EUROVENT 6/12

EUROVENT
AIR HANDLING UNITS
ENERGY EFFICIENCY CLASS
Third Edition – 2012

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Table of contents

Modification as against first edition: Equation regarding correction for electrical heater (1 if heater)
Subscripts regarding last equation

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1. Foreword

In this method the impacts of the various factors are weighted together to establish the final energy class.

Energy to Air Handling Units (AHUs) can be divided into two main groups; thermal energy (for heating and cooling) and electrical energy for fans. Different levels for thermal energy consumption for heating are covered by the consideration of the Heat Recovery System (HRS) efficiency. The climate dependency for the thermal energy consumption is considered and the difference in primary energy between thermal energy and electrical energy is taken into account to evaluate the impact of the pressure drops across the HRS (factors 1 to 2). The thermal energy for cooling is not considered because it will have less impact (negligible for most of Europe). Regarding electrical energy for fans, the method only accounts for the impact of the unit size and efficiency of fan assembly. Other components (e.g. coils) are not individually covered (hence the total pressure increases for fans are not considered) because there is a huge variation in the use of components in different AHU applications. The major influencing factors; velocity, HRS pressure drop, overall static efficiency of the supply and/or the extract air fan and efficiency of the electric motor(s), will give a good estimation of the used energy for fans. The classification, however, cannot be considered as a system energy label. Use LCC calculations to evaluate differences between systems.

The required values for the classes adopted in the calculations are taken from the European Standard EN13053: "Ventilation for buildings – Air handling units – Rating and performance for units, components and sections."

2. Prerequisites

- The calculations shall be made with standard air density = 1.2 kg/m^3
- In the calculations for classification evaluation, the design conditions for winter time shall be used for air flows, outdoor temperature, mixing ratio, heat recovery efficiency, etc.
- The velocities in the calculations are the air velocities in the AHU cross-section based on the inside unit area for supply, respectively extract air flow of the air handling unit. The velocity is based on the area of the filter section of the respective unit, or if no filter is installed, it is based on the area of the fan section
- The relationship between velocity in the cross section of the unit and internal static pressure drop is considered to be exponential to the power of 1.4:

$$\Delta p_{st-1} = \left(\frac{v_1}{v_0}\right)^{1.4} \times \Delta p_{st-0}$$

- The heat recovery dry efficiency at balanced air volume flows shall be used. If the extract (also called "exhaust air in") air volume flow across the heat recovery section diverges from the supply air volume flow through the heat recovery section, the efficiency shall be calculated for both air volume flows equal to the supply air volume flow. For efficiency evaluation the supply air volume for the heat recovery section, winter time shall be taken (the supply air volume flow of the unit can be higher in case of a mixing section).
- For pressure drop evaluation of the heat recovery section the design air volume flows across the heat recovery for winter time shall be taken. Pressure drop increase due to condensation is not considered!

- Heat recovery efficiency figures for run around coil systems shall be based on fluid with the actual ethylene glycol design percentage, design fluid flows and design inlet temperatures.
- Weighting ratio between electric energy and thermal energy is 2 (1 kWh electric energy ≈ 2 kWh (primary) thermal energy).
- An empirical formula for the equivalence between the efficiency and the pressure drop of a heat recovery system, as a function of the outdoor climate, has been derived from numerous energy consumption calculations all over Europe, (see Figure 1 below): $f_{pe} = (-0.0035 \times t_{ODA} - 0.79) \times t_{ODA} + 8.1$ [Pa/%]

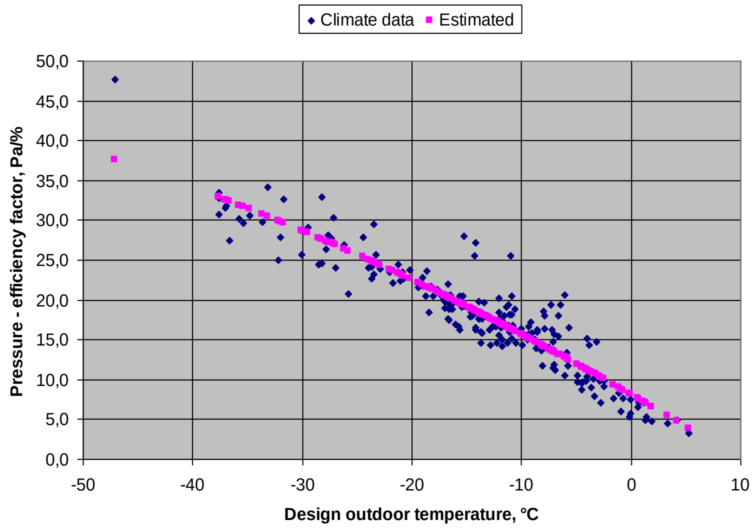

Figure 1: Equivalence Efficiency / Pressure Drop

3. Air Handling Unit subgroups

Three subgroups, with different label signs, are defined:

