The effect of air humidity on the exergy efficiency of domestic heat pumps

Zevenhoven, Ron; Arnas, Özer

Published in:

Publicerad: 01/01/2019

Document Version
Förlagets PDF, även kallad Registrerad version

Please cite the original version:

General rights
Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

Take down policy
If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.
The effect of air humidity on the exergy efficiency of domestic heat pumps

Ron Zevenhoven\textsuperscript{a}, Özer Arnas\textsuperscript{b}

\textsuperscript{a} Åbo Akademi University, Thermal and Flow Engineering Laboratory, Turku, Finland, rzevenho@abo.fi
\textsuperscript{b} U.S. Military Academy at West Point, NY, USA, ozer.arnas@westpoint.edu

Abstract:
Heat pump systems have been used for more than a century in refrigeration and the upgrading of heat to temperature levels demanded by consumers. Private housing in Northern Europe and other countries is moving away from direct electric heating, combustion-based heating and even district heating in favour of heat pumps. These use a cheap renewable heat source and electricity, and the system purchase is motivated by an attractive coefficient of performance (COP), besides the option to reverse the heat pump operation from heating during winter to cooling during summer. Also, industry is increasingly implementing heat pump technology that circumvents the production of CO\textsubscript{2} when producing heat, although reaching sufficiently high temperatures may be challenging. Developments are accelerated by the increasing availability of (cheap) zero-CO\textsubscript{2} electricity. One complication for the assessment of the energy efficiency of heat pump systems using exergy analysis is that besides temperature levels, features of the heat reservoirs, such as humidity of air in a building envelope, become important. In modern buildings, this is affected by exhaust air heat recovery (EAHR) systems that typically replace the air inside a building every few hours. A side-effect of this is a net in- or outflux of humidity. In this paper, an exergy analysis is presented that quantifies energy efficiency of a heat pump system as partly determined by the humidity of a building envelope being heated using a ground source or air source heat pump. Humidity control introduces a significant energy penalty. As shown, an EAHR unit can result in an significantly increased exergy efficiency, adding to the benefits offered by the heat pump, depending on indoor versus outdoor temperature and humidity, and whether a ground source or air source heat pump is used.

Keywords:
Heat pump system, Energy efficiency, Exergy, Domestic heating, Humidity.

1. Introduction
Heat pumps find increased markets for heating of public and private buildings with the 900 000\textsuperscript{th} unit in Finland put into use recently (January 2019), a country with a population below 6 million. While most, \textasciitilde80 \%, of these units use ambient air as the low (during winter) or high (during summer) temperature heat reservoir, \textasciitilde10 \% uses geothermal heat and the remainder uses air ventilation, water bodies or other heat reservoirs [1, 2]. With increased availability of electricity from renewable sources (which on one hand may be intermittent but on the other hand are more
predictable than pricing levels for natural gas) the role of size of district heating for the heating of buildings is being re-evaluated [3]. In principle, heat pumps use a cheap renewable heat source and electricity, and system purchases are often motivated by an attractive coefficient of performance (COP). Also, industry is increasingly implementing heat pump technology that circumvents the production of CO₂ when producing heat, although reaching sufficiently high temperatures may be challenging. A Stirling engine-based heat pump is one approach towards higher output temperatures [4].

From an efficiency point of view the use of electricity for heating may be difficult to motivate, except when costs are low or zero and, as said above, the source is renewable and free of CO₂ emissions. The above-mentioned COP is defined by the heat in- and output, Qₐ at temperature Tₐ (K) and Qₐ at temperature Tₐ (K), respectively, and the power P needed to drive the process:

$$\text{COP}_{HP} = \frac{Q_{\text{out}}}{P} = \frac{Q_{\text{in}} + P}{P} = \frac{Q_{\text{in}}}{P} + 1 = \text{COP}_{R} + 1$$  \hspace{1cm} (1)

Here, HP refers to a heat pump with Qₐ as the target purpose while R refers to a refrigeration process aiming at removing low temperature heat Qₐ. One attractive feature of a heat pump is the option to reverse the heat pump operation from heating during winter to cooling during summer. Typical values for current equipment for building heating application are COPₜₕ = 3-5 depending on the temperatures Tₐ and Tₐ, respectively, of the heat in- and output.

