ANNEX 5

EBC
Energy in Building and Communities Programme

AIVC Technical Note 75

A design guideline of energy recovery ventilation to improve actual contribution to energy saving in buildings

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Preface

The International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an international energy programme. A basic aim of the IEA is to foster international cooperation among the 30 IEA participating countries and to increase energy security through energy research, development and demonstration in the fields of technologies for energy efficiency and renewable energy sources.

The IEA Energy in Buildings and Communities Programme

The IEA co-ordinates international energy research and development (R&D) activities through a comprehensive portfolio of Technology Collaboration Programmes (TCPs). The mission of the IEA Energy in Buildings and Communities (IEA EBC) TCP is to support the acceleration of the transformation of the built environment towards more energy efficient and sustainable buildings and communities, by the development and dissemination of knowledge, technologies and processes and other solutions through international collaborative research and open innovation. (Until 2013, the IEA EBC Programme was known as the IEA Energy Conservation in Buildings and Community Systems Programme, ECBCS.).

The high priority research themes in the EBC Strategic Plan 2019-2024 are based on research drivers, national programmes within the EBC participating countries, the Future Buildings Forum (FBF) Think Tank Workshop held in Singapore in October 2017 and a Strategy Planning Workshop held at the EBC Executive Committee Meeting in November 2017. The research themes represent a collective input of the Executive Committee members and Operating Agents to exploit technological and other opportunities to save energy in the buildings sector, and to remove technical obstacles to market penetration of new energy technologies, systems and processes. Future EBC collaborative research and innovation work should have its focus on these themes.

At the Strategy Planning Workshop in 2017, some 40 research themes were developed. From those 40 themes, 10 themes of special high priority have been extracted, taking into consideration a score that was given to each theme at the workshop. The 10 high priority themes can be separated in two types namely 'Objectives' and 'Means'. These two groups are distinguished for a better understanding of the different themes.

Objectives: The strategic objectives of the EBC TCP are as follows:

– reinforcing the technical and economic basis for refurbishment of existing buildings, including financing, engagement of stakeholders and promotion of co-benefits;
– improvement of planning, construction and management processes to reduce the performance gap between design stage assessments and real-world operation;
– the creation of 'low tech', robust and affordable technologies;
– the further development of energy efficient cooling in hot and humid, or dry climates, avoiding mechanical cooling if possible; the creation of holistic solution sets for district level systems taking into account energy grids, overall performance, business models, engagement of stakeholders, and transport energy system implications.

Means: The strategic objectives of the EBC TCP will be achieved by the means listed below:

– the creation of tools for supporting design and construction through to operations and maintenance, including building energy standards and life cycle analysis (LCA);
– benefitting from 'living labs' to provide experience of and overcome barriers to adoption of energy efficiency measures;
– improving smart control of building services technical installations, including occupant and operator interfaces;
– addressing data issues in buildings, including non-intrusive and secure data collection;
– the development of building information modelling (BIM) as a game changer, from design and construction through to operations and maintenance.

The themes in both groups can be the subject for new Annexes, but what distinguishes them is that the 'objectives' themes are final goals or solutions (or part of) for an energy efficient built environment, while the 'means' themes are instruments or enablers to reach such a goal. These themes are explained in more detail in the EBC Strategic Plan 2019-2024.

The Executive Committee

Overall control of the IEA EBC Programme is maintained by an Executive Committee, which not only monitors existing projects, but also identifies new strategic areas in which collaborative efforts may be beneficial. As the Programme is based on a contract with the IEA, the projects are legally established as Annexes to the IEA EBC Implementing Agreement. At the present time, the following projects have been initiated by the IEA EBC Executive Committee, with completed projects identified by (*) and joint projects with the IEA Solar Heating and Cooling Technology Collaboration Programme by (☼):

Annex 1: Load Energy Determination of Buildings (*)
Annex 2: Ekistics and Advanced Community Energy Systems (*)
Annex 3: Energy Conservation in Residential Buildings (*)
Annex 4: Glasgow Commercial Building Monitoring (*)
Annex 5: Air Infiltration and Ventilation Centre
Annex 6: Energy Systems and Design of Communities (*)
Annex 7: Local Government Energy Planning (*)
Annex 8: Inhabitants Behaviour with Regard to Ventilation (*)
Annex 9: Minimum Ventilation Rates (*)
Annex 10: Building HVAC System Simulation (*)
Annex 11: Energy Auditing (*)
Annex 12: Windows and Fenestration (*)
Annex 13: Energy Management in Hospitals (*)
Annex 14: Condensation and Energy (*)
Annex 15: Energy Efficiency in Schools (*)
Annex 16: BEMS 1- User Interfaces and System Integration (*)
Annex 17: BEMS 2- Evaluation and Emulation Techniques (*)
Annex 18: Demand Controlled Ventilation Systems (*)
Annex 19: Low Slope Roof Systems (*)
Annex 20: Air Flow Patterns within Buildings (*)
Annex 21: Thermal Modelling (*)
Annex 22: Energy Efficient Communities (*)
Annex 23: Multi Zone Air Flow Modelling (COMIS) (*)
Annex 24: Heat, Air and Moisture Transfer in Envelopes (*)
Annex 25: Real time HVAC Simulation (*)
Annex 26: Energy Efficient Ventilation of Large Enclosures (*)
Annex 27: Evaluation and Demonstration of Domestic Ventilation Systems (*)
Annex 28: Low Energy Cooling Systems (*)
Annex 29: ☼ Daylight in Buildings (*)
Annex 30: Bringing Simulation to Application (*)
Annex 31: Energy-Related Environmental Impact of Buildings (*)
Annex 32: Integral Building Envelope Performance Assessment (*)
Annex 33: Advanced Local Energy Planning (*)
Annex 34: Computer-Aided Evaluation of HVAC System Performance (*)
Annex 35: Design of Energy Efficient Hybrid Ventilation (HYBVENT) (*)
Annex 36: Retrofitting of Educational Buildings (*)
Annex 37: Low Exergy Systems for Heating and Cooling of Buildings (LowEx) (*)
Annex 38: ☼ Solar Sustainable Housing (*)
Annex 39: High Performance Insulation Systems (*)
Annex 40: Building Commissioning to Improve Energy Performance (*)
Annex 41: Whole Building Heat, Air and Moisture Response (MOIST-ENG) (*)
Annex 42: The Simulation of Building-Integrated Fuel Cell and Other Cogeneration Systems (FC+COGEN-SIM) (*)
Annex 43: ☼ Testing and Validation of Building Energy Simulation Tools (*)
Annex 44: Integrating Environmentally Responsive Elements in Buildings (*)
Annex 45: Energy Efficient Electric Lighting for Buildings (*)
Annex 46: Holistic Assessment Tool-kit on Energy Efficient Retrofit Measures for Government Buildings (EnERGo) (*)
Annex 47: Cost-Effective Commissioning for Existing and Low Energy Buildings (*)
Annex 48: Heat Pumping and Reversible Air Conditioning (*)
Annex 49: Low Exergy Systems for High Performance Buildings and Communities (*)
Annex 50: Prefabricated Systems for Low Energy Renovation of Residential Buildings (*)
Annex 51: Energy Efficient Communities (*)
Annex 52: ☼ Towards Net Zero Energy Solar Buildings (*)

Annex 53: Total Energy Use in Buildings: Analysis and Evaluation Methods (*)
Annex 54: Integration of Micro-Generation and Related Energy Technologies in Buildings (*)
Annex 55: Reliability of Energy Efficient Building Retrofitting - Probability Assessment of Performance and Cost (RAP-RETRO) (*)
Annex 56: Cost Effective Energy and CO_2 Emissions Optimization in Building Renovation (*)
Annex 57: Evaluation of Embodied Energy and CO_2 Equivalent Emissions for Building Construction (*)
Annex 58: Reliable Building Energy Performance Characterisation Based on Full Scale Dynamic Measurements (*)
Annex 59: High Temperature Cooling and Low Temperature Heating in Buildings (*)
Annex 60: New Generation Computational Tools for Building and Community Energy Systems (*)
Annex 61: Business and Technical Concepts for Deep Energy Retrofit of Public Buildings (*)
Annex 62: Ventilative Cooling (*)
Annex 63: Implementation of Energy Strategies in Communities (*)
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Annex 65: Long-Term Performance of Super-Insulating Materials in Building Components and Systems (*)
Annex 66: Definition and Simulation of Occupant Behavior in Buildings (*)
Annex 67: Energy Flexible Buildings (*)
Annex 68: Indoor Air Quality Design and Control in Low Energy Residential Buildings (*)
Annex 69: Strategy and Practice of Adaptive Thermal Comfort in Low Energy Buildings (*)
Annex 70: Energy Epidemiology: Analysis of Real Building Energy Use at Scale (*)
Annex 71: Building Energy Performance Assessment Based on In-situ Measurements (*)
Annex 72: Assessing Life Cycle Related Environmental Impacts Caused by Buildings (*)
Annex 73: Towards Net Zero Energy Resilient Public Communities (*)
Annex 74: Competition and Living Lab Platform (*)
Annex 75: Cost-effective Building Renovation at District Level Combining Energy Efficiency and Renewables (*)
Annex 76: ☼ Deep Renovation of Historic Buildings Towards Lowest Possible Energy Demand and CO_2 Emissions (*)
Annex 77: ☼ Integrated Solutions for Daylight and Electric Lighting (*)
Annex 78: Supplementing Ventilation with Gas-phase Air Cleaning, Implementation and Energy Implications (*)
Annex 79: Occupant-Centric Building Design and Operation (*)
Annex 80: Resilient Cooling (*)
Annex 81: Data-Driven Smart Buildings (*)
Annex 82: Energy Flexible Buildings Towards Resilient Low Carbon Energy Systems (*)
Annex 83: Positive Energy Districts
Annex 84: Demand Management of Buildings in Thermal Networks (*)
Annex 85: Indirect Evaporative Cooling
Annex 86: Energy Efficient Indoor Air Quality Management in Residential Buildings (*)
Annex 87: Energy and Indoor Environmental Quality Performance of Personalised Environmental Control Systems
Annex 88: Evaluation and Demonstration of Actual Energy Efficiency of Heat Pump Systems in Buildings
Annex 89: Ways to Implement Net-zero Whole Life Carbon Buildings
Annex 90: EBC Annex 90 / SHC Task 70 Low Carbon, High Comfort Integrated Lighting
Annex 91: Open BIM for Energy Efficient Buildings
Annex 92: Smart Materials for Energy-Efficient Heating, Cooling and IAQ Control in Residential Buildings
Annex 93: Energy Resilience of the Buildings in Remote Cold Regions
Annex 94: Validation and Verification of In-situ Building Energy Performance Measurement Techniques
Annex 95: Human-centric Building Design and Operation for a Changing Climate
Annex 96: Grid Integrated Control of Buildings
Annex 97: Sustainable Cooling in Cities
Annex 98: Flexibilization and Optimization of Heat Pump Systems in Existing Buildings through Secondary-Side Digitalization
Annex 99: Air Cleaning for Sustainable and Resilient Buildings

Working Group – Energy Efficiency in Educational Buildings (*)
Working Group – Indicators of Energy Efficiency in Cold Climate Buildings (*)
Working Group – Annex 36 Extension: The Energy Concept Adviser (*)
Working Group – HVAC Energy Calculation Methodologies for Non-residential Buildings (*)
Working Group – Cities and Communities (*)
Working Group – Building Energy Codes

IEA EBC Annex 5: Air Infiltration and Ventilation Centre

EBC Annex 5 was first established in 1979 under the name “Air Infiltration Centre” undertaking technical activities and providing information services with the task of minimizing air infiltration energy losses. In 1986, the name was changed to “Air Infiltration and Ventilation Centre” to reflect the importance of the coupling of a good airtightness with appropriate ventilation. Over time, the AIVC has been continuously evolving to respond to emerging concerns, challenges and opportunities. We have now entered the 46th year of the AIVC’s existence and the Centre’s main goal is to provide reference information on ventilation & air infiltration in the built environment with respect to efficient energy use and good Indoor Environmental Quality (IEQ).

In November 2025, the Executive Committee approved the continuation of the AIVC for the period 2027-2031. Peter Wouters and Arnold Janssens are on behalf of INIVE the operating agents for this period.

The AIVC holds a conference each year in September/October in one of the AIVC participating countries. More information can be found here: www.aivc.org/events/conferences

The AIVC organizes 1 to 2 workshops per year. More information can be found here: www.aivc.org/events/workshops

The AIVC organizes a number of webinars per year. More information can be found here: www.aivc.org/events/webinars

The AIVC has formal collaborations with the TightVent platform (https://tightvent.eu/), the venticool platform (https://venticool.eu/) and the IEQ-GA (https://ieq-ga.net/).

Moreover, there is a close interaction with several ventilation related annexes of IEA-EBC.

If you want to be kept informed on the activities of AIVC and related platforms, you can subscribe here.

