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ANALYSING FREE COOLING POTENTIAL

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Abstract

Increasingly stringent energy directives of the European Union and growing cooling demands driven by climate change emphasize the need for research into energy-efficient cooling solutions. Free cooling is a promising technology for reducing energy consumption; however, its efficiency and potential application across various building types remain unclear. This article aims to minimize the cooling energy demand provided by HVAC systems through the use of direct active free cooling systems in office, residential, and small commercial buildings. The findings of this research are directly applicable to building operations, significantly contributing to enhanced energy efficiency in buildings and the development of cost-effective, sustainable cooling strategies.

Keywords*: free cooling, energy demand, free cooling resistance point.*

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1. Introduction

Reducing energy consumption and utilizing renewable energy sources not only contribute to achieving sustainability goals but also enhance political and economic independence. Accurately assessing the energy demand of buildings is crucial for effective future energy planning. Several solutions exist to reduce energy demand, including improved building insulation, natural ventilation systems, heat storage, and night-time ventilation solutions. These technological innovations significantly enhance energy efficiency. [1, 2, 3, 4]

This paper first introduces the theoretical background derived from previous scientific research and then discusses free cooling and its justification. The study introduces a new factor, ε, or the degree-day ratio, to illustrate and define free cooling zones. Following this, the paper presents a sensitivity analysis of the equation derived for the factor to examine how changes in various parameters affect the degree-day ratio. Finally, the study describes the building (room) used for the research. [5, 6, 7, 8]

2. Theoretical background and description

2.1. Degree-day curve, degree-day, energy demand

If the days of a given year and their corresponding average daily outdoor temperatures are arranged in order based on how many days are below a given external temperature, the resulting curve is called the temperature frequency curve. These degree-day values can be determined based on the temperature frequency curve, the balance point temperature (also known as threshold temperature in Hungarian terminology), and indoor temperature (Fig. 1). [9]

The balance point temperature can be determined using the following method [10]:

$$T_B = T_i - \frac{\dot{Q}_{rad} + \dot{Q}_i}{H_{tr} + c \cdot \rho \cdot V \cdot n}; \quad [K] \tag{1}$$

where:

T_i is the internal temperature, measured in [K];
$\dot{Q}_{rad}$ is the solar heat gain, in [W];
Q_i represents internal heat gains, in [W];

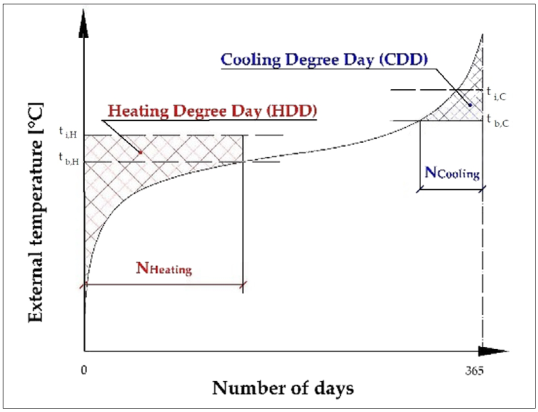

Fig. 1. *Understanding degree days.* [9]

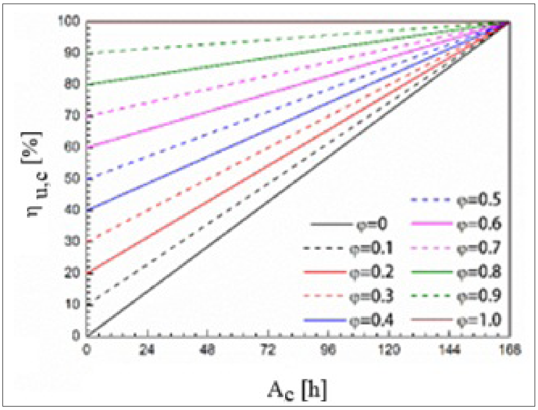

Fig. 2. *Determining the weekly Utilization Efficiency for the Building.* [9]

H_{tr} is the transmission heat loss coefficient, measured in [W·K⁻¹];
c is the specific heat capacity of air in [J/(kg·K)];
ρ is the air density in [kg/m^3];
V is the volume of the space in [m^3];
n is the air exchange rate, measured in [1/h]. [9]

The (cooling) degree-day value (*CDD*) can then be determined using the following equation [9]:

$$CDD = \int_{j=N_{hűtési}}^{365} (\bar{T}_{ej} - T_B)\cdot dt\,;[hK] \qquad (2)$$

where:

$\bar{T}_{ej}$ is the outdoor temperature at the *j*-th hour
T_B is the balance point temperature [K];
CDD represents the cooling degree-day value, measured in [°C·day] or [h·K].