A theoretical maximum exists for the COP, as quantified by Carnot for an adiabatic reversible process. This allows for defining a so-called figure of merit (FOM) that gives an indication of the deviation from theoretically ideal performance of a heat pump:

$$\text{COP}_{\text{rev,HP}} = \frac{T_{\text{out}}}{T_{\text{out}} - T_{\text{in}}} = \text{COP}_{\text{rev,R}} + 1 \quad \text{and} \quad \text{FOM} = \frac{\text{COP}}{\text{COP}_{\text{rev}}}$$ \hspace{1cm} (2a,b)

Typical values for FOM are in the range 0.4 – 0.6 [5].

Energy efficiency is nowadays considered equally important as the sustainability of heat and power generation, since energy not used or needed cannot have costs or detrimental side effects apart from and, not unimportantly, unemployment. A powerful method of using the Second Law of Thermodynamics for quantifying the reversibility of energy conversion processes is an exergy analysis, rating all energy in- and outputs of a process for their capacity to do work [6]. An exergy efficiency ηₑₓ can thus be calculated from the exergies of the in- and output energy, which for a heat pump gives:

$$\eta_{\text{ex. HP}} = \frac{\text{Ex(energy output)}}{\text{Ex(energy input)}} = \frac{\text{Ex(Qₐ)}}{\text{Ex(Qₐ)} + \text{Ex(P)}} = \frac{Q_{\text{out}}(1 - \frac{T_{\text{out}}}{T_{\text{in}}})}{Q_{\text{in}}(1 - \frac{T_{\text{in}}}{T_{\text{in}}}) + P}$$ \hspace{1cm} (3)

where T°(K) is the temperature of the surroundings. All values for heat are multiplied by a Carnot factor (1-T°/T) to give exergy, similar to evaluating the thermal (Carnot) efficiency of a power plant.

Equations (1-3), however, are based on heat and power only and one complication of using a heat pump for heating (or cooling for that matter) a building is that it becomes part of the heating, ventilation and air conditioning (HVAC) system of that building. This brings air quality and specifically the humidity of air inside, entering and leaving a building envelope into the energy efficiency assessment. Thus, the heat pump implicitly is used for keeping temperature but also humidity at a certain level while a thermometer will in practice be used for control, as was also noted in [5]. Exergy analysis has been applied to humidity control and air conditioning in a few earlier studies [7,8], with focus on keeping humidity below a certain level.

Using a heat pump for heating a building to a temperature Tₐ ≈ T° will have a very low exergetic efficiency according to (3), while in fact the heat output required may be increased since it is partly
used for evaporation of water. This applies especially to cold winter days when outside absolute humidity is low and the air conditioning system will have a net effect of sending water vapour out of the building – see also [9]. Thus, the effect of evaporating water, increasing absolute humidity, $\omega$ (kg water / kg dry air) should be distinguished from heating a building space, i.e.

$$Q_{out} = Q_{out,\Delta T} + Q_{out,\Delta \omega} = Q_{out,\Delta T} \left(1 + \frac{Q_{out,\Delta \omega}}{Q_{out,\Delta T}}\right) = Q_{out,\Delta T} (1 + \Omega) \quad (4)$$

Here, $\Omega$ quantifies the (relative) extra heat needed for water evaporation. In this paper, this will be used as starting point for an improved definition of exergy efficiency $\eta_{\text{Ex, HP}}$ using a house in Turku Finland and weather data for year 2018, as given in Figs. 2 and 7. Below, dry air will be noted d.a..

An important feature of the exergy of heat is that it depends on the temperature of the surroundings, $T^\circ$, presenting the question how to separate or at least distinguish the surroundings from the heat reservoirs between which the heat pump is operating. This is extensively discussed in [10], with a discussion on the definition of the “dead state” which defines zero exergy. In this paper, at any time the actual conditions of the surrounding ambient (temperature, pressure, air humidity) is taken as zero exergy reference state.

