8)$$

A higher Δq indicates a larger margin of error, representing the level of inaccuracy for the existing Q_H. Figure 21 illustrates the frequency distribution of Δq throughout all of the 31 projects examined. It shows that by η_{PHIUS}, the margin of error improved by 4 to 8.9% for the majority (77%) of the houses. Considering the HRV/ERV is only a component of the house which contributes to the annual heat demand, the results here demonstrate a very significant impact caused by the new equation.

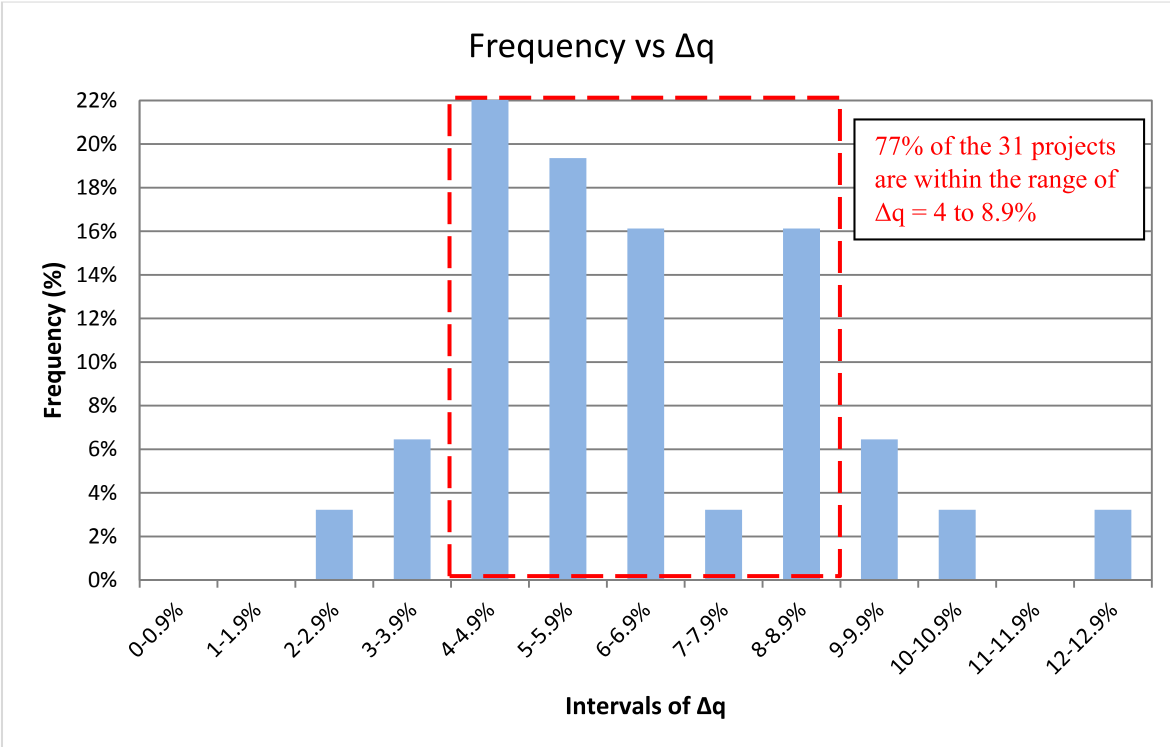

Figure 21. Frequency distribution of Δq

The impact of modelling with η_{PHIUS} yielded varying output results, where Δq is within the overall range of 13%. The greatest reduction in annual heat demand was 12% for House G as shown in Table 12. The smallest reduction was 2.8% for House L. The substantial range shows that the proposed method affects each passive house project differently, dependent on different factors. The most important factor is the regional climate where the house is located. As shown in Figure 22, when houses are built in colder climates with higher heating degree-days (HDD), the margin of error becomes more significant. This margin is a function of climate, and their relationship is given by the equation: $y = 0.00001x + 0.0012$. The reason for the margin increase in relation to HDD can be explained by the prolonged heating season in colder climate areas. A longer heating season results in the HRV/ERV operating and recovering heat or energy for longer periods of time. Hence, the percentage of error will multiply over the longer timespan. This indicates the importance for determining a more accurate method for places with high heat demand, which validates the necessity for PHIUS' proposed equation in the North American climate.

