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Cite as: AIP Conference Proceedings **2191**, 020121 (2019); https://doi.org/10.1063/1.5138854
Published Online: 17 December 2019

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Energy Analysis of Industrial Climatization by an Innovative Radiant Condensing System

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Abstract. Radiant heating plants have become increasingly popular in the last decades, particularly for industrial applications. Generally, they are high temperature radiant heating type, even if more recently low temperature radiant floors have increased their spread due to the increased thermal insulation of buildings. Moreover, radiant floors allow for cooling as well. In this paper, energy performance of a high temperature radiant tubes heating plant coupled to a condensing system for the climatization of an industrial building are investigated by dynamic simulation. A Trnsys type is modified in order to simulate the behavior of the high temperature condensing system: both the radiative (between the heating and the surrounding surfaces) and the convective heat exchanges with suitable devices (like baffles applied in order to reduce convection and consequent thermal stratification) are considered. Energy performance is compared to that of two more traditional plants such as warm air heater, and low temperature radiant floor coupled to condensing boiler. Finally, some considerations about the energy performance of the radiant tubes system in cooling mode are reported.

INTRODUCTION

Nowadays, many different systems are available for the climatization of large indoor environments like industrial buildings. This kind of buildings has different characteristics with respect to residential or commercial ones, such as higher heights (till more than 8 m), the presence of equipment on the walls or ceiling (bridge cranes, pipes and tubes, etc.), very big doors often opened, very large floor surfaces with different kind of occupation by workers, usually scarce thermal insulation, and different comfort condition requests [1].

For such reasons, traditional climatization plants are not commonly used in industrial buildings. Instead, air heater systems and high temperature radiant heating systems are very common because of the easiness of installation and allow quite low construction costs. Concerning the formers, ground or wall-mounted air heaters fueled by natural gas, wall-mounted air heaters supplied by hot water, and controlled mechanical ventilation plants are the most common [1]. Concerning the high temperature radiant systems, many solutions have been proposed (e.g. modular systems equipped with small gas burners, in the form of radiant tubes), in addition to the traditional types of panels heated by steam or pressurized water or the electrical radiant equipment [2] [3]. The advantages of high temperature radiant systems are mainly related to the possibility of addressing the heat flux towards the zone of interest, thus reducing the air temperature required for comfort conditions. Despite of the frequent use of this kind of plants, some uncertainties still remain on the design of this systems, since an overall balance of the room, taking into account each thermal flow, is necessary [4]. Moreover, work is also needed for better evaluating the energy performance and the comfort conditions [5].

Furthermore, during the last years low temperature radiant heating floor systems have become more common in industrial buildings as well, due to the increased thermal insulation of new buildings and the large diffusion of condensing boilers [6].

The aim of this work is to evaluate, by means of dynamic simulations, energy performance and comfort conditions featured by an innovative condensing radiant tubes plant. In this system, the exhausted from the tubes are

74th ATI National Congress
AIP Conf. Proc. 2191, 020121-1–020121-10; https://doi.org/10.1063/1.5138854
Published by AIP Publishing. 978-0-7354-1938-4/$30.00

coupled to a condensing heat exchanger to produce hot water that feeds a wall-mounted air heater, thus enhancing the thermal efficiency of the radiant tubes. The study is based on Trnsys rel. 17 software, and it is structured in three steps:

- modelling of a typical industrial building whose characteristics are based on a real case in order to calculate the annual heating loads;
- modelling of the innovative condensing radiant tubes system (CRT) and analysis of the energy performance and indoor thermal comfort conditions;
- modelling of two alternative heating systems as benchmark: an air heater based system (Air), and a radiant floor coupled to a condensing boiler plant (condensing radiant floor, CRF). For both, energy performance and indoor thermal comfort conditions are analyzed in comparison with the innovative CRT system. In order to consider the variety of real situations, both modern and old plant are considered for both Air and CRF by varying the thermal efficiency of generators in suitable ranges.

Moreover, industrial buildings are usually heated only, even if in the last years interest in assuring acceptable comfort conditions during summer is increasing. For such a reason, it is worth to evaluate the theoretical performance of the radiant tubes when they would be fed by cooled air. Based on real operational data for heating purpose in terms of exhausted flow rate and temperature, the cooling performance is calculated for different cooled air flow rates and inlet temperatures by evaluating both radiative and convective heat exchange with indoor environment.

