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The employment of an earth-to-air heat exchanger as pre-treating unit of an air conditioning system for energy saving: A comparison among different worldwide climatic zones

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Abstract
A great fraction (20–40%) of primary energy is required for building air conditioning, so the use of renewable energy sources is increasing. The geothermal energy for Heating, Ventilating and Air Conditioning (HVAC) systems can be used considering an Earth-to-Air Heat eXchanger (EAHX). This work analyses the performance of an EAHX through a mathematical model (2D), as a function of diameter and length of the air ducts. The problem is solved with finite element method. A case study office building is analyzed. The air conditioning plant is characterized by fan-coil units and primary air; the EAHX is positioned upstream the Air Handling Unit (AHU) to pre-cool/pre-heat the air. The building is virtually placed initially in six Italian cities (different climatic zones according to Italian regulation DPR 412/93) and subsequently in eight worldwide cities according to Köppen climate classification. The following parameters are calculated: air temperature variation and thermal efficiency of the EAHX; the decreasing of cooling and heating capacity of the coils into the AHU. The best results refer to a duct length of 100 m for Ottawa (warm-summer humid continental climate, 65% capacity reduction), the worst ones for Rio de Janeiro (tropical wet and dry climate, maximum 24% reduction).

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1. Introduction
Buildings are attributed the responsibility of accounting for 20% – 40% in the global energy consumption of developed countries, often overcoming the industry and transport fields. Within the building sector, Heating Ventilation & Air Conditioning (HVAC) systems represent the greatest source of energy demand (50% of the total energy consumption of the sector) that represents 10–20% of global energy consumption [1]. Indeed, the energy policies prescribe as primary goal to adopt energetically optimized solutions, also with reference to ventilating and air conditioning systems. Thus, restrictive standards are fixed to keep these prescriptions such providing energy certificates for buildings that incorporate all the aspects concerning with them (materials, the standardization of the checks for the regular maintenance, etc...). In most of the cases, the vapor compression systems could not overcome certain energy limits; consequently, the adoption of HVAC systems exclusively based on vapor compression would not satisfy the energy efficiency guidelines prescribed by these policies. Therefore, many worldwide directives push towards a spread utilization of energy from renewable sources to satisfy the energy constraints, and this also applies to buildings. As a matter of fact, to encourage these aspects, the European Directives on renewable sources issued the “green-building” standards.

Nowadays 14% of the worldwide energy demand is addressed by means of renewable energy sources [2,3] and the global goal is to bring this figure to grow significantly in the next decade. The research for solutions that could supply or integrate vapor compression systems for refrigeration and HVAC, constitutes a spur in the development of renewable energy-based systems for exploiting green energy and, thus, for counteracting the enormous energy demand connected to these fields. Among the most used renewable energy sources (wind, solar, biomass, waves, tides, ...), geothermal energy is very useful for air conditioning systems of buildings. In fact, beginning from certain depths, the soil temperature is almost constant during all the year; moreover, it is frequently higher than the temperature of the outside air in winter, lower in summer. Clearly, the specific value of the temperature is a dependent variable of the geographic location, but the common denominator is the possibility of taking advantage from this property for projecting geothermal systems with the capability of air
cooling during the summer and air heating during the winter [4].

Consequently, the ground assumes a double role: a sink when the system operates in cooling mode; a source when it works in heating mode. The results are that a part of the primary energy could be preserved and the environmental impact [5-7] of the system could be mitigated [8,9]. Specifically, the use of geothermal energy for improving the thermal indoor environment is typically addressed by means of the following three solutions [10]:

- earth homes;
- Ground-Source Heat Pumps (GSHPs);
- Earth-to-Air Heat eXchangers (EAHXs).

The first solution refers to buried buildings, so the contact with the soil reduces their heating and cooling loads. Ground source heat pumps are systems where the secondary fluid (typically water or antifreeze glycol–water mixtures) circulates throughout banks of underground ducts, in closed loop circuits, to exchange heat with the soil.

Earth-to-air heat exchangers are systems formed by a number of ducts, horizontally or vertically placed and buried in the ground at a depth useful to exploit the ground property of exhibiting, under undisturbed conditions, constant temperature during the whole year. The heat transfer fluid to be used in EAHXs is typically water: in the most common configurations the EAHX is inserted in a mechanical ventilation system, more rarely in a primary air circuit. As the ground temperature, below about 10 m depth, is often higher in winter and lower in summer than air temperature [11], the EAHX gives rise to a pre-heating of the external ventilation air in winter and pre-cooling in summer.

The EAHXs should be adequately projected to let the air, during the ducts blowing, brings its temperature close to the undisturbed air in winter and pre-cooling in summer.

**Roman symbols**
- A amplitude of the temperature annual oscillation, °C
- a daily equivalent thermal diffusion of the ground, m² day⁻¹
- AHU Air Handling Unit
- C coefficient of K-ε model
- c specific heat, J kg⁻¹ K⁻¹
- COP Coefficient of Performance
- D diameter of the pipe, m
- d distance between two pipes, m
- E energy, J
- EAHX Earth-to-Air Heat eXchanger
- G incident radiation, W m⁻²
- h specific enthalpy, kJ kg⁻¹
- HVAC Heating, Ventilation & Air Conditioning
- I identity vector, -
- _j component of diffusion flux, kg m⁻³ s⁻¹
- K turbulent kinetic energy, J
- k thermal conductivity, W m⁻¹ K⁻¹
- L length of the pipe, m
- m mass flow rate, kg s⁻¹
- p pressure, Pa
- Q thermal power, kW
- r room (thermohygrometric conditions)
- Re Reynolds number, -
- S orthogonal section area of the pipe, m²
- S source term, kg m⁻³ s⁻¹
- T temperature, °C
- T(g,D,t) ground temperature at a depth D after t days (starting from 1st January), °C
- T_min yearly average temperature of the outdoor environment on the basis of statistical information, °C;
- t time,
- t_min sequential number of the day corresponding to the minimum ground temperature, according to statistical data (1 refers to 1st January);
- U convective heat transfer coefficient, W m⁻² K⁻¹
- u longitudinal fluid velocity, m s⁻¹
- v orthogonal fluid velocity, m s⁻¹
- V volumetric flow rate, m³ h⁻¹
- x longitudinal spatial coordinate, m
- y orthogonal spatial coordinate, m
- z generic property, m

**Greek symbols**
- α absorbance of the surface
- Δ finite difference
- ∂ partial derivative
- ε efficiency, %
- ε turbulent cinematic viscosity, m² s⁻¹
- μ dynamic viscosity, Pa s⁻¹
- ν cinematic viscosity, m² s⁻¹
- ρ density, kg m⁻³
- τ tangential stress, Pa m⁻¹
- φ relative humidity, %
- ψ porosity, %
- ω specific humidity, g v kg⁻¹

**Subscripts**
- 0 phase constant of the lowest average/mean soil surface temperature since the beginning of the year
- AHU Air handling unit
- air air
- db dry bulb
- cc cooling coil
- c convective
- co condensed
- EAHX Earth-to-Air Heat eXchanger
- f fluid
- ground ground
- inlet inlet of the EAHX
- j species
- j liquid liquid
- m annual mean soil
- mass mass
- o outdoor
- outlet outlet of the EAHX
- related to the evaluation of turbulent dynamic viscosity
- p (constant) pressure
- pipe pipe
- PRE pre-heating
- r room
- ReH re-heating
- SA supply air
- sa sun-air
- soil soil
- solid solid
- surface surface
- T turbulent
- w water
plant; number of the ducts; etc. Based on the air circulation mechanism, EAHX are classified into two macro categories: open loop and closed loop systems. In open loop systems the heat transfer fluid is the environmental external air that crosses the buried ducts where it is treated (heated or cooled). Subsequently the air is either sent directly to the building that has to be ventilated/air-conditioned or carried in a conventional air handling unit (inside a HVAC system) to be further heated or cooled and then sent to the building. The open loop system provides that the air, after has completed its “conditioning” task, would be expelled directly from the building into the atmosphere. In the closed loop systems, at the end of the air conditioning process (heat exchange with the ground through flowing in the ducts and heat transfer with the building to be conditioned) the air is fed back at inlet of the EAHX ducts to be recirculated. Therefore, in a closed loop EAHX system, after several circulations, the air needs to exchange a relatively lower amount of heat with the ground compared to an open loop system. In general, the closed loop configuration is energetically more efficient than the open loop one, also allowing to reduce the problem of undesired water condensation in the ducts due to the humidity rate of the external air introduced in the EAHX (a typically summer problem). However, the open loop system is often preferable because it also allows the air exchange in the building, which is not possible for the full air recirculation systems; moreover, the air recirculation can carry to the contamination of the HVAC systems by Coronavirus or other viruses.

EAHX systems can be characterized by vertical or horizontal air ducts. The arrangement of the pipes plays a fundamental role since the portion of the ground required for the installation of the EAHX systems in order to satisfy the heating/cooling demand depends on the design and layout of the pipes; indeed the air conditioning potential of an EAHX system is strongly linked to its geometric configuration. Horizontally oriented pipes are generally used in EAHX systems, mainly because they present a simpler and cheaper type of installation than the vertical ducts, since the former requires a shallower excavation. A further classification in the EAHX system with horizontal ducts can be made between single-layer configuration, with all the ducts buried at a single depth level, and multi-layer configuration where the ducts, horizontally oriented, are buried one on the other at various depths in the ground and separated by vertical drops. Single layer configurations are by far the most used among the solutions proposed in literature [10,12]. At the best of our knowledge, very few are the investigations performed on multilayer EAHX systems but worthy of attention is the work proposed by de Jesus Freire et al. [13]. They made a comparison in an EAHX between multilayer pipes and single layer configurations, on equal number of tubes and distance between one each other, as well as the same were the duct design parameters (diameter, length, air velocity) [13]. On equal amount of heat transfer surface, number of tubes and air velocity, they detected 3% and 6% decreasing in temperature span, respectively considering the two- and three-layer configurations with respect to the single layer one. On the other side, the two- and three-layer configurations analyzed required a reduction of the available flat surface for installation estimated, respectively, on 50% and 67%, if compared to the single-layer one. Indeed, despite of slight energy performances decreasing, the multi-layer configuration could prove very promising in urban contexts with limited installation surfaces.