2. The case study building and weather data

2.1. The case study building

In this paper, the building envelope considered is a 100 m² × 2.75 m volume house in Turku, south-west Finland kept at 22°C with varying inside air humidity. It is ventilated at 0.5 air changes per hour (ACH), corresponding to ~ 0.0382 m³/s (~ 0.0460 kg/s d.a.), with a heat pump used for the heating. Inlet (ambient) air temperature and humidity vary as described below, following weather data. For the air ventilation, cases with and without exhaust air heat recovery (EAHR), rated at 75% [9], are also considered. Only heating, not cooling, is considered here.

The heat pump used is either a ground-source, geothermal, or an air-source heat pump. This gives the four cases illustrated in Fig. 1. For the low temperature heat sources the air-source heat pump used ambient air with temperature varying between -20.0 and 32.8°C, with an hour-based annual average value 7.32 °C. This 7.32°C is taken as the constant low temperature heat source temperature for the ground-source heat pump, located at ~ 100 m depth [11].

![Figure 1. Case studies considered for two different heat pump types, with/without ventilation heat recovery.](image-url)
2.2. Processing weather data for 2018 in Turku, Finland

Hour-averaged data from the Artukainen weather station in Turku [12] is used to calculate absolute humidity, $\omega$, and enthalpy, $h$ for year 2018, defining the inlet air for a building ventilation system during that year. Here, $\omega$ is calculated from temperature $T$ (°C), pressure, $p_{\text{total}}$ (Pa) and relative humidity $\text{RH}$ (%), with water saturation pressure $p_{\text{sat}}$ (Pa) [13]:

\[
\omega = \frac{0.622 \cdot \frac{\text{RH}}{100} \cdot p_{\text{sat}}(T)}{p_{\text{total}} - \frac{\text{RH}}{100} \cdot p_{\text{sat}}(T)}
\]

where $p_{\text{sat}} = 100 \cdot \exp \left( \frac{11.78 \cdot T - 99.64}{T+23} \right)$

(5a,b)

where 0.622 is the molar mass ratio water / dry air = 18 / 29. The specific enthalpy $h$ (kJ/kg d.a.) can be then calculated, combining the properties of air and water vapour using [13]:

\[
h = 1.005 \cdot T + \omega \cdot (2500 + 1.87 \cdot T)
\]

(6)

This gives the values as plotted in Fig. 2 as input data for the calculations given below.

Equation (6) has dry air at 0 °C as the reference point for $h = 0$, explaining the negative values in Fig. 2, for certain combinations of $T$ and $\omega$. Enthalpy $< 0$ occurs when $T < -2500 \cdot \omega / (1.005 + 1.87 \cdot \omega)$.

2.3. Energy balance calculations for the case study building space

Assuming that heat $Q \approx$ enthalpy difference $\Delta H$ where (assuming pressure/volume work $<<$ heat, noting that humid air $\approx$ an ideal gas with enthalpy not depending on pressure), the heat pump output (4) can be rewritten as

\[
Q_{\text{out,}\Delta T} = \dot{m}_{\text{d.a.}} \cdot (1.005 \cdot \Delta T + \omega_{\text{in}} \cdot 1.87 \cdot \Delta T)
\]

\[
Q_{\text{out,}\Delta \omega} = \dot{m}_{\text{d.a.}} \cdot (\Delta \omega \cdot (\Delta h_{\text{vap}} + 2500 + 1.87 \cdot T_{\text{out}}))
\]

with $\Delta \omega = \omega_{\text{out}} - \omega_{\text{in}}$, $\Delta T = T_{\text{out}} - T_{\text{in}}$

and $\dot{m} = \dot{m}_{\text{d.a.}} \cdot (1 + \omega)$

The evaporation heat for water (kJ/kg) vs. $T$ (°C) was calculated using:

\[
\Delta h_{\text{vap}} = 2508.1 - 2.77 \cdot T + 5.173 \cdot 10^{-3} \cdot T^2 - 2.47 \cdot 10^{-5} \cdot T^3
\]