METHODS

Heating operation: industrial building modelling

Energy and thermal comfort performance of the three systems are evaluated on a real industrial building whose characteristics in terms of size and thermal zones' opaque and transparent structures are reported in TABLE 1 and TABLE 2.

Some more hypothesis useful for the determination of heating loads are here reported:

- operation heating plant scheduling: from 6.00 am till 6.00 pm;
- presence of people and lighting scheduling (heating gain fixed at 5 W m^{-2}): from 8.00 am till 6.00 pm;
- presence of people, degree of activity and clothing: 40 persons in zone 1, 8 persons in zone 2, 2 met, 1 clo;
- air infiltration: 0.5 vol h^{-1}.

TABLE 1. General data, size and thermal zones' opaque and transparent structures characteristics.

Type of use of the building (DPR 412/93)		E.8 Building for industrial activity	
Resort (Province – State)		Manta (Cuneo – Italy)	
Altitude a.s.l.		404 m	
Latitude North	44° 36'	Longitude East	7° 29'
Degree Days		2814	
Climatic zone (DPR 412/93)		E	
Outdoor design air temperature		-9.3 °C	
Heating period		15^{th} Sept - 30^{th} Apr	
Thermal trasmittance (W m^{-2} K^{-1})			
External wall		0.389	
Door		3.50	
Main door		3.50	
Wall facing offices		2.954	
Base facing wall		3.220	
Floor facing ground		0.128	
Ceiling		4.086	
Ceiling shed		0.208	
Window		5.0	
Thermal bridge wall – floor facing ground (W m^{-2} K^{-1})		0.353	
Thermal bridge wall – ceiling (W m^{-2} K^{-1})		0.262	

TABLE 2. Main characteristics of the two thermal zones of the building for Trnsys simulation.

	Thermal zone 1	**Thermal zone 2**
Floor area (m^2)	7119	716.5
Net height (m)	8.24	8.22
Indoor air temp. (°C)	18	18
Net volume (m^3)	58669	5886.2

The dynamic simulation of the building with a 0.25 h time step allows to calculate the heating loads (thermal power) and the heating needs (monthly energy, FIGURE 1). The thermal power of the heating generators is limited to 1500 kW and 100 kW for thermal zone 1 and 2 respectively, as to be consistent with the installed power in the real building by the condensing radiant tubes' manufacturer.

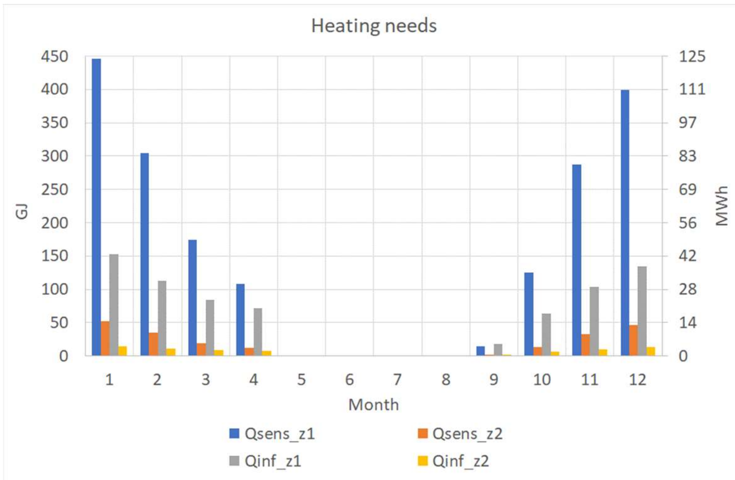


FIGURE 1. Thermal energy for heating needs for the two thermal zones, both sensible and due to infiltration of outdoor air.

Heating operation: CRT, Air and CRF systems modelling

Condensing radiant tubes (CRT) are an innovative system proposed by an Italian manufacturer set up by a radiant tubes system whose exhausted are directed to a condensing heat exchanger to produce hot water (at around 50 °C) that feeds a wall-mounted air heater. The combustion air flow rate is controlled by an exhausted tab that recirculate part of the exhausted (G_{ric}) to keep the air excess at the minimum value at part load operation (i.e. when natural gas fuel is regulated by a proportional valve, FIGURE 2).