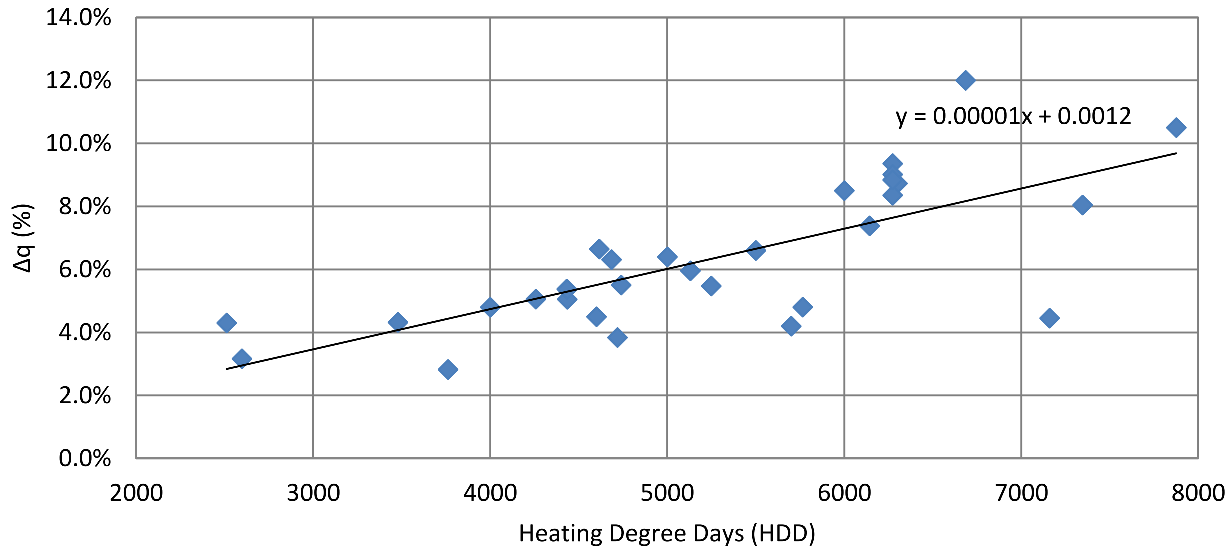

Figure 22. Relationship between heating degree-days and Δq

Another important factor which affects Δq is the size of the house. The results were plotted against the floor area of each corresponding house as shown in Figure 23. For small to medium-sized houses (<250m^2), there is a significant impact and difference in the margin of error is approximately 8%, caused by the proposed calculation method. For large houses (>250m^2), the impact is insignificant, characterized only by a difference of approximately 1%. It should be noted that there are only three large houses in the data sample. With such a small sample size for large houses, the data is not representative and definitive conclusions cannot be drawn. Additionally, several outliers were eliminated from the data sample for one of the following reasons: the house was constructed temporarily for research purposes instead of occupancy, the 12% penalty was not applied to the HRV/ERV as part of the whole house annual heat demand calculation, or the house had both an uncharacteristically large floor area (>300m^2) and constructed in a city with very cold regional climate conditions (>7000 HDD).

