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The effect of a rotary heat exchanger in room-based ventilation on indoor humidity in existing apartments in temperate climates

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Abstract

The investigation constructed and simulated moisture balance equations for single-room ventilation with a non-hygroscopic rotary heat exchanger. Based on literature, the study assumed that all condensed moisture in the exhaust subsequently evaporated into the supply. Simulations evaluated the potential for moisture issues and compared results with recuperative heat recovery and whole-dwelling ventilation systems. To assess the sensitivity of results, the simulations used three moisture production schedules to represent possible conditions based on literature. The study also analyzed the sensitivity to influential parameters, such as infiltration rate, heat recovery, and indoor temperature. With a typical moisture production schedule, the rotary heat exchanger recovered excessive moisture from kitchens and bathrooms, which provided a mold risk. The rotary heat exchanger was only suitable for single-room ventilation of dry rooms, such as living rooms and bedrooms. The sensitivity analysis concluded that varying heat recovery or indoor temperature could limit indoor relative humidity in dry rooms when a moderate risk was present. The rotary heat exchanger also elevated the minimum relative humidity in each room, which could help to avoid negative health impacts. A discussion emphasized the potential benefits of selecting heat recovery to match the individual needs of each room.

Keywords

Decentralized ventilation; single-room ventilation; room-based ventilation; rotary heat exchanger; moisture issues; mold risk; renovated buildings; energy retrofit; temperate climate.

Highlights

- The simulated rotary heat exchanger provided moisture concerns in several rooms.
- The rotary heat exchanger was only suitable for ventilation of so-called dry rooms.
- Varying heat recovery or temperature can limit indoor relative humidity in dry rooms.
- Single-room ventilation allows selection of heat recovery to match the needs of rooms.

Nomenclature

Latin	
e	partial pressure of water vapor in air [hPa]
$G(t)$	mass flows of moisture at time t [g/h]
G_i	moisture release in time step i [g]
m	mass [g][kg]
M	molar mass [g/mol]
N	air changes rate [dt^{-1}][h^{-1}]
p	total barometric pressure [hPa]
R	universal gas constant [J/mol·K]
T	air temperature [°C]
V	volume [m^3]
x	moisture content in mass of water per mass of dry [g/kg]
Greek	
ρ	density [kg/m3]
η	temperature efficiency [-][%]
φ	relative humidity [%]

Subscripts	
amb	ambient air
dmix	mixed dry room exhaust
dp	dew-point
dry	subset of dry rooms
exh	exhaust air
i	time step index
in,out	direction of flow
inf	infiltration air
max	maximum
min	minimum
room	room index
sat	saturation
sources	indoor sources
sup	supply air
vent	ventilation air
wet	subset of wet rooms
wmix	mixed wet room exhaust

1 Introduction

In an effort to mitigate anthropogenic climate change, many governments have targeted energy savings to reduce greenhouse gas emissions. In the temperate climate of Denmark, heating in buildings is responsible for 25% of final energy consumption [1], so renovations provide obvious potential for savings. A Danish national action plan [2] therefore expects to reduce heating consumption in existing buildings by at least 35% before 2050. An assessment by the Danish Building Research Institute provided the basis for these expectations. The assessment [3] also considered a scenario in which renovations improve airtightness and thus require mechanical ventilation with heat recovery. This would further decrease heating consumption and improve indoor climate. To achieve this scenario, the assessment emphasized the need for inexpensive and flexible ventilation systems with heat recovery as well as the necessary knowledge and competence for proper implementation. For that purpose, Smith and Svendsen [4] described a collaborative development of a rotary heat exchanger for room-based ventilation in existing apartments. The development of that prototype led to the current investigation, which simulated its impacts on indoor humidity to obtain knowledge for proper implementation.

The above scenarios assumed that renovations will replace worn out components with new components that that comply with building regulations. The 2010 Danish building regulations require heat recovery with a temperature efficiency of 70% for ventilation of entire buildings and 80% for single dwellings [5]. The 2020 regulations will increase these requirements to 75% and 85%, respectively [6], and the aforementioned prototype targeted the latter value. These regulations emphasize heat recovery, but they neglect the potential coupling of heat and moisture. They only discuss moisture transfer in heat exchangers when specifying conditions for testing. Similarly, a detailed guideline on indoor air quality from the World Health Organization recommended heat recovery to simultaneously retain heat and reduce indoor humidity, but it gave no further guidance on moisture transfer in heat exchangers [7]. In highly efficient heat recovery, the exhaust temperature often decreases below its dew point, so moisture condenses in the heat exchanger. If the amount of condensation is significant, it is important to know whether it will evaporate, drain, accumulate, or freeze, and the type of heat exchanger can influence this behavior.

There are two categories of air-to-air heat exchangers. Regenerators, such as rotary exchangers, intermittently expose airflows to the same medium to store and recover heat, whereas recuperators transfer heat through a membrane between airflows. A recuperator with an impermeable membrane does not transfer moisture. Any condensation on its surfaces must drain from the heat exchanger. Conversely, a regenerator exposes both airflows to the same heat transfer surface, so condensation from exhaust is likely to evaporate into the supply air [8]. This investigation focused on the latter to assess the impact of moisture transfer in a single-room rotary heat exchanger on indoor humidity and moisture issues for different room types.

Moisture removal is an important aspect of residential ventilation in humid temperate climates. According to the World Health Organization, excess indoor humidity can lead to health issues by promoting mold growth and proliferation of dust mites. It can also lead to structural issues by degrading building materials.

Infiltration lowers indoor humidity during the heating season, but its heat loss is excessive, so renovations maximize air tightness. With minimal contributions from infiltration, mechanical ventilation must solely remove sufficient moisture.

In temperate humid climates, the outdoor air is nearly saturated with moisture throughout the heating season. For example, the average relative humidity is 86% from September 16th to May 15th in the 2013 Danish design reference year [9], and the maximum 30-day average is 94%. If a rotary heat exchanger transfers all condensation between airflows, its drying capacity is only the difference in moisture content between the nearly saturated outdoor air and the saturated exhaust air. At low temperatures, the relatively small difference in saturated moisture content may severely limit the drying capacity of mechanical ventilation with a rotary heat exchanger. Figure 1 demonstrates this behavior with psychrometric charts for an uncoated rotary heat exchanger with the average outdoor conditions of 86% relative humidity (RH) and 4°C during the heating season in Denmark. The uncoated rotary heat exchanger has a temperature efficiency of 85% and cools the exhaust air below its dew point temperature for each of the three indoor relative humidities.

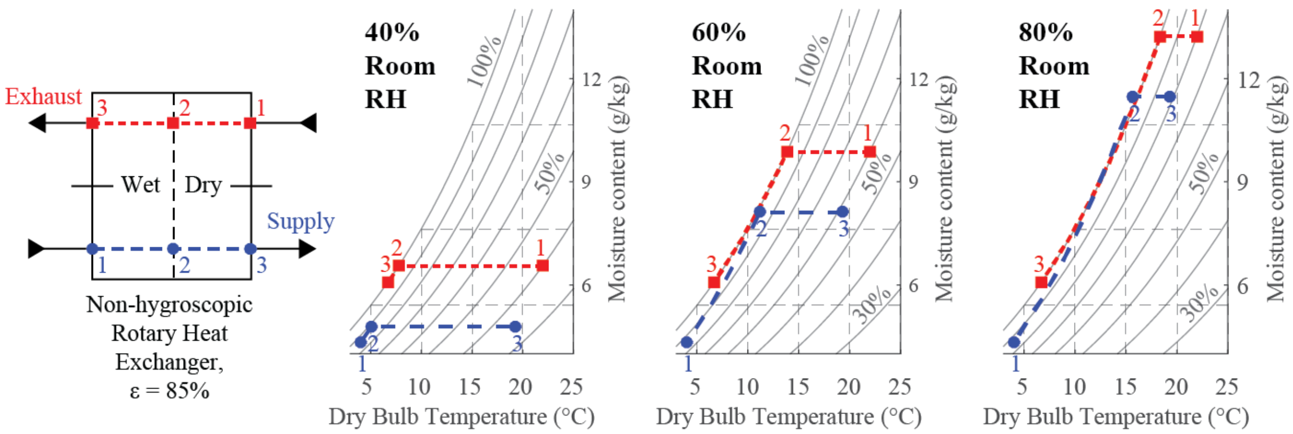

Figure 1. Supply and exhaust airflows through a heat exchanger with 85% temperature efficiency. Outdoor air is 4°C and 86% RH. Room air is 22 °C with three different relative humidities. The dew-point temperature of exhaust air is indicated by the red ‘2’.