Earth-to-air heat exchanger is a very promising technology but mandatory is the optimization of the system to the purpose of appropriately setting the design parameters (such as diameter, length and number of tubs, displacements of the tubes, air velocity), according to the installations specifics and limits as well as to the geographical zone, in order to let the EAHX system showing the highest energy performances [14]. To pursue this goal, before the installation of the system, it is important to widely test the projected EAHX system. This crucial point could be addressed by means of the development of an accurate numerical model able to predict the energy performances under a wide number of working conditions.

A lot of mathematical models are present in the scientific literature: many models are commented in the next paragraph (State-of-the-art).

Based on the literature analyses, it is therefore possible to briefly summarize the main advantages in using and investigating EAHX systems: i) the working fluid is air (unlimited and free available); ii) the energy consumptions of stand-alone EAHX or EAHX/ HVAC-coupled systems are lower than the traditional HVAC systems, as well as higher are the coefficients of performances too; iii) the EAHX system is simple, therefore it requires few maintenance and operating costs; iv) the environmental impact deriving from the operation of the EAHX systems is reduced with respect to the traditional ones, since the former is supplied by a renewable energy source and, furthermore, it requires less use of compressors and high-GWP (Global Warming Potential) refrigerants. However, it must be pointed out that the use of the EAHX is not yet widespread. This is due to both the space problems related to the installation of buried pipes, which can be problematic in widely urbanized contexts, and the excavation costs necessary for burying the pipes.

2. State-of-the art, research gap and goal of the paper

In this section many of the numerical models or computational methods reported in the scientific literature for the analysis of the EAHX are reviewed and summarized.

Bordoloi et al. [15] in their review paper classified and compared the energy performances of the main EAHX systems describing the most relevant analytical and experimental studies on the different combinations of EAHXs, up to the year 2018. Another very appropriate classification was proposed by Bisoniya et al. [16] in their review where, specifically, the numerical models of EAHX were categorized based on the method they are solved through and the dimensions of the geometry investigated. Anyhow, the common denominator of both the reviews is to propose an overview of the worldwide research scenario on this type of geothermal system; the emerging data is the really huge number of EAHX systems and models proposed. An accurate numerical model should be able of evaluating punctually both the conductive (from/to duct/ground) and the convective (related to the air flowing in the duct) heat transfer mechanisms acting in the EAHXs. Even if a number of commercial software like TRNSYS or Energy Plus allows to easily model geothermal systems, providing qualitative data on their performances (“black-box approach”), they are not able to provide punctual (in space and time) indications on the heat transfer and temperature fields in the whole systems. Indeed, to perform accurate heat transfer investigations on the operation of earth-to-air heat exchangers, the most appropriate solution is represented by Computational Fluid Dynamics (CFD)-based models, founded on the discretization of the domain in finite differences/volume/elements despite of the method adopted for solving the differential equations that govern the heat transfer problem.

In open literature, various 1-D, 2-D, and 3-D models of earth-to-air heat exchangers were presented and described. Over the years, the beginning investigations on EAHXs founded on the development of one-dimensional models with simple balances made to derive the inlet–outlet relations for the air parameters. In 2002 Kabashnikov et al. [17] introduced one of the first one-dimensional models of an earth-to-air heat exchanger; the mathematical model was very simple as well few were the
results collected: based on Fourier integral for evaluating the temperature in the system, a mathematical investigation was carried out, by varying length, diameter and depth of the ducts. As main result the model can provide an analytical expression giving the mathematical value of the length and the diameter that optimizes the heat exchange between the air and the ground. Subsequently, in 2003 Kumar et al. [18] presented a parametric analysis carried out through a 1-D finite differences numerical model of an earth-to-air heat exchanger that couples simultaneously the heat and mass transfer equations. The tool was developed through a MATLAB code and validated with experimental data, coming from a system placed in India, and a good agreement (±1.6% relative error) was found. The peculiarity is that the model of Kumar et al. has been one of the first to investigate the transient behaviour of the EAHX for a whole day in summer and in winter, as well as the humidity of the air flowing was considered. In the same year, De Paep and Janssens [19] shared with the scientific community their one-dimensional analytical model where the convective heat exchange is accounted through the calculation of the convective coefficients by means of dimensionless numbers approach. They evaluated the influence of pressure drop as function of volumetric flow rate, of diameter and length of the duct. They noticed that smaller diameters provide higher thermal performance, but greater pressure drops. The solutions they suggested is to project EAHX with more ducts placed in parallel to counteract these contrasting trends. Through the one-dimensional mathematical model proposed in 2008 by Cucumo et al. [20] the effect of burial depth of the tubes on the energy performances of earth-to-air heat exchanger systems was evaluated. The model is able to provide the results following two methods: superposition principle, Green functions. They performed the investigation in a sandy soil, and they asserted that optimal depthness belongs to the range 3–6 m. The effect of burial depth was also investigated by Sehli et al. [21] by means of a finite volume CFD model. The convective heat exchange of the fluid flowing under turbulent motions was evaluated through the k-ε method. They identified 4 m as optimal depth and they noticed that, as soon as Reynolds number increases, the inlet–outlet temperature span decreases due to the less time spent by the air in tube and, consequently, for the heat exchange with the ground. Among the 1-D models that are worthy of mention, there is the one developed by Su et al. [22] since the approach is very unconventional: the EAHX was modelled through a sequential computing algorithm described with a pseudo-code. The model is the coupling of two sub-codes: one accounts transient convection–diffusion sub-model evaluating the temperature and the moisture content of the air; the other computes 1-D transient heat conduction sub-model in a rocky soil.

An intense study is the one of Serageldin et al. [23] that, with their one dimensional CFD transient model of EAHX experimentally validated, asserted that with reference to Egyptian weather: i) the larger are duct diameter, length and distance, the higher is the inlet–outlet temperature span; ii) greater air flow velocities reduce the inlet–outlet temperature span; iii) the duct material does not affect significantly the heat exchange between the air and the ground. Another interesting contribution was given in 2015 by Niu et al. [24] where, through a regression algorithm applied to a 1-D steady state model, the cooling capacity of an earth-to-air heat exchanger was predicted with extreme accuracy. The polynomial regression model bases on six calibration parameters: temperature, relative humidity and inlet velocity of the air, surface temperature, length, and diameter of the tube. The obtained formula could be of wide usage in designing the EAHX systems. Among the latest 1-D models proposed, worthy of note is the one of Cuny et al. [25] published in 2020, where, following the multi-criteria optimization based on genetic algorithms, the Pareto front for an EAHX systems was determined. Three were the criteria selected, two energy and one economic, and they were applied to the operation of an EAHX with reference to French climates. The optimum combination suggests large duct length (97 m), whereas small should be duct diameter (0.15 m) and air flow velocity (0.7 m s⁻¹). A good compromise between performances and cost for burial depth could be 3.2 m. Furthermore, in 2020, a one-dimensional model was used by Lin et al. [26] to quantify the correlation between the moisture of the ground and the log-term energy performances of an earth-to-air heat exchanger. Three different cases were considered: partially and fully saturated, fully dry. The results show that the EAHX energy performance is not affected by the soil moisture when the air velocity is low (up to 1 m s⁻¹) but for higher velocity the effect of the moisture in the soil affects significantly the energy performances, since also basing on their operative condition the flow evolves in turbulent. The performances are higher the more is the soil saturation and a 40% difference in energy performances between the fully dry and fully saturated grounds was appreciated. Moreover, they asserted that the maximum air flow velocity in the tube should not overcome 4.0 m s⁻¹.

One-dimensional models show some limits: they cannot calculate the field of speed and temperature of the air into a transversal section of the duct, neither the temperature field of the soil when the depth varies. These limits can be overcome by the development of two-dimensional models. Most of the 2-D models of earth-to-air heat exchangers presented in literature is solved through finite element methods. An easy and accurate two-dimensional model of a ground to air heat exchanger was introduced by Badescu in 2007 [27]. The numerical model is time-dependent, and it was realized through a computer code in Pirmansens PH. The geometry considered identifies a horizontal single duct (length = 36 m) and the simplified assumption of neglecting the effect of the mass transfer in the heat transfer balance was considered: indeed the air was not considered as humid air and the eventual water condensation was not accounted. The model is implemented through dimensionless numbers approach and solved through finite volume method. The heating and cooling potentials of the EAHX under real climatic conditions (with experimental measured weather data) (year 2000 Chemnitz, Germany), were analysed. In 2016, Ahmed et al. [28] proposed an intense investigation on the optimization of the energy performances of a horizontal earth-to-air heat exchanger with respect to the most salient parameters of the system. The model is 2-D and CFD-based implemented through the software FLUENT 15.0. The Navier-Stokes equations were solved through the finite element method and the effect of the turbulent flow in the above equations was accounted through the k-ε model. The impact of air velocity, as well as duct depth, diameter, length and material on the energy performances was studied with reference to a hot subtropical Australian climate and the results revealed the length of the tube being the most influential factor. Furthermore, the analysis showed that there is not a univocal combination of design parameters to optimize all the energy performances at the same time: i.e. the air flowing with 1.5 m s⁻¹ velocity in a clay duct (0.003 m thickness) with 8.0 m, 60 m, 0.062 m as respectively depth, length and diameter, optimizes the efficiency of the EAHX but not the energy saving. To maximize the latter parameter another combination of such parameters is required.

A further important problem to be analysed by means of 2D models is represented by the soil thermal saturation caused by the presence of the EAHX. This aspect was studied by Niu et al. [29] in 2015 through a numerical model founded on the transient method of control volume. They observed that the effect of saturation of the soil becomes relevant in the degradation of the cooling capacity if the system operates in continuous mode, whereas an intermittent working mode during the day allows the soil thermal
recovery and consequently it mitigates the degradation of the energy performances. The thermal performances of a ground to air heat exchanger operating with a saturated soil were investigated by Zhao et al. [30] through a 2-D model that was simultaneously experimentally validated, too. The saturation was studied through the development of a theoretical model with Darcy natural convection in the soil treated as porous medium and the Keller shooting method was employed for solving the model. It was detected that the saturation of the ground is mostly sensitive to its initial temperature, the air inlet temperature and flow rate.