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Executive Summary

In this design guideline, key points are explained to succeed in reducing energy use for HVAC systems with energy recovery systems. It covers the reduction of the heating and cooling need (load) due to ventilation and infiltration as well as the reduction of energy use for fans and heat generators. As background of the key points, performance indices and testing standards of energy recovery systems are explained.

To depict the key point in a quantitative way, a building energy calculation program integrated with algorithms for energy recovery systems is used for case studies. At a maximum, 18% and 13% reduction of total energy use for HVAC systems have been shown for central heat generator and rotary wheel energy recovery units and for variable refrigerant flow systems and fixed-plate energy recovery ventilators, respectively.

The following are the key points:

1. It is necessary to maintain the air balance between total supply airflow and total exhaust airflow for each floor or section in buildings in order that the mechanical ventilation does not induce infiltration or exfiltration through the building envelope. The air leakage may deteriorate energy saving intended by using energy recovery systems.
2. To maintain the air balance, adjustments of mechanical airflows and/or replacements of exhaust-only ventilation with balanced ventilation are necessary in the design and installation. Those adjustments and replacements should be made taking specific fan power (SFP) of ventilation equipment into consideration.
3. When energy recovery systems are used, the airflow ratio (the ratio of supply airflow to exhaust airflow) should be equal to or closer to 1.0, so that efficiency of the systems is maintained higher.
4. When choosing energy recovery systems, net efficiency should be paid attention rather than just gross efficiency. Unit exhaust air transfer ratio (UEATR) included in the performance data of energy recovery systems should be used to calculate net efficiency.
5. The energy recovery system is one of the methods to reduce heating and cooling need (load). To reduce the energy use of heat generators, it is indispensable to carry out the sizing of the heat generators according to a standardized sizing protocol. Without sizing, the part load ratio of the heat generators may fall in the range, in which no reduction of energy use can be expected.

1. Scope and audiences of this design guideline

This design guideline aims to share the latest technical information (Kan et al., 2025) for fully utilizing the potential of energy recovery ventilation systems so that heating and cooling need (load)[A.15] (see A.15 in Appendix A-Terminology) and energy use for HVAC systems can be reduced as much as possible.

The target readers are HVAC designers, builders, developers and engineers of manufacturers of components relevant to the energy recovery ventilation systems.

This design guideline deals mainly with ducted energy recovery ventilators and energy recovery units integrated with air-handling units (Figure 1).

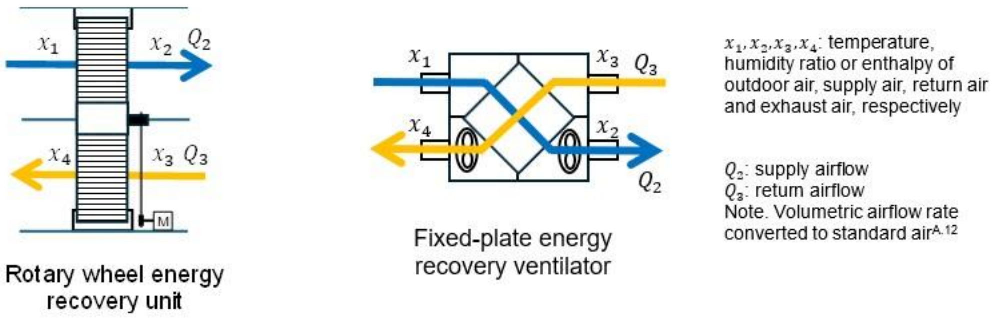


Figure 1: Target types of energy recovery ventilation systems

Figure 2 shows an example of building floor plan with energy recovery ventilation systems, other ventilation equipment and a space heating and cooling system.

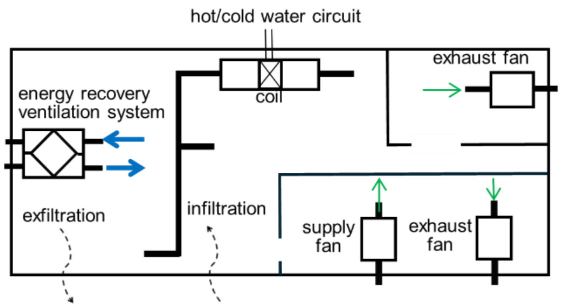


Figure 2: An example of building floor plan installed the HVAC system with an energy recovery ventilator and other mechanical ventilation equipment

There is already a guidebook for residential heat recovery ventilation, where more detailed and overall knowledge is provided on ventilation system sizing, noise calculation, installation, ductwork, commissioning, balancing and maintenance (Kurnitski et al., 2018). Readers are recommended to refer to the guidebook.

The main purpose of the energy recovery ventilation systems is to reduce space heating and cooling need (load) due to outdoor air intake by exchanging heat and moisture between return air and outdoor air. The reduced need is expected to result in the reduction of energy use of heat generators (e.g., heat pumps, boilers) and heat supply systems (e.g., pumps, fans). Actual efficiency of energy recovery ventilation systems such as total efficiency[A.28] is not always equal to the catalogue value, but it depends on designed and actual airflow rates and airflow ratio[A.1] (see blue arrows in Figure 2), as well as on the rate of airflow of other ventilation equipment (e.g., exhaust ventilation in lavatories and balance ventilation in machine room) on the same floor (green arrows in Figure 2). The airflow rates of other ventilation equipment and energy recovery ventilation systems also influence air infiltration and/or exfiltration (broken arrows in Figure 2), which may increase heating and cooling need.

There is another factor influencing energy need and use. It is the energy use of fans, which increases total energy use of HVAC systems and influences cooling and heating need. The heat emission by fans can be added to cooling need or deducted from heating need dealt with such as by air-handling units, which are provided with heat such as by secondary water circuits. The energy efficiency of fans and ventilation systems is represented by the specific fan power, SFP (Equation 1). The SFP can be reduced by using energy efficient fans and motors and by designing low-pressure-loss duct systems.

$$SFP = \frac{P}{Q} \qquad (1)$$

where

SFP: specific fan power (kW/(m^3/s))

P: power consumption of fan (kW)

Q: airflow rate of ventilation system (m^3/s)

2. Terminology

The key technical terms are explained in Appendix A-Terminology in alphabetical order. They are referred to with superscripts (e.g., ventilator[A.31]).

3. Information on performance indices for energy recovery systems

3.1. Energy recovery elements

This design guideline focuses on two types of energy recovery elements applied to building HVAC systems. Thay are 1) rotary wheel energy recovery unit (rotary type) and 2) fixed-plate energy recovery ventilator (fixed-plate type) as shown in Figure 3.

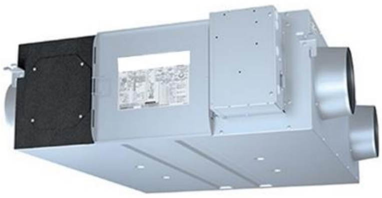
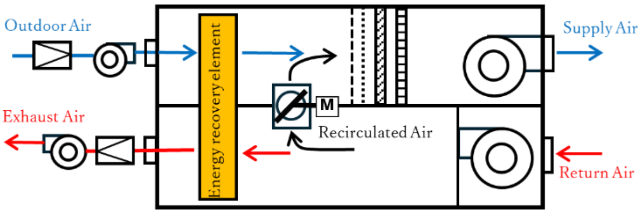


Figure 3: Rotary wheel energy recovery unit (left) and fixed-plate energy recovery ventilator (right)

The energy recovery elements can also be grouped into a) total heat[A.29] recovery element and b) sensible heat[A.24] recovery element. The former group element can recover total heat (i.e., sensible and latent heat), and the latter group element can recover only sensible heat, from return air to supply air. The latter group element is sometimes called *heat* recovery elements, while the former group element is called *energy* recovery element. When it is less necessary to recover latent heat such as in regions with cold and humid winter or with rather dry summer, the *heat* recovery elements are sometimes used. In general, odors are less permeable through the heat recovery element than the energy recovery element (Kusama et al., 2014, and the former type of recovery element is more useful when the permeability of odors should be minimized such as in lavatories. Hereafter in this document, *heat* recovery shall be included in *energy* recovery, but if it is indispensable to deal with them separately, they shall be distinguished in explanations.

3.2. Typical energy recovery ventilation systems dealt with in this design guideline

Two typical ducted energy recovery ventilation systems are dealt with in this document. One system is the combination of air-handling unit and energy recovery unit (Figure 4, left). Another system is the energy recovery ventilator, which is supplied as a unit composed of energy recovery element(s), fans, filters, airflow paths, a drain and a case (Figure 4, right). Hereafter, these two types are collectively called energy recovery systems.



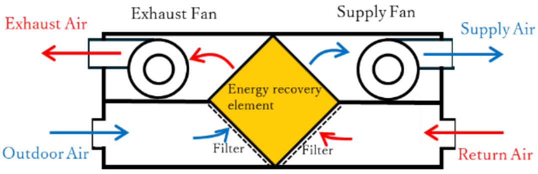


Figure 4: Two typical ducted energy recovery ventilation systems. Air handling unit with energy recovery unit (left) & Energy recovery ventilator (right)

3.3. Tested indices and influencing factors on actual efficiency of energy recovery ventilation systems

By testing and rating standards for energy recovery systems, the following key indices representing their characteristics are prescribed. For e), f) and g), 'net' means indices adjusted by taking air leakage into account in contrast with 'gross' without the adjustment.

a) Gross effectiveness (for temperature, humidity ratio and enthalpy)
b) Gross efficiency (for temperature, humidity ratio and enthalpy)
c) Unit exhaust air transfer ratio (UEATR)
d) Supply and return airflows
e) Net supply airflow ratio
f) Net effectiveness (for temperature, humidity ratio and enthalpy)
g) Net efficiency (for temperature, humidity ratio and enthalpy)
h) Airflow characteristics (relationship between airflow rate and static pressure difference in case of energy recovery ventilators and pressure loss across the energy recovery element in case of rotary wheel energy recovery units)

Examples of standards are listed in Appendix B - Relevant testing and rating standards for energy recovery systems.

3.3.1. Difference between effectiveness and efficiency

In the testing and rating standards for energy recovery systems, gross effectiveness is defined as a ratio of the energy transfer rate (sensible, latent or total) to the maximum possible energy transfer rate, which is equal to the product of the minimum energy capacity rate of two airflows and the maximum difference in temperature, humidity ratio, or enthalpy. For example, gross total effectiveness is defined by Equation 2. The locations of measurement stations 1, 2, 3 and 4 are shown in Figure 5.

$$\varepsilon_{total} = \frac{\dot{m}_2(h_1 - h_2)}{\dot{m}_{min}(h_1 - h_3)} \qquad (2)$$

where

ε_{total}: gross total effectiveness

$\dot{m}_2$: the mass flow rate (kg/s) at station 2 (see Figure 5)

$\dot{m}_{min}$: the lesser of $\dot{m}_2$ or $\dot{m}_3$ (the mass flow rate at stations 2 and 3)

h_1, h_2, h_3: the enthalpy (kJ/kg/K) at stations 1, 2, 3, respectively

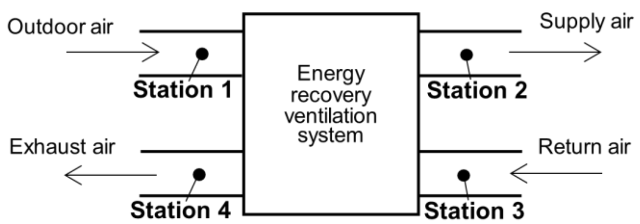


Figure 5: Locations of station 1 to 4 for measuring dry-bulb temperature, humidity ratio and enthalpy

On the contrary, gross efficiency (sensible, latent or total) is defined such as by Equation 3 for the gross total efficiency, for example. The efficiency is not related to the mass flow rates.

$$\eta_{total} = \frac{h_1 - h_2}{h_1 - h_3} \qquad (3)$$

In ISO 16494-1:2022 (ISO 16494-1, 2022) and Eurovent rules (Eurovent, 2025), it is required to measure the effectiveness under the equal mass flow rates at stations 2 and 3. In those standards, the effectiveness is equal to the efficiency.

3.3.2. Gross effectiveness in ISO 16494-1:2022

In ISO 16494-1:2022, gross effectiveness is calculated by using measured dry-bulb temperatures (for sensible effectiveness), humidity ratios (for latent effectiveness) or enthalpies (for total effectiveness) at station 1, station 2 and station 3 (Figure 5) as shown in Equation 4.

$$\varepsilon = \frac{x_1 - x_2}{x_1 - x_3} \qquad (4)$$

where

ε : gross sensible effectiveness, gross latent effectiveness, or gross total effectiveness

x_1: the dry-bulb temperature (for gross sensible effectiveness), °C; the humidity ratio (for gross latent effectiveness), kg water/kg dry air; or the enthalpy (for gross total effectiveness), J/kg, at station 1 (see Figure 5)

x_2: the dry-bulb temperature (for gross sensible effectiveness), °C; the humidity ratio (for gross latent effectiveness), kg water/kg dry air; or the enthalpy (for gross total effectiveness), J/kg, at station 2 (see Figure 5)

x_3: the dry-bulb temperature (for gross sensible effectiveness), °C; the humidity ratio (for gross latent effectiveness), kg water/kg dry air; or the enthalpy (for gross total effectiveness), kJ/kg, at station 3 (see Figure 5)

3.3.3. Unit exhaust air transfer ratio (UEATR) and net supply airflow ratio (NSAR)

For any products of energy recovery system, there are leakages from return air to supply air and from the space surrounding the products to the space inside the products. The leakages occur also during the test for gross effectiveness. The larger the amount of air leakage from return air to supply air and from the space surrounding the products to the space inside the product, the higher the measured gross effectiveness becomes.