In contrast, a desiccant-coated rotary heat exchanger that is “fully hygroscopic” may produce outlet conditions that are on a straight line between inlet conditions on a psychrometric chart, as shown in Figure 2. The term “fully hygroscopic” refers to a rotor with sufficiently high moisture capacity and sufficiently low diffusion resistance such that the moisture transfer efficiency is as high as the temperature efficiency [10].

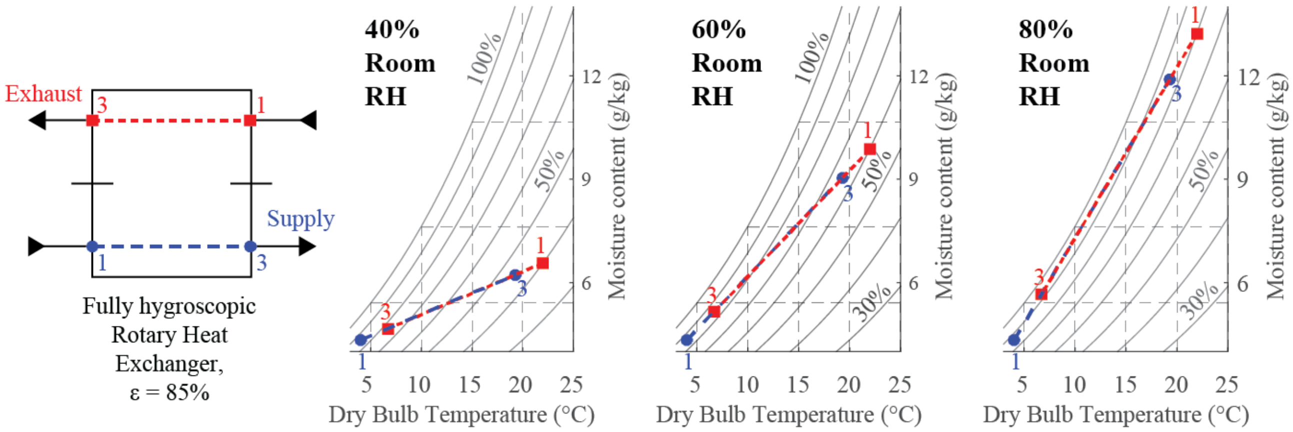

Figure 2. Example of a fully hygroscopic rotary heat exchanger with temperature and moisture efficiencies of 85%. The term “fully hygroscopic” refers to a rotor with sufficiently high moisture capacity and sufficiently low diffusion resistance such that moisture transfer efficiency may equal temperature efficiency.

Recent research has investigated intended moisture transfer in rotary heat exchangers [11][12][13]. These heat exchangers have hygroscopic surfaces to assist moisture transfer between airflows without the need for condensation. However the desirability of moisture transfer depends on context and may not be suitable for all applications. The current study specifically deals with the impacts of moisture transfer in non-hygroscopic heat exchangers with a focus on single-room ventilation in humid temperate climates. In the temperate zones of Sweden, non-hygroscopic rotary heat exchangers are often used in ventilation of entire dwellings, and limited research has indicated potential issues with excessive moisture recovery in certain contexts [14][15][16]. In other temperate climates, single-room ventilation units with various types of heat exchangers are increasingly installed through the façade of renovated buildings to supply fresh air and limit heat loss. These units provide simple installation and inherent advantages in potential efficiency [17], but their impact on indoor humidity has not been adequately researched and compared to standard systems.

This paper presents a preliminary assessment of the moisture impacts from a single-room ventilation unit with a non-hygroscopic rotary heat exchanger in a renovated Danish apartment. The schedule and rates of residential moisture production are clearly influential, so available literature was reviewed to identify suitable schedules. Using moisture balance equations, simulations yielded the sensitivities of indoor humidity to varying levels of moisture production, infiltration, heat exchanger efficiency, and room temperature for ventilation units serving individual rooms or whole dwellings. Since the focus was primarily single-room ventilation, the results compared the rotary heat exchanger to recuperative heat exchangers that do not transfer moisture. If the single-room rotary unit could not meet requirements for temperature efficiency and avoid moisture issues, then the results favored recuperative heat recovery instead.

2 Methods

Simulations applied moisture balance equations to simplified airflows in a renovated apartment in Denmark. The simulations sought to determine the impact on indoor moisture conditions of single-room ventilation with a non-hygroscopic rotary heat exchanger.

2.1 Apartment Description

The simulated apartment assumed new windows and improved sealing to obtain an infiltration air change rate of 0.05 h^{-1}. The gross area of the apartment was 77 m^2, which is the average for low-rise social housing in Denmark [18]. Social housing comprises the largest share of multi-story dwellings in Denmark. Simulated rooms were 2.6 m in height, and Table 1 lists individual room areas. The interior floor area was 67.5 m^2, and Figure 3 shows the floorplan based on an actual apartment. The layout of the apartment assumed that all rooms had access to the façade and that air movement between rooms was fully mixed in a central corridor. The average daily occupancy was 14.2 hours on weekdays, which compared to the recommended attendance time of 14 hours per day for Swedish apartments in Johansson *et al.* [19].

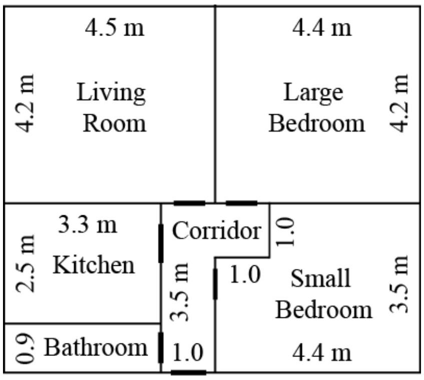

Figure 3. Floorplan of the simulated apartment with interior dimensions. The gross interior and exterior areas were 67.5 m^2 and 77 m^2, respectively. Room heights were 2.6 m.

Table 1. Room summary and occupancy profile for the assumed Danish apartment.

Room Type	Room Area *m²*	Occupancy Schedule *Time interval*	Occupants *No. of adults*
Kitchen	8.3	7:00-8:00 12:00-13:00 17:00-20:00	1
Bathroom	3.0	7:00-9:00	1
Large Bedroom (adult couple)	18.5	22:00-7:00	2
Small Bedroom (child)	14.4	22:00-7:00	0.5
Living Room	18.9	16:00-22:00 (weekdays) 9:00-22:00 (weekends)	1
Corridor	4.4	-	0
Total	67.5	35.5 occupant-hours / weekday (59.2%) 42.5 occupant-hours / weekend day (70.8%)	

2.2 Moisture Production Schedule

Standards and guidelines provide design values for indoor moisture production. This investigation referenced data from BS 5250: Code of practice for control of condensation in buildings [20] and CIBSE Guide A: Environmental Design [21]. However the origins of this data were unclear. Multiple studies have documented moisture production in greater detail. Angell and Olson [22] listed tabular data for individual sources, but some values originated from a study published in 1948 that may be outdated. More recently, TenWolde and Pilon [23] collected and formulated rates and Yik *et al.* [24] comprehensively measured rates for a household in Hong Kong. Reported values have varied substantially, so simulations used three different scenarios to cover greater possibilities.