A two-dimensional model of an earth-to-air heat exchanger was also approached through the concept of artificial neural network by Kumar et al. [31] in 2006. They developed two models, a deterministic and an intelligent one: the former was needed to concurrently study the heat and mass transfer of the EAHX; the latter is the employment of data driven model founded on artificial neural network. The most salient parameters of the ground to air heat exchanger were considered as influencing meters of the energy performances and the investigation revealed that the intelligent model predicted the outlet temperature of the air from the duct with a ±2.6% error with respect to the ±5.3% proper of the deterministic one.

With 3-D models, some other investigations about the EAHX were performed, with reference to: energy performance of the plant for various layouts of horizontal air ducts (grid configuration, radial one, parallel one); thermal interaction between adjacent air ducts. On the other side, also due to the need of accounting the turbulence of the flow, the 3-D models result much more complex in resolution and onerous from computational costs point of view.

Among the first 3-D proposed models, Wu et al. published in 2007 [32] a CDF implicit transient finite volume method code developed through the PHOENICS platform. The design of the model identifies two single layer parallel horizontal tubes and the system was tested by varying length, diameter, buried depth, and air velocity. The results revealed the same parameters trends shown by the above already introduced 1-D and 2-D models (i.e. better results with larger lengths and smaller diameters and air velocities). In 2010 Mnasri et al. [33] introduced a 3-D model of a buried coaxial earth-to-air heat exchanger solved through a hybrid model: the finite element method was applied for the heat transfer problem, whereas the finite volume method was considered for the laminar flow evaluation. This method is called FVM (Finite Volume Method)-BEM (Boundary finite Element Method) and it allows to get faster results even despite of the complexity of the model due to its 3-D nature. In this work the heat transfer coefficients were evaluated and the data collected were compared with the ones provided by a conventional commercial CFD-based 3-D model. The results showed good agreement; indeed, this tool based on hybrid model could represent a fast solution to get the energy performances of EAHX systems. Furthermore, among the novel noteworthy numerical studies on 3D models, there is the one of Zhao et al. [34] where, based on an experimental 1:20 scaled earth to air system, a corresponding 3D model is developed to study the influence of the different parameters on thermal performances (mainly the length and diameter of the tubes and the inlet air velocity). Authors noticed that the thermal extraction efficiency increases with the duct length and decreases for rising values of diameter and air velocity. The highest value (0.96) was detected when the EAHX operates in cooling mode, with air speed of 0.5 m s⁻¹ and a duct diameter of 20 mm, whereas the largest cooling and heating capacities calculated are, in that order, 21.17 kW and 21.72 kW. Furthermore, they asserted that at burial depth below 4 m, the soil temperature is not so accurately determinable. In 2015, Bisoniya et al. [35], through the development of a quasi-steady state CFD 3-D model, focused on the energy payback period, the annual thermal performances and the seasonal efficiency ratio of an experimental EAHX system installed in Bhopal (Central India). They asserted that considering 50 years as lifetime, the EAHX system allows the reduction of 101.3 tons of CO₂ whereas the total carbon credit has estimated being around $ 2838. Always about considering the operation of the EAHX for many years of system life, interesting is the concept of “derating factor” introduced by Bansal et al. [36], that accounts the degradation of the thermal performances over the time. It is defined as instantaneous inlet–outlet temperature span detected on the corresponding steady state one. Indeed, due to the saturation of the soil, the smaller is the ratio, the greater is the degradation of the thermal performances.

The study on the effect of different displacements of the ducts in an earth-to-air heat exchanger is a crucial concept that has been deepened through 3-D models in various works. In 2012, Congedo et al. [37] performed a comparative investigation for evaluating the thermal and energy performances of three different duct configurations: linear, helical and slinky and the effect of the variation of geometrical and functional parameters were studied for each one. The investigation was made by means of a 3-D CFD tool developed in Fluent ambient and the EAHX was supposed to be placed in South Italy. For all the geometrical configuration, the optimal buried depth in terms of costs and performance is 1.5 m. In terms of energy performance, the better design resulted to be the helical one but, on the contrary, the installation costs are higher that the linear one. In 2017 Mathur et al. [38] investigated about the straight and spiral configurations for a ground to air heat exchanger. The performances of the system were analysed over a year while it operates in cooling and heating modes. The comparison was made also in terms of COP and, on equal design and operative parameters, they observed that the cooling/heating mode COP are 6.24 and 2.11 for spiral design whereas 5.94 and 1.92 are the ones proper of the linear ducts system. About the thermal performances of spiral configuration, Benrachi et al. [39] in 2020 reported an interesting comparative investigation where the behaviour of the EAHX was studied for different pitch spaces through a 3D model based on finite volume method, developed in ANSYS FLUENT ambient. Specifically, the impact of air velocity on efficiency and coefficient of performances was analysed for different pitch distances: these two indexes decrease if smaller are the pitch spacing. The inlet–outlet temperature span is greater of 6 °C more for 2 m pitch with respect to 0.2 m. The augmentation of the air flux speed from 2 m s⁻¹ to 5 m s⁻¹ carries to a reduction in efficiency and COP from 60% to 33%. When the air velocity increases from 2 m s⁻¹ to 5 m s⁻¹ the mean efficiency and the COP decrease from 60% to 33% and 2.84 to 0.46, respectively.

As for 1-D and 2-D, the artificial neural network approach was applied also to a 3-D model, through a deterministic model developed by Mihalakakou [40] where the author found that this approach could accurately estimate the outlet temperature of the tubes.

A well-designed EAHX system can be used independently but it can also be coupled to a traditional HVAC system to meet the heating/cooling requirements of the buildings. In the inherent scientific literature, a number of interesting works investigated the energy performances of HVAC plants coupled to/integrated with earth-to-air heat exchangers systems. In 2012, Bansal et al. [41] analysed the energy saving and economic impact deriving from integrating the earth-to-air heat exchanger technology into an evaporative cooling system. Specifically, by means of a CFD tool, they considered four base cases of air-conditioning and electric heater systems characterized by three diverse blowers: energy efficient blower, standard blower, and inefficient blower. With reference to these cases, the energy saving and the payback period related to the use of the EAHX were evaluated. They found that a 2 years payback period for integrating EAHX with an efficient blower evaporative...
system is very convenient. On the other side, for inefficient blowers the integration of an earth-to-air heat exchanger would result in a financially unviable choice. The authors showed that the energy saving was hardly affected by the electricity tariff and the blower efficiency. In 2016 Ascione et al. [42] analysed the effects on energy efficiency and environmental impact of employing an earth–to–air heat exchanger in the air conditioning system enlaving a Nearly Zero Energy Building (NZEB) through the software Energy Plus. The HVAC plant based on an air-to-water heat pump supplying fan-coil units plus mechanical ventilation: the EAHX was employed as pre-treating unit (pre-heating in winter and pre-cooling in summer). They observed that the EAHX integration carried up to a 29% energy saving in winter and 36–46% in summer, for a global yearly saving rate of 24–38%.

In 2020, Li et al. [43] analysed, from energy, environmental and economic point of view the integration of EAHX in an air-to-air heat recovery unit based on mechanical ventilation with respect to the case of coupling a heat pump to a primary air handling unit. The EAHX was tested in a parametric analysis with two parallel horizontal ducts buried at 2.5 m and 5 m with 2 m and 5 m as space between the tubes. The results showed that for severe cold climates, the EAHX-based solution carries remarkable benefits with respect to all the three above aspects. The static and dynamic payback periods for the EAHX-based system were about 2.1 and 2.4 years (with return rate of 8%); a reduction of 17% in equivalent emissions of CO₂ was also calculated. The most promising results were obtained with 5 m distance between the two ducts. D’Agostino et al. [44,45] evaluated the thermal performances of EAHX compared to air-to-air heat exchangers providing promising energy savings also for this configuration but with higher economic costs.

Indeed, the above state-of-the-art on numerical models of earth-to-air heat exchanger systems revealed the limits shown by one-dimensional models, especially in the impossibility of drawing the temperature and velocity profiles of the air flowing in the pipes, as well as the temperature range established around the pipes. As a matter of fact, even if these limits could be overcome through the development of both two- or three-dimensional models, currently, the vast majority of the numerical EAHX tools is 2D because the latter represents a good compromise in accuracy and computational costs. Anyhow, for complex geometries or particular placements of the pipes, where is needed, the 3D model is used.

At the best of our knowledge, the state-of-the-art lacks a worldwide comparison on the performances of a hybrid HVAC system where an EAHX is installed upstream the Air Handling Unit (AHU). Specifically, the present paper aims to fill this gap. At this aim, a case study office building is analyzed and virtually collocated in various climatic zones around the world. The HVAC system is based on fan-coil units and primary air, and the EAHX is installed upstream the Air Handling Unit (AHU) for primary air. For the above considerations on the model dimensions, a 2D mathematical model of the EAHX is developed to obtain the system performance under different outdoor air temperatures. The problem is solved through finite element method. The analyzed EAHX is characterized by 5 horizontal circular ducts disposed in parallel at 2.5 m deep. The EAHX is considered to pre-cool/pre-heat the open-loop airflow (primary air) of the Air Handling Unit. To make a comparison, the office building is virtually placed in six cities of Italy, which belong to different climatic zones according to Italian regulation DPR 412/93. For a further comparison, the case study building is also virtually collocated in eight localities around the world, basing the choice on the climate classification proposed by Köppen. The EAHX is simulated and optimized as a function of the diameter and length of the air ducts. The following parameters are calculated: the variation of air temperature in the EAHX; its thermal efficiency; the decreasing of cooling and heating capacity of the coils into the AHU when comparing with the solution without EAHX. The analysis on the coils of the AHU is performed for winter, summer and for all the year.

