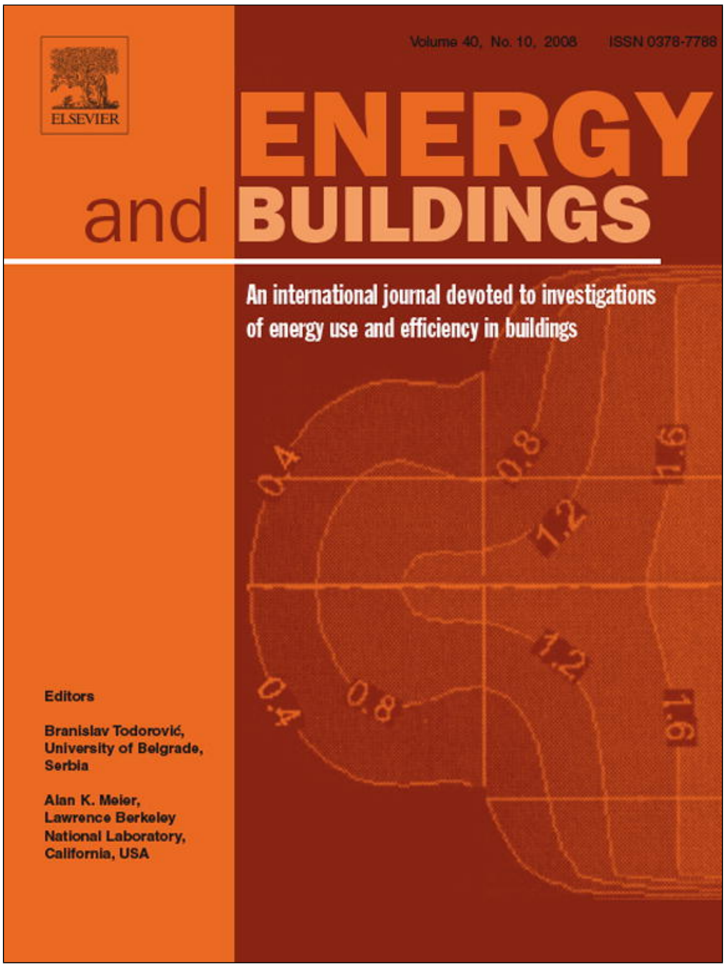
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Energy saving and improved comfort by increased air movement

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ABSTRACT

In this study, the potential saving of cooling energy by elevated air speed which can offset the impact of increased room air temperature on occupants' comfort, as recommended in the present standards (ASHRAE 55 2004, ISO 7730 2005 and EN 15251 2007), was quantified by means of simulations with EnergyPlus software. Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories I, II and III (according to standard EN 15251 2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. The required cooling/heating energy was calculated assuming a perfectly efficient HVAC system. A cooling energy saving between 17 and 48% and a reduction of the maximum cooling power in the range 10–28% has been obtained. The results reveal that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Under the assumptions of this study, the energy saving may not be achieved with the methods for air speed increase, such as ceiling, standing, tower and desk fans widely used today when the power consumption of the fan is higher than 20 W.

1. Introduction

According to the 2007 report of the Intergovernmental Panel on Climate Change [1], warming of the climate system is unequivocal. Therefore, a reduction of greenhouse gases emission is needed. The building sector plays an important role in this challenge. The report states that the residential and commercial building sectors have the greatest global potential for emission reduction among all sectors studied in the report. Energy efficiency options for new and existing buildings can reduce CO_2 emissions considerably with net economic benefit. Energy efficient buildings, while limiting the increase of CO_2 emissions, can also improve indoor and outdoor air quality, improve social well-being and enhance energy security [2].

1.1. Air velocity and maximum operative temperature

In the present international indoor climate standards [3–5] the operative temperature comfort limits are based on an air speed limit of 0.20 m/s. However, according to the standards, elevated air speed can offset the indoor temperature rise and provide occupants with thermal comfort. An air speed increase is necessary in order to maintain the heat exchange between the human body and the environment, this being a prerequisite for thermal comfort. The relationship between the air speed and the upper operative temperature limits, as included in the present standards [3,5], is shown in Fig. 1. The recommended speed increase, as shown in Fig. 1, depends not only on the air temperature but also on the difference between mean radiant temperature (t_{mr}) and air temperature (t_a). When the mean radiant temperature is lower than the air temperature, the elevated air speed is less effective for increasing the heat loss from the body. Conversely, elevated air velocity is more effective for increasing the heat loss when the mean radiant temperature is higher than the air temperature. Fig. 1 is based on a theoretical calculation; however, the neutral curve ($t_a = t_{mr}$) has been verified in human subject experiments [6].