2.2.1 Scenarios

The best-case scenario assumed the lowest estimated values from references, which often resulted from measures to control moisture sources. This included venting of the washer/dryer to the outdoors, cooking with an electric stove, drying inside a dishwasher, and maximum drainage while showering. The typical scenario assumed common modern appliances, recently measured release rates, and common methods for source control. The worst-case scenario mainly referenced standards and design guidelines. It described a scenario with gas stoves, steam-intensive meals, older appliances, wet mopping, and lengthy showers. The assumed aggregate values for each scenario are listed in Table 2.

Table 2. Assumed aggregate values for the release of indoor moisture sources in the simulated apartment.

				Scenarios		
Activity	***Room***	**Frequency**	***Units***	**Best-case**	**Typical case**	**Worst-case**
Cooking method	*Kitchen*	-	-	Electric / Sealed-gas	Electric / Gas	Gas
Cooking load	*Kitchen*	-	*kg/day*	0.24	1.00 / 2.35	5.06
Dishwasher load	*Kitchen*	*daily*	*kg/day*	0.05	0.15	0.45
Cleaning	*All*	*weekly*	*kg/m2*	0.005	0.005	0.15
			kg/day	0.04	0.04	1.32
Shower load	*Bathroom*	3 *showers/day*	*kg/shower*	0.20	0.35	0.53
			kg/day	0.60	1.40	2.12
Clothes method	-	-	-	Dryer vented to outdoors	Fast spinning wash / Hang dry	Slow spinning wash / Hang dry
Clothes drying load	*Bathroom*	3 *loads/week*	*kg/load*	0	1.67	2.9
			kg/day	0	0.72	1.24
Plants	*Living*	*Continuous*	*kg/day*	0	0.06	0.45
Pets	*Living*	*Continuous*	*kg/day*	0	0.12	0.41

2.2.2 Cooking

Cooking on a gas stove releases approximately 0.45 kg/h from combustion [23][24] unless it is sealed and vented to the outdoors. The best-case scenario assumed negligible release from breakfast and lunch and 0.24 kg from a warm dinner cooked with an electric stove. TenWolde and Pilon proposed this dinner using data measured by Yik *et al.*. This scenario also assumed negligible release from an electric kettle. The typical scenario applied data from Hite and Bray [25] that listed loads from three meals as 0.17 kg, 0.25kg, and 0.58 kg, plus 0.28 kg, 0.32 kg, and 0.75 kg from gas combustion. The study noted the wide variability of moisture loads from different meals. The worst-case scenario used measured moisture loads by Yik *et al.* for a family of four in Hong Kong. The loads were approximately 0.25 kg, 0.95 kg, and 3.8 kg for three meals, which included gas combustion.

2.2.3 Dishwashing

Modern washers heat dishes to evaporate moisture and allow vapor to condense on interior surfaces. They include a sensor to indicate complete drying, so minimal moisture remains. The best-case scenario assumed 0.05 kg release from dishwashing. The typical scenario assumed a release of 0.15 kg/day, which agreed with the measured value for hand-washing and drying of 0.144 kg/day by Yik *et al.* as well as the recommended minimum of 0.15 kg/day in CIBSE Guide A. The worst-case scenario assumed 0.45 kg/day, which the CIBSE guide provided as a maximum.

2.2.4 Cleaning

Yik *et al.* measured a release rate from mopping of only 5 g/m^2. Modern mops use microfiber pads and deposit minimal moisture, so the best-case and typical scenarios assumed 5 g/m^2 once per week. The worst-case scenario assumed a value 150 g/m^2 as reported by Hite & Bray and repeated in BS 5250. Simulations assumed that carpets and furniture covered 20% of the floor area.

2.2.5 Showering

Angell and Olson referred to a study from 1985 that estimated the moisture release from a 5 minute shower as 0.25 kg. The estimate did not seem to include all drying, including towels, spillage, bath mats, or hair drying. A study by Unilever N.V. in 2011 determined the average shower length in the UK to be 8 minutes using embedded sensors in shower heads [26]. Yik *et al.* calculated the moisture release from a shower to be 0.53 kg based on ventilation rate, sensor data, and a moisture balance, and their surveyed respondents reported an average shower length of 18 minutes.

In the simulated apartment, the best case scenario assumed 0.25 kg per shower based on the load for a 5-minute shower cited by Angel & Olson. The typical case scenario estimated 0.40 kg per shower based on the same rate over an 8-minute shower as measured by Unilever. The worst case scenario assumed the rate of 0.53 kg per shower as measured by Yik *et al.*.

2.2.6 Washing and drying of clothes

CIBSE Guide A provided outdated release rates of 0.5-1.8 kg for clothes washing and 5-14 kg for drying. Yik *et al.* measured undrained moisture in a clothes washer and regarded it as negligible, so this investigation only considered drying. Hite and Bray reported hand-wringing laundry and measured 12 kg of moisture in a single load. Improvements to washing machines have allowed faster speeds and greater drying. Angell and Olson reported 2.2-2.95 kg per load in 1988 and Yik *et al.* measured 1.66 kg per load in 2004, which may reflect these improvements. Apartments often lack space for a dryer, so Yik *et al.* hung their clothes to dry and measured the release rate over time.

The moisture release from drying laundry depends on the method applied. In this investigation, the best-case scenario assumed full source control and no moisture release. The typical scenario used the total release from Yik *et al.* and assumed similar rates that decreased linearly over 10 hours, as shown in Figure 4. The worst case scenario assumed hang-drying of a wetter load that released 2.9 kg over the same time span.

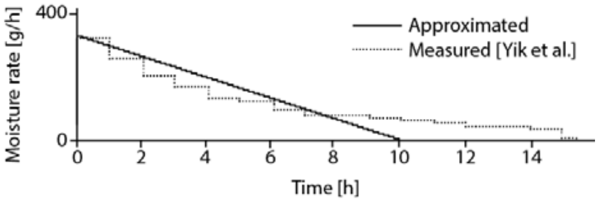

Figure 4. The simulated release of 1.66 kg of moisture from a load of laundry hung to dry based on measurements by Yik et al..

2.2.7 Plants

The best case scenario assumed that the apartment did not contain plants. The typical scenario assumed three plants at 2.5 g/h per plant based on the explanation by TenWolde and Pilon. The worst case scenario assumed seven average sized plants and a total release of 20 g/h as listed by Angell & Olson.

2.2.8 People and pets

TenWolde and Pilon calculated moisture release from an adult person as 0.03 to 0.07 kg/h. This agreed with other reported rates, including 0.04-0.1 kg/h in Yik *et al.* and CIBSE Guide A as well as 0.04 to 0.055 kg/h in BS 5250. For all scenarios, this investigation assumed that an adult of 70 kg released 0.06 kg/h and a child released half this rate. The release for pets was constant and assumed the same release per mass as adults. Their masses were 0 kg, 6 kg, and 20 kg in the best-case, typical, and worst-case scenarios, respectively.

2.3 Moisture Limits

The authors could not specify exact limits to prevent moisture issues due to uncertainty regarding surface temperatures, building materials, and cleanliness. The analysis instead used standards and approximate limits.

2.3.1 Dryness

Reinikainen and Jaakkola [27] studied the impact of relative humidity on human health and determined that low relative humidity can provoke skin symptoms, nasal dryness, and congestion. The standard EN 15251 for indoor climate stated that less than 15%-20% RH can cause these symptoms. The standard recommended greater than 20% RH to achieve the minimum category of air quality and greater than 30% RH to achieve the best category.

2.3.2 Mold Growth

After a comprehensive study, Rowan *et al.* [28] recommended that local surface relative humidity be kept below 75% to limit fungal growth. Johansson *et al.* [29] provided a range of limits above 75% to account for material type and cleanliness. Vereecken & Roels [30] reviewed prediction models for mold growth and found that multiple models used a critical surface RH of at least 80%. These studies demonstrated the variability of mold prediction and risk assessment.