In the test for UEATR representing the percentage of leaking-in air contained in supply air, according to ISO 16494-1, outdoor air (see Figure 5) is kept at a higher concentration of tracer gas (CO_2), while return air and surrounding space is kept at the same concentration as the atmosphere (e.g., in case of ducted set up defined in ISO 16494-1). Due to the leakages, the tracer gas concentration of outdoor air is diluted by air infiltration from return air and the space surrounding the product, and supply air concentration becomes lower than that of outdoor air. By using measured tracer gas concentrations at stations 1, 2 and 3, UEATR can be calculated by Equation 5. As shown in Figure 6, when the concentration at station 2 is closer to the one at station 1, UEATR is closer to 0%, while the concentration at station 2 is lower due to the dilution by the mixture of outdoor air and return air and leaking-in air from the space surrounding the product, UEATR becomes higher such as 10% or 15%.

$$UEATR = \frac{C_2 - C_1}{C_3 - C_1} \times 100 \qquad (5)$$

where

C_1: tracer gas concentration at station 1 of outdoor air (see Figure 6)

C_2: tracer gas concentration at station 2 of supply air (see Figure 6)

C_3: tracer gas concentration at station 3 of return air (see Figure 6)

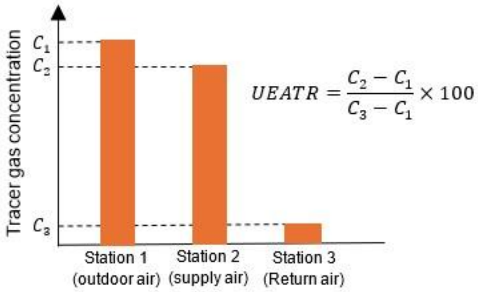


Figure 6: Relationship between UEATR and tracer gas concentrations at stations 1, 2 and 3 (exemplified measurement results in case of ducted set up of ISO 16494-1)

Net supply airflow (m^3/s) can be quantified as supply airflow minus transferred airflow from return and the space surrounding the energy recovery system (m^3/s). Net supply airflow ratio and net supply airflow are calculated by Equations 6 and 7, respectively.

$$NSAR = 100 - UEATR \qquad (6)$$

$$Q_{2,net} = \frac{NSAR}{100} \times Q_2 \qquad (7)$$

where

$NSAR$: net supply airflow ratio (%)

$Q_{2,net}$: net supply airflow (m^3/s)

Q_2: supply airflow expressed in m^3/s of standard air (dry air with a density of 1.204 kg/m^3)

3.3.4. Net effectiveness and net efficiency

When the gross effectiveness or efficiency is measured, the larger UEATR is, the larger the gross effectiveness or efficiency is. It is because the thermal condition of the air at station 2 (x_2) becomes closer to that of the air at station 3 (x_3) and the air of the surrounding space due to more mixture of them into supply air.

That is the reason why the measurement of UAETR must be carried out under the same conditions for the measurement of gross effectiveness and efficiency. Net effectiveness by taking UAETR into consideration and the way of its derivation are provided such as in ASHRAE Standard 84-2018 (ANSI/ASHRAE, 2018) and AHRI 1060/1061-2018 (AHRI, 2018) as in Equations 8 to 11. Equation 8 means that supply air condition x_2 is a weighted average of the net condition of air after the energy recovery element x_{net} and the condition of transferred return air x_3.

$$x_2 = (1 - UAETR/100) \cdot x_{net} + UAETR/100 \cdot x_3 \qquad (8)$$

where

x_2: the dry-bulb temperature (for gross sensible effectiveness), °C; or the humidity ratio (for gross latent effectiveness), kg water/kg dry air; or the enthalpy (for gross total effectiveness), kJ/kg, at station 2

x_{net}: the net dry-bulb temperature, °C; or the net humidity ratio, kg water/kg dry air; or the net enthalpy, kJ/kg, at station 2, if there is no leakage from return air and the space surrounding the energy recovery element

x_3: the dry-bulb temperature (for gross sensible effectiveness), °C; or the humidity ratio (for gross latent effectiveness), kg water/kg dry air; or the enthalpy (for gross total effectiveness), kJ/kg, at station 3

By using x_{net} defined by Equation 8, net effectiveness ε_{net} is calculated by Equation 9.

$$\varepsilon_{net} = \frac{x_1 - x_{net}}{x_1 - x_3} \qquad (9)$$

Using Equations 4, 8 and 9, the following relationship among ε, ε_{net} and UAETR is obtained.

$$\varepsilon_{net} = \varepsilon - \frac{UAETR}{100 - UAETR} \cdot (1 - \varepsilon) \qquad (10)$$

In the same way, net efficiency is calculated by Equation 11.

$$\eta_{net} = \eta - \frac{UAETR}{100 - UAETR} \cdot (1 - \eta) \qquad (11)$$

where

η_{net}: net efficiency

η: gross efficiency

3.3.5. Supply and return airflows when gross effectiveness or gross efficiency is measured

Supply airflow Q_2 and return airflow Q_3 largely influence measured gross effectiveness and efficiency. Therefore, testing standards require records of supply and return airflows during tests of gross effectiveness and efficiency.

ISO 16494-1 requires that supply airflow and return airflow volumes at standard air[A.26] conditions equal within 0.006 m^3/s plus 1 % of the measured airflow volume. It is because the condition of equal airflow volumes is the most usual condition when energy effectiveness of energy recovery systems is labelled and presented in technical documents (e.g., brochures).

AHRI Standard 1060/1061 requires that supply and return mass airflows shall be equal to the specified mass airflow within ±1.5% or 0.002 L/s of standard air, whichever greater. This tolerance should be accepted because of difficulty in controlling the airflow rates completely equal.

The Eurovent rule for air-to-air regenerative heat exchangers also requires that supply airflow and return airflow are the same volume flow at standard air conditions.

On the other hand, ISO 21773:2021 (ISO 21773, 2021) mainly for rotary wheel energy recovery units requires that supply and return mass airflow or airflow volumes when the effectiveness is measured shall be included in test data without requirement for equal airflow volumes. It is because manufacturers are requested by users to provide effectiveness under uneven airflow volumes for their design conditions.

3.3.6. Airflow characteristics of energy recovery ventilators

Energy recovery ventilators contain fans as shown in Figure 4 (right). Based on designed supply and return airflow requirements, designers select a product of energy recovery ventilator. Thus, test requirements for airflow characteristics (i.e., a relationship between airflow and external static pressure difference) are also prescribed in standards, such as ISO 16494-1.

An important advancement of ISO 16494-1, which was originally developed in 2014, is to prescribe static pressure conditions for stations 1, 2, 3 and 4, which are applied to all tests (for airflow characteristics, effectiveness and UEATR). The requirement for static pressure conditions is shown in Table 1. ISO 16494-1 includes two test methods, ‘Ducted setup’ and ‘Two-rooms setup’ and testers can choose one from the two setups. By either setup, static pressure conditions for all four stations shall be controlled symmetrically for two airflows (supply and exhaust) when tests are carried out. Examples of static pressures at four stations are included in Table 1. This requirement has improved the reliability of test results for energy recovery ventilators.

Table 1: Requirement for static pressure conditions at Station 1, 2, 3 and 4 (ISO 16494-1)

<table>
<tr><th>Setup
(test methods)</th><th>Test</th><th>Static pressure conditions at station 1, 2, 3 and 4 (see Figure 5)
$(P_{si}, i = 1,2,3,4, in\ Pa)$</th></tr>
<tr><td rowspan="3">Ducted setup</td><td>Airflow characteristics</td><td rowspan="3">For the maximum and minimum rated airflows
$|(|P_{s1}| - |P_{s2}|)| \leq max(10Pa, max(|P_{s1}|, |P_{s2}|) \times 5\%)$
$|(|P_{s3}| - |P_{s4}|)| \leq max(10Pa, max(|P_{s3}|, |P_{s4}|) \times 5\%)$
For each intermediate test point
$max(|P_{s1}|, |P_{s2}|, |P_{s3}|, |P_{s4}|) - min(|P_{s1}|, |P_{s2}|, |P_{s3}|, |P_{s4}|) \leq max(10Pa, max(|P_{s1}|, |P_{s2}|, |P_{s3}|, |P_{s4}|) \times 5\%)$
Example: $P_{s1} = -150Pa, P_{s2} = 150Pa, P_{s3} = -150Pa, P_{s4} = 150Pa$</td></tr>
<tr><td>UEATR</td></tr>
<tr><td>Effectiveness</td></tr>
<tr><td rowspan="3">Two-rooms setup</td><td>Airflow characteristics</td><td rowspan="3">$P_{s1} < 0, \quad P_{s3} < 0$
$|(|P_{s1}| - |P_{s3}|)| \leq max(10Pa, max(|P_{s1}|, |P_{s3}|) \times 5\%)$
$|(|P_{s2}| - |P_{s4}|)| \leq max(10Pa, max(|P_{s2}|, |P_{s4}|) \times 5\%)$
Example: $P_{s1} = P_{s3} = -5Pa, P_{s2} = P_{s4} = 300Pa$</td></tr>
<tr><td>UEATR</td></tr>
<tr><td>Effectiveness</td></tr>
</table>

For rotary wheel energy recovery elements, pressure drop across the elements from station 1 to station 2 and station 3 to station 4 shall be measured according to ISO 21773. The test results are used when sizing fans taking the pressure loss across energy recovery elements into consideration.

The static pressure difference between station 2 and station 3 shall also be recorded, because it influences air leakage from return air to supply air. According to Eurovent rules, the static pressure difference shall be kept very small, namely between 0 Pa and 20 Pa to limit the air leakage from return airflow to supply airflow and the test result of UAETR.

3.4. Allowance of deviation of test results

Some testing and rating standards prescribe allowance of deviation of test results of a product from published ratings of the product.

For example, AHRI 1060/1061 prescribes that test results for sensible effectiveness shall not be lower than the published rating by more than: (1) the sum of four relative percentage points and one absolute percentage point, or (2) two absolute percentage points below the published rating, whichever is greater. Eurovent rules have the acceptance criteria of -3 absolute percentage points for gross temperature efficiency and -5 absolute percentage points for gross humidity efficiency. JIS B 8628:2017 (JIS B 8628, 2017) prescribes that test results of efficiency and net supply airflow ratio shall not be lower than 90% and 100% of the published ratings, respectively.

The allowance should be considered when actual energy performance of energy recovery systems is estimated by building engineers and HVAC designers.

4. Key points to increase energy saving by energy recovery systems

Based on information for products of energy recovery systems, more systematic consideration for energy recovery ventilation systems should be taken hereafter.

In this chapter, the following key points are discussed:

1. Overview of relevant factors on energy saving by energy recovery ventilation systems
2. Keep balance between outdoor air supply and air exhaust to outdoor for each floor or section of buildings
3. Extract air for energy recovery from well heated and/or airconditioned space
4. Exhaust air through energy recovery elements as much as possible
5. Choose energy recovery systems with higher net effectiveness and net efficiency
6. Choose energy recovery ventilators and units with larger capacity of energy exchange
7. Choose fans with smaller specific fan power (SFP) and duct systems with smaller pressure loss
8. Demand control ventilation
9. Cleaning of energy recovery elements and filters
10. Implement the sizing of heat generators.

4.1. Overview of relevant factors on energy saving by energy recovery ventilation systems

Figure 7 depicts a cluster of spaces, such as a floor or a section, which is partitioned from other parts of the building. The cluster of spaces consists of a main space (Space A) for longer occupancies and two supplementary spaces (Space B and Space C).

Office rooms, meeting rooms, shop spaces, classrooms, dining rooms of restaurants, living rooms, etc. are examples of Space A.

Space B represents spaces, which tend to be underpressurized such as by exhaust-only mechanical ventilation to avoid transfer of pollutants and odor to Space A. Examples are lavatories, kitchenettes, garbage storage rooms, bathrooms, etc.

Space C represents spaces, where occupancy is rare, and it is not necessary to provide space heating nor cooling for occupants, but ventilation is indispensable. Examples are machine rooms, electric rooms, storage rooms, etc.