(8)
Using this – as an example for the type of calculations made for the analysis given in section 3 – the enthalpy change to come from inlet air with properties as in Fig. 2 to inside air, eventually ventilated, at 22°C and 50 % RH is given in Fig. 3. It distinguishes between heat needed for heating up the air including the humidity it enters with, and the heat needed to bring water to 22°C and evaporate it, resulting in 50% RH. Temperature control would require heat varying from ~ 35 kJ/kg d.a. during winter to ~15 kJ/kg d.a. during summer while energy requirements for humidity control would vary from – 20 to +20 kJ/kg d.a.

Note that these are calculated numbers for an imaginary situation when aiming at 50% humidity, since a low indoor humidity during winter is generally accepted. This is apparently the result of a limited supply of water to be evaporated, making summer time humidity levels impossible to reach during winter.

**Figure 3.** Enthalpy increase to incoming ambient air for increasing temperature and for increasing humidity, respectively, to 22°C, 50% RH.

Fig. 4 gives the ratio $\Omega = \frac{Q_{\text{out},\Delta \omega}}{Q_{\text{out},\Delta T}}$ as given in (4) to illustrate the relative importance of humidity versus temperature control, with values varying in the range 0 to 1 for most of the year. The HP must supply the extra heat needed for humidity control. During late spring and summer months, however humidity is varying violently, presumably as a result of plants and trees growth (which in southern Finland continues during the then short nights).

As Figs. 3 and 4 also show: at the end of summer, ambient humidity > indoor humidity and the negative enthalpy effect of changing humidity supplies some of the heat needed for maintaining indoor temperature. This could imply moisture condensation inside the building envelope.
2.4. Exergy (efficiency) analysis for the case study building space

The exergy efficiency of changing the temperature and humidity of the indoor air of a building space can be based on the physical and chemical exergy of humid air, as worked out by Wepfer et al. [14]:

\[
e_{\text{phys}} = \left( 1.005 + 1.87 \cdot \frac{\omega}{0.622} \right) \cdot T^0 \cdot \left( \frac{T}{T^0} - 1 - \ln \left( \frac{T}{T^0} \right) \right) + \left( 1 + \frac{\omega}{0.622} \right) \cdot \frac{R}{M_{\text{d.a.}}} \cdot T^0 \cdot \ln \left( \frac{p}{p^0} \right)
\]

and

\[
e_{\text{chem}} = \frac{R}{M_{\text{d.a.}}} \cdot \left( 1 + \frac{\omega}{0.622} \right) \cdot \ln \left( \frac{1 + \frac{\omega}{0.622}}{1 + \frac{\omega^0}{0.622}} \right) + \frac{\omega}{0.622} \ln \left( \frac{\omega}{\omega^0} \right)
\]

with unit kJ/kg d.a. Here, \( R \) is the universal gas constant 8.314 kJ/kmol·K, \( M_{\text{d.a.}} \) is the molar mass of dry air \( \approx 29 \) kg/kmol, with pressure \( p \), temperature \( T \) and absolute humidity \( \omega \). Values with subscript \( ^\circ \) represent the ambient surroundings. In (9), the specific heat constants are those also used in (6).

For the case study of this paper, air enters the building at \( T = T^\circ \), \( \omega = \omega^\circ \) and \( p = p^\circ \), and thus exergy \( \text{Ex}_{\text{inlet air}} = 0 \). Moreover, it is assumed that indoor pressure \( p \approx p^\circ \), putting the most right-hand part of (9) equal to 0. The exergy of humid air at 22°C, 50% RH, at pressure \( p = p^\circ \), is given in Fig. 5, distinguishing between physical and chemical exergy.
The exergy efficiency of the process where a heat pump is used to provide the heat for raising temperature and evaporating water as to increase humidity can thus be defined by:

$$\eta_{Ex, HP} = \frac{E_{phys} + E_{chem}}{E_{Q_{in}} + P} = \frac{E_{phys} + E_{chem}}{Q_{in}(1 - \frac{T_0}{T_{in}}) + P} \approx \frac{E_{phys} + E_{chem}}{P}$$

where $P = \frac{Q_{out}}{COP_{HP}}$ \hspace{1cm} (11a,b)

It is assumed that the exergy of the input heat $<<$ HP input electricity $P$, being equal to (for an air source HP), or very close to (for a ground source HP) the surroundings temperature $T_0$. Neglected is also the exergy of the liquid water needed for the humidity increase $\Delta\omega$.