The conditions defined in Fig. 1 may be applied only to a lightly clothed person with a clothing insulation between 0.5 and 0.7 clo (0.08–0.1 m^2 K/W) who is engaged in near sedentary physical activity with metabolic rates between 1.0 and 1.3 met (58.15–75.6 W/m^2). The effect of elevated speed on the heat loss from the human body increases at high activity and lighter clothing [3]. Moreover, the increase in operative temperature cannot be higher than 3.0 °C above the values for the comfort zone and the elevated air speed must not be higher than 0.8 m/s. Large individual differences exist between people with regard to the preferred air speed [7]. Therefore the standards require personal control over the speed, the benefit of which was also confirmed in [8]. Thus it

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Nomenclature

COP	coefficient of performance of the chiller
$E_{N,C}^{v=i}$	energy need for cooling ($E_{N,C}$) obtained when the air velocity is $i \leq 0.2$ or $i = 0.5$ or $i = 0.8$ m/s (kWh/(m^2 y))
$E_{N,C}$	energy need for cooling (kWh/(m^2 y))
$E_{el,Cool}$	electrical energy consumed by the chiller (kWh/(m^2 y))
$E_{el,Fan}$	electrical energy consumed by the fan (kWh/(m^2 y))
$E_{el,Net}$	net electrical energy saved (kWh/(m^2 y))
h_i	annual number of hours that the fan is operating for increasing the air velocity. It is calculated for an air velocity of 0.5 m/s ($h_{0.5}$) and 0.8 m/s ($h_{0.8}$) (h)
h_{tot}	the total occupant working hours (h)
P_{Fan}	the electrical input power of the fan (W)
t_a	air temperature (°C)
t_{mr}	mean radiant temperature (°C)
t_{op}	operative temperature (°C)

Greek symbol

η	energy losses from emission, distribution and storage for cooling. It is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce

may not be appropriate to offset a temperature increase by increasing the air speed within a centrally controlled air system [8].

The possibility of increasing the upper operative temperature limit may reduce the energy consumption without significantly affecting occupants' thermal comfort. The individual control of air movement can be achieved with personalized ventilation systems, task/ambient systems, desk, standing, tower or ceiling fans, and under some conditions with operable windows. The energy consumption for air movement generation by these methods is different. The purpose of this study is to quantify, by means of simulations with EnergyPlus software, the potential savings of energy need for cooling (defined in EN 15615 [9]) achieved by elevated air speed without reducing occupants' thermal comfort conditions.

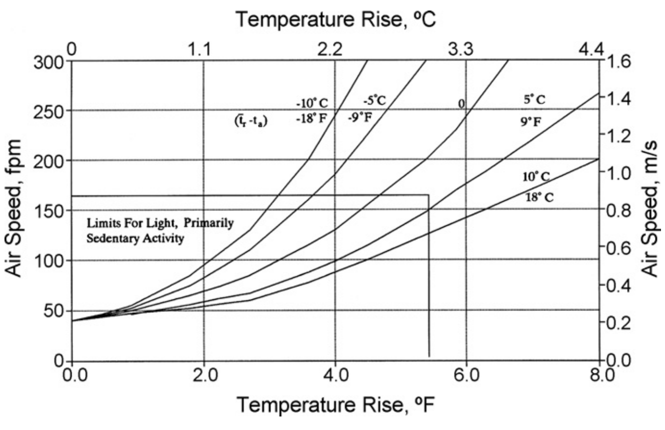

Fig. 1. Air speed required to offset increased temperature (Fig. 5.2.3 from ASHRAE [3]).

2. Methods

The European standard 15265 [10] recommends a format for reporting the input data of an energy simulation. The following presentation of input data complies with the guidance in the standards.

2.1. Building locations and weather data

The energy simulations were performed for the same single office sited in six European and Mediterranean cities listed in Table 1. The cities were chosen in order to describe in a homogeneous way different climate conditions. The focus was on summer conditions. The Cooling Degree Days [11] with a base temperature of 18 °C were used as an indicator of the intensity of the summer period. The ASHRAE IWEC Weather Files were used as input data in the simulation model.

2.2. Description of the office room

The single office room has a floor surface area of 4 m by 2.5 m. The room height is 3 m. The external walls are constructed with 20 mm of plaster, 100 mm of glasswool, 240 mm of brick and 10 mm of internal plaster. The window has an external low-emissivity glass pane (thickness 6 mm), 13 mm of air and an internal glass pane (thickness 6 mm). It has a *U*-value equal to 1.72 W/(K m^2) and a g-factor or Solar Heat Gain Coefficient equal to 0.56. The window has a total area of 2.4 m^2 (24% of the floor area, height of 1.2 m and width of 2 m). The window faces south. There is an external shading device. It has a shading coefficient of 0.48 (g-factor equal to 0.43), and it is activated when the total irradiance on the windows is higher than 400 W/m^2. The internal walls, floor and ceiling are adiabatic. The effect of thermal mass is taken into account.