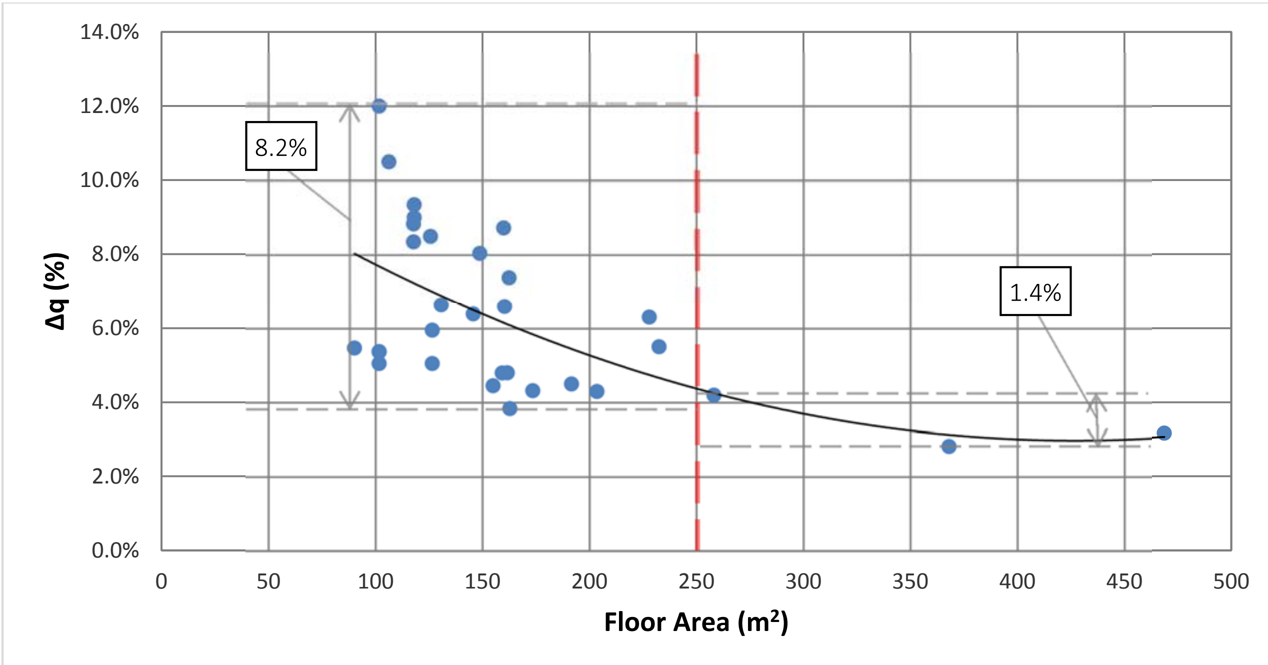

Figure 23. Relationship between total floor area and Δq

As the house gets bigger, the efficiency of the heat exchanger becomes less important. This can be explained by the example below.

EXAMPLE 1.0

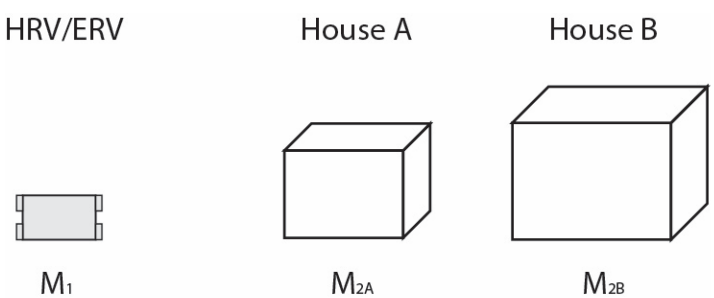

Figure 24. Example showing efficiency of heat exchanger for different size houses

Consider the same HRV/ERV unit used in two scenarios, for House A and House B as shown in Figure 24. House A has a smaller floor area and requires a mass flow rate of supply air at Station 2, M_{2A}. House B has a larger floor area and requires a mass flow rate of supply air at Station 2, M_{2B}. The HRV/ERV unit has a mass flow rate of outdoor air at Station 1, M_1. The efficiency of the unit is given by Equations 9 and 10:

$$\eta_A = \frac{M_1}{M_{2A}} \tag{9}$$

$$\eta_B = \frac{M_1}{M_{2B}} \tag{10}$$

Due to the larger size of House B, the mass flow rate of supply air at House B is larger than at House A, $M_{2B} > M_{2A}$. As a result, the efficiency of the HRV/ERV at House B is less than that in House A, $\eta_B < \eta_A$. In other words, as the house gets bigger, the efficiency of the heat exchanger decreases and becomes less important. The typical North American passive house is constructed with a small to medium floor area (<250m^2), therefore the proposed calculation method is relevant and affects the accuracy of modelling the majority of certified houses.

6.0 Conclusion

The analyses in this project identified the major differences in testing procedures and requirements between the relevant performance testing standards for HRV/ERVs currently employed in single-family North American passive houses. It is evident that the European certification program by PHI upholds a higher level of excellence relative to that of the North American CSA and HVI standards. However, PHI's calculation method for heat recovery efficiency and CSA test value penalty does not accurately reflect the true winter performance of North American units. An alternate method and equation, η_{PHIUS}, which accounts for fan power was proposed by PHIUS in order to reconcile the penalty and provide more accuracy to determining on-the-field HRV/ERV performance. It combines both the stated efficiency of E_{SHR} provided by manufacturer and adds the fan power measured on the field, increasing the accuracy of energy modelling. The consequence of this proposal greatly affected the modelled annual heat demands for the 31 houses examined, resulting in an improved margin of error in the range of 2.8% to 12%. The margin of error between the existing and newly modelled annual heat demand caused by the adjustment in HRV/ERV heat recovery efficiency rating was large enough to justify the new equation. This margin was closely correlated to the regional climate and the total floor area of the house. As houses are constructed in regions with higher number of heating degree-days, the margin increased. In addition, the largest difference in margin of error occurred for small to medium-sized houses ($<250m^2$) at approximately 8%. These observations further validates the need for a more accurate way to determine HRV/ERV heat recovery efficiency in the winter. This is because, by and large, North America as a whole is colder than Europe and the average North American passive house have small to medium-sized floor areas.