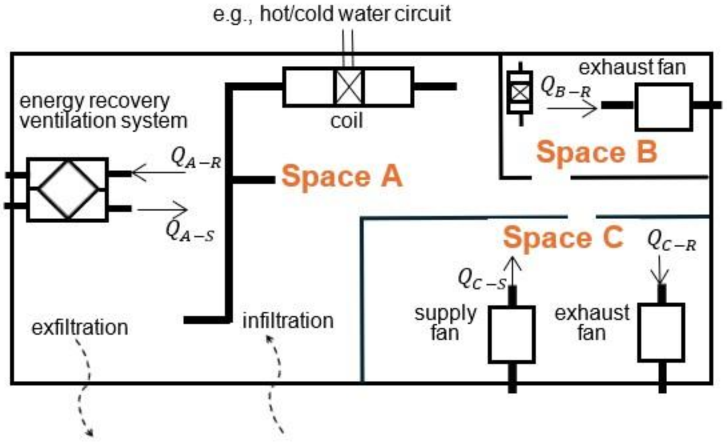


Figure 7: An example of a floor or a section of buildings, where spaces with different needs of space heating and cooling and ventilation are contained

The objective of energy recovery ventilation is to reduce heating and cooling need (load) due to outdoor air supply for ventilation needs, as already mentioned.

The balance of outdoor air and exhaust air for this floor or section is expressed by Equation 12.

$$\Delta Q = (Q_{A-S} + Q_{C-S}) - (Q_{A-R} + Q_{B-R} + Q_{C-R}) \quad (12)$$

where

ΔQ: balance of outdoor air supply and air exhaust driven by mechanical ventilation (kg/s)

$Q_{A-S}, Q_{A-R}, Q_{B-R}, Q_{C-S}$ and Q_{C-R}: Supply airflows or return airflows (kg/s) for Space A, Space B and Space C.

If $\Delta Q = 0$ (complete balance condition), mechanical ventilation does not drive any infiltration nor exfiltration. In that case, infiltration or exfiltration can be induced only by buoyancy and/or wind pressure.

To achieve balance, the first way is to reduce return airflow of the energy recovery system Q_{A-R} to make the following relationship among Q_{A-S}, Q_{A-R} and Q_{B-R}, which are included in Equation 12.

$$Q_{A-S} = Q_{A-R} + Q_{B-R} \quad (13)$$

When reducing Q_{A-R}, if it is possible to lower the inverter frequency of the fan motor, energy use for the fan can be decreased. When this way is taken, airflow ratio Q_{A-S}/Q_{A-R} increases and efficiency of the energy recovery system decreases. An example of test result for an energy recovery ventilator is shown in Figure 8 (Kan et al., 2025).

If the reduction of Q_{A-R} is done by increasing airflow resistance such as by a manual damper, it can make almost no decrease of the fan energy use. Therefore, this way with a manual damper should be avoided.

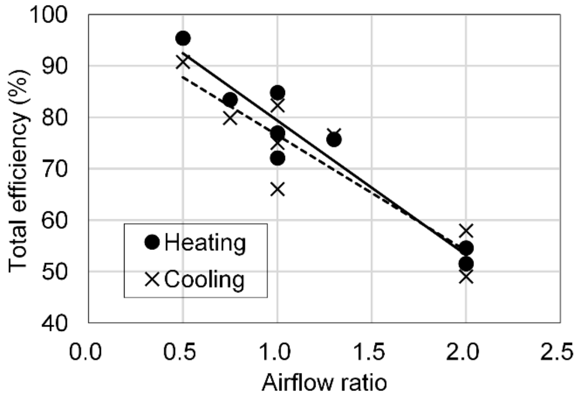


Figure 8: An example of the relationship between airflow ratio (supply airflow rate to exhaust airflow rate) and total efficiency of an energy recovery ventilator (Kan et al., 2025)

The second way to achieve the balance is to exhaust air from the energy recovery system to indoors at the location near Space B as shown in Figure 9. To depressurize Space B to avoid transfer of pollutants and odor to outside Space B, Q_{B-R} should be designed a little bit more than Q_{A-E}. There is usually no ventilation requirement for corridor (hall) in front of Space B, but if there is any requirement (e.g., corridor includes any place like a resting area for longer occupancy), that requirement can be added to the ventilation requirement of Space A.

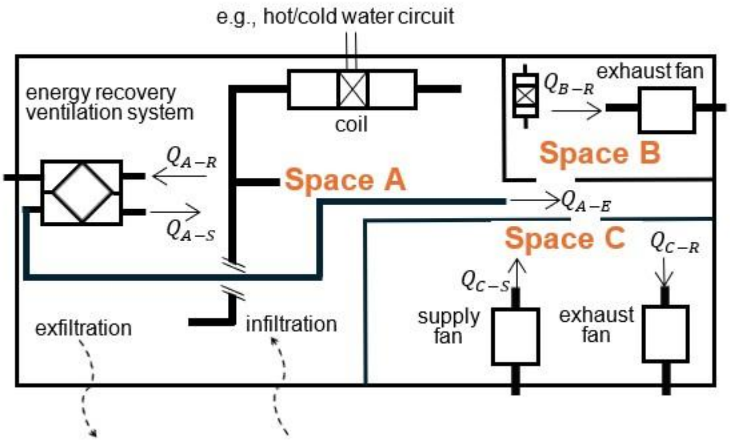


Figure 9: An example of a floor or a section of buildings with air exhaust of energy recovery system located near Space B, which is ventilated by exhaust-only ventilation

The third way to achieve the balance is to replace the exhaust-only ventilation of Space B with a balanced ventilation. It can be an energy or heat recovery ventilation system. To minimize the return of odor or contaminated air through energy recovery elements, heat recovery elements with very low permeability of vapor and gases could be recommended. Since energy recovery systems tend to have higher SFP due to plural fans and pressure loss by energy recovery elements, careful selection of products with higher energy efficiency is needed.

The heating or cooling need (load) due to outdoor air supply can be calculated by Equation 14, assuming setpoints of temperature and humidity ratio of all spaces are equivalent for simplicity.

$$L = \begin{cases} \{Q_{A-S}(1-\eta_{tot}) + Q_{C-S}\} \times \Delta h & \Delta Q \geq 0 \\ \{Q_{A-S}(1-\eta_{tot}) + Q_{C-S} - \Delta Q\} \times \Delta h & \Delta Q < 0 \ \text{(infiltration occurs)} \end{cases} \qquad (14)$$

where

L: space heating and cooling need (load) due to outdoor air supply (kW)

η_{tot}: net total efficiency of energy recovery under Q_{A-S} and Q_{A-R}(-)

Δh: enthalpy difference between outdoors and indoors (kJ/kg)

If Q_{A-S} and Q_{C-S} are fixed to comply with ventilation requirement for Space A and Space C, respectively, there are the following two options to minimize L of Equation 14.

Option 1: To let ΔQ be closer to zero (balance condition) so that heat loss due to infiltration can be minimized. The three ways to achieve the balance have already been described.

Option 2: To let η_{tot} be larger by keeping the balance between Q_{A-S} and Q_{A-R}, and by choosing an energy recovery system with higher net total efficiency. By Equation 11 in 3.3.4, the net total efficiency can be calculated from the rated total efficiency and UEATR.

4.2. Keep balance between outdoor air supply and air exhaust to outdoor for each floor or section of buildings

It is basically very important to make a balance between the total amounts of outdoor air supply and indoor air exhaust, for each floor or each of any section partitioned from other parts of the building. Even though it is not easy for equipment installers to measure airflows by fans and to adjust them to designed airflows, the adjustment practice is fundamental when an energy saving by using energy recovery system is intended. The sizing of fans is based on pressure loss calculation of duct systems usually with safety-side assumptions on pressure loss coefficients of components. Such assumptions usually result in airflows larger than designed ones. This is a reason why adjustments of airflow is needed.

The imbalance of the mechanical ventilation between the total supply airflow rate and the total exhaust airflow rate inevitably causes air infiltration or exfiltration. Higher air tightness of the building envelope alleviates air infiltration and exfiltration due to the imbalance to some extent, but it cannot change mechanical ventilation rates nor alleviate the air leakage to a large extent (e.g., 10-15% at most roughly).

In residential buildings, not like non-residential buildings, whole spaces are connected to each other, and the air balance should be considered for a single whole space.

For air handling units with a variable air volume control[A31] for the temperature control of air-conditioned rooms (see Figure 4, left), the balance between outdoor airflow and exhaust airflow must be controlled such as by using variable air volume units (with airflow measurement and airflow control functions) in the outdoor air duct and the exhaust air duct and by using a motorized return air damper or an outdoor air fan (Yoshida and Matsushita, 2024).

4.3. Extract air for energy recovery from well heated and/or airconditioned space

The primary conditioned space (i.e., Space A) for heating and cooling in buildings is defined as the space provided with comfortable conditions (e.g., higher temperature in winter, lower humidity in summer) than other spaces (secondary conditioned space, Space B). For example, office rooms, classrooms in schools and living rooms in detached houses are examples of such primary conditioned space, while halls, lobbies and lavatories in the same buildings may not be heated nor cooled so well as the primary conditioned spaces. It may be partly because occupants do not stay longer in those secondary conditioned spaces.

To fully recover the enthalpy in exhausted air by energy recovery ventilation, it is recommended to extract air from the primary conditioned spaces with higher temperature in winter and with lower enthalpy in summer during cooling.

In residential buildings, halls are not so well heated nor cooled as living rooms and dining rooms especially when they are partially heated and cooled. In those cases, return air should be taken not from the halls but from the living rooms and dining rooms.

4.4. Exhaust air through energy recovery elements as much as possible

As many experts know, the higher the envelope thermal performance becomes, the larger the portion of the heating and cooling need due to the ventilation in the total need becomes. The longer the heating and cooling system is operated and/or the larger the difference between outdoor and indoor enthalpies becomes, the higher the cost effectiveness of using energy recovery systems becomes.

To reduce heating and cooling need (load) due to outdoor air intake, energy recovery between exhaust air and outdoor air should be utilized as much as possible by exhausting indoor conditioned air through energy recovery systems.

4.5. Choose energy recovery systems with higher net effectiveness and net efficiency

As previously described, the higher the unit air transfer ratio (UEATR) is, the higher the gloss effectiveness and efficiency becomes as a test result. Therefore, the value of the UEATR must be used when comparing net effectiveness and efficiency of products by using Equations 10 or 11.

The transfer of the return air and/or the infiltration of the air of the space surrounding the energy recovery systems occurs also during the actual operation in buildings. The transfer results in a decrease of outdoor air supply to conditioned spaces (see Equation 7) and may deteriorate the actual specific fan power, that is the efficiency of introducing outdoor air. Therefore, the installation and maintenance of energy recovery systems should be carefully made to reduce the air leakage into supply air.

4.6. Choose energy recovery ventilators and units with larger capacity of energy exchange

The same energy recovery system shows different effectiveness and efficiency depending on airflow rate, as can be seen in Figure 10. Assuming equal supply and exhaust airflow rates, the lower the airflow rate, the higher the effectiveness and efficiency becomes. Therefore, there is an idea to choose energy recovery systems with the larger rated airflow rates and set the design airflow rate lower than the product's rated airflow rate, so that higher effectiveness and efficiency can be obtained.

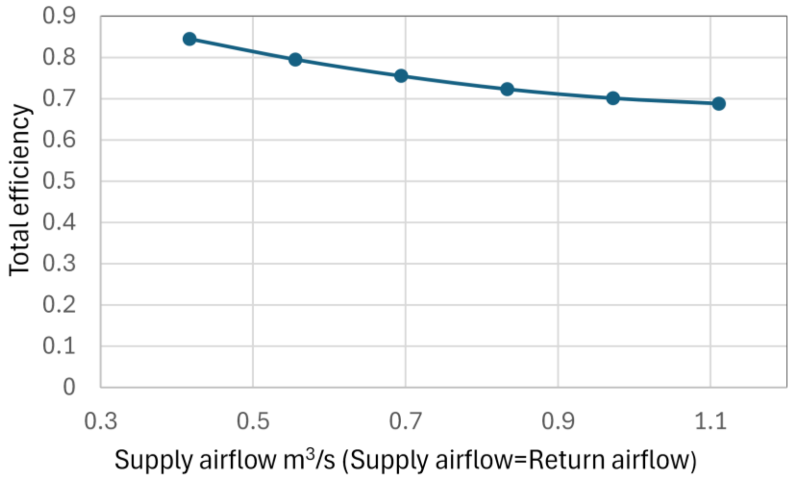


Figure 10: Airflow rates and total efficiency (rotary wheel energy recovery units), based on a calculation method described in C.3.1 of Appendix C - Review of relevant standards to calculate space heating and cooling need (load) reduction by energy recovery systems

4.7. Choose fans with smaller specific fan power (SFP) and duct systems with smaller pressure loss

Even if energy recovery systems can reduce heating and cooling need and energy use, total energy use for the HVAC system may increase without careful design for fan system's energy efficiency. Therefore, designers should stick to the specific fan power SFP (see Equation 1). The SFP reflects efficiencies of fan, motor and inverter as well as pressure loss through airstreams (duct and relevant components, energy recovery element, etc.).