2.3. Internal temperature, ventilation and infiltration rate

The thermal comfort conditions and ventilation specifications were chosen in order to guarantee the values defined in EN 15251 [4] for the categories I, II and III for indoor environment in the room during occupation. From 7:00 till 18:00 the heating and cooling system kept the internal operative temperature within a range between the minimum operative temperature below which heating is required (Min t_{op} for heating) and the maximum operative temperature above which cooling is required (Max t_{op} for cooling). The minimum and maximum operative temperatures are shown in Table 2. During weekends and night-time the temperature set-back was 12 °C in winter and 40 °C in summer. The design ventilation rates are shown in Table 2. The design airflow rate was supplied during occupation hours. The airflow rates during unoccupied periods were 7% of the design values, i.e. from 0.06 to 0.14 l/s m^2 (the standard suggests a minimum airflow rate for

Table 1
Cities where the office is sited

City	Country	Latitude	Cooling degree day $-t_{base}$18 °C
Helsinki	Finland	60°19′	33
Berlin	Germany	52°28′	170
Bordeaux	France	44°49′	263
Rome	Italy	41°47′	508
Jerusalem	Israel	31°46′	647
Athens	Greece	37°54′	1076

The intensity of the summer period is described using the Cooling degree days with a base temperature of 18 °C.

Table 2
Simulated cases: category of indoor environment, airflow rates, minimum and maximum operative temperatures

Category according EN 15251 2007	Airflow per person (l/(s person))	Airflow per floor area[a] (l/(s person))	Min t_{op} for heating (°C)	Velocity (m/s)	Temperature increase (K)	Max t_{op} for cooling (°C)
I	10	1	21	<0.2	0	25.5
				0.5	1.7	27.2
				0.8	2.5	28
II	7	0.7	20	<0.2	0	26
				0.5	1.7	27.7
				0.8	2.5	28.5
III	4	0.4	19	<0.2	0	27
				0.5	1.7	28.7
				0.8	2.5	29.5

The maximum operative temperatures for cooling are increased according to the air velocity.
[a] Recommended values from Annex C of EN 15251 [4] for low polluting buildings.

unoccupied hours in the range 0.1–0.2 l/s m^2). The infiltration is considered null.

2.4. Internal heat gains, occupancy and description of the HVAC system

One occupant was present in the room (10 m^2 per person). She/he contributed to both sensible and latent heat loads. The activity level of the occupant was 1.2 met (1 met = 58.15 W/m^2), and the total heat produced per occupant was thus around 125 W. The balance between sensible and latent heats was calculated by the software used. The occupant was present in the room from Monday to Friday, from 9:00 to 18:00 with an hour as break at noon. Saturday and Sunday were free days and no public holidays were involved. The heat load due to office equipment was 5.4 W/m^2. According to ASHRAE [11], this value corresponds to a "light load office". The loads follow the schedules of the occupant. The lighting load was 6 W/m^2, a common value used in practice for an office. The lighting load was at 90% of its capacity from 9:00 to 10:59, at 70% from 11:00 to 12:59 and from 14:00 to 15:59, and at 100% from 16:00 to 17:59. In the other hours the light was switched off. The energy needed was calculated assuming a perfectly efficient HVAC system. The airflow network and the heating and cooling plants were not modelled; therefore the airflow needed was supplied at outdoor conditions. The humidity level was monitored but not controlled.

2.5. Simulated cases

From Fig. 1, assuming that the air temperature is equal to mean radiant temperature ($t_a = t_{mr}$), it is shown that the increase allowed in operative temperature is equal to 1.7 °C for an airflow of 0.5 m/s and 2.5 °C for an airflow of 0.8 m/s. These values were added to the maximum summer operative temperatures for the three categories as specified in EN 15251 [4]. The values shown in Fig. 1 were obtained for a comfort limit of 26 °C, which is the comfortable temperature limit for category II in EN 15251 [4]. It is reasonable to assume that the same increments in operative temperature can be applied for the comfortable temperature limits for categories I and III, i.e. 25.5 and 27 °C. In total, 54 cases, covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories (I, II and III) and three air velocities (<0.2, 0.5 and 0.8 m/s) as listed in Table 2, were simulated. The summer design day simulation was performed for 54 cases in order to calculate the maximum power needed for providing the comfort conditions. The maximum power is used to size the chiller. The summer design day conditions were taken from ASHRAE [11]. The cooling design days used in the simulation were characterized by an annual percentile of 1.0% for the dry-bulb temperatures and the mean coincident wet-bulb temperatures. These are suggested for use by ASHRAE [11] when sizing cooling equipment such as chillers or air-conditioning units.