Considering the building's function, it is advisable to account for the weekly Utilization Efficiency for the Building $\eta_{U,C}$ when determining the building's energy demand (Fig. 2) [9]:

$$\eta_{U,C} = \frac{A_C + (168 - A)\cdot \bar{\varphi}_C}{168}\cdot 100;\quad [\%] \qquad (3)$$

where:

A_C is the number of active hours per week from a human usage perspective, measured in hours,
ϕ is the passivity operating ratio during inactive periods compared to active periods, dimensionless [-]. [9]

Taking these factors into account, the building's energy demand (E_C) can be determined using the following equation [9]:

$$E_C = \frac{\eta_{U,H}}{100}\cdot (H_{tr} + c\cdot\rho\cdot V\cdot n)\cdot CDD;\quad [Wh] \qquad (4)$$

The relationships described here are typically used on an annual basis but can also be adapted for a single day. The difference lies in the fact that the degree-day curve and the balance point temperature curve will become more segmented, as they are constructed from fewer measured data points (e.g., only hourly temperature values may be available.

2.2. The Viability of Free Cooling

Free cooling is a technology that utilizes the lower temperature of ambient air for cooling purposes, reducing or eliminating the need for conventional compressor-based cooling. This results in significant energy savings, as free cooling requires less energy than compressor-driven cooling. It is particularly applicable during transitional periods when the external temperature is lower than the indoor temperature that needs to be maintained. [6, 11, 12]

Free cooling systems can be categorized into two main types: active and passive free cooling. [12, 13]

In active free cooling, outside air is introduced directly into the building's cooling space, typically through ventilation systems. This method allows a high degree of air exchange; however, air pollution can affect efficiency. [8, 12]

Passive free cooling is where heat exchangers facilitate heat transfer, where a refrigerant (e.g. water or other cooling medium) absorbs heat and transfers it to the building air through a heat exchanger. This method is particularly effective when used in conjunction with liquid-based cooling towers. [8, 12]

In building services engineering, free cooling technology is becoming increasingly important, especially as energy efficiency standards and environmentally conscious design come to the fore. It has many advantages from both an environmental and an economic point of view. [11]

The use of free cooling significantly reduces the cooling energy demand of buildings, as it does not require the operation of machines that are necessary for the cooling equipment. From an environmental point of view, the reduced energy consumption results in lower carbon dioxide emissions. In terms of economic efficiency, the energy savings will reduce operating costs, which will accelerate the return on investment. The use of free cooling can also increase the lifetime of the chillers, as they need to be operated for less time under heavy load. [11, 12]

Free cooling is the ideal solution for facilities that require continuous cooling, such as office buildings, data centres or industrial facilities. It is also an excellent solution for industrial facilities where constant cooling is required due to process cycles. [11, 12]

The use of free cooling technology is becoming increasingly popular in the building services sector, especially as energy efficiency and sustainability needs grow. As the demand for cooling buildings gradually increases with changing weather conditions, urbanisation and evolving building standards, the use of free cooling offers a significant opportunity for sustainable cooling solutions. [12, 13]

The development of innovative free cooling solutions and the expansion of the regulatory environment can facilitate the further uptake of free cooling. In the future, we can expect intelligent systems that use free cooling in combination with other cooling technologies, automatically controlling the system according to the external and internal environmental conditions. Free cooling therefore not only offers a cost-effective solution for building services engineers, but can also make a significant contribution to environmental sustainability. [12, 14]

3. Results

3.1. The Viability of Free Cooling

Free cooling, is applicable during transitional periods and cooling-demand phases of the year. It is advisable to examine the free cooling potential on a daily basis. Consider the external temperature values of a specific day when free cooling might be feasible. Arrange these temperatures by the number of hours with external temperatures lower than a given value. This results in the degree-day curve for that day. The daily balance point temperature curve can also be plotted on this graph, as shown Fig. 3.

Based on Eq. 1, the balance point temperature curve is not a horizontal line (as radiative gains vary by hour). Instead, it forms a straight line during periods without solar radiation, while during the daytime, it trends downward with minor fluctuations due to varying radiative gains.

The potential free cooling zone can be interpreted using Fig. 3 The 'engine' of free cooling will be the difference between the internal and external temperature, and therefore the degree-day characterising free cooling can be plotted on a thermal frequency curve. This will be the bounded by the balance point temperature, the external temperature and an auxiliary curve. This auxiliary curve is obtained by looking at the difference between the outside and inside temperature at the given hour and projecting downwards from the outside temperature. The cooling temperature bridge is divided into three zones by this curve. For a given air exchange rate, zone I is purely free cooling, zone II requires the use of mechanical cooling in addition to free cooling, and zone III is free cooling impossible.