After having analysed the state of the art, it is evident that the thermal and energy performances of the EAHX are greatly influenced not only by the characteristics of the soil, but also by the boundary climatic conditions of the place where the exchanger is located. For these reasons, the main innovative contributions of this paper are as follows:

- a worldwide comparison about the thermal performance of an EAHX, which, at the best authors’ knowledge, lacks in the scientific literature;
- the proposed comparisons are not only based on cities with various climates, but also considering different classifications of climatic conditions in the world;
- the EAHX has been commonly investigated as a component added to a usual mechanical ventilation system, while this paper analyses a hybrid air conditioning system in which the EAHX is inserted upstream the air handling unit to minimize the energy requirements;
- for various climatic zones around the world, the thermal efficiency of the EAHX is evaluated, and also the decreasing of cooling and heating capacity of the coils into the AHU.

### 3. Methodology

The methodology of this paper is based on a 2D mathematical model of an EAHX to obtain the system performances under different outdoor air temperatures. The EAHX is considered not only as an air pre-treatment device placed inside a mechanical ventilation system, but as a component to pre-treat the air to be conditioned into an air handling unit inside a HVAC system. In this way it can ensure a relevant energy saving.

The investigation is conducted on a HVAC system for an office building. The building is spread over two floors for a total area equal to 260 m² and a volume equal to 910 m³. In Fig. 1a and Fig. 1b the ground floor and first floor are shown.

The investigated HVAC system is based on fan-coil units and primary air, with an upstream-placed horizontal-ducts EAHX system. According to Italian technical standard UNI 10,339 [46], the design ventilation outdoor airflow has been set at 11·10⁻³ m³ s⁻¹ per person, for a total amount of 3.61·10⁻² m³ s⁻¹ (1300 m³ h⁻¹).

The analyzed EAHX is characterized by five buried horizontal air ducts, installed in parallel at 2.5 m depth. The section of the buried pipes is circular, and the analysis is carried out referring to a maximum length of 100 m and to three different diameters (0.2 m, 0.3 m, and 0.5 m). Two adjacent ducts are spaced 2.5 m apart, then the system will extend over a rectangular surface with a maximum size of 12.5 m × 100 m.

The model considers only one of the ducts; the soil which surrounds it is 20 m deep. The finite element method is used to solve the mathematical model formed by the equation of the mass conservation for fluid, the equations of momentum conservation of fluid, and the energy equations for the fluid and for the solid. The heat transfer processes between the fluid (humid air) and the solid (ground) are conjugated. The equations are solved with appropriate boundary and initial conditions. Specifically, at the bottom of the soil domain, a 1st type condition is used to fix the undisturbed temperature of the ground, identified through the Kusuda relation [47]. The sun-air temperature is forced as 1st type condition on upper boundary of the volume. The model is then validated by means of experimental data reported in the scientific literature.

The climatic conditions of the localities where the system is installed, affect the EAHX thermal performance. Therefore, to make
a comparison, the office building is virtually placed in six different cities of Italy, chosen to belong to six different climatic zones identified by D.P.R. 412/93 [48]. For a further comparison, the building is subsequently placed in eight cities of the world following the Köppen climate classification [49] (the Italian cities are also included).

During the analysis on the EAHX, the diameter of the air ducts is varied to optimize the system, but the airflow rate necessary for the building must remain constant, so the speed of the air varies consequently. The temperature of the air at the exit of the EAHX is evaluated: this air is then sent to the air handling unit before it is supplied to the building. The following parameters are evaluated: the variation of air temperature in the EAHX; its thermal efficiency; the decreasing of cooling and heating capacity of the coils included.

To make a comparison, six Italian localities have been considered in this analysis (Lampedusa, Catania, Naples, Rome, Milan, Pian Rosa) belonging to the different six climatic zones. The weather data considered were identified through ASHRAE climatic data [50].

Table 1 shows, for each of the six localities, the design values of the outside air temperature, the relative humidity and the solar incident radiation in winter and summer.

4. Climate description

4.1. Italian climatic zones

According to D.P.R n. 412 of 1993 [48], as shown in Fig. 2, the Italian territory is divided into six (A-F) climatic areas based on the heating degree-days of the localities. For each climatic zone, over one year, the period and the maximum number of hours per day where heating may be switched on are fixed. The heating degree-days are the unit used to assign a climatic zone to each municipality: they are the sum, extended to all days in a conventional annual heating period, of only positive differences between indoor temperature (conventionally fixed at 20 °C) and the mean daily outdoor temperature. To the greatest values of the degree-days correspond the coldest climate zones in winter. The degree-days vary from a minimum of 600 (for zone A) to over 3000 (for zone F).

To make a comparison, six Italian localities have been considered in this analysis (Lampedusa, Catania, Naples, Rome, Milan, Pian Rosa) belonging to the different six climatic zones. The weather data considered were identified through ASHRAE climatic data [50].

Table 1 shows, for each of the six localities, the design values of the outside air temperature, the relative humidity and the solar incident radiation in winter and summer.

4.2. Climatic zones according to Köppen climate classification

The Köppen climate classification [49] is based on the evaluation of the local vegetation in each zone, since it was known, from the first publication in 19th century, that in a certain region the concentration of the vegetation depends on both the temperature and precipitation. The Köppen classification subdivides the Earth area into five main climatic zones based on temperature criteria, apart from the second zone (B) in which it is assumed that the dryness of the zone is the main key factor for vegetation’s concentration.

The principal zones are identified with a capital letter as follows [49]:
- Zone A: equatorial or tropical climates (the minimum monthly temperature value during the year is equal to or greater than 18 °C). This zone includes the warmest climates.
- Zone B: dry climates (annual mean value of precipitation is less than a specific limit). This zone includes deserts and steppes.
- Zone C: mild temperate climates (monthly average temperature of the warmest month is equal or greater than 10 °C, monthly average temperature of the coldest month ranging from –3°C to 18 °C).
- Zone D: continental climates (monthly average temperature of the warmest month is equal or greater than 10 °C, monthly average temperature of the coldest month is equal or lower than –3°C).
- Zone E: polar climates (monthly average temperature of the warmest month is less than 10 °C).

Each climatic area can be also divided in subareas by means of a second letter to take into account precipitations; in some cases, also another sub-criterion (based on temperature) is considered, by adding a third letter.

Fig. 3 shows the Italian map according to Köppen classification. Based on this classification, the Italian localities belong to the climatic zones reported in Table 2. Pian Rosa was omitted since it is characterized by extreme and not very generalizable climatic conditions. Furthermore, in the summer season it does not require a cooling system.

As Table 2 shows, Lampedusa can be classified according to Köppen classification as Bsh (hot semi-arid climate); Catania, Naples and Rome as Csa (hot-summer Mediterranean climate); Milan as Cfb (temperate oceanic climate).

Moreover, three further localities (Rio de Janeiro, Dubai, Ottawa) are analysed. They belong to the A, B and D zones, respectively, based on the classification proposed by Köppen, whereas zone E (polar area) is not considered. Table 3 shows, for these three
towns, the design values of outside air temperature, relative humidity and solar incident radiation in winter and summer.

According to this classification, Rio de Janeiro belongs to Aw (tropical wet and dry climate) climate zone; Dubai to Bwh (hot desert climate) and Ottawa to Dfb (warm-summer humid continental climate).

5. The HVAC system description

The air conditioning system is characterized by fan coils and primary air. A reversible (invertible) heat pump provides hot and cold water for both the coils of an air handling unit, in order to treat the primary air, and the fan-coil units located in each room of the building. The design external (or outdoor) air flow has been set at $11 \times 10^{-3}$ m$^3$ s$^{-1}$ per person, for a total of 1300 m$^3$ h$^{-1}$. The design thermo-hygrometric conditions to be guaranteed inside each room are:

- indoor air: temperature of 20 °C for winter and 26 °C for summer, relative humidity ($\phi$) of 50% for both winter and summer;
- supply primary air: temperature of 20 °C and $\phi$ of 50% for winter, 15 °C and $\phi$ of 85.2% for summer (this value of $\phi$ is calcu-

![Fig. 2. The different Italian climatic zones according to D.P.R. n. 412 of 1993 [48].](image)

**Table 1**

Winter and summer design outdoor parameters for the six localities chosen [48,50].

| Climatic Zone - locality | Geographic coordinates | Winter design parameters | Summer design parameters |
|-------------------------|------------------------|--------------------------|--------------------------|
|                         | T[°C]            | $\phi$ [%] | G[Wm$^{-2}$] | T[°C] | $\phi$ [%] | G[Wm$^{-2}$] |
| A - Lampedusa (AG)      | Lat. 35° 30' 05'' N Long. 12° 30' 34'' W | 9.80 | 54.60 | 874 | 31.00 | 65.80 | 831 |
| B - Catania             | Lat. 37° 10' 46'' N Long. 15° 3' 27'' W | 2.90 | 71.20 | 824 | 32.80 | 44.10 | 819 |
| C - Naples              | Lat. 40° 51' 22'' N Long. 14° 14' 47'' W | 1.90 | 52.00 | 808 | 31.90 | 48.60 | 825 |
| D - Rome                | Lat. 41° 9' 27'' N Long. 12° 49' 23'' W | 0.10 | 60.40 | 783 | 32.60 | 37.80 | 829 |
| E - Milan               | Lat. 45° 46' 46'' N Long. 9° 18' 85'' W | -3.20 | 65.10 | 743 | 32.00 | 4.70 | 805 |
| F - Pian Rosa (AO)      | Lat. 45° 93' 33'' N Long. 7° 70' 00'' W | -22.20 | 44.60 | 1026 | 8.10 | 51.70 | 994 |

- indoor air: temperature of 20 °C for winter and 26 °C for summer, relative humidity ($\phi$) of 50% for both winter and summer;
- supply primary air: temperature of 20 °C and $\phi$ of 50% for winter, 15 °C and $\phi$ of 85.2% for summer (this value of $\phi$ is calcu-
lated after evaluating the specific humidity $\omega$ by means of a mass balance for each room, referred to water.

Two air conditioning systems are analyzed: Fig. 4 shows the traditional one characterized by only the AHU for primary air (without EAHX), whereas Fig. 5 shows the system where the EAHX is placed upstream the AHU. The first one is a usual HVAC system with only the AHU (without EAHX) and fan-coil units; the air treated in the AHU is outdoor air. The second air conditioning system is instead characterized by the EAHX which pre-heats or pre-cools the outside air before being handled into the AHU.