Below, only the building space conditioned to 22°C, 50% RH is analysed – see [9] for other air conditioning specifications.

3. Exergy efficiency calculation results

3.1. Exergy efficiency based on HP heat flows

From a First Law of Thermodynamics energy efficiency point of view there is no effect of evaporating water as to raise indoor humidity. With the ratio $\Omega$ as in equation (4), the efficiency of energy use can be written as

$$\eta_{HP} = \frac{\text{energy output}}{\text{energy input}} = \frac{Q_{out}}{Q_{in} + P} = \frac{Q_{out}(\Omega=0)}{Q_{in}(\Omega=0) + P(\Omega=0)} \cdot \left(1 + \Omega\right)$$ \hspace{1cm} (12)

This is independent of $\Omega$, i.e. whether the HP must supply more or less heat. That this is an incorrect assessment of efficiency of energy use follows from (7b) where enthalpy of evaporation $\Delta h_{vap}$ implies that some work must be done, as the water evaporation $\Delta\omega$ will result in a volume increase $\Delta v$. This required work is equal to $p^0 \cdot \Delta V$, making the claim $Q = \Delta H$ invalid although $Q \approx \Delta H$ may still hold. Secondly, transferring pure liquid water into a gas phase mixture implies generation of mixing entropy [6], as described by (10) for humid air.

3.2. Exergy efficiency of bringing ambient air to 22°C, 50% RH

3.2.1. Case a: Ground source HP, no exhaust air heat recovery

The input exergy for COP = 4, using (11b) and the exergy efficiency calculated using (11a) are given in Fig. 6. This corresponds to a commercial heat pump available for the case study house (NIBE F-1226-8, max 8 kW rated heat output) which specifies $COP_{HP} = 3.96$ for 7°C at the low temperature heat reservoir. (For clarity, the axes in Fig. 6 are taken the same as for Fig. 8, below).

![Exergy input and exergy efficiency: ground source heat pump](image)

Figure 6. Exergy input) and exergy efficiency of bringing humid air to 22°C, 50% RH, for a ground source heat pump with $COP_{HP} = 4$. (Case a.)
The exergy input values in Fig. 6, disregarding strong variations as a result of strongly varying ambient humidity, are < 11.3 kJ/kg d.a. during the coldest winter days, which corresponds to < 500 W for case study house ventilated at ~ 0.046 kg/s d.a., at a maximum exergy efficiency of ~ 34%. This drops to < 2.2 kJ/kg d.a. during summer, with an exergy efficiency as low as 0.2%.

3.2.2. Case b: Air source HP, no exhaust air heat recovery

A similar calculation can be made for an air source HP, where the low temperature heat source temperature varies significantly. Using a recent study made at our institute (ÅAU), comparing nine brands of commercial air source HPs [15] an averaging expression

\[
COP_{HP} = 3.45 \cdot e^{0.037T}
\]  

(13)

can be produced, for ambient air temperature \(T\)°(°C). Using the weather data [12] this gives the varying \(COP_{HP}\) values shown in Fig. 7, dropping to ~ 2 in winter and rising to ~ 7 during summer. Also shown in the Fig. 7 are \(COP_{HP} = 4\) which is the value used for the ground source HP and the 22°C ambient temperature during which no heating would be needed and a thermostat would put the HP on stand-by.