The aim from the industry perspective is to minimise the actual installed cooling capacity of the machines. As shown in the figure, this can be achieved by minimising the free cooling zone. To eliminate the free cooling zone, the values of T_i and T_b must be equal. That is, either the internal temperature or the balance point temperature must be changed. The value of the internal temperature has a direct effect on human comfort. Thus, its appropriate value can be determined by accurate testing, taking into account a number

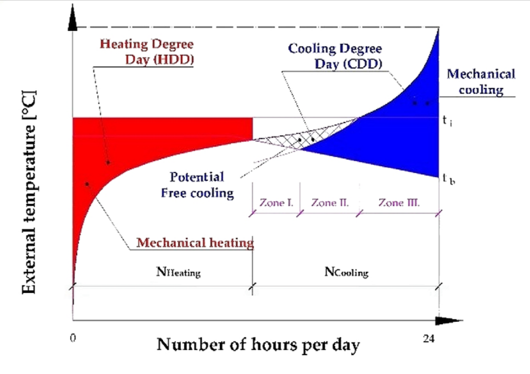

Fig. 3. *Interpretation of the free cooling zone on an average transitional day.*

of aspects. For this reason, the appropriate values for design are laid down in standards, from which deviations should only be made with great care. This is therefore not the case in the following.

The other option is to the balance point temperature. By varying this, we can achieve the elimination of the potential free cooling zone (according to equation 1) if $n = \infty$ [1/h]. By increasing the air exchange rate, the balance point temperature increases and thus the number of machine cooling hours decreases. The problem is that there are technological and economic limitations to increasing the air exchange rate.

We cannot technologically create infinitely large air exchanges, because we would need to be able to get air into the room with infinitely large volume flow rates, and this would require infinitely large fan blades and/or infinitely large shaft speeds.

From an economic point of view, the problem is that such an increase in air exchange rate can only be produced by mechanical ventilation. An air exchange rate value is reached at which the power demand for mechanical ventilation is already higher than the power demand for mechanical cooling. This is the subject of this article.

3.2. The degree-day ration

3.2.1. The definiton of the degree-day ration

Introduce the following factor to characterise the free cooling zones (zones I and II, i.e. where $T_i > T_e$) in **Fig. 3**:

$$\varepsilon = \frac{(T_i - T_B)^2 - (T_B - T_e)^2}{(T_i - T_e)^2 + \epsilon} \approx \frac{(T_i - T_B)^2 - (T_B - T_e)^2}{(T_i - T_e)^2} = \frac{T_i^2 - T_e^2}{(T_i - T_e)^2} - \frac{2}{(T_i - T_e)} \cdot T_B \quad ;[-] \tag{5}$$

where ϵ is an infinitely small quantity so that there can be no zero in the denominator.

If $-1 < \varepsilon < 0$, it is in zone I (100% free cooling), if $0 \leq \varepsilon$, it is in zone II (combination of free cooling and mechanical cooling), and if $\varepsilon \leq -1$, it is in zone III (100% mechanical cooling).

For the future usefulness of the indicator, it is proposed to provide two additional contexts. For the first one, we can write down the relationship at a given moment with a fixed air exchange rate:

$$\varepsilon = \frac{T_i^2 - T_e^2}{(T_i - T_e)^2 + \epsilon} - 2 \cdot \frac{(T_i - T_e)}{(T_i - T_e)^2 + \epsilon} \cdot T_i + 2 \cdot \frac{(T_i - T_e)}{(T_i - T_e)^2 + \epsilon} \cdot \frac{\dot{Q}_{rad} + \dot{Q}_i}{H_{tr} + c \cdot \rho \cdot V \cdot n} \tag{6}$$

$$\varepsilon = \frac{T_i^2 - T_e^2}{(T_i - T_e)^2} - \frac{2 \cdot T_i}{T_i - T_e} + \frac{2 \cdot \left(\dot{Q}_{rad} + \dot{Q}_i\right)}{(T_i - T_e) \cdot (H_{tr} + c \cdot \rho \cdot V \cdot n)} ;[-]$$

where:

T_i is the internal temperature [K],
T_e is the external temperature [K],
$\dot{Q}_{rad}$ is the solar heat gain [W],
$\dot{Q}_i$ is the internal heat gain [W],
H_{tr} is the transmission heat loss coefficient, [W/K],
c is the specific heat capacity of the air [J/(kg·K)],
ρ is the air density [kg/m^3],
V is the volume of the space [m^3],
n is the air exchange rate[1/h].