In addition, especially when the energy recovery ventilation system is carefully designed for its fan efficiency and smaller pressure loss by thicker, shorter ductwork and rigid duct (not flexible), the operational energy use becomes more limited to contribute to improving cost effectiveness.

4.8. Demand control ventilation

Energy recovery ventilation systems shown in Figure 4 may have a function of demand control ventilation such as by using CO_2 concentration measured in the return air duct or in the ventilated room. When applying CO_2 concentration control, one issue to be considered is the reliability of CO_2 censors and a regular calibration of the sensors. For regular calibration, the sensors must be more easily accessible.

4.9. Cleaning of energy recovery elements and filters

In the narrow airflow paths of energy recovery elements, dust accumulates rapidly. To avoid accumulation, filters are installed but they also need regular cleaning. An important issue to be considered is accessibility to the elements and filters. Energy recovery ventilators are frequently installed behind the ceiling, which must be accessed through inspection openings. When they are installed in residential buildings, one way to relieve the issue is to install them on the floor of the mechanical space, on the wall or below the floor, which may be more easily accessible especially for elder residents.

4.10.Implement the sizing of heat generators

The energy use of heat generators is not necessarily proportional to the heating and cooling need (load) to be dealt with by the heat generators. When the part load ratio, which is defined as a ratio of heating or cooling need dealt with at each time to the rated (full) capacity of the heat generator, is low such as 5% to 30%, COP at that time can be much lower than the rated COP. Therefore, it is possible that the energy use of the heat generator does not decrease even when the heat need decreases thanks to energy recovery systems. To avoid such situations, it is indispensable to implement the sizing of heat generators according to any standardized protocols for the sizing of heat generators. Following the protocol for sizing is indispensable not only for heat generators, but also for fans and pumps, especially when their speeds can be controlled by the amount of heat transported by them (i.e., in case of Variable Air Volume and Variable Water Volume controls).

5. Case studies with energy use calculation

5.1. Assumptions of case studies and energy use calculations

To make decisions for the design of HVAC systems, building energy use calculation methods are helpful because they can quantify all energy uses for heat sources and fans.

As described earlier, the air balance between supply airflow and exhaust airflow has an impact on the efficiency of energy recovery ventilation systems (ERVs). To keep the overall air balance of the floor or the section of the building, the make-up air for exhaust-only ventilation for Space B (e.g., lavatories) is a key issue. Three ways to keep the balance (See Equation 12, $\Delta Q = 0$) has been described in Section 4.1.

In this series of case studies, the following four patterns are analyzed by using a building energy calculation program, which integrate the algorithms for energy recovery systems described in Section C.3 of Appendix C - Review of relevant standards to calculate space heating and cooling need (load) reduction by energy recovery systems.

- Pattern 1: no energy recovery system,
- Pattern 2: reduction of exhaust airflow through ERVs to compensate for air exhaust by exhaust-only ventilation of Space B,
- Pattern 3: exhaust air from ERVs to a position near Space B with exhaust-only ventilation (see Figure 9), and
- Pattern 4: change exhaust-only ventilation of Space B to balanced ventilation.

General characteristics of the building energy calculation method were reviewed in references (MLIT, NILIM, BRI and JSBC, 2025, Yanai et al., 2011, Sawachi (ed.), 2020).

In Appendix C - Review of relevant standards to calculate space heating and cooling need (load) reduction by energy recovery systems, a European building energy calculation standard (EN 16798-5-1) and an American calculation method (EnergyPlus) are also reviewed. Other building energy calculation methods could be used if they can calculate energy use of heat generators, fans and pumps of HVAC systems based on heating and cooling need calculations. It is also required to be able to calculate the net total efficiency of energy recovery systems, taking airflow rates into account.

The four patterns were calculated of their annual energy use not only for heat generators (heat pumps are assumed) but also for ventilation systems. For each of the four patterns, two kinds of HVAC systems are studied as follows.

- Case 1: central heat pump chillers with air handling units and rotary energy recovery units,
- Case 2: variable refrigerant flow split air-conditioners with fixed-plate type energy recovery ventilators)

As a total, for eight types of HVAC system (2 Cases by 4 Patterns), energy calculations have been done. Characteristics of Case 1 and Case 2 are summarized in Table 2.

A seven-storied office building located in Tokyo is assumed, and its floor plan for 2nd to 7th floors is shown in Figure 11 and Figure 12 with layouts of air-handling units, energy recovery systems, other ventilation equipment, ducts and air terminals for Pattern 3. In Figure 11, the exhaust duct of air-handling unit 'AHU-2-2' integrated with an energy recovery unit is extended to near the lavatories located on the south-east corner of the floor (design exhaust airflow rate is 2700 m^3/h). In Figure 12, the exhaust ducts of four energy recovery ventilators installed for a northern large office room is extended to the hall connected with the lavatories (750$m^3/h \times 4$=3000m^3/h).

In Table 2, specifications of heat generators are described such as COPs, total capacities, capacities per unit air-conditioned floor area. For example, the capacity of electric heat pump chillers for Case 1 and electric VRF systems for Case 2 are 0.177 kW/m^2 and 0.122 kW/m^2 for cooling, respectively. The specific fan power of various fans is also described. The sizing of the heat generators, fans and pumps was carried out according to a Japanese HVAC system design standard (MLIT, 2018).

As for energy recovery systems, rotary wheel energy recovery units and fixed-plate energy recovery ventilators are assumed for Case 1 and Case 2, respectively. Their rated gross total efficiencies, rated airflow rates and UEATR are also described.

Table 2: Assumptions on characteristics of building and HVAC systems including energy recovery ventilation systems

Building characteristics: Office building, seven-storied, total floor area: 10,192m^2 Location: Climatic Region 6 (1750 Heating Degree-Day) including Tokyo Envelope: exterior wall R=1.25m^{2}K/W, roof R=2.5 m^{2}K/W, window U=2.59W/m^{2}K, η=0.32		
	Case 1: Central heat generators and Air handling unit (see Figure 8)	**Case 2**: Variable refrigerant flow split air-conditioner (see Figure 9)
Heat generators	Electric heat pump chiller module type: 180kW (COP cooling 3.22, heating 3.38)×4 and 150kW (COP cooling 3.5, heating 3.54)×1 Capacity per unit floor area of conditioned space: 0.117kW/m^2 for heating and cooling The assumed performance curve of input energy and part load ratio: See Appendix D - Assumed performance curves for heat generators.	Electric heat pump outdoor units: 16 units with capacities 894kW as a total for cooling and 1028kW for heating (COP cooling 3.39, heating 2.96) Capacity per unit floor area of conditioned space: 0.122kW/m^2 for cooling and 0.141kW/m^2 for heating The assumed performance curve of input energy and part load ratio: See Appendix D - Assumed performance curves for heat generators.
Emitters	Specific fan power of air handling unit: Supply air & return air fans 1.688 kW/(m^3/s) Outdoor air & exhaust air fans 1.667 kW/(m^3/s) Specific fan power of fan coil unit: 0.134 kW/(m^3/s)	Specific fan power of indoor units: 0.236 kW/(m^3/s)
Heat transfer medium	Water	Refrigerant
Energy recovery systems	Rotary wheel type Total efficiency: 0.74 (cooling and heating) Airflow condition (supply and return airflow) 0.903 m^3/s UEATR 90%	Fixed-plate type Total efficiency: 0.65 (cooling), 0.75 (heating) Airflow condition (supply airflow and return airflow) 0.22 m^3/s UEATR 92%

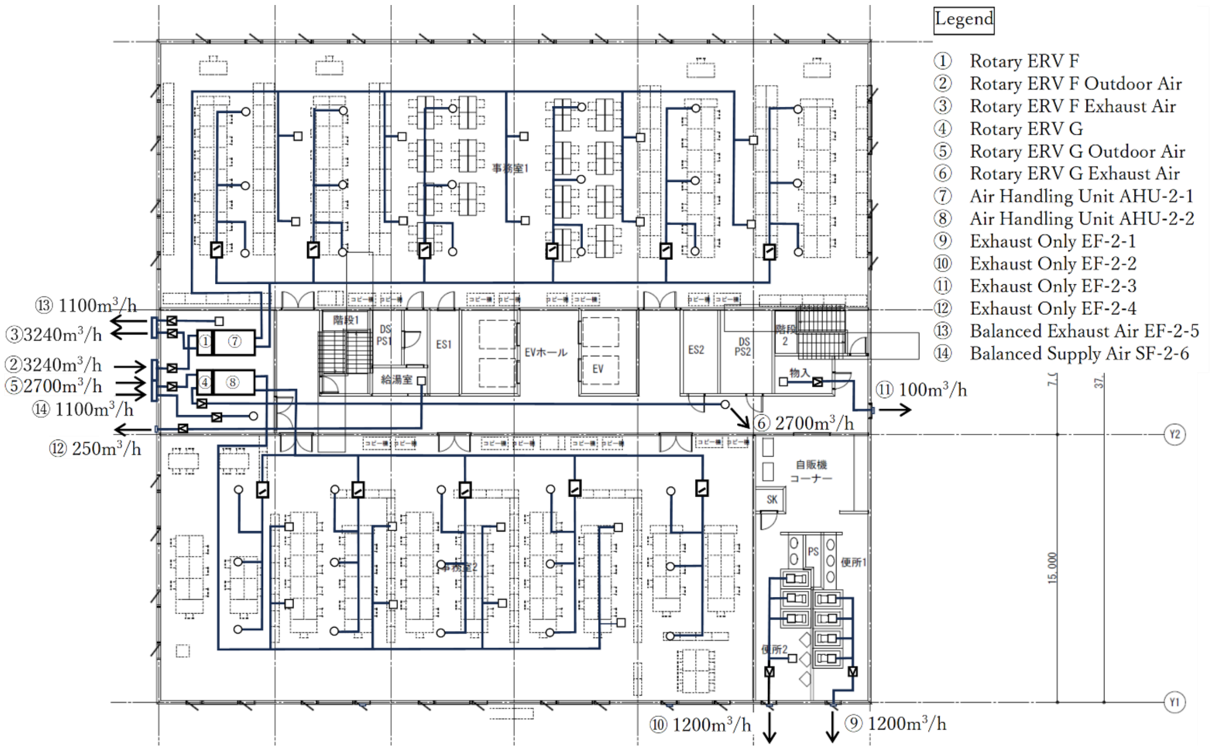


Figure 11: Pattern 3 of Case 1, layout of air handling units, energy recovery units, ducts, air terminal and other ventilation fans (Note: this is a plan for the 2nd to 7th floor of the building described in Table 2)

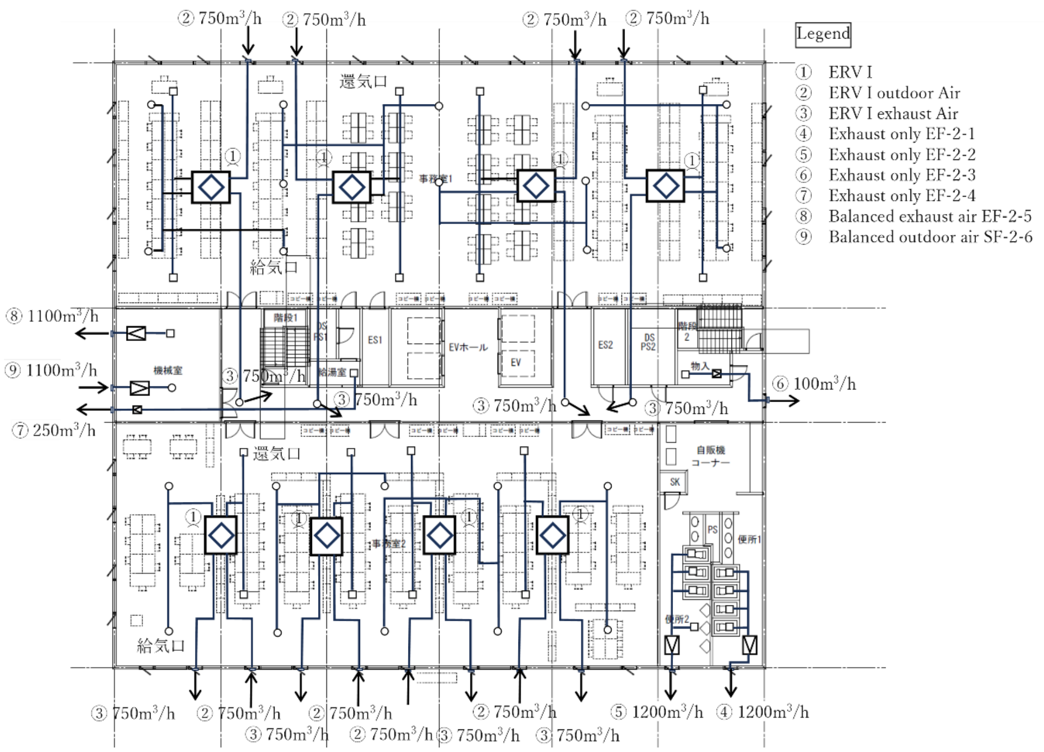


Figure 12: Pattern 3 of Case 2, layout of energy recovery ventilators, ducts, air terminals and other ventilation fans