![COP air source heat pump and ambient temperature](image)

Figure 7. Air source heat pump COP\(_{HP}\) and ambient temperature (Case b & d)

![Exergy input and exergy efficiency: air source heat pump](image)

Figure 8. Exergy input) and exergy efficiency of bringing humid air to 22°C, 50% RH, for an air source heat pump with COP\(_{HP} = f(T)\). (Case b).
The exergy input values in Fig. 8, again disregarding strong variations as a result of strongly varying ambient humidity, are < 34.9 kJ/kg d.a., during the coldest winter days which corresponds to < 1600 W for case study house ventilated at ~ 0.046 kg/s d.a., at a maximum exergy efficiency of ~ 10.7%. This drops to < 0.8 kJ/kg d.a. during summer, with an exergy efficiency as low as 0.4%. The HP would switch off when T° reaches 22°C, then operating at an exergy input ~ 2 kJ/kg d.a. at an exergy efficiency < 1 %.

Clearly the efficiency of energy use by the ground source HP is much better than that of an air source HP, which is a well-known fact for which exergy analysis can give the corresponding numbers.

### 3.2.3. Cases c & d: Ground or air source HP, 75% exhaust air heat recovery

Modern buildings and houses are often equipped with exhaust air heat recovery (EAHR) systems, which for so-called passive energy houses are rated at ~ 75% heat recovery [16]. In practice this implies heat exchange $Q_{HX}$ (W) between incoming ambient air and outgoing ventilation air:

$$Q_{HX} = 0.75 \cdot \dot{m}_{d.a} \cdot (h_{exhaust \ air (22^\circ C, 50\% RH)} - h_{inlet \ air (T^o, \omega^o)})$$  \ (14)

Similar to the analysis in section 3.2.1, a First Law of Thermodynamics energy use efficiency calculation would not give a different efficiency if $Q_{out}$ in (3) and (4) is reduced by 75%. For a proper energy efficiency based on the Second Law of Thermodynamics an increased efficiency as a result of heat recovery the exergy of the incoming air should be calculated. For this, the physical exergy of the incoming air is calculated using (9), while (10) is not needed here since the absolute humidity of the incoming air is unchanged at $\omega = \omega^o$. Thus, the exergy efficiency for HP integration with EAHR becomes, based on (11):

$$\eta_{Ex, HP+EAHR} = \frac{Ex_{phys,HP} + Ex_{chem,HP} + Ex_{phys,EAHR}}{Ex_{(Q_{in})} + p} \approx \frac{Ex_{phys,HP} + Ex_{chem,HP} + Ex_{phys,EAHR}}{p} \text{ where } p = \frac{Q_{out}}{COP_{HP}}$$  \ (15)

by introducing the exergy of the incoming air after the heat transfer in the EAHR. Thus, some exergy enters the building space besides what is delivered by the HP. Here, $T_{indoor} = 295K$ and the temperature $T$ at which incoming air enters the building space follows from energy balances (14) and (7a):

$$Q_{HX} = \dot{m}_{d.a} \cdot (1.005 + 1.87 \cdot \omega^0) \cdot \Delta T$$

$$\rightarrow T_{preheated air} = T^o + \left(\frac{Q_{HX}}{\dot{m}_{d.a}}\right) \cdot \frac{1.005 + 1.87 \cdot \omega^0}{1.005 + 1.87 \cdot \omega^0}$$  \ (16)
Several constraints apply here: 1) the inlet air won’t be cooled by the outgoing ventilation air in the EAHR, i.e. $Q_{HX} \geq 0$ and as a result 2) $T_{preheated\ \text{air}} \geq T^\circ$. Also, 3) a minimum driving force of $\Delta T = 10^\circ\text{C}$ is needed for effective heat transfer (which here implies a maximum temperature of $12^\circ\text{C}$ for the incoming ambient air), otherwise the heat transfer surface is by-passed and is fed to the building space as such. Finally, 4) at ambient temperature -8°C the flow through the EAHR is periodically lowered as to keep the inlet air entering the building space at at least 15°C. (See [9] for more detail on a typical EAHR for the case study building considered here). Thus calculated values for $T_{preheated\ \text{air}}$, see Fig. 9, are used in (9) to calculate $E_{\text{phys,EAHR}}$ with $\omega = \omega^\circ$.