Fort the second one, we can enter the air exchange rate value at which the air excanhge rate needed to reach the given value ε is obtained at a given moment:

$$n = \frac{2}{c \cdot \rho \cdot V} \cdot \frac{\dot{Q}_{rad} + \dot{Q}_i}{2 \cdot T_i + \left[\frac{(T_i - T_e)^2 + \epsilon}{T_i - T_e}\right] \cdot \varepsilon - \frac{T_i^2 - T_e^2}{T_i - T_e}} - \frac{H_{tr}}{c \cdot \rho \cdot V} ;[-]$$

$$n = \frac{2}{c \cdot \rho \cdot V} \cdot \frac{\dot{Q}_{rad} + \dot{Q}_i}{2 \cdot T_i + (T_i - T_e) \cdot \varepsilon - \frac{T_i^2 - T_e^2}{T_i - T_e}} - \frac{H_{tr}}{c \cdot \rho \cdot V} ;[-] \tag{7}$$

3.2.2. Sensitivy test of the degree-day ratio

Presented here is a factor sensitivity study of the degree-day ratio described above. The sensitivity test is based on equation *(6)*. The idea of the test is to illustrate the extent to which the value of the factor changes when one of its members is changed. Starting from the same initial data set for each factor change, the value of the highlighted factor was changed by percentage.

Fig. 4 shows the variation of the degree-day ratio value determined by equation *(6)* with changes in internal and external temperature.

The horizontal axis shows the temperature difference (Δt), and the vertical axis shows the percentage change in the degree-day ratio with a unit change in temperature. The internal temperature (T_i) is increased from 22 °C to 27 °C per half degree, while the external temperature (T_e) is varied between 25,5 and 30,5 °C kC per half degree.

Fig. 5 lshows the effect of changing the air exchange rate, transmission heat loss coefficient and percentage change in room volume on the degree-day ratio value based on equation *(6)*.

The horizontal axis shows the percentage change in the factors and the vertical axis shows the change in the degree-day ratio. The percentage change in air exchange rate is shown, but the starting point in the basic equation was 0.5 1/h. The air exchange rates tested are 0; 0.125; 0.25; 0.375; 0.5; 0.625; 0.74; 0.875 and 1 [1/h].

The transmission heat loss coefficient in the basic equation is 16.49 W/K, which is shown in the figure as a variation from 0-32.

When the volume of the room was changed, the initial value was 174.35 m^3, which is shown in the diagram with a 25 % decrease or increase per unit.

3.3. Limits of application of free cooling - performance side

In the following, we investigate what happens when the air exchange rate is increased by Δ*n*. Let us now take as a starting point the air exchange rate n_{min}, which is the result of natural filtration. The corresponding cooling balance point temperature can be determined by the following relation **[10]**:

$$T_{B,min} = T_i - \frac{\dot{Q}_{rad} + \dot{Q}_i}{H_{tr} + c \cdot \rho \cdot V \cdot n_{min}} \tag{8}$$

If this air exchange rate is increased by the value of n_{min} and Δ*n* this will also result in a change in the cooling balance point temperature. The rate of change is:

$$\Delta T_B = \frac{T_i - T_{B,min}}{1 + \frac{n_{min}}{\Delta n} + \frac{H_{tr}}{c \cdot \rho \cdot V \cdot \Delta n}} \tag{9}$$

The increased air exchange rate results in more electrical work required by the fan **[15]**:

$$\Delta W_{ve} = \frac{\Delta P_{hasznos}}{\eta_{ve}} \cdot \Delta\tau_{ve} = \frac{\Delta p_{ö} \cdot \mathrm{V} \cdot \Delta\mathrm{n}}{\eta_{ve}} \cdot \Delta\tau_{ve} = \left(\frac{\Delta p_{ö} \cdot \mathrm{V}}{\eta_{ve}}\right) \cdot \Delta\mathrm{n} \cdot \Delta\tau_{ve} \tag{10}$$

The bracketed tag represents the fan pressure rating, efficiency and room volume. In principle, these are assumed to be constant as the fan air exchange rate is increased. The $\Delta\tau_{ve}$ will be the fan operating time.

By using free cooling, the electrical work needed to run the chiller is saved. The electrical work saved:

$$\Delta W_{HG} = \frac{\Delta E_c}{SCOP_R} \tag{11}$$

Due to the increased air exchang (Δ*n*) the balance point temperature (ΔT_B)changes and therefore the number of cooling hours changes ($\Delta N_{c,max}$ *decreases to* ΔN_C i.e. it decreases by exactly $\Delta\tau_{ve}$). There is no mathematical relationship between the change in the balance point temperature and the number of cooling hours, the exact relationship has to be investigated for individual days. **[16]**

The use of free cooling is justified if the increased ventilation work is less than a predetermined amount more than the electrical work saved (this can be taken into account, for example, for simpler process designs.