7.0 Future Work

This project has evaluated the impact of the proposed method for calculating HRV/ERV heat recovery efficiency in North American passive houses during the heating season only. It would be insightful to analyze an additional method for calculating efficiency in the summer, and its impact on annual cooling demand. In the future as more passive houses are being constructed in North America, the data sample can be expanded to include more homes which have larger floor areas ($>250m^2$).

Furthermore, all North American units certified to the CSA/HVI standards to date are the same model. Other models may have different configurations, including different placements for fan drives. This means the motor waste heat will raise the temperatures at different outlets of the airstream. Also, there are residential HRV units already utilizing dual heat exchanger cores designed for very large homes (Lifebreath, 2014). Further studies may be required to analyze such units with different component configurations.

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9.0 Appendix A – PHIUS Certified Projects for Single Family Passive Houses

$\eta_{HR,eff}$	HRV/ERV Model	Heating System	Annual Heating Demand (kWh/m²a) (monthly or annually)	Heat Load (W/m²)	HDD	Room Ventilation Volume (m³)	Location	Treated Floor Area (m²)
85%	Zehnder Comfo Air 200	Electric Radiant Matts	13.79	6.62	2568	242.47	San Francisco, CA	n/a
81%	Zehnder Comfo Air 350	Electric	13.47	11.67	4577	272.92	Portland, OR	n/a
84%	Ultimate Air RecoupAerator 200DX	Electric	14.60	12.20	5700	645.00	Salt Lake City, UT	258
82%	Ultimate Air RecoupAerator 200DX	Heat Pump	12.87	9.15	5000	363.84	Salem, OR	146
82%	Ultimate Air RecoupAerator 200DX	MiniSplit Heat Pump - 1 per floor	13.41	15.99	4600	478.75	Chapel Hill, NC	192
82%	Ultimate Air RecoupAerator 200DX	Daiken Air Source Heat Pump with Electric Backup	12.00	21.60	7875	266.00	Belfast, ME	106
77%	Ultimate Air RecoupAerator 200DX	Daiken Mini Split.	13.00	15.46	5500	400.62	Kansas City, KS	160
80%	Ultimate Air RecoupAerator 200DX	In duct micro compressor heat pump system	12.65	23.97	6000	117.57	Champaig n, IL	47
79%	Zehnder Comfo Air 350	Heat Pump with Electric Backup	13.31	13.88	7400	345.26	Charolette , VT	n/a
90%	Zehnder Comfo Air 200	Space Heat with Electric Backup	12.00	13.70	6900	364.00	Hudson, NY	n/a
83%	Ultimate Air RecoupAerator 200DX	Electric	13.09	11.99	7000	739.35	Shrewsbur y, MA	296
80%	Ultimate Air RecoupAerator 200DX	Air Source Heat Pump with Electric Backup	14.20	11.90	6000	314.00	Falmouth, MA	126
87%	Ultimate Air RecoupAerator 200DX	1000W (3,400 Btu/h) electric resistance element post heater	15.70	15.70	6359	512.00	Urbana, IL	210
80%	LUEFTA LS 300 DC-K	Nuheat Electric Radiant Heating Mats	10.60	17.40	8000	411.00	Hudson, WI	n/a
80%	Ultimate Air RecoupAerator 200DX	Mitsubishi MSZ-FE09 HyperHeat Inverter-driven heat pump	14.98	12.62	6685	254.00	Chicago, IL	102