The AHU is composed of the following main components:

- filters;
- pre-heating water coil;
- cooling and dehumidifying water coil;
- humidifying section;
- re-heating coil;
- supply fan.

In the Fig. 6(a) for summer and 6(b) for winter the transformation in the AHU on the psychometric chart are reported. During the summer (Fig. 6(a)) the processes that the humid air undergoes are: cooling and dehumidification from point “‘$o$” (outdoor air conditions) to point “‘A$” and subsequent re-heating from point “‘A$” to point “‘$s$” (supply air conditions). The cooling coil is supposed to be ideal with a by-pass factor equal to 0% (i.e., $\phi_A = 100\%$). During the winter (Fig. 6(b)) the processes are: pre-heating from point “‘$o$” to point “‘A$”, humidification with liquid water from point “‘A$” to

### Table 3

Winter and summer design outdoor parameters for Rio de Janeiro, Dubai and Ottawa.

| Climatic Zone Locality | Geographic Coordinates | Winter design parameters $T[°C]$ | $\phi[\%]$ | $G[Wm^{-2}]$ | Summer design parameters $T[°C]$ | $\phi[\%]$ | $G[Wm^{-2}]$ |
|------------------------|------------------------|-------------------------------|------------|------------|-------------------------------|------------|------------|
| Aw         | Rio de Janeiro | Lat. 22° 54’ 29.9988”S Long. 43° 11’ 46.9968” W | 17.1 | 76.8 | 826 | 32.8 | 47.5 | 923 |
| Bwh       | Dubai          | Lat. 25° 16’ 37.1532” N Long. 55° 17’ 46.4964” E | 14.2 | 43.9 | 795 | 41.8 | 21.9 | 661 |
| Dfb       | Ottawa         | Lat. 45° 25’ 28.9956” N Long. 75° 41’ 42.0000” W | -20.8 | 64.9 | 858 | 29 | 50.1 | 846 |
point "B" and re-heating from point "B" to point "s" which coincides with the thermohygrometric conditions to be maintained in the room (point "r"). The humidifier is supposed to be ideal (with saturation efficiency of 100%). When the EAHX is used for pre-cooling/pre-heating the air flow, the point "o" (outside air) is substituted with the point EAHX (air conditions at the exit of the EAHX, individuated through the 2D model below described).

To evaluate the coils capacity, the mass and energy balances are carried out on the control volumes shown in Fig. 7(a) and Fig. 7(b). During the summer (Fig. 7(a)) the running components of the AHU are: the cooling coil and the re-heating coil. The energy balance equation for calculating the cooling capacity (with reference to control volume 1 of Fig. 7(a)) is:

\[
\dot{Q}_{CC} = \dot{m}_o(h_o - h_A) - \dot{m}_{coh}h_0
\]

(1)

The re-heating coil power (control volume 2 of Fig. 7 a) can be evaluated as:

\[
\dot{Q}_{ReH} = \dot{m}_c(T_A - T_o)
\]

(2)

During the winter, as shown in Fig. 7(b), the active components are: the pre-heating coil, the humidifier with liquid water and the re-heating coil. The pre-heating coil capacity (control volume 3 of Fig. 7(b)) can be evaluated as:

\[
\dot{Q}_{PRE} = \dot{m}_c(T_A - T_o)
\]

(3)

The mass flowrate of humidification water (control volume 4 of Fig. 7(b)) can be evaluated as:

\[
\dot{m}_w = \dot{m}(\omega_B - \omega_A)
\]

(4)

The re-heating coil capacity (control volume 5 of Fig. 7(b)) is obtained from the equation:

\[
\dot{Q}_{ReH} = \dot{m}_c(T_A - T_o)
\]

(5)

When the EAHX is in use, the reduced capacity of the AHU coils both for summer and winter has been evaluated. Consequently, the reduction of the coils' capacity obtained by the introduction of the EAHX technology compared to the AHU without this heat exchanger is calculated, considering the coils operating in winter season, summer season and all over the year.

6. Mathematical model of the EAHX

In this research, the open-loop earth-to-air heat exchanger was 2D modeled through a finite element method software. The EAHX was made of five horizontal ducts. The horizontal disposition was chosen since vertical one usually involves with higher installation and maintenance costs. The number of tubes has been chosen to obtain, at fixed air volumetric flowrate, a range of air velocity...
between 0.4 and 2.5 m s\(^{-1}\) that is a good compromise between effectiveness of heat transfer and pressure drops. 2.5 m is the distance \(d\) stemming between two adjacent ducts: this value is chosen to avoid thermal interaction between the two air ducts. The computational domain of the model consists of one circular buried duct (for air flowing) surrounded by a ground volume of 20 m deep. This value of deepness was chosen to consider the ground as undisturbed [51,52].

The buried duct is installed at 2.5 m deep from the soil surface because, in agreement with other studies [11], for deepness more than 2 m, the soil temperature is about undisturbed and close to the annual mean values of the outdoor air. Burying the pipe between 2 m and 3 m is a good compromise [27,53] between yearly temperature excursion and excavation costs.

The mass flowrate of the air entering each pipe is evaluated as:

\[
\dot{m}_{pipe} = \frac{\dot{m}}{5} \tag{6}
\]

Various values of the diameter and length of the air ducts are considered:

\[
L = [20; 50; 60; 80; 100] m \tag{7}
\]

\[
D = [0.2; 0.3; 0.5] m \tag{8}
\]

The outside airflow of 1300 m\(^3\) h\(^{-1}\) must be provided to the building, so the modification of the duct diameter leads to a modification in the air speed. In Table 4 for each diameter the air velocity and the Reynolds number are reported. The table clearly shows that the airflow rate can be always considered in fully turbulent developed regime.

In Fig. 8 the computational domain and the used mesh with a triangular shape are reported. The model is solved through finite element method and the domain was divided into 70,625 free-triangular elements: a higher concentration of elements can be found inside and surrounding the duct where more marked is the temperature gradient. The diameter and the length of the ducts are indicated with \(D\) and \(L\), respectively. The air entering the tube has temperature and relative humidity proper of the external air.

The following assumptions are used in the present study:

- two-dimensional model;
- no thermal interaction between the buried ducts;
- a longitudinal section of the domain (composed by the ground and the duct) is modeled for symmetry;
- the air flow velocity ensures the full turbulent regime;
- the thermal resistance of the tube is neglected;
- the study is time-dependent and the model runs until reaching the steady-state;
- temperature in soil undisturbed at 20 m depth;
- constant properties of the soil in the whole domain;
- constant properties of the air flow in the ducts;
- the soil considered as an isotropic medium.

The air entering the ducts is humid air. The thermodynamic properties of humid air (dry bulb temperature; relative and specific humidity) can be punctually evaluated, in time and space, through the model. Therefore, the condensed water flow rate can be also evaluated. For the fluid domain the following differential equations can be numerically solved:

\[
\frac{\partial \rho}{\partial t} + \nabla \left( \rho \mathbf{v} \right) = \dot{S}_m \tag{9}
\]

![Fig. 7. Schematic of AHU for primary air in: (a) summer and (b) winter.](image)
where \( S_m \) is a negative term that represents the mass of water vapor condensed;

- the conservation momentum of the air flow is guaranteed by the Navier-Stokes equations for turbulent flow:

\[
\frac{\partial \rho \vec{v}}{\partial t} + \rho (\vec{v} \cdot \nabla) \vec{v} = \nabla \cdot \left[ -p I + \left( \mu + \mu_t \right) \left( \nabla \vec{v} + \left( \nabla \vec{v} \right)^T \right) \right]
\]  

(10)

where \( \mu_t \) is the turbulent viscosity defined as:

\[
\mu_t = \rho C_{\mu} \frac{k}{\varepsilon}
\]  

(11)

with \( C_{\mu} \), that is one of the constants of the K-\( \varepsilon \) model for turbulent flow [29];

- the energy equation for the air flow:

\[
\frac{\partial (\rho E)}{\partial t} + \nabla \cdot \left[ (\rho E + p) \vec{v} \right] = \nabla \cdot \left[ k_{\text{eff}} \nabla T - \sum_j h_j \vec{J}_j + \left( \tau_{\text{eff}} \cdot \vec{v} \right) \right]
\]  

(12)

where \( k_{\text{eff}} \) is the effective conductivity defined as the sum of the conventional thermal conductivity of the fluid \( (k_f) \) and the thermal conductivity of the turbulent flow \( (k_t) \) and thus modeled as:

\[
k_{\text{eff}} = k_f + k_t
\]  

(13)

- Using the K-\( \varepsilon \) model for turbulent flow, the turbulence kinetic energy equation is:

\[
\frac{\partial (\rho k)}{\partial t} + \rho \vec{v} \cdot \nabla k = \nabla \cdot \left( \left( \mu + \frac{\mu_t}{\sigma_k} \right) \nabla k \right) + P_k - \rho e
\]  

(14)

where \( P_k \) can be evaluated as:

\[
P_k = \mu_t \left( \nabla \vec{v} : \left( \nabla \vec{v} + \left( \nabla \vec{v} \right)^T \right) \right) - \frac{2}{3} \left( \nabla \cdot \vec{v} \right)^2 - \frac{2}{3} \rho \kappa \nabla \cdot \vec{v}
\]  

(15)

The specific dissipation rate equation is:

\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \rho \vec{v} \cdot \nabla \varepsilon = \nabla \cdot \left( \left( \mu + \frac{\mu_t}{\sigma_e} \right) \nabla \varepsilon \right) + C_{\epsilon 1} \frac{e}{k} P_k - C_{\epsilon 2} \rho \kappa \varepsilon^2
\]  

(16)

The experimental constants of the K-\( \varepsilon \) model are reported in Table 5.

The differential equation of conduction in solid domain numerically solved is:

\[
\frac{\partial (\rho c_{\text{soil}} T_{\text{soil}})}{\partial t} = \nabla \cdot \left( k_{\text{eff}} \nabla T_{\text{soil}} \right)
\]  

(17)

The soil humidity is considered balancing water and solid properties throughout the porosity \( (\psi) \) with the following equation:

\[
z_{\text{soil}} = \psi z_{\text{soil}} + (1 - \psi) z_{\text{solid}}
\]  

(18)

For each locality of the analysis the thermal properties of the soil are evaluated and reported in Table 6. For all the Italian localities, the mean value of the soil thermal properties is considered corresponding to a porosity of 37%.