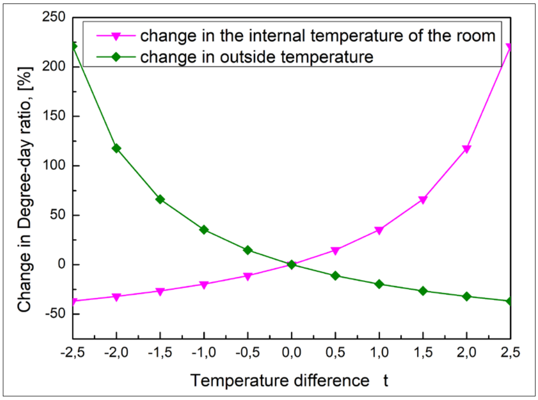

Fig. 4. *Variation of degree-day ratio as a function of internal and external temperature.*

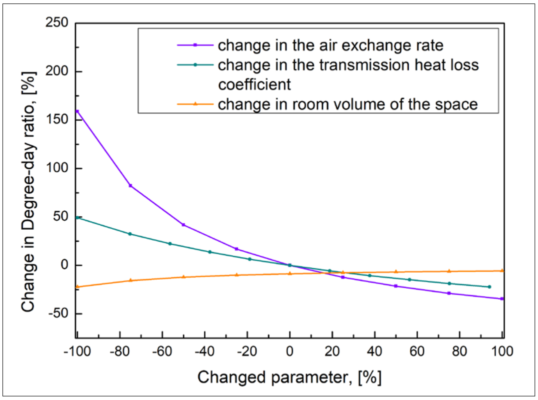

Fig. 5. *Variation of the degree-day ratio as a function of air exchange rate, transmission heat loss coefficient and room volume change.*

$$\Delta W_{ve} \le z \cdot \Delta W_{HG} \tag{12}$$

As you can see, the applicability of free cooling should be examined through a case study.

4 Case study

4.1. The analysed room and the changed parameter

During the analysis, we examined a room (**Fig. 6**), located on the third floor of a four-story building (ground floor and three upper floors). The room has two external walls ($U = 0.24$ W/(m^2·K)), part of which is glazed ($U_{üveg} = 1.1$ W/(m^2·K)). These values comply with the standards set forth in the Hungarian ÉKM Decree 9/2023 regulation. The room has a floor area of 61.45 m^2 and an air volume of 174.35 m^3. The ceiling height near the glazed surface is approximately 3 m, while it is 2.5 m closer to the door. In **Fig. 6** the area below the suspended ceiling is marked with a green outline. The transmission heat loss coefficient from the room's structures H_{tr} is 16.49 W/m^2. The dimensions of the room are shown in **Fig. 6.** **[16, 17]**:

The design and characteristics of the space were examined according to three functions, which are commercial (e.g., a convenience store with mixed trade), office, and residential (living room). **Fig. 6** currently shows the interior design corresponding to the office function. For residential and commercial uses, the furniture design and layout may vary.

Parameters changed during the room test:

- orientation (North, East, West, South;
- glazing rate 0-100% (40%);
- building function: office, commercial, residential;
- meteorological: extreme summer day, extreme heat day, extreme hot day.

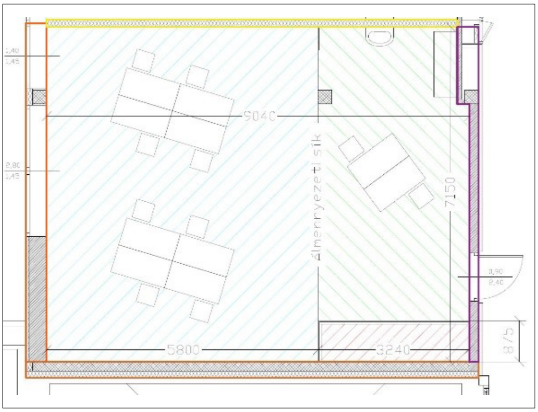

Fig. 6. *The design of the analysed room: Orange - external wall, Yellow - Internal wall facing the room, Purple - Internal wall facing the corridor, Blue - Ceiling height of 3 m, Green - Ceiling height of 2,5 m.*

In the cases tested, the internal temperature was taken as a uniform 24.5 °C, based on the recommendation of MSZ CR 1752. The outdoor conditions of the period potentially affected by cooling were characterised by three meteorological days (extreme hot day, extreme heat day and extreme summer day). For the selection, the Debrecen database 2009-2019 was used, firstly isolating the three types of meteorological days from each year, then, selecting the days with the largest daily temperature fluctuations from each of the three temperature groups in each of the years studied. Finally, in the third round, the year with the smallest daily temperature fluctuation from the three temperature groups was always selected. With the second round, the aim was to find the days where the energy savings were expected to be the greatest, and with the third round, to filter out sudden changes in the weather (e.g. sun in the morning, rain in the afternoon. **[18, 19, 20]**