81%	Ultimate Air RecoupAerator 200DX	Mitsubishi multi-split heat pump with two wall-mounted indoor units	11.10	15.46	4000	397.85	Charlottesville, VA	159
86%	Comfo Air 500 StorkAir	Hot water from DHW boiler runs to heat exchanger in ductwork. Back-up heat available from heat pumps..Mitsubishi City Multi minisplit heat pump.	14.38	14.83	2944	853.09	Bethesda, MD	n/a
81%	Zehnder Comfo Air 350	Primary: Hydronic System (1st Floor only) fed by Instantaneous Condensing Gas Boiler (Navien) fueled by propane. Secondary: Heat Pump / Air Conditioning via Ducted Mini-Splits (Mitsubishi).	12.50	11.80	3698	1054.00	Onancock, VA	n/a
81%	Zehnder Comfo Air 350	Fujitsu air to air heat pump	14.23	10.73	6109	630.27	Salt Lake City, UT	n/a
75%	Ultimate Air RecoupAerator 200DX	9000 BTU Ductless Fujitsu mini split heat pump, and 12,000 BTU ducted Mitsubishi mini split heat pump	14.98	13.25	4434	254.00	Williamsburg, KY	102
82%	Zehnder Comfo Air 550	Daikin Altherma air-water heat pump	13.12	11.99	5165	757.10	Newberg, OR	n/a
78%	Ultimate Air RecoupAerator 200DX	n/a	14.80	11.99	4718	406.71	Ironto, VA	163
86%	Zehnder Comfo Air 350	n/a	14.48	7.26	2301	689.20	Menlo Park, CA	n/a
75%	Zehnder Comfo Air 350	Mitsubishi Mini-split, one outdoor unit, (3) indoor units.	12.93	17.03	4832	755.40	Stuart, VA	n/a
78%	Comfo Air 500 StorkAir	Heat Pump - Climate Master mod# TMW036AGC00C0CS Radiant Tank:	6.78	1.26	8156	778.08	Stowe, VT	n/a

		Vaughn 80 gal, mod# S-80						
90%	Zehnder Comfo Air 200	Mitsubishi ASHP MSZ-FE12NA Hyper heat MUZ-FE12NA 1500W Electric Resistance Baseboard Back-up	9.81	15.46	7345	264.76	Knox, ME	n/a
78%	Zehnder Comfo Air 350	Lennox Seer 19 Heat Pump w/first stage only connected; approx. 16kBTU	11.51	11.04	2472	644.54	San Jose, CA	n/a
81%	Ultimate Air RecoupAerator 200DX	n/a	14.10	11.99	6143	405.78	Yellow Springs, OH	162
79%	Ultimate Air RecoupAerator 200DX	Climate Master GSHP 3 ton to 2.2 variable	11.17	10.73	3761	919.67	0	368
81%	Ultimate Air RecoupAerator 200DX	Fujitsu 9,000 Btu ductless mini split heat pump	14.98	16.09	5248	225.34	Berea, KY	90
69%	Venmar EKO 1.5	Hydronic radiant floors, 500 square feet, Daikin Altherma combined heat source	12.43	9.15	2584	490.70	Palo Alto, CA	n/a
82%	Ultimate Air RecoupAerator 200DX	Electric baseboard with Electrical Thermal Storage Unit in living area.	14.16	12.62	7160	387.14	n/a	155
79%	Ultimate Air RecoupAerator 200DX	Mitsubishi mini-split (12,000 BTU) in line with ventilation air.	13.79	11.04	5765	403.77	Heidelberg, PA	162
74%	Ultimate Air RecoupAerator 200DX	Mini split	14.61	16.72	3478	433.56	Chapel Hill, NC	173
80%	Ultimate Air RecoupAerator 200DX	9000 BTU Ductless Fujitsu mini split heat pump, and 12,000 BTU ducted Mitsubishi mini split heat pump	14.67	12.93	4432	254.00	Williamsburg, KY	102
91%	Zehnder Comfo Air 200	mini-split heat pump	13.69	15.14	4580	259.89	Dundee, OR	n/a
83%	Zehnder Comfo Air 350	Morso 3142 woodstove. 7 - 475 watt Envi resistance heaters.	14.95	11.99	7485	449.61	Newry, ME	n/a
87%	Zehnder Comfo Air 200	Convectair Electric	13.19	10.41	4577	390.03	Portland, OR	n/a