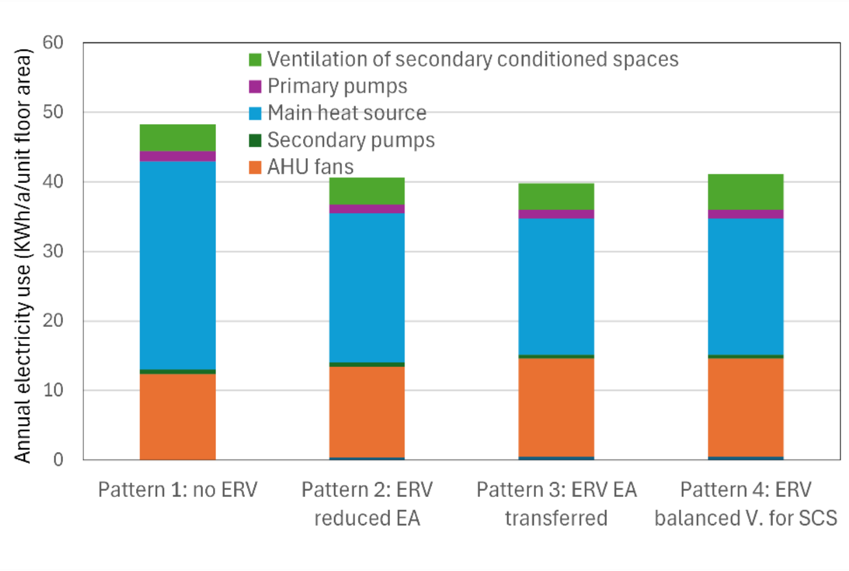


Figure 13: Result of energy use calculation for HVAC systems (Case 1)

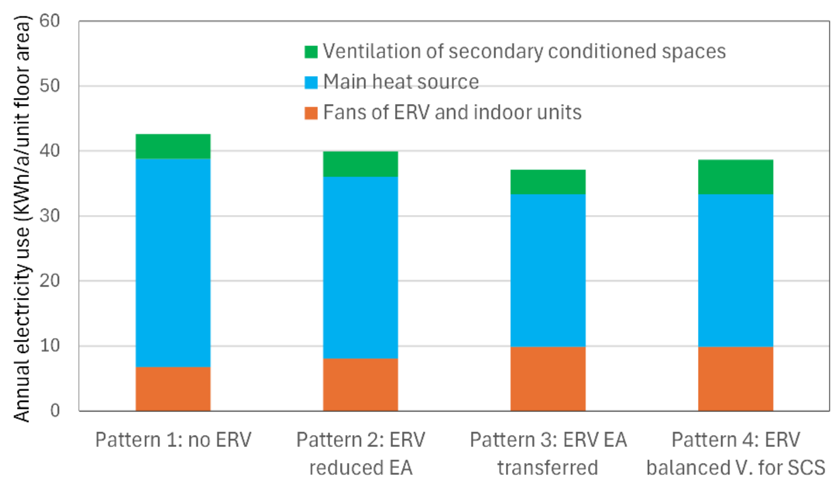


Figure 14: Result of energy use calculation for HVAC systems (Case 2)

Table 3: Result of energy use calculation for HVAC systems in Case 1 (Figure 13)

Pattern	HVAC total	AHU fans	Secondary pumps	Heat generators	Primary pumps	Ventilation for SCS
1	48.3 kWh/m^2/a	12.4 kWh/m^2/a	0.65 kWh/m^2/a	29.9 kWh/m^2/a	1.5 kWh/m^2/a	3.8 kWh/m^2/a
2	40.6 (84%)	13.5 (109%)	0.54 (83%)	21.5 (72%)	1.3 (87%)	3.8 (100%)
3	39.8 (82%)	14.6 (118%)	0.52 (80%)	19.6 (66%)	1.3 (87%)	3.8 (100%)
4	41.1 (85%)	14.6 (118%)	0.52 (80%)	19.6 (66%)	1.3 (87%)	5.2 (137%)
	Note: each % in brackets is based on energy use of Pattern 1 for each component					

Table 4: Result of energy use calculation for HVAC systems in Case 2 (Figure 14)

Pattern	HVAC total	ERV, indoor unit fans	Heat generators	Ventilation for SCS
1	42.6 kWh/m^2/a	6.8 kWh/m^2/a	32.0 kWh/m^2/a	3.8 kWh/m^2/a
2	39.9 (94%)	8.0 (118%)	28.0 (88%)	3.8 (100%)
3	37.1 (87%)	9.9 (146%)	23.4 (73%)	3.8 (100%)
4	38.7 (91%)	9.9 (146%)	23.4 (73%)	5.4 (142%)
	Note: each % in brackets is based on energy use of Pattern 1 for each component			

Table 5: Net total efficiency of representative energy recovery unit under design airflows

Case	Pattern	Rated net total efficiency	Rated airflow condition Supply/Return (m^3/s)	Net total efficiency under design airflow condition	Design airflow condition Supply/Return (m^3/s)
1	2	0.71	0.903/0.903	0.49	0.900/0.483
	3 and 4			0.71	0.900/0.900
2	2	0.62	0.222/0.222	0.45	0.208/0.111
	3 and 4			0.63	0.208/0.208
		Note: values in the table are for cooling			

5.2. Result for a HVAC system with electric heat pump chillers for central heat generators (case 1)

The results for Case 1 are shown in Figure 13 and Table 3.

This type of HVAC system has an advantage to have the number control of multiple heat generators (in this case, five electric chillers) to improve energy efficiency under part load conditions. From the heat sources, thermal energy is carried by cold and hot water through primary and secondary water circuits to emitters (i.e., air handling units (AHU) and fan coil units). Fan energy use of fan coil units is much smaller than AHU, for which pressure loss is much larger (i.e., 0.134 kW/(m^3/s) versus 1.688kW/(m^3/s) as shown in Table 2). In Case 1, air handling units (AHU) are used to transport thermal energy and outdoor air through ducts to conditioned spaces. Another advantage of this type is to be able to have high performance filters in the AHUs, while a disadvantage of this type is to use air to transport thermal energy and the specific heat (in J/K/kg) of the air is smaller than water and refrigerants.

By using ERV, energy use for main heat sources (generators) decreases to 72% (Pattern 2) or 66% (Pattern 3 and Pattern 4) of Pattern 1 with no ERVs, while energy use for fans of AHUs increases to 109% (Patter 2) or 118% (Pattern 3 and Pattern 4). This increase of fan energy use is due to pressure loss by energy recovery units. Throughout these case studies, it is assumed to resize fans for exhaust air of AHU as well as of energy recovery ventilators (as recommended in 4.1, just below Equation 13). As for Pattern 2, to reduce exhaust airflow through the ERV to compensate for exhaust airflow from lavatories, etc., it is assumed to size down the exhaust fan in AHU compared with Pattern 3. If resizing the fans is not carried out and dampers are used instead to regulate the exhaust airflow, the fan energy use for Pattern 2 may stay almost equal to Pattern 3. In Pattern 3 and Pattern 4, net total efficiency of ERV is improved by balancing supply airflow and return airflow (from 0.49 to 0.71, see Table 5), and energy use for main heat source is further reduced to 66% of Pattern 1 in contrast with Pattern 2 (72%).

In Pattern 4, to maintain the air balance of each floor, the exhaust-only ventilation in lavatories (secondary conditioned space, SCS) is replaced with balanced ventilation with heat recovery. Due to the replacement, energy use of ventilation for SCS increases to 137% as shown in Table 3. Due to this increase, the total energy use of the HVAC system of Pattern 4 increases when compared with Pattern 3.

In Case 1 for a HVAC system with electric heat pump chillers for central heat generators, by applying energy recovery systems, total energy use of the HVAC system is reduced by 15 to 18 %.

In the total energy use of HVAC systems, the energy use for AHU and ventilation equipment for secondary conditioned spaces is 19.8 kWh/m^2/a, which is larger than the energy use for main heat sources (19.6 kWh/m^2/a) for Pattern 4.

By using ERV and by balancing airflows through the ERV space heating and cooling need (load) and energy use for heat sources can be reduced thanks to the improvement of the net total efficiency. However, for HVAC system with AHUs, it is critically necessary to be careful with sizing fans and choosing energy efficient fans and their drive system (motor and inverter). Without such careful design of fans and total HVAC system, there is a risk of failing in reducing total energy use even if energy recovery units are installed.

5.3. Result for a HVAC system with VRF systems (Case 2)

The results for Case 2 are shown in Figure 14 and Table 4.

This type of HVAC system has an advantage to save energy for heat transportation by using refrigerant between outdoor units and indoor units. This type has become popular in North America and Asia-Oceania, but there is an issue due to the refrigerant, of which global warming potential (GWP) is still large. Recently, refrigerant R32 (GWP=771) is replacing R410A (GWP=2256) (UNEP, 2022). The VRF systems for Case 2 use R32, while heat pump chillers for Case 1 use R410A.

It has become popular to combine VRF multi-split air conditioners and energy recovery ventilators such as in the Japanese market, like Case 2. For Case 2, there is no water circuit and its pump nor ducts for heat transportation. Ducts are used only for supplying outdoor air and exhausting indoor air. That is the reason why the energy use for indoor units of VRF systems and ventilation equipment for secondary conditioned spaces in Case 2 (10.6 to 15.3 kWh/m^2/a) is smaller than the energy use for AHUs and other ventilation equipment in Case 1 (16.2 to 19.8 kWh/m^2/a).

As described in the row for 'Heat generators' of Table 2, designed capacities of the VRF system are 0.122 KW/m^2 and 0.141 kW/m^2, for space cooling and heating, respectively. The sizing has been done with energy-oriented considerations and according to the design method of building services published by the Public Buildings Association (MLIT, 2018). According to a questionnaire survey to Japanese HVAC designers (Eto et al., 2023), the average capacity of VRF systems for office buildings in Region 6 is 0.168 kW/m^2 for cooling (number of samples is 121), which is 1.38 times of the assumption for these case studies.

In Case 2, energy use of heat generators in Pattern 1 (with no ERVs) is successfully reduced to 88% of Pattern 1 in Pattern 2 and 73 % of Pattern 1 in Pattern 3, even though the reduction is not as large as in Case 1, as shown in Table 4. The net total efficiencies of ERVs in Case 2 are 0.45 in Pattern 1 and 0.63 in Pattern 3 and Pattern 4 as shown in Table 5. The fan energy use for energy recovery ventilators in Pattern 2 and Pattern 3 increases compared with Patter 1 to 118% and 146%, respectively. It is mainly because fan power for energy recovery ventilators needs to be increased to overcome pressure loss due to energy recovery elements. In Pattern 2, the fan power is lower than Pattern 3, because the return airflow rate is deducted by the amount of air exhausted by exhaust-only ventilation for lavatories etc. and the fan for the exhaust has been resized according to the deducted exhaust airflow.

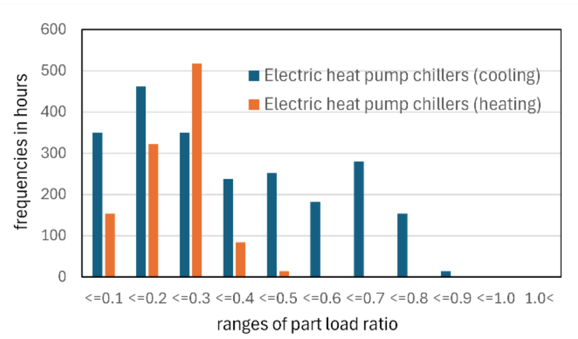
In Pattern 4, air balance for each floor is maintained by replacing exhaust-only ventilation for lavatories with balanced heat recovery ventilators, of which fan power is larger than the exhaust-only ventilation (142% of Pattern 1 and Pattern 2) as a total of ventilation for secondary conditioned spaces.

5.4. Assumed characteristics of heat generators

According to the assumed characteristics of VRF systems (see Figure D.1 in Appendix D - Assumed performance curves for heat generators), the input energy stays the same in the range of the part load ratio (the ratio of output at each time divided by the full capacity of the heat sources) between 0 and 0.3. It means that energy use does not decrease even if heating or cooling need is reduced in the range of part load ratio.

The distribution of part load ratio of a VRF system with Pattern 1 for a northern office room on the 4th floor is shown in Figure 15 (right). During more than 40% of cooling operation period, this VRF system operates under the part load condition below 0.3. In Pattern 2 and Pattern 3, where the heat need is reduced thanks to ERVs compared with Pattern 1, it is impossible to expect the decrement of energy use at least for that period.