The associated thermal boundary conditions for the domain are:

- at the side boundaries of the soil domain, adiabatic (2nd type) conditions are used disregarding any thermal influence from the ground beyond these boundaries;
- at the bottom of the soil domain, since the ground temperature remains constant beyond this boundary (deep 20 m), a 1st type

![Diagram](image_url)

**Fig. 8.** Computational mesh domain of the investigation on a single buried duct of the EAHX.
condition is imposed, following the undisturbed temperature evaluated with the Kusuda [47] equation:

\[
T_g(D, t) = T_m - A \cdot \exp \left[ -\text{Depth} \cdot \sqrt{\frac{\pi}{365 \cdot a}} \right] \cdot \cos \left[ \frac{2\pi}{365} \left( t - t_{\text{min}} - \frac{\text{Depth}}{2} \cdot \sqrt{\frac{365}{\pi \cdot a}} \right) \right]
\]  
(19)

Table 7 reports the weather data used in equation (19) and the resulting undisturbed ground temperature for each locality of the present analysis:

- at the top of the soil domain (the surface), a 1st type boundary condition is assumed: the sun-air temperature. This temperature takes into accounts both the incident solar radiation on the ground surface and the convective heat exchange with the external air, and is expressed with the following equation:

\[
T_{sa}(x, 0, t) = T_{\text{air, ext}}(t) + \frac{\alpha G(t)}{h_c}
\]  
(20)

- the air temperature at the inlet of the duct is fixed according to the values of external air reported in Tables 1 and 3 for the different localities in summer and winter season;
- inlet air velocity (in \( \times \) direction) is chosen depending on the tube diameter.

After the calculation of the outside air at the exit of the EAHX (by means of the described mathematical model), the thermal efficiency of the EAHX is calculated through the typical equation (21) for heat exchangers (as the ratio between the EAHX temperature span and the ideal temperature difference):

\[
\varepsilon = \frac{T_{\text{outlet}} - T_{\text{inter}}}{T_{\text{ground}} - T_{\text{inter}}}
\]  
(21)

6.1. Grid independence study

A time dependent solver is used to solve the mathematical model, while the implicit BDF (Backward Differentiation Formula) is used as time step procedure. The implicit BDF procedure utilizes backward differentiation equations that present accuracy from one (named as the backward Euler method, too) to five. BDF procedures were often utilized due to their stability characteristics. On the other hand, they could show some damping effects, mainly when considering the lowest order methods (some high frequencies are often damped). Although one could expect a solution with sharp gradient, a frequently smooth solution is obtained due to the above-mentioned damping effects. The use of BDF could be characterized by high order if possible, and lower order when it is indispensable to reach stability. The strategy of the solver selected for the model used in this work is BDF with “Free time stepping”: in this way, the solver can set greater or smaller time steps to satisfy the required tolerances. In fact, the solver tries to calculate with the largest possible time step, but, when the solution starts to rapidly vary and therefore the (relative and absolute) tolerances are not verified, it decreases the timestep size until it is indispensable. The values of the absolute and relative tolerances fixed for the solver of the introduced model are \( 5 \times 10^{-4} \) and \( 1 \times 10^{-2} \), respectively.

The independence of the spatial grid is evaluated after the model was simulated, on equal initial, boundary and operative conditions with three different meshing geometries: the system is evaluated with the domain divided into 11124, 15827, 70,625 triangular elements, following free triangular meshing.

| City          | \( T_m [^\circ \text{C}] \) | \( A [^\circ \text{C}] \) | Depth[m] | \( \alpha [\text{m}^2 \text{ day}^{-1}] \) | \( t [\text{day}] \) | \( t_{\text{min}} [\text{day}] \) | \( T_{\text{ground}}[^\circ \text{C}] \) |
|---------------|-----------------------------|-----------------------------|----------|-----------------------------------|---------------------|-----------------------------|-----------------------------|
| Lampedusa     | 20                          | 5.4                         | 100      | 0.0821                             | 365                 | 15                          | 20.00                       |
| Catania       | 17.6                        | 7.8                         | 100      | 0.0821                             | 365                 | 15                          | 17.60                       |
| Naples        | 17.0                        | 8.5                         | 100      | 0.0821                             | 365                 | 15                          | 17.0                        |
| Rome          | 15.9                        | 8.9                         | 100      | 0.0821                             | 365                 | 15                          | 15.90                       |
| Milan         | 12.3                        | 9.4                         | 100      | 0.0821                             | 365                 | 15                          | 12.30                       |
| Pian Rosa     | -5.0                        | 6.95                        | 100      | 0.0821                             | 365                 | 15                          | -5.10                       |
| Rio de Janeiro| 25.1                        | -2.8                        | 100      | 1.028                              | 365                 | 15                          | 25.0                        |
| Dubai         | 29.3                        | 7.9                         | 100      | 0.795                              | 365                 | 15                          | 29.0                        |
| Ottawa        | 12.1                        | -1.2                        | 100      | 0.064                              | 365                 | 15                          | 12.0                        |

Fig. 9. Average air temperature at the outlet of the buried pipes as a function of time for three different meshing (11124, 15827, 70,625 elements).
The average temperature profiles at the outlet of the buried ducts for different mesh size are reported in Fig. 9 as a function of time. The small box reported in the Figure represents a zoom of the end of the time interval to better highlight the spacing of the curves at different meshes.

As clearly visible from the figure we found a substantial overlapping between the temperature profiles with 15,827 and 70,625 elements and a good agreement between the solution with 11,124 elements (maximum difference lower than 0.04 K). Since the computational time for elaborating the solution does not differ appreciably if we simulate with 15,827 and 70,625 elements, we opted for the finer meshing (70,625 elements).

6.2. Model validation

To ensure the reliability of the results obtained with the numerical code, the model is validated by comparing it with experimental data available in open literature. Specifically, three case studies of horizontal EAHX installed in three different countries (Algeria, Morocco, Egypt) are chosen for validation.

The experimental data published by Hatraf et al. [58] have been obtained through an EAHX installed at the University of Biskra, Algeria (Longitude 5° 44'E, Latitude 34° 48'N), a location characterized by a hot and dry summer, typical of the Saharan climate. The experimental facility consists of four horizontal ducts buried at a depth of 3 m; each duct is 60 m long with an inner diameter of 0.11 m. The external air at the inlet of the EAHX was at a temperature of 36°C with two different volumetric air flowrates (135 m$^3$ h$^{-1}$ and 156 m$^3$ h$^{-1}$). The corresponding Reynolds numbers (2.63$\times$10$^4$ and 3.03$\times$10$^5$) ensure for the air flow in the ducts a fully developed turbulent regime. Fig. 10 shows a comparison between the experimental and the numerical air temperature at different location along the tube length for both the volumetric flowrates considering the same geometrical parameters and boundary conditions.

In Table 8 is reported the absolute and relative error on the outlet air temperature: the maximum relative deviation between the experimental and the numerical data is 3.5% (at 17 m of tube length). That is, the maximum difference between the predicted and the experimental air temperature at the outlet of the EAHX is 1°C.

In addition, the EAHX model has also been validated by means of some of the experimental results provided by Khabbaz et al. [59], related to an Earth-to-Air Heat Exchanger system located in Marrakech (Morocco) (Latitude 31° 38'02" N, Longitude −7° 59'59" E), location characterized by hot semi-arid climate. The experimental EAHX is constituted of three horizontal parallel ducts (length: 77.7 m; inner diameter: 0.15 m), buried at 2.2 m and 3.5 m depth; the air enters the ducts with a velocity of 5 m s$^{-1}$ (corresponding to a Reynolds number of 4.55$\times$10$^4$ that ensures a turbulent flow regime). Measure uncertainties on temperature is ± 0.5°C. The experimental tests have been carried out at the air inlet temperature of 34.5 and 35.9°C. In Fig. 11 the numerical and experimental air temperature values are reported alongside the buried pipes.

In Table 9 is reported the absolute and relative error on the outlet air temperature. It can be noted that the maximum relative deviation between the experimental and the numerical data is

| L (m) | $\varepsilon_{135}^\circ\text{C}$ | $\varepsilon_{135}\%$ | $\varepsilon_{156}^\circ\text{C}$ | $\varepsilon_{156}\%$ |
|-------|-----------------|----------------|----------------|----------------|
| 1     | 0.14            | 0.41           | 0.03           | 0.084          |
| 3     | 0.12            | 0.35           | 0.13           | 0.37           |
| 7     | 0.030           | 0.091          | 0.06           | 0.18           |
| 15    | 0.22            | 0.72           | 0.35           | 1.12           |
| 31    | 0.07            | 0.25           | 0.20           | 0.71           |
| 63    | 0.3             | 1.2            | 0.11           | 0.45           |
| 72    | 0.26            | 1.1            | 0.25           | 1.04           |

Table 9

| L (m) | $\varepsilon_{34.9}^\circ\text{C}$ | $\varepsilon_{34.9}\%$ | $\varepsilon_{35.9}^\circ\text{C}$ | $\varepsilon_{35.9}\%$ |
|-------|-----------------|----------------|----------------|----------------|
| 1     | 0.14            | 0.41           | 0.03           | 0.084          |
| 3     | 0.12            | 0.35           | 0.13           | 0.37           |
| 7     | 0.030           | 0.091          | 0.06           | 0.18           |
| 15    | 0.22            | 0.72           | 0.35           | 1.12           |
| 31    | 0.07            | 0.25           | 0.20           | 0.71           |
| 63    | 0.3             | 1.2            | 0.11           | 0.45           |
| 72    | 0.26            | 1.1            | 0.25           | 1.04           |

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1.2% (at 63 m of tube length). The maximum difference between the predicted and the experimental air temperature at the outlet of the EAHX is 0.3 °C (lower than the experimental uncertainty on the measured temperature).

A further validation of the model is presented with experimental data of Serageldin et al. [23]. The experimental system is an EAHX installed at the University of Science and Technology of Borg El Arab (Egypt) (Latitude 3° 55' N Longitude 29° 42' E), a location characterized by a desert climate, generally hot, sunny and dry throughout the year.