With inexact limits on room RH, analyses can gauge relative mold risks with either the degree or the duration of violated limits. ASHRAE Standard 160:2009 attempts to evaluate both with one simple measure by limiting the maximum 30-day moving-average of surface relative humidities to 80% [31]. Surface temperatures depend on local effects, such as convective heat transfer coefficients, thermal transmittance of building components, and indoor and outdoor temperatures. Consequently, the minimum surface temperature in each room may be highly uncertain. During the heating season, a thermostat controls the average air temperature in each room, which increases its certainty. Simulations may assume fully mixed room air, which enables a simple and accurate calculation of room RH for known air temperatures. To simplify analysis, this investigation estimated an approximate limit on room RH using an 80% limit on surface RH. The author assumed a 1.5°C temperature difference between the room air and the coldest interior surface. For fully mixed air, an increase in air temperature of 1.5°C roughly corresponds to a 10% decrease in RH, so the author estimated a limit of 70% for room RH. Analysis evaluated the 30-day moving-average of room RH against this limit. The results section displays the maximum annual value in each room during the heating season, which indicates a compliance or violation of this limit.

The evaluation only considered surface relative humidities in the heating season since the summer period provides uncertain conditions. Most Danish apartment buildings turn off space heating in summer periods and do not use active cooling, so a lack of thermostatic control provides varying indoor air temperatures. Additionally, higher outdoor temperatures result in warmer interior surfaces, which also depend on the thermal inertia of the building construction. Lastly, the outdoor moisture content is high in summer and may dominate other influences. Many occupants open their windows in the summer period, which increases this effect.

Maximum 30-day moving average RH may roughly correspond to a steady state. Figure 1 shows that indoor humidity does not affect the drying capacity of ventilation when the exhaust is saturated in an uncoated rotary heat exchanger, and all simulated airflows may be fairly constant. However steady state simulations cannot capture the effects of fluctuating indoor RH, and Section 2.2 showed that indoor moisture sources vary over time. Since mold only grows above critical limits, dynamic simulations can improve risk characterization by quantifying the total duration above limits. This ensures that results are not disproportionately influenced by warmer months with high outdoor moisture content. Simulations of these months carry the greatest uncertainty due to the aforementioned issues in the summer period. The duration above limits captured the cumulative risk for the whole heating season. This allowed a visual representation of the relative influence from varied parameters.

2.3.3 Dust Mites

Dust mites require relative humidity above 45%-50% and multiply faster at higher levels [32]. They feed on dust that is abundantly available in beds and carpets, so relative humidity is the primary factor driving their growth [7]. As a result, indoor air should be maintained below 50% during the heating season, particularly in bedrooms and living rooms.

2.4 Moisture Balance Equations

The authors derived and simulated balance equations to describe the properties and dynamics of moisture flows in a renovated Danish apartment. Simulations used Matlab software to perform calculations with time steps of 10 minutes. Figure 5 shows the steps of the simulation and their associated equation numbers.

2.4.1 Weather data

The simulation imported hourly data from the 2013 Danish design reference year and copied it into 10 minutes intervals to capture the dynamic effects from short and intense moisture sources. The imported values included ambient air temperature, relative humidity, and pressure. Table 3 shows the quartiles of hourly values of temperature and relative humidity for the months of January, April, July, and October.

	Temperature					Relative Humidity				
Month	Min.	25%	50%	75%	Max.	Min.	25%	50%	75%	Max.
January	-8	0	1	3	5	58	84	91	96	100
April	0	4	7	10	20	34	67	78	87	100
July	9	15	18	20	28	38	63	79	91	98
October	1	8	10	12	16	62	82	89	94	100

Table 3. The minimum, 1st quartile, median, 3rd quartile, and maximum of hourly values of temperature and relative humidity in the Danish design reference year for the months of January, April, July, and October.

2.4.2 Infiltration

In the simulated apartment, the nominal infiltration rate was only 0.05 air changes per hour since infiltration should be minimized to warrant investment in heat recovery [33]. This assumed that infiltration rate was constant and proportional to room volume. In reality, various factors influence infiltration, such as wind pressure, indoor temperature, ventilation flows, and leakage in the building envelope [34], so it may not be uniformly distributed.

2.4.3 Ventilation requirements

The minimum ventilation rate was 0.5 air changes per hour, as recommended in a multidisciplinary review of literature on ventilation and health by Sendell *et al*. based on limited data [35]. Danish regulations require exhaust capacity in kitchens and bathrooms of 20 L/s and 15 L/s respectively, so simulations assumed these maximum rates. The ventilation rate in kitchens and bathrooms underwent a controlled increased from minimum to maximum capacity based on indoor relative humidity. The proportional increase occurred from 50% to 70% RH, which took the following form in simulations:

$$N_{vent,room,i} = N_{vent,room,min} + \left[\left(N_{vent,room,max} - N_{vent,room,min}\right) \cdot min\left\{1, max\{\varphi_{room,i} - 50\%, 0\}/(70\% - 50\%)\right\}\right] \qquad (1)$$

where $\varphi_{room,i}$ is the relatively humidity of the *room* at time step *i*, and $N_{vent,room}$ is the minimum, maximum, or variable ventilation air change rate in *room* denoted by *min*, *max*, and *i*, respectively. Simulations compared room-based ventilation to whole-dwelling ventilation to assess the impact of a local rotary heat exchanger in each room.

2.4.4 Room-based ventilation

Room-based ventilation was balanced and assumed no exchange of air between rooms. Simulations applied the following moisture balance equations for each room:

$$\left[m\frac{dx}{dt}\right]_{room} = G_{sources}(t) + G_{in}(t) - G_{out}(t) \qquad (2)$$

where *m* is the mass of dry air, *x* is the moisture content per mass of dry air, *G(t)* are mass flows of moisture at time *t*, *in* and *out* denote inward and outward airflows respectively, and the subscript *sources* denotes moisture from indoor sources. Expanding Eq. (2) yielded

$$(\rho V)_{room}\frac{dx}{dt} = G_{sources}(t) + N_{inf}(\rho V)_{room}[x_{amb}(t) - x_{room}(t)] + N_{vent,room}(t)(\rho V)_{room}[x_{sup}(t) - x_{room}(t)] \qquad (3)$$

where ρ and V are the dry air density and volume respectively, N_{inf} is the air change rates per time increment *dt* from infiltration, and the subscripts *amb* and *sup* denote ambient air and supply air respectively. Eq. (3) was discretized and took the following form for simulated iterations:

$$x_{room,i+1} = x_{room,i} + \frac{G_{room,i}}{(\rho V)_{room}} + N_{inf}\left(x_{amb,i} - min\{x_{room,i}, x_{sat,room}\}\right) + N_{vent,room,i}\left(x_{sup,room,i} - min\{x_{room,i}, x_{sat,room}\}\right) \qquad (4)$$

where $x_{room,i}$ is the moisture content in mass of water (i.e. vapor and condensation) per mass of dry air at the start of time step *i*, N_{inf} and $N_{vent,room,i}$ are the air change rates per time step, $x_{sat,room}$ is the saturation moisture content of room air, and $G_{room,i}$ is moisture release in *room* during time step *i*. Infiltration air change rates were specified at dry air densities and indoor air temperatures.