On the contrary, the assumed characteristics of central heat pump chillers allows energy use reduction down to the part load ratio of 0.1 (see also Figure D.1). The range with no reduction of energy use for this chiller is narrower than that for VRF systems. In addition, the total capacity of the chiller is split into five chillers and the number of operating chillers is controlled according to the total heat need (see Table 2 Table 2). The control of the number of chillers contributes to raising the part load ratios of operating chillers. The difference of the assumed characteristics and the control between Case 1 and Case 2 is the reason for the difference of energy use reduction of heat generators (see Table 3 and Table 4).



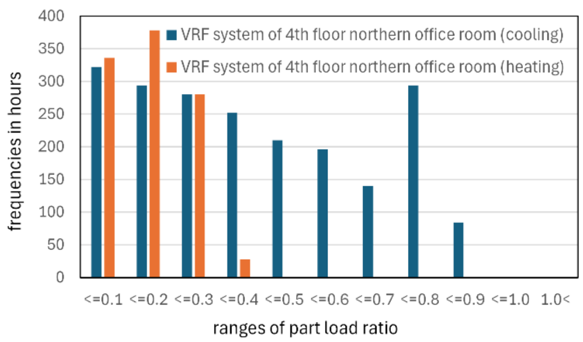


Figure 15: Frequencies of part load ratio for electric heat pump chillers (Case 1, left) and VRF system of an office room on 4th floor (Case 2, right)

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ANSI/ASHRAE Standard 84-2018 Method of Testing Air-to-Air Heat/Energy Exchangers

AHRI Standard 1060/1061-2018 Performance Rating of Air-to-Air Heat Exchangers for Energy Recovery Ventilation Equipment

ISO 21773:2021 Methods of test and characterization of performance of energy recovery components

JIS B 8628:2017 Air to air heat and energy exchanger and ventilators

JIS B 8639:2017 Heat and energy recovery ventilators – Method of test for performance of flowrate, net supply airflow and gross effectiveness

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7. Appendix A-Terminology

The key technical terms are explained below in alphabetical order.

A.1 Airflow ratio: A ratio of supply airflow rate to return airflow rate (see Figure 1 and Figure 4)
A.2 Allowance of test results: Acceptable deviation of test results from published ratings of products
A.3 COP, Coefficient of performance: A ratio of the net refrigerating or heating capacity to the total Input power at any given set of rating conditions
A.4 Energy (heat) recovery element: Core part of energy (heat) recovery ventilator or unit, through which two isolated airstream pass and exchange both heat and moisture (only heat)
A.5 Energy (heat) recovery unit: Unit containing energy (heat) recovery element
A.6 ERV, Energy (heat) recovery ventilation system, energy (heat) recovery system: Ventilation system which is designed to transfer both heat and moisture (only heat) between two isolated airstreams
A.7 Energy (heat) recovery ventilator: Ventilator which is designed to transfer both heat and moisture (only heat) between two isolated airstreams
A.8 Energy use for heating and cooling: Energy input to the heating and cooling system to satisfy the heating and cooling need (load)
A.9 Energy use for ventilation: Electric energy input to a ventilation system for air transport and energy recovery
A.10 Exhaust airflow (rate): Airflow rate of indoor air after passing through the energy (heat) recovery element or other ventilation equipment
A.11 Fixed-plate exchanger: Exchanger with multiple alternate airflow channels, separated by a heat or heat and water vapor transfer plate(s) and connected to supply and exhaust airstreams
A.12 Gross effectiveness: A ratio of the energy transfer (sensible, latent or total) to the product of the minimum energy capacity rate and the maximum difference in temperature, humidity ratio, or enthalpy
A.13 Gross efficiency, gross ratio: A ratio of the difference of temperature, humidity ratio, or enthalpy between outdoor air and supply air to the difference of temperature, humidity ratio, or enthalpy between outdoor air and return air
A.14 Heat capacity rate: Product of mass airflow rate and specific heat of the air
A.15 Heating and cooling need (load): Heat to be delivered to or extracted from a thermally conditioned space to maintain the intended space temperature and humidity conditions during a period
A.16 Latent heat: Heat that is transferred to or from a substance during a phase change, such as from a liquid to a gas or from a gas to a solid
A.17 Net effectiveness: A ratio of the actual energy transfer (sensible, latent, or total) to the product of the minimum energy capacity rate and the maximum difference in temperature, humidity ratio, or enthalpy, adjusted to account for that portion of the psychometric change in the supply airflow that is the result of leakage of return airflow and air surrounding the ventilator rather than exchange of heat and/or moisture between the airstreams
A.18 Net efficiency, net ratio: A ratio of the difference of temperature, humidity ratio, or enthalpy between outdoor air and supply air to the difference of temperature, humidity ratio, or enthalpy between outdoor air and return air, adjusted to account for that portion of the psychometric change in the supply airflow that is the result of leakage of return airflow and air surrounding the ventilator rather than exchange of heat and/or moisture between the airstreams
A.19 Net supply airflow ratio (NSAR): 100% minus the unit exhaust air transfer ratio (UEATR)
A.20 Outdoor airflow (rate): Airflow rate of outdoor air entering the energy (heat) recovery element or other ventilation equipment

A.21 Part load ratio, partial load ratio: A ratio of the actual capacity of a heat source to its rated capacity
A.22 Return airflow (rate): Airflow rate of indoor air entering the energy (heat) recovery element or other ventilation equipment
A.23 Rotary (wheel) type, rotary (wheel) exchanger: Exchanger with porous discs, fabricated from materials with heat or heat and water vapor retention capacity, that are regenerated by collocated supply and exhaust airstreams
A.24 Sensible heat: Heat that is transferred to or from a substance without causing a change in phase
A.25 Specific fan power: A ratio of power consumption to airflow rate driven by the fans (see Equation 1)
A.26 Standard air: Dry air at 20°C and 101,325 kPa absolute with a density of 1,204 kg/m^3
A.27 Supply airflow (rate): Airflow rate of outdoor air after passing through the energy (heat) recovery element or other ventilation equipment
A.28 Total efficiency: Gross efficiency or net efficiency for enthalpy exchange
A.29 Total heat: Sum of sensible heat and latent heat
A.30 Unit exhaust air transfer ratio (UEATR): Tracer gas concentration difference between the supply air and the outdoor air divided by the tracer gas concentration difference between the return air and the outdoor air
A.31 Variable air volume control, VAV control: A control of air volumes supplied by air-handling units to air-conditioned spaces by using motorized dampers in supply ducts and by variable frequency drives of supply and return fans. Its main purposes are to control room temperatures and to save power consumption of fans.
A.32 Ventilator: Self-contained unit that includes fans to move air through the heat/energy exchanger

8. Appendix B - Relevant testing and rating standards for energy recovery systems

Major relevant testing and rating standards are listed below. Among them, ISO 16494-1, JIS B 8628 and JIS B 8639 are mainly for fixed-plate energy recovery ventilators, while ISO 21773 and Technical certification rules of the Eurovent certified performance mark (hereafter, referred as ‘Eurovent rule’) are focused on rotary energy recovery units. EN 308, ANSI/ASHRAE Standard 84 and AHRI Standard 1060/1061 deal with both rotary wheel type and fixed-plate type.

1) ISO 16494-1:2022 Heat recovery ventilators and energy recovery ventilators – Method of test for performance – Part 1: Development of metrics for evaluation of energy related performance
2) ISO 21773:2021 Methods of test and characterization of performance of energy recovery components
3) EN 308:2022 Heat exchangers – Test procedures for establishing the performance of air to air and flue gases heat recovery devices
4) Technical certification rules of the Eurovent certified performance mark 2025, air-to-air regenerative heat exchangers/air-to-air plate and tube heat exchangers
5) ANSI/ASHRAE Standard 84-2018 Method of Testing Air-to-Air Heat/Energy Exchangers
6) AHRI Standard 1060/1061-2018 Performance Rating of Air-to-Air Heat Exchangers for Energy Recovery Ventilation Equipment
7) JIS B 8628:2017 Air to air heat and energy exchanger and ventilators
8) JIS B 8639:2017 Heat and energy recovery ventilators – Method of test for performance of flowrate, net supply airflow and gross effectiveness

All standards deal with air leakage from return air to supply air, but there are two different objectives to utilize the information on air leakage. One is to utilize the index of leakage to know net effectiveness or efficiency, while the other is to grasp how much air leakage occurs when a large static pressure difference exists in actual situations between supply and return sides due to fan pressurization or depressurization. For the first objective, static pressure conditions at outdoor-air, supply-air, return-air and exhaust-air stations (see Figure 5) are prescribed and controlled under the tests so that the static pressure difference between supply-air airstream and return-air stream stays closer to operational condition. On the contrary, the Eurovent rule includes a special test for exhaust air transfer to supply air under the extremely large static pressure difference of 250 Pa.

9. Appendix C - Review of relevant standards to calculate space heating and cooling need (load) reduction by energy recovery systems

C.1 EN 16798-5-1 (BS EN 16798-5-1, 2017)

In EPBD-related standards, EN 16798-5-1 provides a procedure to calculate the temperature and moisture content of the supply air leaving the energy recovery unit for non-residential buildings.

$$\theta_2 = \theta_1 + \eta_{hr}(\theta_3 - \theta_1) \qquad \text{(C.1)}$$

$$x_2 = x_1 + \eta_{xr}(x_3 - x_1) \qquad \text{(C.2)}$$

where

η_{hr}: temperature recovery efficiency

η_{xr}: humidity recovery efficiency

θ_2: temperature of supply air leaving energy recovery unit

θ_1: temperature of outdoor air entering energy recovery unit

θ_3: temperature of extract (return) air entering energy recovery unit

x_2: humidity ratio of supply air leaving energy recovery unit

x_1: humidity ratio of outdoor air entering energy recovery unit

x_3: humidity ratio of extract (return) air entering energy recovery unit

The temperature recovery efficiency η_{hr} is obtained as the product of nominal temperature recovery efficiency and three correction factors as equation C.3.

$$\eta_{hr} = \eta_{hr;nom} \cdot f_q \cdot f_v \cdot f_n \qquad \text{(C.3)}$$

where

$\eta_{hr;nom}$: nominal temperature recovery efficiency at a fixed equivalent mass flow rate for supply and return (mean face air velocity of 2m/s for rotary energy recovery unit).

f_q: correction factor for the mass flow ratio other than 1

f_v: correction factor for the air velocity (supply airflow)

f_n: correction factor for the rotation speed of the rotor (only for rotary recovery units)

The equations C.2 and C.3 cover both fixed-plate type and rotary type, and they can be used to calculate energy need (air conditioning load) due to outdoor intake and reduction thanks to energy recovery ventilation.

C.2 EnergyPlus (U.S. DOE, 2019, Wetter, 1999)

EnergyPlus provides two models for energy recovery systems.

The models are for 1) air-to-air sensible and latent effectiveness heat exchanger and 2) air-to-air flat plate heat exchanger.

The model for air-to-air sensible and latent effectiveness heat exchanger requires sensible and latent effectiveness for heating and cooling conditions with balanced airflow (supply airflow equal to the exhaust airflow) at two airflow rates: 75% and 100% of the nominal supply airflow.

To obtain the “operating” effectiveness, the equations C.4 and C.5 are used. In the equations, the ratio of the average operating volumetric airflow (the average of supply airflow and exhaust airflow) to the nominal supply airflow $HX_{flowratio}$.

$$\varepsilon_{operating,sen} = \varepsilon_{sen,75\%flow} + \left(\varepsilon_{sen,100\%flow} - \varepsilon_{sen,75\%flow}\right)\left(\frac{HX_{flowratio} - 0.75}{1 - 0.75}\right) \quad (C.4)$$

$$\varepsilon_{operating,lat} = \varepsilon_{lat,75\%flow} + \left(\varepsilon_{lat,100\%flow} - \varepsilon_{lat,75\%flow}\right)\left(\frac{HX_{flowratio} - 0.75}{1 - 0.75}\right) \quad (C.5)$$

where

$\varepsilon_{operating,sen}$ $\varepsilon_{operating,lat}$: the operating sensible and latent effectiveness, respectively

$\varepsilon_{sen,75\%flow}$ $\varepsilon_{sen,100\%flow}$: test results of the sensible effectiveness at 75% and 100% airflow condition, respectively

$\varepsilon_{lat,75\%flow}$ $\varepsilon_{lat,100\%flow}$: test results of the latent effectiveness at 75% and 100% airflow condition, respectively

$HX_{flowratio}$: the ratio of the average operating volumetric airflow (the average of supply airflow and exhaust airflow) to the nominal supply airflow

The supply air conditions leaving the heat exchanger are determined using the heat exchanger operating effectiveness calculated by equations C.4 and C.5, the ratio of the minimum heat capacity rate (among those of supply and exhaust airflow) to the supply airflow heat capacity rate, and the difference in temperature and humidity ratio between the supply air and return air, as given by equations C.6 and C.7.

$$\theta_2 = \theta_1 + \varepsilon_{operating,sen}\left(\frac{\dot{m}c_{p,min}}{\dot{m}c_{p,sup}}\right)(\theta_3 - \theta_1) \quad (C.6)$$

$$x_2 = x_1 + \varepsilon_{operating,lat}\left(\frac{\dot{m}c_{p,min}}{\dot{m}c_{p,sup}}\right)(x_3 - x_1) \quad (C.7)$$

$$\dot{m}c_{p,min} = MIN\left(\dot{m}c_{p,sup}, \dot{m}c_{p,exh}\right) \quad (C.8)$$

where

$\dot{m}c_{p,min}$: the minimum heat capacity rate obtained by Equation C.8 (W/K)

$\dot{m}c_{p,sup}$: the heat capacity rate of the supply air (W/K)

$\dot{m}c_{p,exh}$: the heat capacity rate of the exhaust air (W/K)

As equations C.1 and C.2 in EN 16798-5-1, Equations C.6 and C.7 can be used to calculate heating and cooling need (load) due to outdoor intake and reduction thanks to energy recovery ventilation.