81%	Zehnder Comfo Air 350	9000Btu Fujitsu Halcyon 9RLS Mini Split w/ Heat Pump	10.66	14.83	647	342.04	Austin, TX	n/a
81%	Ultimate Air RecoupAerator 200DX	2 Fujitsu Ducted mini split systems 18,000 btu for first & second floor and 9,000 but for basement. Systems provide heating, cooling, dehumidification & fan only modes.	14.32	17.35	4739	581.00	Falls Church VA	232
81%	Ultimate Air RecoupAerator 200DX	Electric Resistance	13.72	14.20	7346	294.18	Ithaca, NY	149
83%	Ultimate Air RecoupAerator 200DX	American Standard 19 SEER 2-ton heat pump	12.97	9.46	2597	1171.43	Wilmington, NC	469
79%	Ultimate Air RecoupAerator 200DX	Ducted Fujitsu Minisplit - see enclosed mechanical drawings and specs	14.83	18.93	5130	315.96	Abingdon, VA	126
83%	Zehnder Comfo Air 350	n/a	12.62	n/a	7293	626.93	Norwich, VT	n/a
82%	Zehnder Comfo Air 550	Ducted heat pump: Mitsubishi MXZ2B20NA with SEZKD09NA4, HSPF=8.5, SEER=15.5	14.90	8.90	2796	732.00	Sonoma, CA	n/a
82%	Zehnder Comfo Air 550	Daikin Altherma coupled with Messana ray magic radiant ceiling and wall panels	18.04	7.89	2301	1240.12	Palo Alto, CA	n/a
82%	Zehnder Comfo Air 350	n/a	14.26	12.30	6336	681.41	Cleveland, OH	n/a
80%	Ultimate Air RecoupAerator 200DX	9000 BTU Ductless Fujitsu mini split heat pump, and 12,000 BTU ducted Mitsubishi mini split heat pump	14.10	12.93	4434	254.00	Williamsburg, KY	102
80%	Life Breath 195 Max	Fujitsu 9RLS ductless mini-split and electric baseboards	15.30	9.78	7666	632.65	Waubaushene, ON	n/a
80%	Ultimate Air RecoupAerator 200DX	Mitsubishi multi-split heat pump - 2 wall mount units, 1 ducted unit	9.50	12.62	4685	569.67	Charlottesville FAA, VA	228

74%	Zehnder Comfo Air 350	First Co. Hydronic Fan Coil	17.80	8.80	6292	623.00	Adams County, CO	n/a
81%	Ultimate Air RecoupAerator 200DX	Daiken Point Source	11.01	8.52	2511	508.45	Hereford, AZ	203
76%	Zehnder Comfo Air 350	Hydronic coil in line w/ ventilation ducts; electric mats in bathrooms	16.59	8.20	2376	331.42	Carmel-by-the-Sea, CA	n/a
90%	Zehnder Comfo Air 200	12,000 BTU Mitsubishi heat pump system provides conditioned air to the main living area. An 8" duct system and in-line Panasonic fan route conditioned air to the rest of the house.	14.38	12.93	6011	321.22	Danbury Municipal CT	n/a
76%	Ultimate Air RecoupAerator 200DX	Mitsubishi ductless mini-split	23.85	10.09	6300	399.35	Elk, WA	160
81%	Zehnder Comfo Air 350	In-line duct heater (Electro EM-WX0212R)	13.38	7.57	4413	608.49	Portland, OR	n/a
74%	Ultimate Air RecoupAerator 200DX	n/a	9.97	11.92	4615	326.38	Thaxton, VA	131
77%	Zehnder Comfo Air 550	2.5 kW Electric Resistance Inline Duct Heat in Ventilation Air Stream	16.70	7.00	2984	770.00	San Francisco, CA	n/a
79%	Ultimate Air RecoupAerator 200DX	Ducted Fujitsu Minisplit	14.35	13.94	4257	315.96	South Boston, VA	126
81%	Ultimate Air RecoupAerator 200DX	electric resistance	10.79	13.22	6273	294.86	Ithaca, NY	118
81%	Ultimate Air RecoupAerator 200DX	electric resistance	13.22	14.01	6273	294.18	Ithaca, NY	118
81%	Ultimate Air RecoupAerator 200DX	electric resistance	10.85	13.22	6273	294.95	Ithaca, NY	118
81%	Ultimate Air RecoupAerator 200DX	electric resistance	11.42	13.94	6273	294.18	Ithaca, NY	118