2.6. Simulation software

A robust building energy simulation program, EnergyPlus, was used for the simulations. This software allows for performing simulations of the building and the HVAC system as a whole. It calculates the thermal loads to be satisfied and defines the system strategy needed to fulfil the required comfort conditions. In the present research, EnergyPlus is used mainly in order to predict the energy need for keeping the room operative temperature within the comfort limits (specified in Table 2).

3. Results

The energy need for cooling ($E_{N,C}$) [9] of the room when located in each of the selected six cities for the three categories (Table 2) at the three levels of velocity (0.2, 0.5 and 0.8 m/s) and the corresponding operative temperatures (Table 2) is listed in Table 3. The energy need for cooling is the annual amount of cooling energy that must be supplied to the room to keep the operative temperature below the maximum summer operative temperature limit. The cooling energy for the control of humidity and the energy losses in the system are not included.

The heating energy need is not affected by the air velocity increase. It depends on the outdoor conditions (climate zone) and the required category of the indoor environment. The maximum heating energy need is in Helsinki for category I (83 kWh/m^2 y). In Rome, Jerusalem and Athens the heating demand is covered by the internal heat load, and there is therefore no need for a heating system.

The fan operation total hours (h_i) are shown in Table 2 as well. It is assumed that when the indoor operative temperature is higher than the maximum operative temperature limit (without any increase of the air velocity) the occupant switches on the fan. Thus the fan operation hours were calculated as the sum of hours during which the operative temperature was higher than the maximum operative temperature limit and the occupant was in the room, e.g. an hour is counted if the occupant was in the room and the room operative temperature was above 25.5 °C for category I, or above 26 °C for category II, or above 27 °C for category III. The total number of hours that the fan is in operation is proportional to the energy consumption of the fan. In Table 3 the ratio between the fan operation hours and the total yearly occupant working hours is reported. The total occupant working hours (h_{tot}) per year (260 working days) is 2080.

Table 3
Energy need for cooling ($E_{N,C}$) per unit of floor and fan operating hours at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climate conditions

City	C[a]	Velocity								
		<0.2 m/s	0.5 m/s				0.8 m/s			
		Reference case	Energy		Fan		Energy		Fan	
		$E_{N,C}$[b]	$E_{N,C}$[b]	Saved[c] (%)	$h_{0.5}$[d]	$h_{0.5}/h_{tot}$[e] (%)	$E_{N,C}$[b]	Saved[c] (%)	$h_{0.8}$[d]	$h_{0.8}/h_{tot}$[e] (%)
Helsinki	I	18	12	34	636	31	9	48	645	31
	II	21	15	29	765	37	12	41	788	38
	III	24	18	24	859	41	16	35	867	42
Berlin	I	24	16	32	814	31	13	45	826	31
	II	26	19	28	848	37	16	40	864	38
	III	27	21	23	907	41	18	34	916	42
Bordeaux	I	39	28	27	1080	52	24	38	1091	52
	II	41	31	24	1184	57	27	34	1204	58
	III	42	33	21	1345	65	29	31	1368	66
Rome	I	52	40	23	1300	63	35	33	1308	63
	II	53	42	21	1406	68	37	30	1420	68
	III	53	43	19	1499	72	38	27	1509	73
Jerusalem	I	65	51	21	1483	71	45	30	1491	72
	II	66	52	20	1722	83	47	29	1746	84
	III	66	54	19	1909	92	48	27	1928	93
Athens	I	75	61	18	1419	68	56	25	1439	69
	II	74	61	17	1555	75	56	25	1579	76
	III	73	61	17	1888	91	55	24	1921	92

The energy saved due to the increase of air velocity (or relative increase of upper operative temperature limits) is listed.
[a] C = category according EN 15251 [4].
[b] $E_{N,C}$ = energy need for cooling (kWh/(m^2 y)).
[c] Saved = percentage of the saved energy need for cooling compared to the reference case.
[d] h_i = annual number of hours that the fan is operating for increasing the air velocity.
[e] h_i/h_{tot} = annual number of hours that the fan is operating (h_i) over yearly occupant working hours (h_{tot}).

The maximum cooling power per unit of floor area and the percentage of time that the relative humidity requirements are fulfilled when the occupant is in the room are shown in Table 4.