The model for air-to-air flat plate heat exchanger in EnergyPlus is expressed by equations C.9 together with Equation C.6. This model can be applied only to sensible effectiveness not to latent effectiveness.

$$\varepsilon_{opearating,sen} = 1 - exp\left(\frac{e^{-NTU^{0.78}\left(\frac{\dot{m}c_{p,min}}{\dot{m}c_{p,max}}\right)} - 1}{\left(\frac{\dot{m}c_{p,min}}{\dot{m}c_{p,max}}\right)NTU^{-0.22}}\right) \quad (C.9)$$

$$\dot{m}c_{p,max} = MAX\left(\dot{m}c_{p,sup}, \dot{m}c_{p,exh}\right) \quad (C.10)$$

where

NTU: number of transfer unit (-) for operating condition, which is obtained by using a test result of ε, test conditions (airflows and temperatures) and operating conditions for airflows and temperatures.

To obtain NTU, Equations C.11 and C.12 are used.

$$NTU = NTU_0 \times \frac{\dot{m}c_{p,min,0}}{\dot{m}c_{p,min}} \times \frac{1 + r}{\left(\frac{\dot{m}c_{p,sup,0}}{\dot{m}c_{p,sup}}\right)^{0.78} + r\left(\frac{\dot{m}c_{p,exh,0}}{\dot{m}c_{p,exh}}\right)^{0.78}} \qquad (C.11)$$

$$r = \left(\frac{\dot{m}c_{p,sup,0} \times (\theta_{1,0} + 273.15)}{\dot{m}c_{p,exh,0} \times (\theta_{3,0} + 273.15)}\right)^{0.78} \qquad (C.12)$$

$$\varepsilon = 1 - exp\left(\frac{e^{-NTU_0{}^{0.78}\left(\frac{\dot{m}c_{p,min,0}}{\dot{m}c_{p,max,0}}\right)} - 1}{\left(\frac{\dot{m}c_{p,min,0}}{\dot{m}c_{p,max,0}}\right) NTU_0{}^{-0.22}}\right) \qquad (C.13)$$

where

NTU_0: number of transfer unit (-) for test condition

$\dot{m}c_{p,min,0}, \dot{m}c_{p,max,0}$: smaller and larger capacity rate (J/Ks) of supply and exhaust airflows under test, respectively

$\dot{m}c_{p,sup,0}, \dot{m}c_{p,exh,0}$: capacity rate (J/Ks) of supply and exhaust airflows under test, respectively

$\theta_{1,0}$, $\theta_{3,0}$: temperature of outdoor air and return air under test, respectively

By inserting NTU_0, which is obtained by solving the equation C.13, and r from the equation C.12 into the equation C.11, NTU can be obtained. By inserting NTU into the equation C.9, and by using the equation C.6, θ_2 can be calculated to be used for calculating energy need (air conditioning load) due to outdoor intake and reduction thanks to energy recovery ventilation.

C.3 Web program for BECS (MLIT, NILIM, BRI and JSBC, 2025, Sawachi (ed.), 2020)

Since 2013, computer programs to make calculations of building energy use have been provided online to practitioners (e.g., designers and building owners), who need to get building permits by complying with legal minimum requirements or to get an energy performance certification for higher energy performance such as net zero energy buildings. The programs are called “Web program”, and they have been developed and maintained by the Ministry of Land, Infrastructure, Transportation and Tourism (MLIT), National Institute for Land and Infrastructure Management (NILIM), Building Research Institute (BRI) and Japan Sustainable Building Consortium (JSBC).

In the Web program, the impact of energy recovery ventilation systems has been evaluated by using the following theories to take designed airflows into consideration.

C.3.1 Rotary energy recovery elements

This type of energy recovery element used for building HVAC systems is called regenerative heat exchanger. In general, thermal effectiveness (i.e., sensible effectiveness, latent effectiveness and total effectiveness) of the regenerative heat exchangers are equivalent. The total effectiveness is dependent upon supply airflow and return airflow, and it can be estimated by equations C.14 and C.15 (Murata, 2004).

$$R < 1: \quad \varepsilon = \frac{1 - e^{k}}{1 - R \cdot e^{k}} \qquad k = -\varepsilon_{eq}\frac{1 - R}{1 - \varepsilon_{eq}} \qquad (C.14)$$

$$R > 1: \quad \varepsilon = \frac{1 - e^{k'}}{R - e^{k'}} \qquad k' = -\varepsilon_{eq}\frac{R - 1}{1 - \varepsilon_{eq}} \qquad (C.15)$$

where

$R = \frac{Q_2}{Q_3}$

Q_2: supply airflow (m^3/s), Q_3: return airflow (m^3/s)

ε_{eq}: total effectiveness for equivalent supply and return airflow ($Q_2 = Q_3$)

The rotary energy recovery elements are usually used as a component of air-handling units, and supply and return airflows are determined by HVAC designers as design values. The estimation of ε_{eq} can be provided with test results for three different equivalent airflow conditions, $Q_{2,min}$, $Q_{2,med}$ and $Q_{2,max}$. For any supply and return airflow conditions ($Q_2 = Q_3 = Q$) between $Q_{2,min}$ and $Q_{2,max}$, total effectiveness at can be estimated by the following quadratic functions C.16.

$$\varepsilon_{eq,Q} = a_2 Q^2 + a_1 Q + a_0 \qquad (C.16)$$

C.3.2 Static plate energy recovery elements

The same basic relationship between sensible effectiveness and airflow conditions as the model for flat plate heat exchanger in EnergyPlus is used as shown in equations C.17, C.18 and C.19[22].

$$\varepsilon_{opearating,sen} = 1 - exp\left(\frac{e^{-NTU^{0.78}\left(\frac{V_{min}}{V_{max}}\right)} - 1}{\left(\frac{V_{min}}{V_{max}}\right) NTU^{-0.22}}\right) \qquad (C.17)$$

$$V_{max} = MAX(V_{SA}, V_{RA}) \qquad (C.18)$$

$$V_{min} = MIN(V_{SA}, V_{RA}) \qquad (C.19)$$

where

NTU: number of transfer unit (-) for operating condition, which is obtained by using a test result of ε, the rated conditions (airflows and temperatures), number of transfer unit for the rated conditions NTU_{rtd} and operating conditions for airflows and temperatures.

The following relationship is assumed for simplification.

$$NTU = NTU_{rtd} \times \frac{MIN\left(V_{rtd,SA}, V_{rtd,RA}\right)}{MIN(V_{SA}, V_{RA})} \qquad (C.20)$$

For the rated conditions and test result (effectiveness), the following relationship exists.

$$\varepsilon_{rtd,sen} = 1 - exp\left(\frac{e^{-NTU_{rtd}{}^{0.78}\left(\frac{V_{rtd,min}}{V_{rtd,max}}\right)} - 1}{\left(\frac{V_{rtd,min}}{V_{rtd,max}}\right) NTU_{rtd}{}^{-0.22}}\right) \qquad (C.21)$$

$$V_{rtd,max} = MAX\left(V_{rtd,SA}, V_{rtd,RA}\right) \qquad (C.22)$$

$$V_{rtd,min} = MIN\left(V_{rtd,SA}, V_{rtd,RA}\right) \qquad (C.23)$$

where

$V_{rtd,SA}$: rated supply airflow expressed in standard air when the test was carried out (m^3/s)

$V_{rtd,RA}$: rated return airflow expressed in standard air when the test was carried out (m^3/s)

V_{SA}: design supply airflow expressed in standard air (m^3/s)

V_{RA}: design return airflow expressed in standard air (m^3/s)

By equations C.15, C.16 and C.17 with measured $\varepsilon_{rtd,sen}$, $V_{rtd,SA}$ and $V_{rtd,RA}$, NTU_{rtd} can be obtained. By inserting design airflows and NTU, which is obtained by equation C.14 into equation C.11, $\varepsilon_{operating,sen}$ can be calculated. Furthermore, sensible efficiency at the design condition η, which is defined as equation B18, can be obtained by equation C.19.

$$\eta = \frac{\theta_1 - \theta_2}{\theta_1 - \theta_3} \qquad (C.24)$$

where

η: sensible efficiency

θ_1: outdoor air temperature (°C)

θ_2: supply air temperature (°C)

θ_3: return air temperature (°C)

$$\eta = \begin{cases} \varepsilon_{operating,sen} & if\ V_{RA} > V_{SA} \\ \varepsilon_{operating,sen} \times \frac{MIN(V_{SA}, V_{RA})}{MAX(V_{SA}, V_{RA})} & if\ V_{RA} \leq V_{SA} \end{cases} \qquad (C.25)$$

10. Appendix D - Assumed performance curves for heat generators

The state of the art of energy calculation methods for heat pump systems has been reported in a recent internationally collaborative R&D project (Neto et al. (eds.), 2024). The report introduces examples of calculation methods. One of the methods is the one for Japanese standard called BECS (MLIT, NILIM, BRI and JSBC, 2025, Yanai et al., 2011, Sawachi (ed.), 2020).

In the BECS, relationships between the part load ratio (defined as the ratio of the thermal need deal with at a time to the full capacity of the heat generator) and input ratio (defined as the ratio of input energy at a time to the energy input at the full capacity) is prescribed as shown in Figure D.1, which contains the relationships for electric water chillers (module type air-to-water heat pumps) and electric VRF (variable refrigerant flow) systems.

It should be noted that there are still no common relationships which are agreed upon internationally. More detailed information is described in a reference (Neto et al. (eds.), 2024).

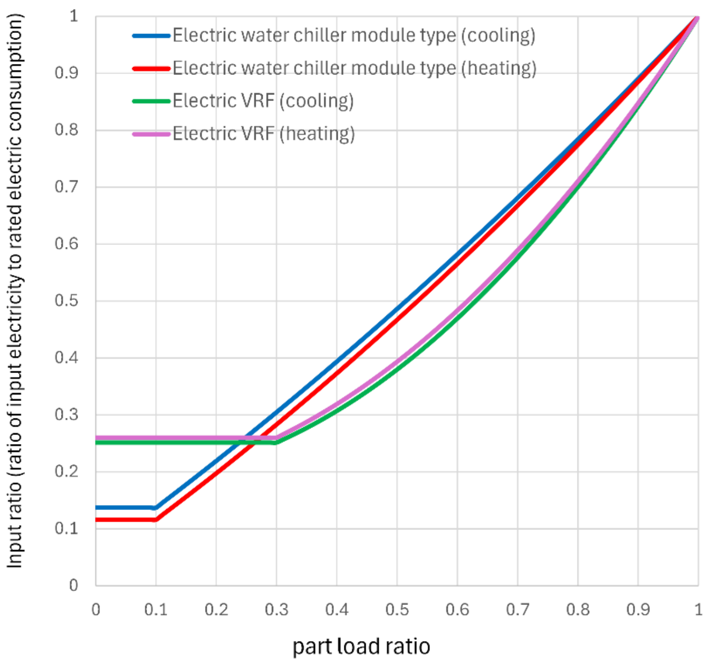

Figure D.1. Assumed characteristics of heat generators (Electric water chiller module type and electric VRF system, which are assumed for Case 1 and Case 2. See Table 2) (MLIT, NILIM, BRI and JSBC, 2025, Yanai et al., 2011)

In the range of 0.3 to 1.0 of part load ratio, the input ratio of electric water chillers module type is higher than electric variable refrigerant flow systems. Below 0.3 of part load ratio, the input ratio of electric variable refrigerant flow systems stays the same value, which means electric consumption does not decrease even if heating and cooling need (load) decreases. On the other hand, for electric water chillers module type, the input ratio decreases in a wider range of part load ratio between 0.1 to 1.0.

Because of these characteristics of heat generators for lower part load ratio, sizing practice according to standardized protocols is indispensable to make HVAC systems energy efficient.

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