The internal heat gains (heat dissipation of people, machines, etc.) are recorded according to MSZ EN ISO 13790. For the calculation of the energy demand, the initial air exchange rate $n_0 = 0.5$ 1/h, was taken as a value typical of natural filtration. A minimum air exchange rate for each function was also established, based on the number of occupants and the fresh air demand per person. The characteristic values for artificial ventilation and mechanical cooling were based on the recommendations of the ECM. These values are summarised in **Table 1**. **[17, 21]**

Table 1. *Data considered in relation to engineering*

	Sign	Value
All	η_{ve}; [%]	80%
	$\Delta_{pö}$; [Pa]	500 Pa
	SEER; [-]	4
Office	n_{min}; [1/h]	1.0324
	A_C; [h]	7:00–17:00 (10 h)
	φ; [%]	30%
Commercial	n_{min}; [1/h]	4.3017
	A_C; [h]	7:00–19:00 (12h)
	φ; [%]	0%
Residential	n_{min}; [1/h]	0.6883
	A_C; [h]	0:00–24:00 (24h)
		0%

4.2. Preliminary analysis before calculating energy demand

Two types of preliminary analysis should be carried out before considering potential energy savings. One is an analysis of the heat load and the cooling balance point temperature, and the other is an analysis of the ratio of the newly introduced thermal bridge.

4.2.1. Effect of glazing ratio, function and orientation on heat load and cooling balance point temperature

Before analysing the energy demand, the effect of glazing ratio, function and orientation on the cooling balance point temperature and heat load was examined. For simplification, only the extreme heat day was considered. Thus, a total of 36 cases were developed as follows:

- 4 orientations: North, East, South and West;
- 3 functions: office, residential and commercial;
- 3 glazing rates: 20%, 40% and 80%.

For the cases tested, the internal temperature was recorded at 24.5 °C according to the recommendation of MSZ CR 1752. The internal heat gains (heat dissipation of people, machines, etc.) were determined according to MSZ EN ISO 13790 and varied according to the function 5 W/m^2 for laboratories, 7,4 W/m^2 for offices, 10 W/m^2 for shops and 9 W/m^2 for living rooms. **[18, 21, 22]**

The air exchange (natural and artificial) values are based on the ventilation air requirements of comfort class "A" of the MSZ CR 1752 standard, hence 2.50 h^{-1} for offices, 5.3 h^{-1} for shops and 9.00 [1/h] for living room. **[18]**

The following figure shows the value of the cooling balance point temperature (**Fig. 7.a**), and the value of the heat load for the 36 cases studied, grouped according to different aspects (**Fig. 7. b,c,d**).

Fig. 7.a shows that the cooling balance point temperature is significantly influenced by a number of factors and that the function of the building