4. Discussion

In all simulated cases, increasing the air velocity implied a reduction of the energy consumption (Table 3). A saving of the

Table 4
Maximum cooling powers per square metre and percentage of time that the relative humidity requirements are fulfilled at the three velocity levels for the three categories of indoor environment when the room is located in the six cities with different outdoor climate conditions

City	C[a]	Velocity							
		<0.2 m/s (reference case)		0.5 m/s			0.8 m/s		
		Max power[b]	RH percentage[c]	Max power[b]	Saved[d] (%)	RH percentage[c]	Max power[b]	Saved[d] (%)	RH percentage[c]
Helsinki	I	49	65	42	15	65	38	22	65
	II	48	94	41	13	93	39	19	92
	III	45	100	41	10	100	38	15	100
Berlin	I	55	82	47	14	83	44	20	82
	II	51	98	45	11	98	42	17	98
	III	46	100	42	10	100	39	15	100
Bordeaux	I	60	71	52	13	77	49	18	78
	II	54	95	48	10	95	45	16	95
	III	47	100	43	9	100	41	14	100
Rome	I	60	57	53	12	67	50	18	72
	II	55	96	49	10	96	46	16	96
	III	48	100	44	10	100	41	14	100
Jerusalem	I	56	90	49	13	88	46	18	85
	II	50	99	45	11	98	42	16	97
	III	44	100	40	10	100	38	15	100
Athens	I	73	74	66	10	80	63	14	81
	II	65	97	59	9	96	56	13	96
	III	56	100	51	8	100	49	12	100

[a] C = category according EN 15251 [4].
[b] Max power = maximum cooling power (W/m^2).
[c] RH percentage = percentage of time that the relative humidity requirements are fulfilled when the occupant are in the room.
[d] Saved = percentage by which the maximum cooling power is reduced compared to the reference case.

energy need for cooling between 17 and 48% is obtained. The highest percentage of energy saving was obtained in Helsinki for category I of the indoor environment. The lowest percentage of energy saving was obtained in Athens for category III of the indoor environment. The percentage of savings decreases when the quality of the indoor environment category decreases, e.g. in Bordeaux for category I the saving was 27% and for category III it was 21%. The percentage of savings decreases with the increase of the cooling degree days (defined in Section 2.1). The percentage of savings increases when the air velocity increases. In fact, the higher savings have been obtained for an air velocity equal to 0.8 m/s. These conclusion can be drawn from Fig. 2. In summary, increasing the air velocity to compensate for the higher room temperature is an energy-saving solution that gives a higher performance in high quality indoor environment offices located in a cold climate. It is interesting to note that, in Helsinki, Berlin and Bordeaux, the energy need for cooling increased with the reduction of the quality of the indoor environment due to the free cooling effect of the outdoor air.

The fan operation hours are listed in Table 3. The fan operation hours increase with an increase in the number of cooling degree days (defined in Section 2.1) and with a reduction of the indoor environment category. The fan operation hours are almost independent of the increase of air velocity. In Table 3 the ratio between the fan operating hours and the yearly occupant working hours is shown. The ratio varies between 31 and 93%. High values of the ratio mean that the fan would work also during winter-time, when it is presumed that people dress with a clothing insulation equal to 1 clo. In this case the graph, as shown in Fig. 1, cannot be applied. However, the fan is working during winter-time in warm climates (Jerusalem and Athens), where the occupant would probably have lighter clothing. Moreover, during winter-time, it is reasonable to think that other techniques would be used to cool the room, such as night free-cooling, or increasing the shading capacity or the thermal mass of the building.

The relative humidity in the environment was not controlled by the system but it was monitored. From Table 4 it can be seen that for all simulated cases with category III the requirements for indoor relative humidity ($20\% < \mathrm{RH} < 70\%$) were always fulfilled. For cases with category II the requirements ($25\% < \mathrm{RH} < 60\%$) were fulfilled from 92 to 99% of the time, depending on the outdoor conditions and the relative increase of air velocity. For the cases with category I the requirements ($30\% < \mathrm{RH} < 50\%$) were fulfilled from 57 to 90% of the time, the rest of the time the humidity conditions were mostly within the range $25\% < \mathrm{RH} < 60\%$. A humidification and dehumidification system would be needed to keep the relative humidity always in accordance with the requirements in the standards for category I and II.

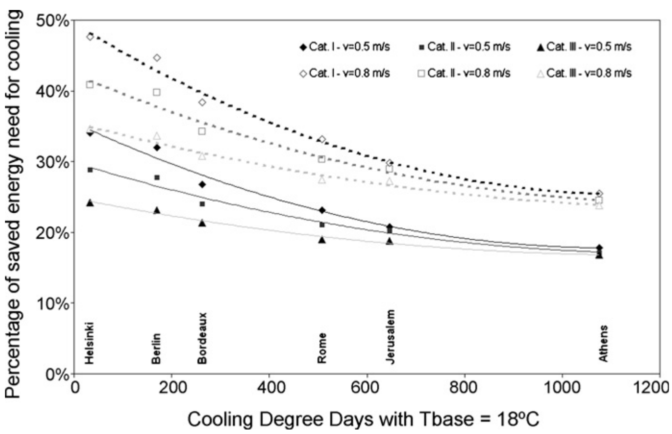

Fig. 2. Percentage of saved energy need for cooling vs cooling degree days. The points are the values obtained from the simulations. The lines are second order polynomial interpolations of the calculated data. The reference case for each category and city is the one without any increase in air velocity (<0.2 m/s).