1) <u>Units for full or partial outdoor air at design winter temperature ≤ 9°C.</u>
This subgroup comprises units connected to outdoor air with the design outdoor temperature, winter time ≤ 9°C. If the unit contains a mixing section; it will be treated within this group as long as the amount of recirculation air is less than 85 %. If more recirculation is claimed, the calculation value for 85% shall be used in the applicable equation for pressure correction Δp_z. This subgroup will consider the velocity in the filter cross section, the HRS efficiency and pressure drop and the mains power consumption to the fan(s). The class signs are **A** to **<E**.
2) <u>Recirculation units or units with design inlet temperatures always > 9°C.</u>
This subgroup includes units with 100% recirculation air, units connected to outdoor air for which the design outdoor temperature during winter time > 9°C or units with (pre-conditioned) inlet temperature > 9°C emanating from a make-up air unit up-stream. This subgroup will only consider the cross section velocity of the filter section and mains power consumption to the fan(s). The class signs are from **A↺** to **<E↺**.
3) <u>Stand-alone extract air units.</u>
Subgroup for pure extract air units (First reason to allocate an energy label to this kind of unit application is that they could include heat recovery. Another reason is that the design outdoor temperature has no relevance for such units). This subgroup will only consider the cross section velocity of the filter section and mains power consumption to the fan(s). The class signs are from **A↑** to **<E↑**.

4. Reference table

Table 1: Table for energy efficiency calculations

CLASS	All Units	Units for full or partial outdoor air at design winter temperature ≤ 9°C		
	Velocity	Heat recovery system		Absorbed power factor
	v_{class} [m/s]	η_{class} [%]	Δp_{class} [Pa]	$f_{class–Pref}$ [-]
A / A⤷ / A↑	1.8	75	280	0.9
B / B⤷ / B↑	2.0	67	230	0.95
C / C⤷ / C↑	2.2	57	170	1.0
D / D⤷ / D↑	2.5	47	125	1.06
E / E⤷ / E↑	2.8	37	100	1.12
<E / <E⤷ /<E↑	No calculation required			No requirement

The lowest classes <E, <E⤷ and <E↑ have no requirements.

5. Methodology

The principle is to establish whether the selected unit with different energy parameters will consume no more energy than a unit that would exactly meet the requirements for the aimed class in Table 1.

Perform the four following steps for respective air sides, supply and/or extract:

1. Assume an AHU is designed to meet the requirements for a particular class, so apply the corresponding class values (subscript “class”) from Table 1:
    - for velocity v_{class}
    - absorbed motor power $f_{class–Pref}$

    If subgroup 1 (units for full or partial outdoor air at design winter temperature ≤ 9°C), apply also:
    - heat recovery efficiency η_{class}
    - pressure drop Δp_{class}
2. Use, for the actual air handling unit to be classified at design air flow, winter time, the actual selection values (subscript “s”) values:
    - fan static pressure increase $\Delta p_{s\text{-}static}$
    - external pressure drop $\Delta p_{s\text{-}external}$
    - velocity v_s
    - power supplied from mains to selected fan $P_{s\text{-}sup}$ if supply air side else $P_{s\text{-}ext}$

    If subgroup 1 use also:
    - HRS dry efficiency η_s
    - HRS pressure drop $\Delta p_{s\text{-}HRS}$
3. Calculate the pressure correction due to velocity Δp_x (see 6)

    If subgroup 1, then calculate:
    - pressure correction due to HRS pressure drop Δp_y (see 7)
    - pressure correction due to HRS efficiency Δp_z (see 8)
4. Calculate fan reference power $P_{air\ side\text{-}ref}$ for the actual air handling unit side, i.e. $P_{sup\text{-}ref}$ if supply air side or $P_{ext\text{-}ref}$ if extract air side (see 9).

Final check consists in verifying whether the selected unit meets the absorbed power consumption criterion for the aimed class. So calculate the absorbed power factor; f_{s-Pref} (see 10). If the value f_{s-Pref} is equal or lower than the value $f_{class-Pref}$ in Table 1 for the aimed class, the unit meets the requirements for the class. If not, the same calculation procedure shall be repeated for a lower class.

6. Pressure correction due to velocity; Δp_x

$$\Delta p_x = \left(\Delta p_{s\text{-internal}} - \Delta p_{s\text{-HRS}}\right) \times \left\{1 - \left(\frac{v_{class}}{v_s}\right)^{1,4}\right\}$$

where:
- Δp_x = pressure correction due to velocity [Pa]
- $\Delta p_{s\text{-internal}}$ = $\Delta p_{s-static}$ - $\Delta p_{s-external}$ internal pressure drop across components; exclusive system effect pressure drops [Pa]
- $\Delta p_{s\text{-static}}$ = useful fan static pressure increase measured between fan inlet and fan outlet [Pa]
- $\Delta p_{s\text{-external}}$ = external (ductwork system) pressure drop [Pa]
- $\Delta p_{s\text{-HRS}}$ = HRS pressure drop [Pa] (0 if no HRS or subgroup 2 or 3)
- v_{class} = value from Table 1 [m/s]
- v_S = velocity in AHU filter (fan if no filter) cross section [m/s]

With pressure drop correction for velocity, the equivalence figures for primary energy and the corrections for heat recovery it is possible to make a conversion to static pressure surplus or deficit compared to a unit fully compliant with the energy class. A surplus of static pressure means that the actual unit demands a higher static pressure; a deficit of static pressure means that the actual unit needs a lower static pressure than the class compliant unit. Hence, a surplus of static pressure means higher energy consumption while a deficit of static pressure will mean lower energy consumption!