2.4.5 Whole-dwelling ventilation

For ventilation of whole-dwellings, the term *dry rooms* describes bedrooms and living rooms while *wet rooms* describes kitchens and bathrooms. Fresh air enters dry rooms and exhaust exits from wet rooms. The moisture balance equations for the whole-dwelling were similar to Eq. (4), but the exhaust from dry rooms mixed completely in the corridor and entered wet rooms as supply air. Simulations assumed that the flow rate from each dry room was proportional to its volume, which took the form of

$$N_{vent,room,i} = \frac{V_{room}}{\sum V_{dry}} \cdot \sum N_{vent,wet,i} \ \forall\, room \subset dry \qquad (5)$$

where *dry* and *wet* denote subsets of dry and wet rooms respectively. This enabled a calculation of the moisture content of the mixed exhaust from dry rooms, $x_{dmix,i}$, at the beginning of each iteration as

$$x_{dmix,i} = \frac{\sum(V_{room} \cdot x_{room,i})}{\sum V_{room}} \ \forall\, room \subset dry \qquad (6)$$

The equation for the moisture content of the supply air to wet rooms followed as $x_{sup,wet,i} = x_{dmix,i}$. The exhaust flows from wet rooms provided a combined minimum air change rate of 0.5 h^{-1} for the entire apartment. The simulation assumed that the minimum exhaust airflows from each wet room kept the same proportion as their maximum capacities. Therefore the 107.8 m^3/h divided into minimum rates, $N_{vent,wet,min}$, of 46.2 m^3/h and 61.6 m^3/h for the bathroom and kitchen respectively. Similar to the room-based ventilation, the whole-dwelling ventilation increased exhaust from the bathroom and kitchen up to their capacities, $N_{vent,wet,max}$, based on relative humidity. Moisture and sensible heat were transferred from mixed exhaust to mixed supply. The equation for the moisture content of mixed wet room exhaust, $x_{wmix,i}$, took the form:

$$x_{wmix,i} = \sum\left(N_{vent,wet,i} \cdot x_{wet,i}\right) / \sum N_{vent,wet,i} \qquad (7)$$

2.4.6 Variable calculations

The August-Roche-Magnus formula calculates the saturation vapor pressure of air as

$$e_{sat} = C_1 exp\left(\frac{A_1 T}{B_1+T}\right) = 610.94\ Pa \times exp\left(\frac{17.625T}{243.04°C+T}\right) \quad (8)$$

where T is air temperature, e is the partial pressure of water vapor in air, and the subscript *sat* denotes saturation. Alduchov and Eskridge [36] suggested coefficients of A_1 = 17.625, B_1 = 243.04°C, C_1 = 610.94 Pa, which provide accuracy within 0.4% over the range -40°C to 50°C.

The equation $\varphi=100\cdot e/e_{sat}$ relates relative humidity, φ, to partial vapor pressure. Inserting Eq. (8) yielded:

$$e = e_{sat}\frac{\varphi}{100} = 610.94\ Pa \times exp\left(\frac{17.625T}{243.04°C+T}\right) \times \frac{\varphi}{100} \quad (9)$$

At initialization, simulations calculated $e_{sat,room}$ and $e_{amb,i}$ for all time steps. Simulations also calculated the ambient moisture content, $x_{amb,i}$, for all time steps. The moisture content, x, is the mass ratio of water vapor to dry air given by

$$x = \frac{M_{H_2O}}{M_{dry\ air}} = \frac{18.0\ g/mol \cdot \frac{e}{RT}}{29.0\ g/mol \cdot \frac{p-e}{RT}} = 0.622\frac{e}{p-e} \quad (10)$$

where M is the molar mass, p is the total barometric pressure, e is the vapor pressure as calculated above, and R is the universal gas constant.

Simulations then performed iterations for each time step of ten minutes. The indoor moisture content, $x_{room,i}$, was allowed to exceed saturation, and the surplus moisture represented condensation on surfaces that immediately evaporated when possible. The simulation used moisture content from the previous iteration and limited relative humidity to 100%. Each iteration calculated relative humidity from the moisture content with the following equation:

$$\varphi_{room,i} = min\{x_{room,i}/x_{sat,room}, 100\} \quad (11)$$

Simulations then used the following formula from Lawrence [37] to calculate the dew point in each room, $T_{dp,room,i}$,:

$$T_{dp,room,i} = \frac{B_1 ln\left(\frac{e_{room,i}}{C_1}\right)}{A_1 - ln\left(\frac{e_{room,i}}{C_1}\right)} = \frac{B_1\left[ln\left(\frac{\varphi_{room,i}}{100}\right)+\frac{A_1 T_{room}}{B_1+T_{room}}\right]}{A_1 - ln\left(\frac{\varphi_{room,i}}{100}\right)-\frac{A_1 T_{room}}{B_1+T_{room}}} = \frac{243.04°C\left[ln\left(\frac{\varphi_{room,i}}{100}\right)+\frac{17.625\,T_{room}}{243.04°C+T_{room}}\right]}{17.625 - ln\left(\frac{\varphi_{room,i}}{100}\right)-\frac{17.625\,T_{room}}{243.04°C+T_{room}}} \quad (12)$$

where $\varphi_{room,i}$ and T_{room} are the relative humidity and temperature in the room respectively.

Simulations then used temperature efficiency and temperature differential to calculate the exhaust temperature leaving the heat exchanger. Exhaust had a lower limit of 0.5°C to avoid freezing inside the heat exchanger. Additionally, the exhaust temperature could not exceed the indoor room temperature, so it was determined by

$$T_{exh,i} = min\left\{T_{room}, max\{T_{room} - \eta_{exh}(T_{room} - T_{amb,i}), 0.5°C\}\right\} \qquad (13)$$

where η_{exh} is the exhaust temperature efficiency, which was assumed to be equal to the supply temperature efficiency**.**

If the exhaust was warmer than the dew point temperature inside the room, heat recovery was a dry process. If the exhaust was colder, then it was saturated and vapor condensed inside the heat exchanger. Simulations of the rotary heat exchangers assumed that all condensation evaporated into the supply air, and equations for moisture content were

$$x_{exh,room,i} = \begin{cases} x_{sat,exh,i} = 0.622\frac{e_{sat,exh,i}}{p_{atm}-e_{sat,exh,i}}, & T_{exh,i} \leq T_{dp,room,i} \\ min\{x_{room,i}, x_{sat,room}\}, & T_{exh,i} > T_{dp,room,i} \end{cases} \qquad (14)$$

$$x_{sup,room,i} = \begin{cases} x_{amb,i} + min\{x_{room,i}, x_{sat,room}\} - x_{exh,room,i}, & T_{exh,i} \leq T_{dp,room,i} \\ x_{amb,i}, & T_{exh,i} > T_{dp,room,i} \end{cases} \qquad (15)$$

where the subscripts *exh*, *sup*, and *amb* denoted exhaust, supply, and ambient air respectively.

Simulations of recuperative heat recovery assumed that all condensate drained from the heat exchanger. Therefore the equations for moisture content of exhaust were the same as Eq. (14) and (15), except that the supply air had $x_{sup,i} = x_{amb,i} \; \forall \; T_{exh,i}$.

For both types of heat recovery, each iteration lastly updated moisture content according to Eq. (4).

SIMULATION STEPS	**Variables** $\varphi_{room,i}$ $e_{room,i}$ $x_{room,i}$ $T_{dp,room,i}$ $x_{exh,room,i}$ $x_{sup,room,i}$ $N_{vent,room,i}$ $e_{exh,room,i}$ $x_{dmix,i}$ $x_{wmix,i}$ $\varphi_{wmix,i}$ $T_{dp,wmix,i}$
Parameters $G_{room,i}$ T_{room} η N_{inf} $T_{amb,i}$ $\varphi_{amb,i}$ V_{room} $N_{vent,room,min/max}$	

Initial equations	**Eq. #**
1: $\{T_{room}\ \varphi_{room,1}\} \rightarrow e_{room,1}$	(9)
2: $e_{room,1} \rightarrow x_{room,1}$	(10)
3: $T_{room} \rightarrow e_{sat,room} \rightarrow x_{sat,room}$	(8,10)
4: $\{T_{room}\ T_{amb,i}\ \eta\} \rightarrow T_{exh,i}\ \forall\ i$	(13)
5: $T_{exh,i} \rightarrow e_{sat,exh,i}\ \forall\ i$	(8)
6: $\{T\ \varphi\}_{amb,i} \rightarrow e_{amb,i} \rightarrow x_{amb,i}\ \forall\ i$	(9,10)