The dynamic operation of the system is simulated in Trnsys by coupling type 607 and 659. The former has been modified in order to simulate the behavior of the high temperature radiant tube system: both the radiative and the convective heat exchanges with suitable baffles applied in the upper part in order to reduce convection and consequent thermal stratification are considered. At the start of operation, the CRT burner turns on at maximum power in order to get the maximum exhausted temperature. When indoor air temperature raises approaching the set-point, thermal power by the CRT burner is modulated by controlling the natural gas proportional valve in order to have the fuel mass flow rate (G_{fuel}) that is necessary to produce the heating load requested at that time step (*Heating_load*):

$$G_{fuel} = \frac{Heating\_load}{\eta_{th,HHV} \cdot HHV_{NG}} \tag{1}$$

In Eq. 1, $\eta_{th,HHV}$ is the system's thermal efficiency based on the high heating value of natural gas (HHV=39 MJ Sm^{-3}) (FIGURE 3). The fuel modulation decreases the exhausted temperature, and the exhausted tab is regulated in order to have the correct minimum air excess in the burner (FIGURE 3).

FIGURE 4 reports the Trnsys project of the building and the condensing radiant tubes heating plant.

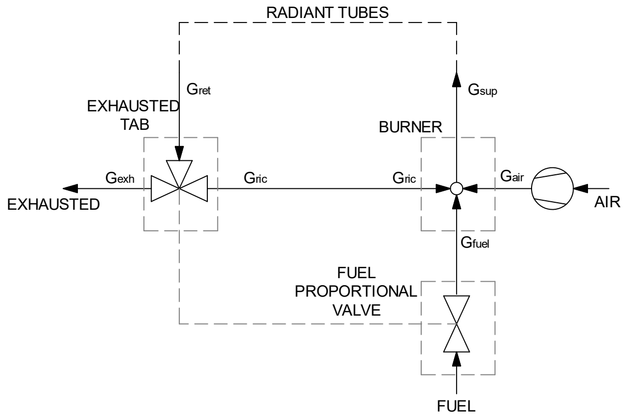


FIGURE 2. Mass flow balance for the condensing radiant tube. *G* is the mass flow rate, the meaning of the subscripts is: sup=supply; ret= return; exh=exhausted; ric=recirculated; air=combustion air; fuel=combustion fuel.

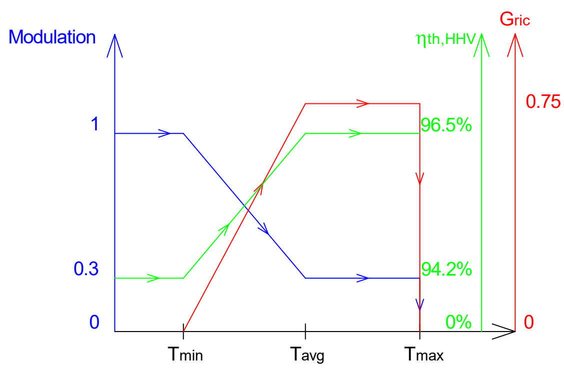


FIGURE 3. Control logic of the natural gas proportional valve (blue) and the exhausted tab (red). G_{ric} is expressed in terms of fraction of the radiant tubes return flow (G_{ret} in FIGURE 2). Thermal efficiency (on HHV) vs modulation is reported as well (green). In the simulations the hypotheses are T_{min}=17 °C, T_{avg}=17.5 °C, T_{max}=18 °C.