The maximum cooling power per unit of floor area is shown in Table 4. The reduction of the maximum cooling power due to the increase of air movement is in the range 8–22%. It is higher for an air velocity equal to 0.8 m/s, for the cold climates and for higher quality level of indoor environment. The most effective parameter is the level of air velocity. As a consequence, smaller chillers may be installed, which will lead to a reduction of the initial (investment) costs.

4.1. *Energy consumption of the fan*

The air movement increase can be produced by ceiling fans (common nameplate power consumptions around 70 W), standing fans (50 W), tower fans (40 W), desk fans (30 W), personal ventilation systems and under certain conditions with operable windows. Measurements of several fans, performed in this study, confirm that the effective input fan power is equal to the value stated on its nameplate.

In order to check whether the electrical consumption of the fan is a critical factor for energy saving, the difference between the saved (in the chiller) and consumed (by the fan) energy is calculated. The saved electrical energy for running the chiller is named $E_{\mathrm{el,Cool}}$ and the electrical energy consumed by the fan is named $E_{\mathrm{el,Fan}}$. The difference between $E_{\mathrm{el,Cool}}$ and $E_{\mathrm{el,Fan}}$ is hereafter named net electrical energy saved ($E_{\mathrm{el,Net}}$). The saved electrical energy for running the chiller ($E_{\mathrm{el,Cool}}$) depends on the saved energy need for cooling (see $E_{\mathrm{N,C}}$ in Table 3), on the energy losses from emission, distribution and storage (taken into consideration in the calculations by η) and on the coefficient of performance (COP) of the chiller. COP and η depend on the type of cooling system used and on the building characteristics. The electrical energy consumed by the fan ($E_{\mathrm{el,Fan}}$) depends on the electrical input power of the fan (P_{Fan}) and on the number of fan operating hours (h_i). The net electrical energy saved ($E_{\mathrm{el,Net}}$) is defined by Eq. (1).

$$E_{\mathrm{el,Net}} = E_{\mathrm{el,Cool}} - E_{\mathrm{el,Fan}} = \frac{(E_{\mathrm{N,C}}^{v \le 0.2\,\mathrm{m/s}} - E_{\mathrm{N,C}}^{v=i})(1+\eta)}{\mathrm{COP}} - 10^{-4} P_{\mathrm{Fan}} h_i \quad (i = 0.5\ \mathrm{or}\ 0.8\ \mathrm{m/s}) \tag{1}$$

where $E_{\mathrm{el,Net}}$ is the net electrical energy saved (kWh/(m^2 y)); $E_{\mathrm{N,C}}^{v=i}$ is the energy need for cooling ($E_{\mathrm{N,C}}$) obtained when the air velocity is $i \le 0.2$ or $i = 0.5$ or $i = 0.8$ m/s (kWh/(m^2 y)); P_{Fan} is the electrical input power of the fan (W); h_i is the number of hours that the fan is operating (Table 3) (h); η is the ratio between the energy need for cooling and the thermal energy that the chiller has to produce; COP is the coefficient of performance of the chiller.

Practical experience shows that the COP can vary within the range between 2.5 and 4.5 with a best guess value of 3.5 and the η can vary within the range between 0 and 0.15 with a best guess value of 0.05. The influence of these two parameters on the net electrical energy saved, $E_{\mathrm{el,Net}}$, was calculated for Helsinki in the case of the indoor environment category I for velocity elevated to 0.5 and 0.8 m/s. From the results shown in Fig. 3 it can be seen that $E_{\mathrm{el,Net}}$ varies as a function of the COP and η for the two air velocities.

The results in Fig. 3 reveal that COP has a significant influence on the net electrical energy saved, and η has less impact. Moreover, it can be seen that $E_{\mathrm{el,Net}}$ is lower for higher values of COP, is due to the fact that the required electrical energy for producing a certain amount of cooling energy decreases with the increase of the COP.