Single-room vent. iteration	**Eq. #**
1: $x_{room,i} \rightarrow \varphi_{room,i} \rightarrow T_{dp,room,i}$	(11,12)
2: $T_{exh,i} < T_{dp,room,i} \rightarrow x_{exh,room,i},\ x_{sup,room,i}$	(14,15)
3: *Wet room iteration* $\{room \subset wet\}$	
$\varphi_{room,i} \rightarrow N_{vent,room,i}$	(1)
4: $\{x\ G\ N_{vent}\ x_{sup}\}_{room,i} \rightarrow x_{room,i+1}$	(4)

Whole-dwelling vent. iter.	**Eq. #**
1: $x_{room,i} \rightarrow \varphi_{room,i}$	(11)
2: *Mixed dry room exhaust*	
$x_{room,i} \rightarrow x_{dmix,i}\ \forall\ room \subset dry$	(6)
3: *Wet room iteration* $\{room \subset wet\}$	
$\varphi_{room,i} \rightarrow T_{dp,room,i}$	(12)
$\varphi_{room,i} \rightarrow N_{vent,room,i}$	(1)
$x_{dmix,i} = x_{sup,room,i}$	
4: *Mixed wet room exhaust*	
$\{N_{vent}\ x\}_{wet,i} \rightarrow x_{wmix,i}$	(7)
$x_{wmix,i} \rightarrow \varphi_{wmix,i} \rightarrow T_{dp,wmix,i}$	(11,12)
$T_{exh,i} < T_{dp,wmix,i} \rightarrow x_{wmix,i}$	(14)
5: *Dry room iteration* $\{room \subset dry\}$	
$T_{exh,i} < T_{dp,wmix,i} \rightarrow x_{sup,room,i}$	(15)
$N_{vent,wet,i} \rightarrow N_{vent,room,i}$	(5)
6: $\{x\ G\ N_{vent}\ x_{sup}\}_{room,i} \rightarrow x_{room,i+1}$	(4)

Figure 5. Schematic of simulation steps, including equation numbers for reference. The simulation declared variables for all time steps *i*, specified parameters, calculated the initial equations, and then performed the iterations. The iterations only show variables.

2.4.7 Heat recovery

The 2020 Danish building regulations will require 85% temperature efficiency for ventilation serving single dwellings. The authors obtained similar efficiencies for a range of flow rates using a prototype single-room ventilator with a rotary heat exchanger intended for use in existing apartments [4]. Its modelled temperature efficiency accounted for leakage and predicted 90% to 78% for balanced flow rates of 3.6 L/s to 13.0 L/s respectively, and experiments agreed adequately despite some uncertainty.

To enable a comparison with whole dwelling ventilation, the temperature efficiency was set to 85% for all flow rates in both cases. In reality, higher flow rates result in decreased temperature efficiencies and even greater drying capacity. In this investigation, only the kitchens and bathrooms allowed variable fan flow. With whole-dwelling ventilation, the allowable range of flow rates was much smaller because 0.5 h^{-1} applied to the whole apartment and required at least 30 L/s. Conversely, the minimum flow rates with single-room ventilation were much smaller as the air change rate applied to each room. To further simplify analysis, heat recovery only operated in the heating season, which ran from September 16th to May 15th in the simulation.

2.5 Parameter Variations

Simulations varied sensitive parameters to demonstrate the impact of different conditions. Based on the moisture balance equations, the authors identified infiltration, heat exchanger efficiency, and room temperature as three potentially influential parameters. Their standard values were 0.05 h^{-1}, 85%, and 22°C, respectively. Variable moisture sources were also influential, but their influence was assessed through the three scenarios described in Section 2.2.1.

3 Results

The following section shows the results of the reference case, which simulated recuperative heat recovery with the typical moisture production scenario. The subsequent section shows the results of the test case, which simulated a rotary heat exchanger with all moisture production scenarios. In all figures the dashed lines represents the standard case as listed in Section 2.5, which is used with other parameter variations.

3.1 Recuperative Heat Recovery

Ventilation with recuperative heat recovery provided the reference case for comparison. With the typical moisture production scenario, Table 4 shows the minimum moving-average relative humidities for ventilation serving single-rooms or whole-dwellings. The table compares these values to the recommended design minimum in standard EN 15251 of 15%-20%. The data represents the standard simulation with 85% temperature efficiency, an infiltration rate of 0.05 h^{-1}, and an indoor temperature of 22°C. The results indicate that the relative humidity in the living room and bedrooms may be insufficient for short durations with recuperative heat recovery.

Table 4. Minimum moving-average relative humidities with the standard simulation parameters and recuperative heat recovery.

Ventilation type	EN 15251 Annex B.3 Criteria	Minimum moving average	Minimum moving average RH in heating season [%]				
			Kitchen	Bathroom	Large bedroom	Small bedroom	Living room
Single-room	> 15-20%	1-day	26	16	19	13	15
		7-day	28	26	22	16	18
		30-day	32	30	26	21	23
Whole-dwelling	> 15-20%	1-day	20	16	12	11	11
		7-day	22	23	15	14	14
		30-day	27	28	20	19	19

With the standard simulation parameters, Table 5 shows that the maximum 30-day moving averages did not exceed 60% RH. All values were less than the estimated limit of 70% room RH, which implied that mold risk was not an issue. As described in Section 2.3, the authors assumed equivalence between this room air RH limit and the 80% surface RH specified in ASHRAE 160. Figure 6 shows the percentage of time steps with greater than 70% RH for each ventilated zone with the typical moisture production scenario. Ventilation with recuperative heat recovery adequately removed moisture from all rooms for both ventilation types. In terms of varied parameters, temperature efficiency did not influence indoor relative humidity, and infiltration had a very minor effect over the simulated range. Cooler room temperatures provided slightly higher relative humidities, but none of the simulated cases provided 30-day moving-averages greater than 70% room RH.

Table 5. Maximum 30-day moving-average relative humidities with standard simulation parameters and recuperative heat recovery.

Ventilation Type	Maximum Moving Average	ASHRAE Surface Limit	Adjusted Room Limit	Kitchen [%]	Bathroom [%]	Large Bedroom [%]	Small Bedroom [%]	Living Room [%]
Single-room	30-day	< 80%	< 70%	57	56	56	51	53
Whole-dwelling	30-day	< 80%	< 70%	57	57	49	48	49

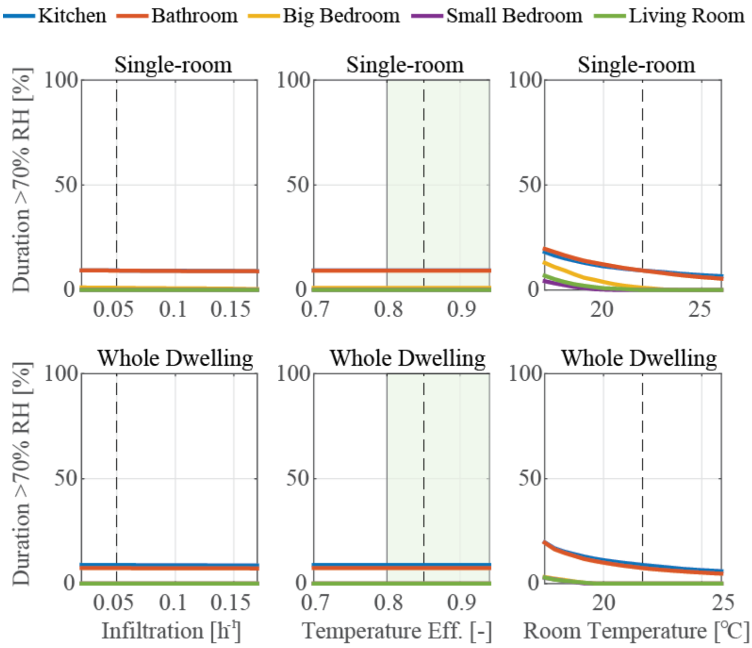

Figure 6. Recuperative heat recovery. Duration curves for the percentage of time steps with greater than 70% room RH for simulations with varied parameters.