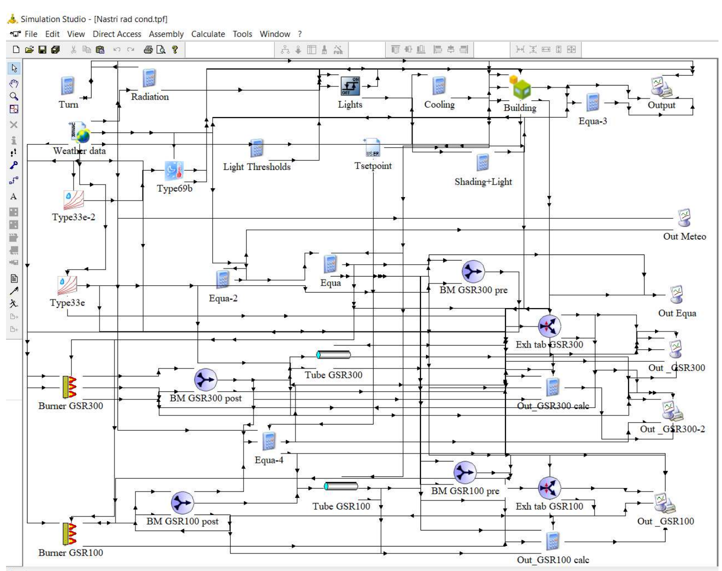
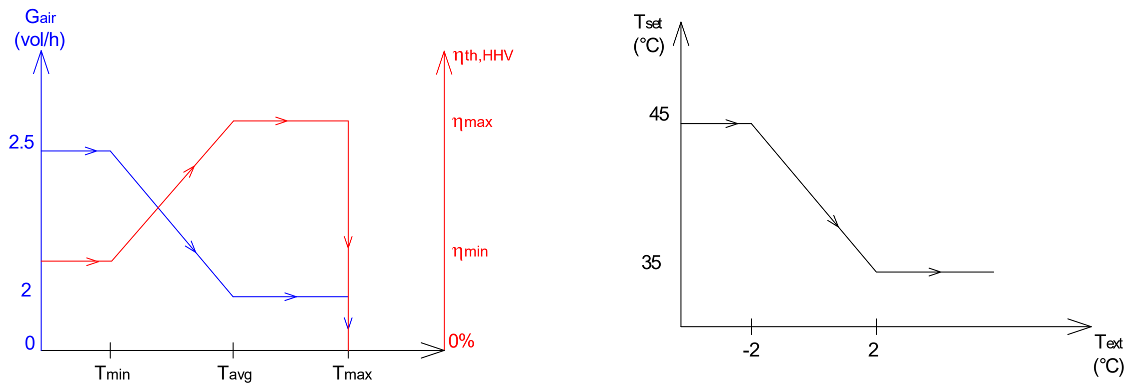


FIGURE 4. Trnsys project of the building and the condensing radiant tubes heating plant.

As already stated in the Introduction, two alternative heating systems are considered as benchmark to evaluate the energy and thermal comfort performance of the CRT system. A first solution is a ground air heater system (Air), the hypotheses considered in this case are:

- total nominal thermal power of the ground air heater installed 1600 kW (equal to the CRT case);
- thermal efficiency and air flow rate variable as function of the indoor air temperature as reported in FIGURE 5a (2 vol h^{-1} is considered the minimum value useful to keep suitable uniformity of indoor air temperature). Thermal efficiency is considered varying in different ranges to take into account a more recent generator case ($0.80 \leq \eta_{th,HHV} \leq 0.84$) or the case of an older one ($0.70 \leq \eta_{th,HHV} \leq 0.72$);
- supply air temperature variable as function of outdoor air temperature (FIGURE 5b).

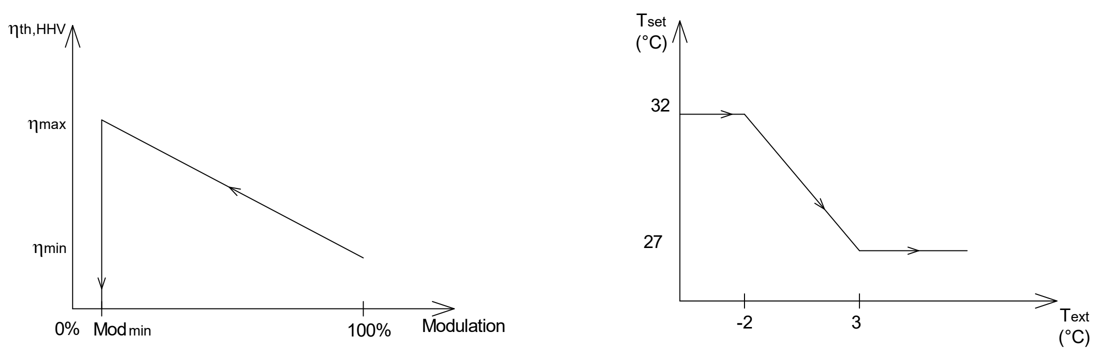


(a) (b)

FIGURE 5. Air heater system: (a) thermal efficiency (on HHV, red) and supply air flow rate (blue, in vol h^{-1}) (η_{max} is equal to 0.84 or 0.72, η_{min} is equal to 0.80 or 0.70 for modern and old generator respectively); (b) supply air temperature as function of outdoor air temperature. In the simulations the hypotheses are T_{min}=17 °C, T_{avg}=17.5 °C, T_{max}=18 °C.