The experimental heat exchanger is a horizontal PVC duct of 5.5 m long, with an inner diameter of 0.0508 m, buried at a depth of 2 m. The comparison refers to the summer operating mode, with an inlet air temperature at 30.2 °C and a volumetric flow rate of 11 m³ h⁻¹ (the corresponding Reynolds number is 5.05 · 10⁴). The Reynolds number follows in the transition between laminar and turbulent flow. In this range the numerical model uses the laminar solution. The experimental temperature was detected with T-type calibrated thermocouples, with an error following in the normal range with deviations between the thermocouples reading and that of a standard one (beta Calibrator TC-100) of ± 0.1–0.5 °C. In Fig. 12 is reported a comparison between experimental and numerical air temperature as a function of the tube length: the figure clearly shows that the numerical model always overpredicts the experimental data.

In Table 10 the absolute and relative error on the outlet air temperature is reported. It can be noted that the maximum relative deviation between the experimental and the numerical data is 2.55% (at 4.67 m of tube length). The maximum difference between the predicted and the experimental air temperature at the outlet of the EAHX is 0.70 °C.

From all these analyses we can conclude that the presented model is able to predict the thermal performance on a horizontal EAHX not only in fully developed turbulent flow but also in laminar or transition regime.

7. Results and discussion

In this paper the thermal and energy performances of an EAHX pre-treating unit coupled to an AHU are evaluated. The thermal behaviour of an EAHX is not the same on the globe but depends on the climatic context, the soil temperature, and the configuration of the EAHX. The soil temperature is very similar to the annual mean temperature of the place in which EAHX is installed; therefore, it is often higher than air temperature in winter and lower in summer. The aim of this paper is to compare the performances of an EAHX in: i) six localities of Italy belonging to different climatic zones according to the Italian D.P.R. 412/93 classification; ii) nine cities with different climatic conditions based on the classification proposed by Köppen. The described EAHX is tested by means of a mathematical model; each simulation is carried out until the steady state regime is obtained. At this point, all the parameters are calculated.

7.1. A comparison among the six different Italian localities

In Fig. 13 the temperature variation along the tube length in winter and in summer varying the inner tube diameter for the six localities is reported. Fig. 13(f) represents the temperature profile for Pian Rosa; for this locality only the winter conditions are reported since the cold temperatures of the area do not require summer operation of HVAC system. On the Fig. 13 the dew point temperatures (only for summer) and the ground temperature are also reported for each locality. Note that in Milan and in Lampedusa, in summer, the air temperature at the exit of the EAHX is minor than the dew point, and this means that the air specific humidity is reduced, and the air is dehumidified. The maximum value of the condensed water mass flow rate is achieved with the lower inner diameter (0.2 m) and the highest tube length (100 m) and is 1.16 g s⁻¹ for Lampedusa and 1.51 g s⁻¹ for Milano. The tube length corresponding to the beginning of vapour condensation decreases with the reduction of the inner tube diameter. Indeed, with a diameter of 0.2 m, the condensation begins at 50 m length. For all the figures a similar trend can be observed: the air flow temperature increases/decreases through the tube length in winter/summer. This increment/decrement is faster for the initial length of EAHX (the first 20–30 m) and then becomes moderate. A temperature of the air close to undisturbed ground temperature (Knee point) is obtained at a duct length of about 80 m for all the localities. The Knee point represents the length of the tube at which more than 90% of the global increase or decrease of the air temperature has been obtained. Moreover, note that the air speed rises with the reduction of the duct diameter, and this implies the rising of the heat transfer coefficient for convection and, finally, the improvement of the heat exchange. Therefore, with the lowest value of the inner tube diameter (D = 0.2 m corresponding to an inlet air velocity of 2.3 m s⁻¹) a greater temperature variation can be obtained in the EAHX.

Fig. 14 shows the air temperature span in summer between the inlet and the exit of the EAHX, for a duct diameter of 0.2 m, when varying the length of the air duct, considering the six analyzed localities.

The Figure clearly shows that temperature variation is more marked for Milan, a city characterized by cold winter and hot sum-

![Fig. 12. Experimental and numerical air temperature alongside the pipe of the earth-to-air heat exchanger.](image)
mer (with a maximum value greater than 17 K). The lowest values are those pertaining to Lampedusa, a locality characterized by very mild climate (maximum value lower than 10 K).

In Fig. 15 is reported the variation between the outlet and the inlet temperature of the EAHX for the six localities in winter season.

It could be noted that the greatest temperature variation can be obtained for Pian Rosa, a locality characterized by very cold winters (maximum value of 14.7 K), whereas the lowest is for Lampedusa (maximum value of 8.9 K).

From the data plotted in Figs. 14 and 15 the following considerations can be drawn:

- the temperature variation that can be obtained with an EAHX at fixed tube length is always greater in the summer than in the winter season. This is due to the greater temperature difference between the external air and the soil during the summer for each of the tested localities;
- the temperature of the undisturbed ground is almost constant and very similar to the yearly average value for the outside air. So, the lowest temperature of the ground is for zone F, while the highest occurs for the zone A. The temperature difference between the ground and the outside air represents the principal driving force in the heat exchange process. Where the difference between the air temperature at the inlet of the heat exchanger (corresponding at L = 0 in Fig. 13) and the undisturbed soil is more marked, the more efficient is the heat transfer process. The greatest values of driving force can be obtained in the localities with greater temperature excursions between summer and winter (Pian Rosa, Milan).
Lampedusa shows the lowest temperature span because it has very mild winters with moderate rainfall and hot, dry summers. So, the temperature span along the EAHX is minor than $10 \degree C$, although a relevant length of 100 m is considered for the air duct ($\Delta T$ belongs to $3.7 \div 9.5 \degree K$ in summer and to $3.5 \div 8.9 \degree K$ in winter).

In Fig. 16 is reported the efficiency of the EAHX as a function of the tube length for the six localities for an inner tube diameter of 0.2 m. From the results obtained, the efficiency in winter and summer is almost the same, this is the reason why there is only one graph that can be used for both seasons. The graph also shows that at fixed tube length the efficiency is independent on the climatic zone where the EAHX is installed. Indeed, the effectiveness of the heat exchange mainly depends on the convective heat transfer coefficient that at fixed inner tube diameter is almost constant for the different localities (because constant is the air flow velocity too).

Furthermore, the Fig. 16 clearly shows that the increase of the efficiency is very pronounced up to about 80 m: for longer lengths, the increase becomes moderate. An optimal efficiency value of about 86% is ensured with a length duct of 100 m. This result is also relevant in the possible comparison between the analyzed EAHX and an air-to-air heat recovery unit. In fact, the latter is characterized by a mean efficiency of about 65%- 80% [60-62]. Moreover, air-to-air heat exchangers are usually more dangerous due to the risk of spreading SARS-CoV-2 or other viruses.

Fig. 14. Air temperature variation in the EAHX as a function of the tube length for an inner tube diameter of 0.2 m in summer.

Fig. 15. Air temperature variation in the EAHX as a function of the tube length for an inner tube diameter of 0.2 m in winter.

Fig. 16. EAHX efficiency as a function of tube length for an inner tube diameter of 0.2 m.

Fig. 17. EAHX efficiency as a function of tube length varying the inner tube diameter in summer for Milan.
The efficiency is a strong function of the inner tube diameter (and of the consequent fluid velocity). As an example, in Fig. 17 is reported the efficiency for the different tube diameter for the city of Milan during summer. Similar results can be obtained in winter and for the other examined Italian localities.

We can observe the rising of the EAHX efficiency when the inner diameter of the air duct reduces (with a constant value of the duct length). In fact, when the duct diameter is reduced the air speed rises improving the heat transfer coefficient for convection (the air speed rises from 0.37 to 2.3 m s\(^{-1}\) when the duct diameter rises from 0.5 m to 0.2 m).

The EAHX also implies a relevant decreasing of the heating and cooling capacity of the coils inside the air handling unit. Fig. 18 reports the heating and cooling capacity of the operating coils during winter (i.e. preheating coil and reheating coil) (a), during summer (i.e. cooling coil and reheating coil) (b), all over the year (c), as a function of the tube length for an inner tube diameter of 0.5 m for the different localities. Fig. 18 also shows the heating and cooling capacity of the operating coils during winter (a), during summer (b), all over the year (c), as a function of the tube length for an inner tube diameter of 0.5 m.
capacity values for the system without the EAHX. Finally, the figure also reports the decreasing (in percentage) of these capacity values when the EAHX is considered, for various lengths of the air ducts.

From the Fig. 18 the following considerations can be drawn:

- increasing the tube length carries to an augmentation of the capacity reduction, too. Therefore, the best results can be obtained with the duct 100 m long;

- during the winter (Fig. 18(a)) the capacity reduction using the EAHX to pre-heat the air flow is more marked in zone A (maximum value of 40%) than in zone F (maximum value of 19%). Indeed, in Lampedusa, the southernmost point of Italy with a very mild winter and hot, dry summer, the capacity reduction is greater than in Pian Rosa (with a short and cool summer and a long, freezing, and snowy winter);

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**Fig. 19.** Coils capacity of the operating coils as a function of tube length for an inner tube diameter of 0.3 m: (a) during winter, (b) during summer, (c) all over the year.
an opposite trend is observed during the summer (Fig. 18(b)): the capacity reduction using the EAHX to pre-cool air flow is more marked in zone E (maximum value of 48%) than in zone A (maximum value of 21%);

with reference to winter and summer (Fig. 18(c)), the highest total decrease in capacity of the coils occurs for zone E (Milan – decrease of 38% for a duct length of 100 m), while the lowest value occurs for zone A (Lampedusa - maximum decrease of capacity equal to 27%). So, in Italy the annual utilization of the EAHX linked to an air handling unit is useful in all the national territory, even if preferable in zone E (i.e. in the climatic areas showing a high temperature excursion between winter and summer) compared to zone A.

Similar trends are observed with the other inner tube diameters that are reported in the following figures.

Fig. 20. Coils capacity of the operating coils as a function of tube length for an inner tube diameter of 0.2 m: (a) during winter, (b) during summer, (c) all over the year.
In Fig. 19 are reported the winter (a), summer (b), global (c) coil capacities as a function of the tube length for an inner tube diameter of 0.3 m.