**Figure 9. Ambient and inlet air temperature after heat exchange with outgoing ventilation air (22°C, 50% RH) and the transferred heat.**

With the thus computed additional exergy input to the building space the increased exergy of HP + EAHR compared to HP only can be calculated: results are given in Figs. 10 and 11 for a ground source HP (case c.) and an air source heat pump (case d.), respectively. The additional exergy input is identical for both HP types but is of the same order of magnitude as the exergy supplied by the HP directly for the ground source HP (case a.) but significantly lower than the direct exergy supplied by the air source HP (case b.). Nonetheless, the overall benefit of combining an EAHR system with a HP for space heating is significant and most likely will be economically attractive from an investment point of view, looking at a significantly more expensive (to install) ground source HP system.

**Figure 10. Additional exergy into the building space after heat exchange with outgoing ventilation air (22°C, 50% RH) and the increased exergy efficiency (%-points) of HP + EAHR vs. HP only. Ground source heat pump ($\text{COP}_{HP} = 4$). (Case c).**
As a result of the lower exergy efficiency and higher exergy input requirements of an air source HP compared to a ground source HP, the increase in exergy efficiency when integrated with an EAHR unit is more modest for the air source heat pump. The relative order of efficiency increase as %-points is ~3× better for a ground source HP. In both cases the contribution of EAHR is zero during summer simply because it switches off.

Figure 11. Additional exergy into the building space after heat exchange with outgoing ventilation air (22°C, 50% RH) and the increased exergy efficiency (%-points) of HP + EAHR vs. HP only. Ground source heat pump (COP_{HP} = f(T°)). (Case d).

Thus, while a First Law of Thermodynamics assessment would not reveal the increased energy efficiency that can be computed using exergy analysis, the assessment given in this section 3.2.3 quantifies how the energy input requirement (i.e. electricity) of the HP can be reduced when integrated with an EAHR system. Finally, note that possible condensation of outgoing ventilation air moisture in the EAHR is not specifically addressed here – see an earlier paper for that [9].

4. Conclusions

This paper showed how an exergy analysis based on the Second Law of Thermodynamics can quantify the efficiency of input energy use, for cases studies that involving heat pump-driven building heating integrated with exhaust air heat recovery. A specific yet very important feature is that besides temperature also humidity of a building space will change, with temperature measurement typically being the control parameter. One important finding is the significantly higher energy (exergy) efficiency of a ground source heat pump compared to an air source heat pump. Seasonal effects are very important with much higher energy use but also higher efficiency during winter compared to summer. Energy input requirements for increasing humidity are typically similar to what is needed to raise temperature. The implementation of exhaust air heat recovery seems certainly worthwhile as it gives a significant efficiency improvement, again primarily during winter months. Finally, this study is limited to conditioning a building space to a pleasant 22°C, 50% RH but the findings can be readily recalculated to other conditions, as well as other climate zones different from southwest Finland.

**Nomenclature**

- ex: specific exergy, kJ/kg (d.a.)
- Ex: exergy stream, kW
- h: specific enthalpy, kJ/kg (d.a.)
- H: enthalpy, kW
- m: mass flow rate, kg/s (d.a.)
M  molar mass,
p  pressure (Pa)
R  universal gas constant, 8.314 kJ/(kmol·K)
T  temperature, K or °C

Greek symbols
Δ  difference
η  efficiency
ω  absolute humidity kg water vapour / kg d.a.
Ω  enthalpy ratio humidity increase / temperature increase, -

Subscripts and superscripts
chem  chemical
in  in
out  out
phys  physical
rev  reversible
sat  saturation
vap  evaporation
°  ambient environment

Abbreviations
COP  coefficient of performance
d.a.  dry air
EAHR  exhaust air heat recovery
FOM  figure of merit
HP  heat pump
HX  heat exchange
RH  relative humidity

References
[2] Taloussanomat 17.1.2019. In Finland already 900 000 heat pumps – these pumps were bought the most (in Finnish), Available at: <https://www.is.fi/taloussanomat/oma-raha/art-2000005968577.html> [accessed 20.1.2019].