3.2 Rotary Heat Exchanger

Results compared single-room ventilation with a rotary heat exchanger to the reference case. The results include simulations of whole-dwelling ventilation with a rotary heat exchanger for supplemental reference.

3.2.1 Best case scenario

At the nominal conditions in this scenario, the moving average relative humidities never exceeded the limits of ASHRAE 160 for any of the simulated cases, even after applying the 10% deduction described in Section 0. This standard protects against mold growth [38], so any concerns about dust mites still applied.

Figure 7 presents the results of simulations for single-room ventilation with the best-case moisture scenario. Air did not mix between rooms, so results were distinct. Only the bathroom and large bedroom provided potential concerns. In reality these two rooms have very different critical humidities, so they cannot be directly compared using this evaluation. As described in Section 2.3, dust mites proliferate in fabrics at relative humidities greater than 50%, whereas the interior surfaces of bathrooms may be resistant to mold growth, which raises their critical humidity. As such, the high humidity in bedrooms was more concerning.

Figure 7 also presents the results of rotary heat exchanger with the whole-dwelling ventilation system. The results were similar to the reference case with recuperative heat recovery. In this system, moisture transfer applied to the bulk properties of the mixed supply and exhaust airflows so recovered moisture was distributed evenly throughout the dwelling.

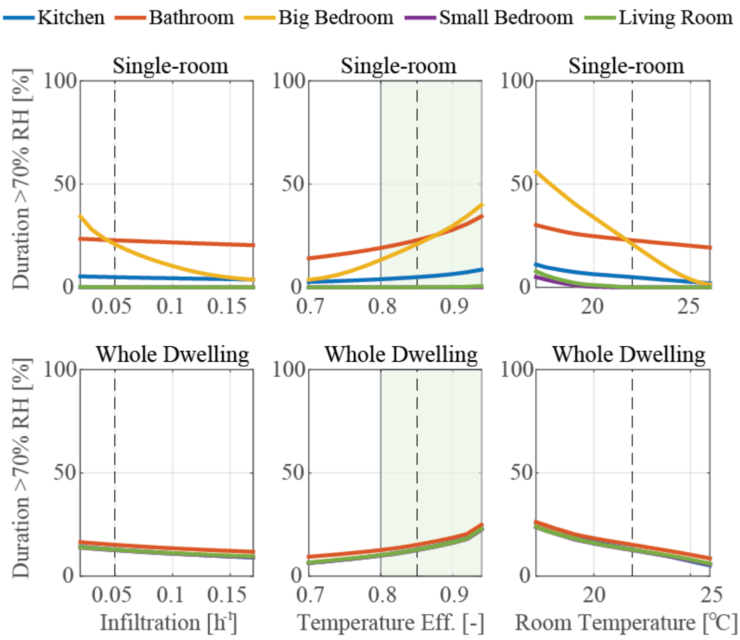

Figure 7. Simulation of regenerative heat recovery with the best-case moisture production scenario. Duration curves for percentage of time steps with greater than 70% room RH for simulations with varied parameters.

3.2.2 Typical scenario

With the typical moisture production scenario, Table 6 shows the minimum moving-average relative humidities for ventilation serving single-rooms or whole-dwellings with a rotary heat exchanger. Compared to the reference case with recuperative heat recovery, nearly all simulations provided better categories of relative humidity according to standard EN 15251. This demonstrates a potential benefit of moisture recovery.

Table 6. Minimum moving-average relative humidities with the standard simulation parameters and a rotary heat exchanger.

			Minimum moving average RH in heating season [%]				
Ventilation type	**EN 15251 Annex B.3 Criteria**	**Minimum moving average**	**Kitchen**	**Bathroom**	**Large bedroom**	**Small bedroom**	**Living room**
Single-room	1-day	> 20%	43	40	27	13	15
	7-day		48	51	32	16	19
	30-day		53	53	42	21	25

Whole-dwelling	1-day	> 20%	40	39	33	32	33
	7-day		47	47	40	39	39
	30-day		57	57	50	49	50

Table 7 compares the maximum 30-day moving averages to the adjusted ASHRAE limits to predict mold growth at nominal conditions. The single-room ventilation did not violate limits in any dry rooms.

Table 7. Maximum 30-day moving-average relative humidities with standard simulation parameters and a rotary heat exchanger.

Ventilation Type	**Maximum Moving Average**	**ASHRAE Surface Limit**	**Adjusted Room Limit**	**Kitchen [%]**	**Bathroom [%]**	**Large Bedroom [%]**	**Small Bedroom [%]**	**Living Room [%]**
Single-room	30-day	< 80%	< 70%	87	94	64	51	53
Whole-dwelling	30-day	< 80%	< 70%	97	97	91	90	90

Figure 8 shows that all simulations of single-room ventilation, including parameter variations, provided excessive humidity in kitchens and bathrooms with this moisture scenario.

In the best-case scenario reported above, whole-dwelling ventilation provided less risk of excessive humidity by combining airflows and evenly distributing recovered moisture to all rooms. In the typical scenario, the same mixing recovered moisture to all rooms, but the contributions from wet rooms were much more significant.

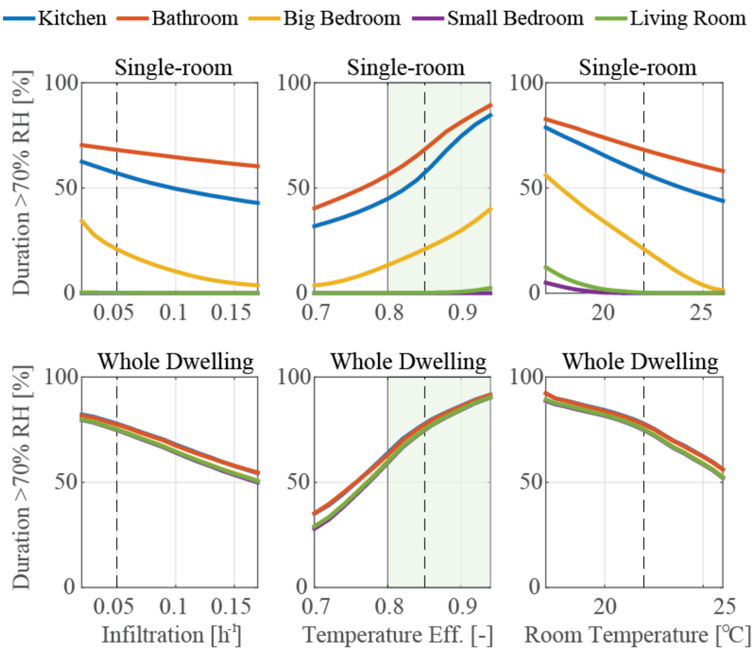

Figure 8. Simulation of regenerative heat recovery with the typical moisture production scenario. Duration curves for percentage of time steps with greater than 70% room RH for simulations with varied parameters.

Figure 9 shows the cumulative distribution curve for indoor RH during representative months to assess seasonal differences. A rightward or downward shift provided an unfavorable change in RH. The curves are relatively similar in all the displayed months. However January provided the least favorable conditions for the kitchen and bathroom and the most favorable conditions for the small bedroom and living room. Humidity in the adult bedrooms was the most critical in October.