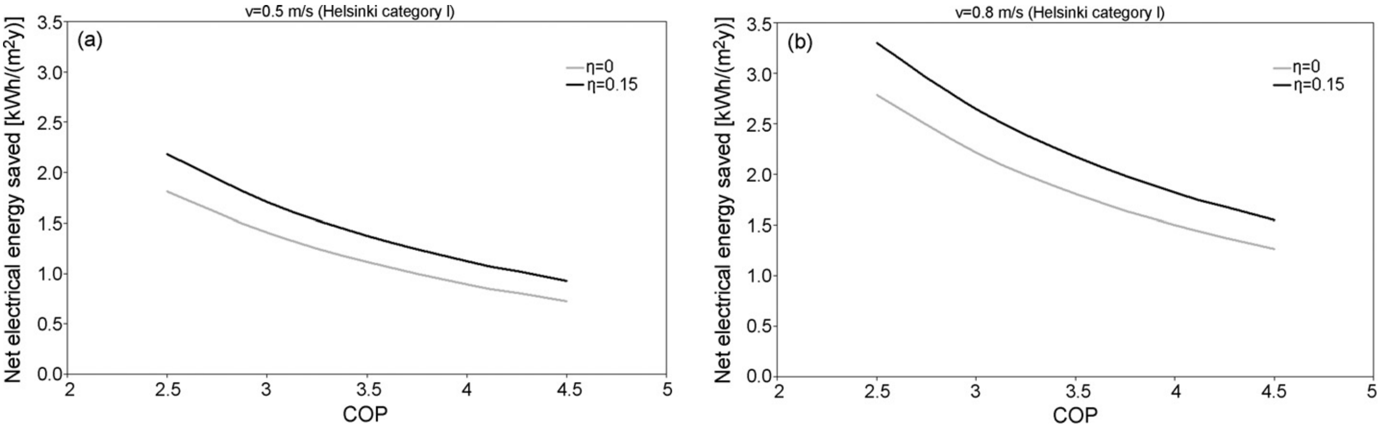

Fig. 3. The net electrical energy saved ($E_{el,Net}$) calculated for Helsinki for category I vs the COP for η equal to 0 or 0.15 for air velocity of 0.5 m/s (a) and 0.8 m/s (b).

Easy-to-use graphs for checking, as a rule of thumb, how much energy can be saved as a function of the fan input power are shown in Fig. 4. Four cases are reported, including two air velocities (0.5 and 0.8 m/s) and two combinations of COP and η. The combinations of COP and η were chosen in order to calculate the extreme cases. With COP = 2.5 and η = 0.15 the $E_{el,Net}$ is the highest, while with COP = 4 and η = 0 the $E_{el,Net}$ is the lowest. The net electrical energy saved ($E_{el,Net}$) was calculated for a fan input power within the range 2–70 W for all the 54 simulated cases. The maximum and minimum values for each fan input power has been plotted. The use of these graphs is explained in the following example. If the input power of the fan is 20 W, the COP is equal to 2.5, η = 0.15 and the air velocity is 0.8 m/s (Fig. 4a), the expected net electrical energy saved is then at minimum 2.1 kWh/(m^2 y) and at maximum 5.9 kWh/(m^2 y). On the other hand, with the same fan input power, if the COP is equal to 4, η = 0 and the air velocity remains the same (Fig. 4b), the expected net electrical energy saved is then at minimum 0.4 kWh/(m^2 y) and at maximum 1.9 kWh/(m^2 y). If the input power of the fan is still 20 W, the COP is equal to 4, η=0 and the air velocity is 0.5 m/s (Fig. 4d), the expected net electrical energy saved is then at maximum 0.5 kWh/(m^2 y). In this case, the minimum is not plotted because there is no energy saving but energy waste. The values plotted in Fig. 4 were obtained from computer simulations where the human behaviour was not