7. Pressure correction due to HRS pressure drop; Δp_y

$$\Delta p_y = \Delta p_{s\text{-HRS}} - \Delta p_{class}$$

where:
- Δp_y = pressure correction due to HRS pressure drop [Pa]
- $\Delta p_{s\text{-HRS}}$ = HRS pressure drop (0 if no HRS or subgroup 2 or 3) [Pa]
- Δp_{class} = value from Table 1 [Pa] (0 if subgroup 2 or 3)

8. Pressure correction due to HRS efficiency; Δp_z

$$\Delta p_z = \left(\eta_{class} - \eta_s + 5 \times cf_{heater}\right) \times \left(1 - \frac{mr}{100}\right) \times f_{pe}$$

where:
- Δp_z = pressure correction due to HRS efficiency [Pa]
- η_s = HRS dry efficiency winter [%] (0 if no HRS or subgroup 2 or 3)
- η_{class} = value from Table 1 [%] (0 if subgroup 2 or 3)
- mr = mixing ratio, winter (recirculation air / supply air; maximum), allowed range 0 to 85 [%]
- f_{pe} = pressure – efficiency factor
  = $(-0.0035 \times t_{ODA} - 0.79) \times t_{ODA} + 8.1$ [Pa/%]
- t_{ODA} = design outdoor temperature, winter [°C]
- cf_{heater} = correction for electrical heater (reheater, i.e. heater downstream the HRS).
  = 0 when there is no electrical heater
  = 1 when there is an electrical heater

9. Fan reference power; $P_{sup-ref}$ if supply air side or $P_{ext-ref}$ if extract air side

The total static pressure correction $\Delta p_x + \Delta p_y + \Delta p_z$ has a negative or positive value. A negative value means that the required static pressure for the selected unit is lower than the static pressure for the class compliant unit would be. For a positive pressure value it is just the other way round. Now the fan reference power for a class compliant unit has to be derived from the available static pressure of the selected unit by taking into account the calculated pressure corrections.

$$P_{air\,side\text{-}ref} = \left(\frac{\Delta p_{s\text{-}static} - \left(\Delta p_x + \Delta p_y + \Delta p_z \right)}{450} \right)^{0.925} \times \left(q_{v\text{-}s} + 0.08 \right)^{0.95}$$

where: $P_{air\,side\text{-}ref}$ = fan reference power [kW] (use $P_{sup-ref}$ for supply air side or $P_{ext-ref}$ for extract air side)
$q_{v\text{-}s}$ = air volume flow rate [m^3/s]

10. Absorbed power factor; fs–Pref

$$f_{s\text{-}Pref} = \frac{P_{s\text{-}sup} + P_{s\text{-}ext}}{P_{sup\text{-}ref} + P_{ext\text{-}ref}}$$

where: $f_{s\text{-}Pref}$ = absorbed power factor
$P_{s\text{-}sup}$ = active power supplied from the mains, including any motor control equipment, to selected supply air fan [kW]
$P_{s\text{-}ext}$ = active power supplied from the mains, including any motor control equipment, to selected extract air fan [kW]
$P_{sup\text{-}ref}$ = supply air fan reference power [kW]
$P_{ext\text{-}ref}$ = extract air fan reference power [kW]

11. Heat recovery for run around coil systems

The following applies for run around coil systems.

Regarding the glycol or temperature, no corrections of efficiency shall be considered: efficiency shall be evaluated on the actual glycol percentage and actual temperatures.

For the efficiency at balanced airflows a correction shall be applied. If the real correction can be obtained from the selection software, it is always possible using it. Otherwise, the following equation shall be used:

$$\varphi_{1:1} = \varphi_s x \sqrt{\frac{\dot{m}_{ODA}}{\dot{m}_{ETA}}}$$

where: $\varphi_{1:1}$ = efficiency for balanced airflows [%]
φ_s = actual efficiency for unbalanced airflows [%]
$\dot{m}_{ODA}$ = outdoor *(supply)* air flow [kg/s]
$\dot{m}_{ETA}$ = extract air flow [kg/s]

Equation is valid for a minimum extract air flow of 0.6 x supply air side or a maximum extract air flow of 1.2 x supply air side. If ratio is out of the limits, the 0.6 and 1.2 corrections shall be used.