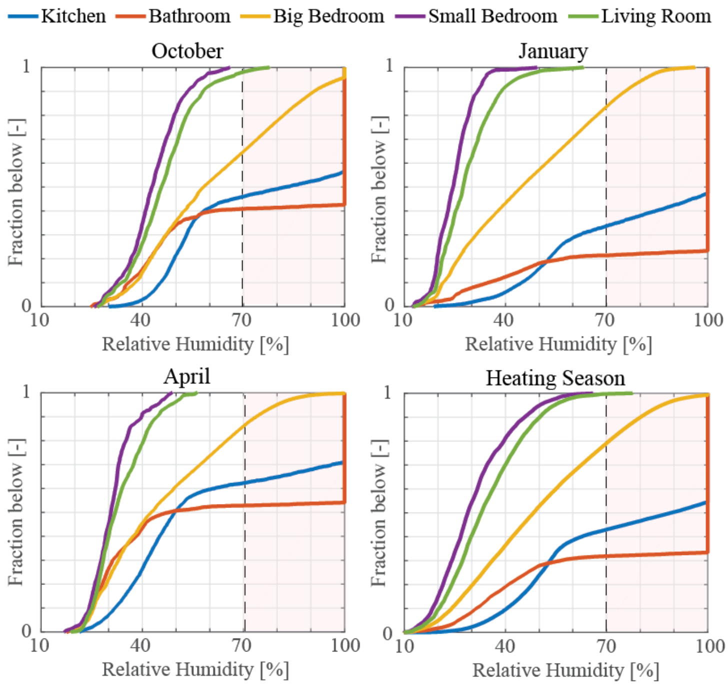

Figure 9. Cumulative distribution function of indoor relative humidities in each room during the months of October, January, April, and the whole heating season with a rotary heat exchanger in single-room ventilation and the typical moisture production scenario.

3.2.3 Worst-case scenario

With the worst-case moisture scenario, Figure 10 shows that ventilation serving only wet rooms provided an extremely high mold risk, but ventilation serving dry rooms yielded a moderate risk. With nominal parameters in the worst-case scenario, all 30-day moving averages exceeded the limits from ASHRAE 160 except for the case of the living room and bedrooms with single-room ventilation, which exceeded none. Whole-dwelling ventilation with a rotary heat exchanger yielded excessive relative humidity for the majority of the heating season in all simulated rooms for all parameter variations.

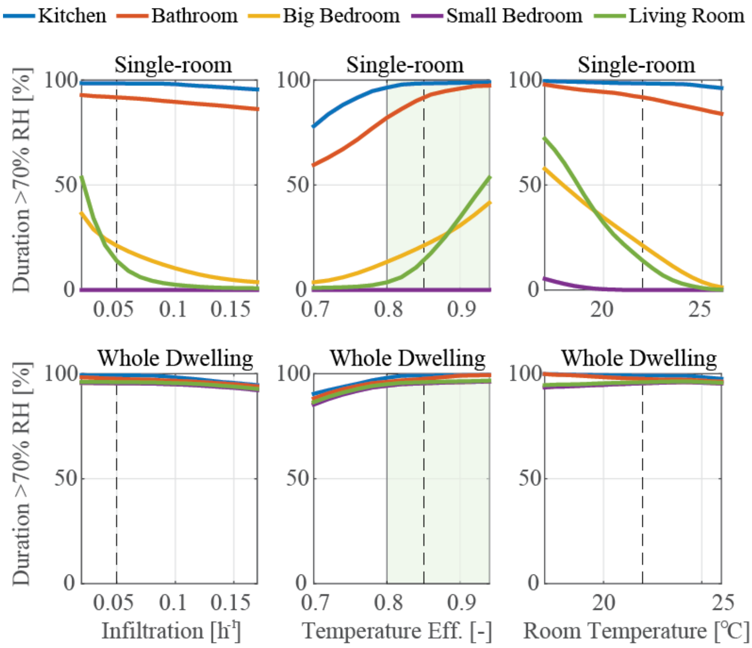

Figure 10. Simulation of regenerative heat recovery with the worst-case moisture production scenario. Duration curves for percentage of time steps with greater than 70% room RH for simulations with varied parameters.

4 Discussion

The results indicated that highly efficient rotary heat exchangers were unsuitable for wet rooms under the assumed conditions due to excessive moisture recovery. The results also indicated that rotary heat exchangers may provide low to moderate mold risk with single-room ventilation of dry rooms for a range of probable conditions.

The authors speculate that an adequate solution could include rotary heat exchangers in dry rooms and recuperative heat exchangers in wet rooms. A rotary heat exchanger transfers condensation to the supply air, so it does not require drainage. It may also prevent negative health impacts from dryness, as indicated by Table 6. Recuperative heat exchangers require drainage, and installation in kitchens and bathrooms would allow easier access to plumbing. This combination utilizes the inherent advantages of each heat exchanger for the specific demands of individual rooms.

The moisture production schedule in dry rooms was similar for all three scenarios, so the rotary heat exchanger consistently provided a low to moderate mold risk with single-room ventilation. This could allow finer adjustments to minimize mold risk. The figures in Section 3.2 demonstrated the clear influence of varied parameters on the duration of excess relative humidity. Interestingly, two of the varied parameters are controllable during operation. This realization yields potential options to adjust relative humidity in dry rooms to maintain appropriate levels. A rotary heat exchanger relies on cyclical regeneration, so a controller could reduce its cyclical speed to reduce heat transfer. Less heat transfer implies greater exhaust temperatures and drying capacity. Similarly, higher room temperatures result in lower relative humidities and mold risk, so a controller could maintain sufficient room temperatures to avoid risk. Both options could negatively affect occupant thermal comfort. The former option could generate local discomfort due to cool draughts from lower supply temperatures, while the latter could affect whole-body comfort with changes in room temperature. However Figure 10 indicates that the required reduction in heat recovery or increase in room temperature may be small to limit relative humidities to acceptable levels.

This paper focused on single-room ventilation, but the same concerns may apply to whole-dwelling ventilation that extracts exhaust from wet rooms. The whole-dwelling simulation included many significant assumptions regarding air flows. It also assumed ambitious infiltration rates and temperature efficiencies, so the results are not conclusive. The results merely suggest that whole-dwelling ventilation with a highly efficient rotary heat exchanger should be researched in greater detail to assess potential issues from moisture recovery.

This investigation simplified implementation with many assumptions. Simulations did not account for moisture buffering from walls, which could dampen variations on indoor humidity and reduce the duration of exceeded limits. Salonvaara *et al.* [39] and Mortensen *et al.* [40] determined that typical interior paints can act as vapor barriers and effectively limit moisture transfer between construction materials and room air, so this assumption was reasonable. However, the simulation did not account for dampening from furniture, books, and textiles, and Svennberg *et al.* [41] measured a reduced daily peak of 10% RH and an increased daily trough of 5% RH after fully furnishing a room. Additionally, simulations did not distinguish between interior surface materials, which provide different resistances to mold growth and different critical humidities. The investigation also assumed approximate surface temperatures, which highly influence surface relative humidities. Greater knowledge of the average Danish apartment could therefore improve the assessment of mold risk with these ventilation systems.

This study also assumed that rotary heat exchangers transfer all condensation in the exhaust to the supply air. This point is commonly advertised by manufacturers to emphasize that drainage is not required. However, Holmberg [10] presents the possibility of excess moisture in the heat exchanger. If cold outdoor air is nearly saturated upon entry to the heat exchanger then condensation may not be able to completely evaporate. This small longitudinal region would then accumulate moisture and its movement is difficult to predict. This study assumed that any accumulated moisture moved to a warmer section of the heat exchanger and evaporated into the supply air. This can only be confirmed experimentally.

5 Conclusion

The investigation constructed and simulated moisture balance equations for a single-room ventilation unit with a non-hygroscopic rotary heat exchanger. Its assessment focused on its moisture impacts in a typical renovated apartment in a humid temperate climate. The rotary heat exchanger recovered excess moisture in kitchens and bathrooms and provided a serious mold risk. The rotary heat exchanger was only suitable for single-room ventilation of dry rooms, such as living rooms and bedrooms. In these rooms, the risk of mold depended on moisture production. The sensitivity analysis concluded that varying heat recovery or indoor temperature could limit indoor relative humidity in dry rooms when a moderate risk was present. The rotary heat exchanger also elevated the minimum moving-average relative humidities, which may help to avoid negative health impacts from dry air. A discussion emphasized the potential benefits of selecting heat recovery to match the needs of individual rooms.

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