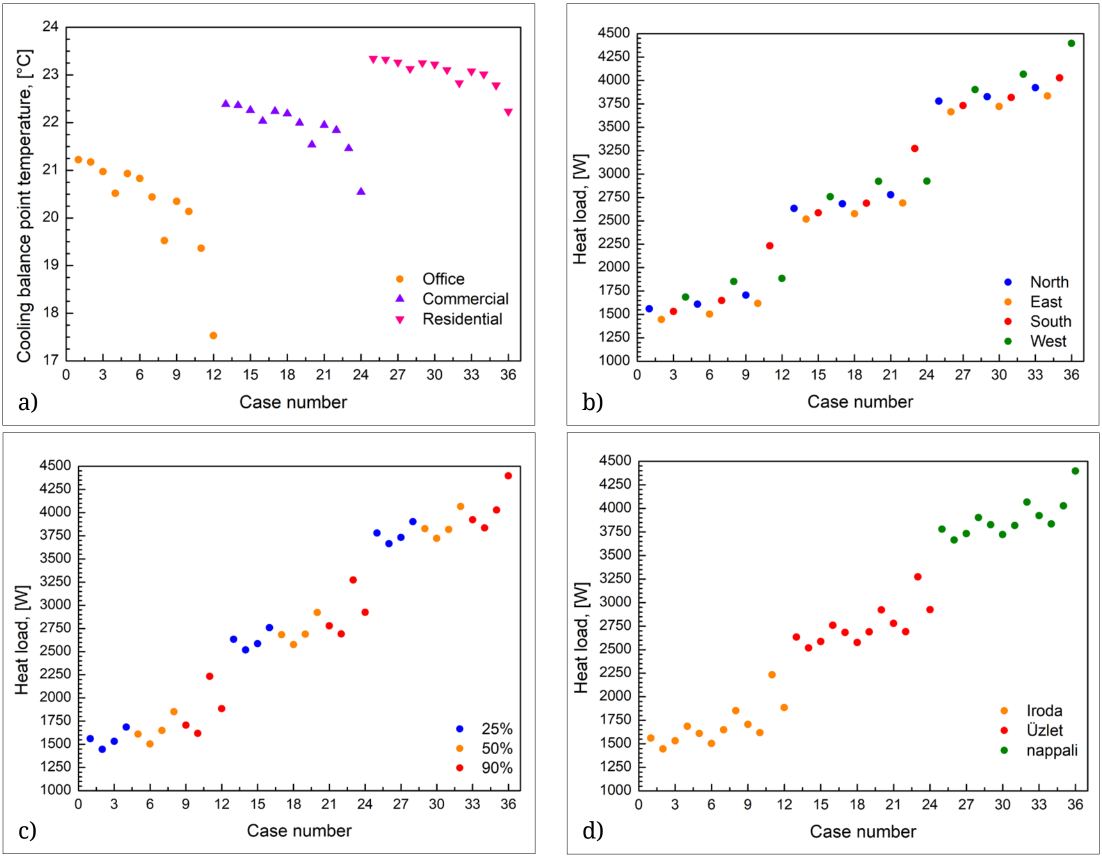

Fig. 7. *a) Cooling balance point temperature, b) Summer heat load value, same orientation cases in one colour, c) Summer heat load value, glazing ratio cases in one colour, d) Summer heat load value, same function cases in one colour.*

is not the only or the most important consideration. It can be observed from the figure that the higher the glazing ratio (the higher the glazing ratio associated with higher case numbers within a given function), the lower the cooling balance point temperature. For the 48 cases studied, the cooling balance point temperature varies between 17,53 °C and 23,66 °C.

In the heat load analysis, cases with the same orientation are shown in one colour in **Fig. 7.b** cases with the same glazing ratio in **Fig. 7.c** and cases with the same function in **Fig. 7.d.** The figures show that for a given function, the heat loads will be higher for west-facing rooms and secondarily for south-facing rooms with a large glass area.

It appears that this preliminary study did not provide enough information to clearly identify the dominant aspects, and therefore further studies should be carried out.

4.2.2. Preliminary analysis using the degree-day ratio

Before determining the energy that can be saved by using free cooling, let's consider at what time of day, in what orientation and under what external meteorological conditions significant energy savings can be expected.

To do this, a few parameters had to be fixed. Firstly uniform glazing of 40% is assumed, and secondly, the minimum (nmin) was taken as the value of the air exchange rate for a given function.

For ease of reference and to emphasise the results, the data for the summer days are shown in a separate figure.

The results for the summer sun are shown in **Fig. 8.**

The results for heat days and hot days are presented in **Fig. 9.**

For clarity, the N-S and E-W orientations are shown separately in the figure. The ε = 0 and ε = –1 lines are also shown to help. If ε < –1, then only mechanical cooling is an option. If greater than zero, it is possible to use free cooling with reduced machine cooling.

Based on the results, significant savings are expected in summer days, followed by heat days and finally hot days. For the orientations, a sequence of east, south, west and north can be observed. For function, a sequence of residential, office, and commercial is expected.

5. Conclusions

The prospect of future energy crises is increasingly pushing efforts to determine the expected energy demand of buildings as accurately as possible. This paper presents novel approaches that can contribute significantly to this goal. In addition to theoretical concepts, their applicability through a concrete case study is also illustrated.

In the first section the theoretical background of the methods used is presented. The heat rate curve, the definition of the balance point temperature and the function dependent efficiency are presented, (which has been dealt with in previous scientific work), and the concept and importance of free cooling is described.

The second section presents the justification for free cooling, illustrating the interpretation of the free cooling zone on the heat rate curve and, consequently, the heat rate coefficient. Following presentation of the factor by equation, a sensitivity study was carried out, showing the extent to