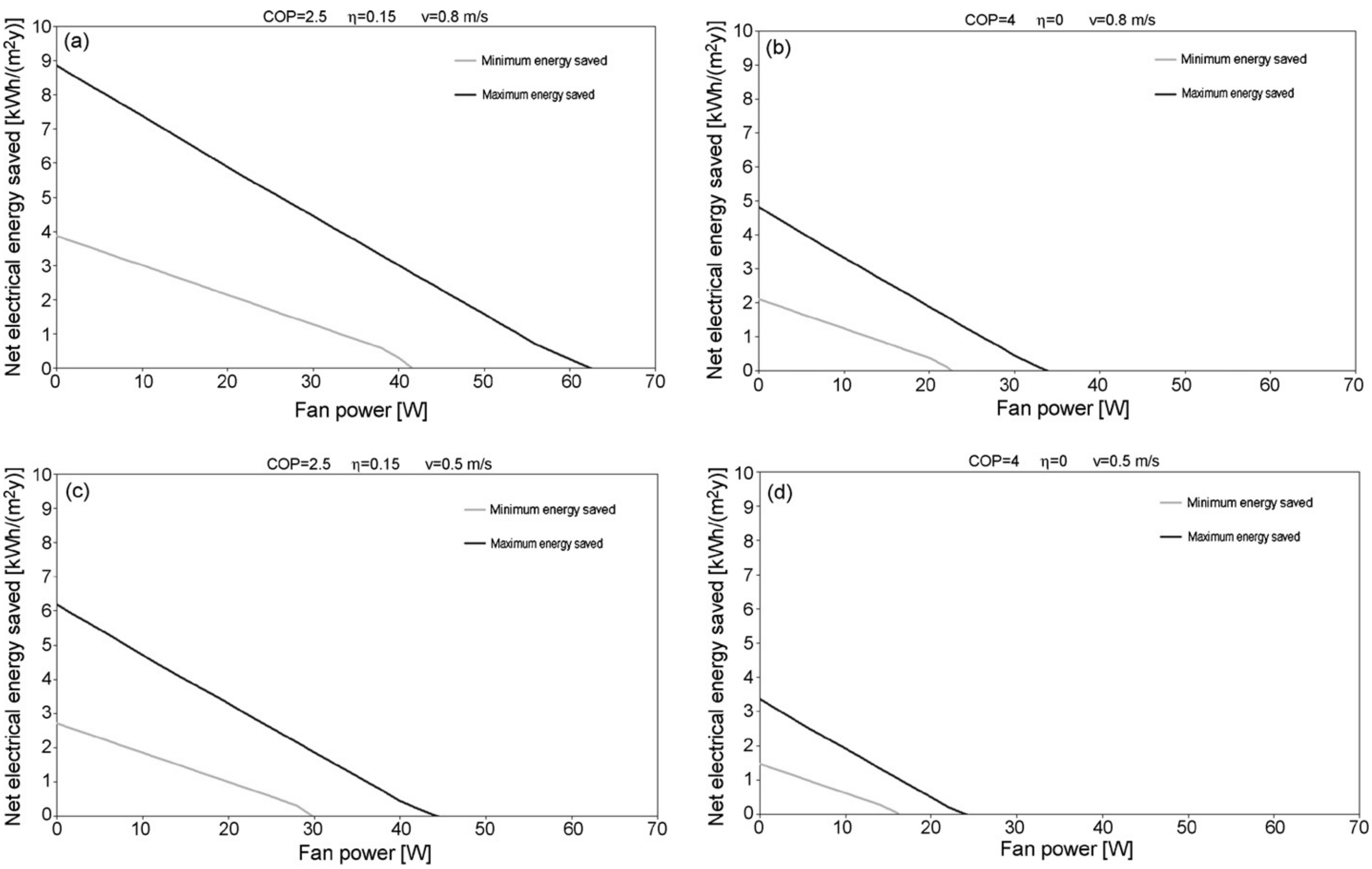

Fig. 4. The net electrical energy saved vs fan input power when: (a) COP = 2.5, η = 0.15 and air velocity = 0.8 m/s; (b) COP = 4, η = 0 and air velocity = 0.8 m/s; (c) COP = 2.5, η = 0.15 and air velocity = 0.5 m/s; and (d) COP = 4, η = 0 and air velocity = 0.5 m/s.

modelled. The human behaviour (e.g. leaving the fan switched on when the occupant is out of the office) affects the possibility of saving energy by using the technological solution studied in this paper. The main advantage of the presentations in Fig. 4 is that the graphs are independent of the location and of the indoor environment category and can therefore give a first estimation of the saving. For example, if the fan power input is 60 W, then it can be easily seen that energy savings cannot be achieved. From the figures, it can be concluded that traditional systems, such as ceiling fans (70 W) and standing fans (50 W), cannot be used to save energy on the basis of assumptions made in this study. From Fig. 4 it can be seen that for the conditions considered in this study (outdoor climate, indoor environment category, air velocity increase) and for the range of COP and η used, it is never possible to reach a net energy saving with a fan input power higher than 60 W. On the other hand, it is always possible to save energy if the input power is lower than 15 W. Calculations made for the best guess values for COP and η, respectively 3.5 and 0.05, reveal that energy savings will not be achieved with fans using more than 20 W. This can be done using a small desk fan or a personal ventilation system. The main conclusion is that the fan input power is a critical factor for the applicability of this solution in practice.

The results in Fig. 1 were obtained and verified with an airflow over the whole body [6] while personal ventilation systems or desk fans typically provide cooling only to the upper part of the body. Nevertheless, the authors believe that the difference would not be significant, because most of the heat loss occurs in the upper part of the body (the head is a strong dissipater of heat). Another advantage of the personal ventilation system is that it will increase the inhaled air quality and this will improve occupants' health and productivity [12].

4.2. Limitations of the study

The HVAC system was not modelled; therefore the interaction between the building and the system could not be predicted. The moisture control was not modelled either. These simplifications may change the range of saved energy need for cooling. Sensitivity analyses for internal and external heat loads and behaviour of the occupant have not been performed.

5. Conclusions

The main conclusions of this study are:

- Cooling energy savings in the range of 17–48% have been obtained in the case of increased room temperature and elevated velocity. The percentage of savings increases when: the air velocity increases, the indoor environment category level increases, and the number of cooling degree days decreases.
- The required power input of the fan is a critical factor. Traditional systems, such as ceiling, standing, tower and desk fans may not be applied to save energy under the assumptions made in this study.

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