In Fig. 20 are reported the winter (a), summer (b), global (c) coil capacities as a function of the tube length for an inner tube diameter of 0.2 m for the different localities.

From a comparison among Figs. 18, 19 and 20, the emerging data is the greatest coils capacity reduction that can be achieved considering the smallest diameter of 0.2 m (maximum global power reduction of 55% in Milano).

7.2. A comparison among the different Köppen climatic areas

The previous analysis has shown that the best results can be obtained with the smaller inner tube diameter considered (0.2 m). Therefore, in this analysis the tube diameter is fixed at 0.2 m.

In Fig. 21 is reported the temperature variation in EAHX as a function of the tube length in summer (a) and in winter (b) season.

According to the Köppen classification, the analyzed Italian localities belong to the zone C (mild temperature climates) except Lampedusa that belongs to the zone B (dry climates). The figures clearly show that:

- the temperature span between the inlet and the exit of the EAHX, in the Italian localities, is lower in winter than in summer, while an opposite result is obtained for Ottawa, Dubai and Rio de Janeiro. This depends on the temperature span between the soil and the outside air. This difference in Italy is not so dissimilar between summer and winter and slightly greater in summer. Instead, the contrary is found for Dubai, Rio de Janeiro and Ottawa (in this last case there is a strong variation between summer (14.5 K) and winter (27.4 K));
- in summer conditions, the maximum EAHX temperature difference between the inlet and the exit is obtained for Milan (higher than 17 °C), while the minimum temperature difference is obtained for Rio de Janeiro (maximum value lower than 7 °C);
- during summer, the best results can be obtained in Milan (maximum value greater than 17 K), whereas the worst results are registered in Rio de Janeiro (maximum value lower than 7 K);
- during winter Ottawa shows the greatest temperature variation in the EAHX (maximum value of 27.4 K), on the contrary Rio shows the lowest (maximum value 6.8 K);
- the temperature of the undisturbed ground is almost constant and very similar to the yearly average temperature of the outside air, in all the considered climatic areas. The temperature difference between ground and outside air is the principal driving force in the heat transfer process related to the EAHX; its highest values occur in the climatic areas with higher temperature excursion between winter and summer. So, the most relevant results occur for zones C and D (mild temperate climatic areas and continental climatic areas, respectively), while the less relevant results occur for equatorial or tropical climatic areas (zone A).

Fig. 22 shows the EAHX efficiency in summer, when varying the air duct length, for the eight analyzed towns. Indeed, in winter the
results are almost the same. It can be shown that the efficiency depends slightly on the climatic and soil characteristics of the area in which the exchanger is installed. For all the climatic areas the efficiency exceeds 80% at 100 m tube length.

Fig. 23 reports the heating and cooling capacity of the operating coils during winter (a), summer (b), all over the year (c), varying the tube length for an inner tube diameter of 0.2 m for the different localities. At the top of each histogram bar, the percentage reduction of coils capacity with the use of EAHX at each tube length is also reported.

From Fig. 23 (a) the following considerations can be drawn about the percentage capacity reduction calculated for the winter season:

- in Dubai and Rio de Janeiro, the temperature of the air at the exit of the air duct is higher than 20 °C, even when the duct length is only 20 m. Moreover, the couple dry bulb temperature - specific humidity of the air at the exit of the air duct is very similar to those required for comfort conditions in winter. So, the air exiting the EAHX can be supplied to the building without any HVAC system (only a suitable filtration of the air is obviously required). Therefore, in these cases the decreasing of the heating capacity of the coils rises 100% for the air handling unit working in heating operating conditions;
- the heating capacity reduction using the EAHX to pre-heat the air flow is more marked in A and B zones (maximum value of 100%) than in C or D zones (maximum value of 47%).

![Fig. 23. Coils capacity of the operating coils as a function of tube length for an inner tube diameter of 0.2 m: (a) during winter, (b) during summer, (c) all over the year.](image)
From Fig. 23 (b), referred to summer, one can notice that:

- the air temperature in the duct decreases below the dew point temperature (17.6 °C) in Ottawa, for duct lengths higher than 50 m; it means that the air has been dehumidified. The related mass flowrate values of the condensed water vapor \((m_{\text{wv}})\) at the EAHX exit are [0.06; 0.63; 0.99] g s\(^{-1}\) at [60;80;100] m length;
- in Ottawa, with a tube length of 100 m the air flow reaches temperature and relative humidity values such that the air can be directly conveyed in the building without using the air conditioning plant. Therefore, the percentage cooling capacity reduction in cooling mode is 100%;
- the capacity reduction using the EAHX to pre-cooling the air is more marked in D and C zones (maximum value 100% for Ottawa) than for A and B zones (maximum value of 18% for Rio de Janeiro).

From Fig. 23 (c) it can be seen that the highest decreasing of cooling plus heating capacity for the coils occurs for Dubai while low values of the length of the air duct are considered (up to 60 m), whereas for higher lengths of the air ducts the best results occur for Ottawa. Therefore, one can conclude that the best results can be obtained for tube length of 100 m in the city of Ottawa (reduction of 65% of heating + cooling capacity using the EAHX) that belongs to the Dfb zone according to the Köppen classification. This city is characterized by the greatest temperature span between winter and summer season, with a very cold winter (with frequent snowfalls) and a hot-humid summer.

### 8. Conclusions

In this paper the thermal and energy performance of an earth-to-air heat exchanger are investigated.

A two-dimensional unsteady numerical model of a horizontal EAHX has been developed. The EAHX is formed by 5 horizontal circular ducts, displaced in parallel at 2.5 m depth. Two adjacent ducts are 2.5 m spaced apart. The 2D model represents one of the five circular horizontal buried ducts of the EAHX surrounded by a ground volume 20 m deep; the problem is solved through finite element method. The model has been validated with experimental results found in literature: the maximum relative deviation between the experimental and the numerical data is 3.5%, the absolute deviation is always lower than 1 °C.

The EAHX is considered as a component of an air conditioning system for an office building. The air pre-heated or pre-cooled in the EAHX is not directly supplied to the building, but it is successively treated into the air handling unit. Since the thermal performance of the EAHX depends on the climatic conditions of the place where it is installed, the office building was firstly virtually placed in six different localities of Italy (Lampedusa, Catania, Naples, Rome, Milan, Pian Rosa), which belong to different climatic zones according to the Italian law D.P.R. 412/93, based on heating degree-days. For a further comparison, the building was subsequently placed in eight cities of the world according to Köppen climate classification (Dubai, Rio de Janeiro, Ottawa, plus five of the abovementioned Italian localities).

The EAHX is simulated and optimized as a function of the diameter and length of the air ducts. The following parameters are calculated: the variation of air temperature in the EAHX; its thermal efficiency; the decreasing of cooling and heating capacity of the coils into the AHU when comparing with the solution without EAHX. The analysis on the coils of the AHU is performed for winter, summer and for all the year. The following main conclusions are obtained:

- At the EAHX outlet, a temperature of the air close to the undisturbed ground temperature (Knee point) is obtained for tube length of about 80 m for all the localities. Therefore, a duct length of 80 m represents an acceptable compromise considering thermal performances, pressure drops and EAHX costs.
- Decreasing the tube diameter, the air velocity increases enhancing the convection heat transfer coefficient and, as a result, the heat exchange becomes more efficient. Therefore, with the lowest value of the inner tube diameter (0.2 m) a greater air temperature variation can be obtained in the EAHX.
- For all the analyzed climatic zones the undisturbed soil temperature is about constant and close to the annual mean values of the external air. Temperature gradient between ambient air and soil is the main driving force for the heat transfer in the EAHX. The greatest values of driving force can be obtained in the locality with the greatest temperature excursions between summer and winter. Therefore, the worst results in terms of temperature variation in the EAHX can be obtained in the zone A (according to Köppen climate classification, it refers to tropical or equatorial climates) and the best in zones D (continental climates) and C (mild temperate climates).
- Among the Italian localities, Lampedusa shows the lowest temperature span between the inlet and outlet of the EAHX, always smaller than 10 K even with a tube length of 100 m (AT belongs to 3.7 ÷ 9.5 K in summer and to 3.5 ÷ 8.9 K in winter), Milan (with a maximum value greater than 17 K) and Pian Rosa in winter (whose maximum temperature span is 14.7 K) show the highest temperature spans.
- According to the climate classification of Köppen, during summer the best results can be obtained in Milan (maximum value greater than 17 K), whereas the worst results are registered in Rio de Janeiro (maximum value lower than 7 K). During winter Ottawa shows the greatest temperature variation in the EAHX (maximum value of 27.4 K), on the contrary Rio de Janeiro shows the lowest (maximum value of 6.8 K).
- The efficiency of the EAHX is almost independent on the climatic zone where the EAHX is installed and its increase is very pronounced up to about 80 m: for longer lengths, the increase becomes moderate. Indeed, 100 m as length of each duct ensures the achievement of an optimal efficiency value, around 86%.
- Considering the reduction of heating and cooling capacity of the coils inside the AHU (deriving from the placement of EAHX upstream the AHU) for the whole year, the best case is Milan (zone E) with a heating + cooling capacity reduction of 55% for a tube length of 100 m, whereas the worst case is Lampedusa (zone A) with a maximum value of reduction equal to 39%. Therefore, when the yearly operation period of the AHU coupled to an EAHX is considered, in Italy the use of an EAHX is recommended in all the climatic zones, but more in E than in A zone. The best results can be obtained in the localities with a great temperature excursion between summer and winter.
- Considering the reduction of heating and cooling capacity of the coils inside the AHU based on Köppen climatic zones, one can conclude that the best results can be obtained for tube length of 100 m in the city of Ottawa (reduction of 65% when using the EAHX) that belongs to the Dfb zone. This city is characterized by the greatest temperature excursion between winter and summer season, with a very cold winter (with frequent snowfalls) and a hot-humid summer. On the contrary, the worst results can be obtained in Rio de Janeiro (Aw zone) with a maximum value of reduction of 24%.

### Author contributions

All the authors have contributed in the same manner to the various aspects of the research activity described in the paper. All...
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