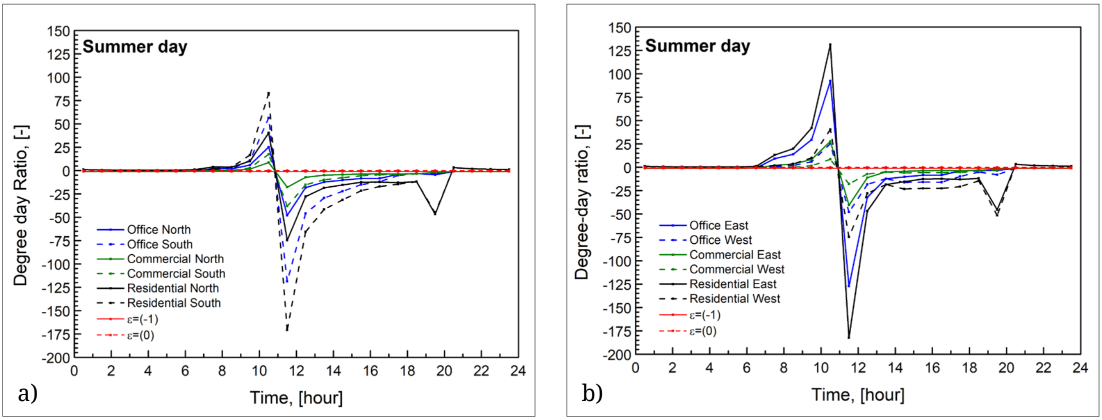

Fig. 8. *Evolution of the degree-day ratio at 40% glazing and n_{min} air exchange rate. a) Summer day, N and S orientation; b) Summer day, E and S orientation.*

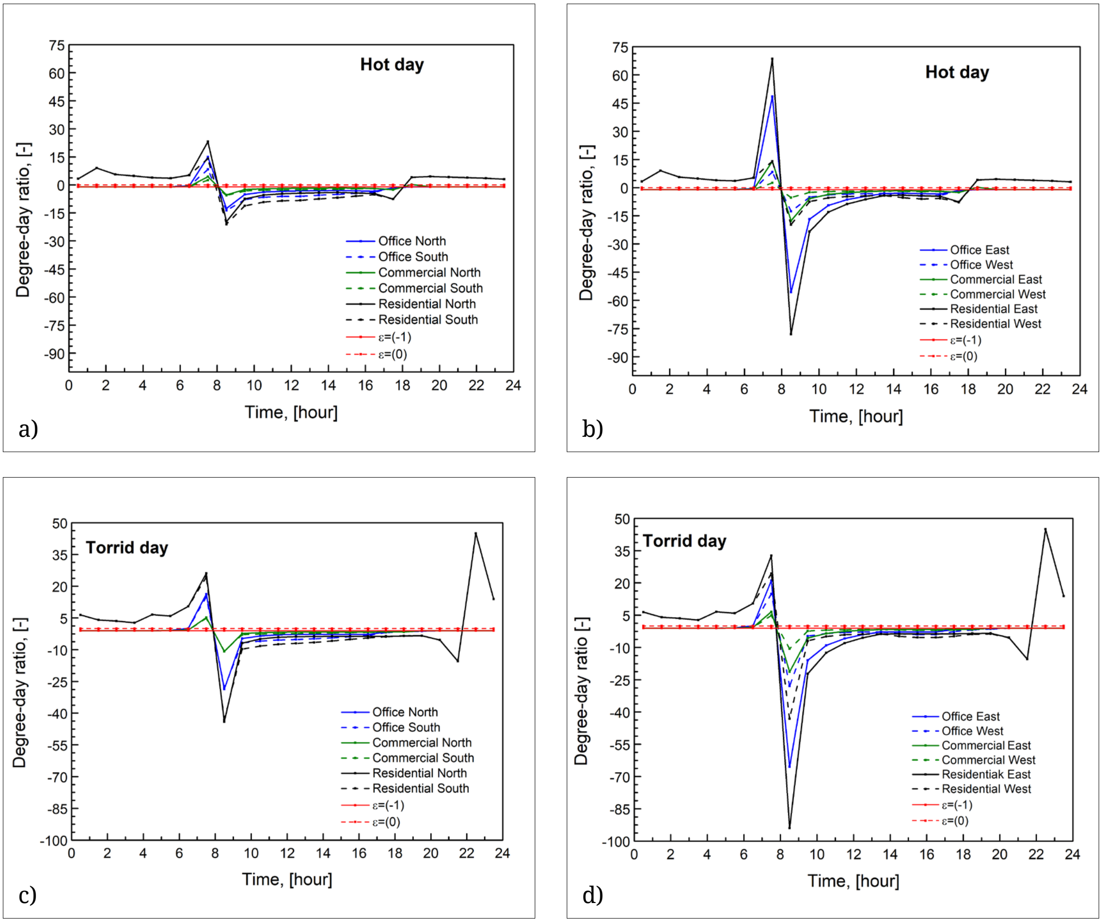

Fig. 9. *Evolution of the degree-day ratio at 40% glazing and n_{min} air exchange rate. a) Heat day, N and S orientation; b) Heat day, E and N orientation; c) Hot day, N and S orientation; d) Hot day, E and N orientation.*

which the value of the factor changes as the value of a term in the basic equation is changed.

Also in this section, the limits of free cooling were addressed, examining what can achieved by increasing the air exchange rate per unit.

Finally, the case study is given, presenting the building properties and calculation values used in the study, followed by illustrated results obtained by calculating the ε variation (e.g. summer day, heat day, hot day).

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