The second benchmark system is a radiant floor coupled to a condensing boiler plant (CRF), the hypotheses considered in this case are:

- distance between tubes 0.3 m, outer diameter and thickness of tubes 0.02 m and 0.002 m respectively;
- radiant floor water flow rate 30 kg h^{-1} m^{-2};
- nominal thermal power of the condensing boiler 1600 kW (equal to the CRT case);
- thermal efficiency variable as function of the burner modulation (FIGURE 6a). As previously, thermal efficiency is considered varying in different ranges to take into account a more recent generator ($0.90 \leq \eta_{th,HHV} \leq 0.96$) or an older one ($0.87 \leq \eta_{th,HHV} \leq 0.93$);
- supply water temperature variable as function of outdoor air temperature (FIGURE 6b).



(a) (b)

FIGURE 6. Condensing radiant floor system: (a) thermal efficiency of the condensing boiler (on HHV) (η_{max} is equal to 0.96 or 0.93, η_{min} is equal to 0.90 or 0.87, Mod_{min} is equal to 0.10 or 0.30 for modern and old generator respectively); (b) supply water temperature as function of outdoor air temperature.

Cooling operation of the radiant tubes

The cooling capacity of the radiant tubes is worth of investigation. Based on the calculation of the exhausted mass flow rate in heating operation, the internal convective heat transfer coefficient is determined in order to evaluate the cooling capacity of the system when fed by cooled air.

The starting point of calculation are the experimental data concerning the tubes' wall temperature. The data provided by the tubes' manufacturer concern a dual-tube radiant system with 180 kW useful thermal power (on LHV of natural gas). A first attempt of the internal convective coefficient for this kind of application can be 40 W m^{-2} K^{-1}. Given the thermal power to be exchanged and the dimensions of the tubes (length 72+72 m, outer diameter 300 mm), a 60 °C temperature difference between exhausted and external wall of the tubes can be fixed. As an air excess between 60 % and 70 % is declared by the manufacturer, the burner that produce 180 kW needs around 19 Sm^3 h^{-1} of natural gas and gives around 300 Sm^3 h^{-1} of exhausted at around 1550 °C. The exhausted experimental temperature is max 400 °C (in the initial part of the tubes), and min 190 °C (in the last part of the tubes). The consequence is that the recirculated exhausted flow rate is around 2200 kg h^{-1}, giving a supply exhausted mass flow rate around 2600 kg h^{-1} for this model of radiant tubes.

On this basis, an initial exhausted velocity of 20 m s^{-1} is calculated, decreasing till 13 m s^{-1} in the last part of the tube. This allows to calculate the convective coefficient between the exhausted and the internal wall, based on the Dittus-Boelter equation (Eq. 2):

$$Nu = 0.023\ Re^{0.8}\ Pr^{1/3} \tag{2}$$

The result is around 30 W m^{-2} K^{-1}. The cooling performance of the tubes when supplied by a similar mass flow rate of cooled air (2600 kg h^{-1} supposed at 5 °C) can now be calculated. The thermal (cool) exchanges are between cooled air and tubes' wall, and between the latter and indoor environment. This second exchange is the sum of two contributions, radiative and convective. Based on further hypothesis on the emissivity of the tube (0.9) and the external convective coefficient (6 W m^{-2} K^{-1}), an iterative procedure allows to calculate the tubes' wall temperature and, finally, the cooling capacity of the tubes. Some very preliminary results of such procedure are reported in next section.

RESULTS AND DISCUSSION

Heating operation: energy performance and indoor thermal comfort analysis

The monthly energy performance of the three heating systems are reported in FIGURE 7 and FIGURE 8. The former reports the primary energy consumption (natural gas) in absolute terms, and the energy needs of the building. The latter reports the primary energy consumption in specific terms (per square meter of floor area), and the mean monthly thermal efficiency based on HHV of natural gas.

For all the months of the heating period, the Air system features the worst performance in terms of primary energy consumed, even if considering the modern generator case. Note that the saving of the "condensing solutions" (CRT and CRF) with respect to the Air plant is greater, in relative terms, during mild months (from April to October) (FIGURE 7). This is apparent in FIGURE 8, where the greater distances between the "condensing solutions" efficiencies and the Air systems efficiencies occur during the period with the lowest heating demand. This is due to the positive characteristic of the condensing heaters to increase their efficiency when operating at partial conditions.

When comparing CRT vs CRF, the radiant tubes feature lower energy consumption during the coldest months (from December to February, when the heating needs are the greatest) (FIGURE 7). Instead, energy performance of the two systems are quite similar during milder months. Obviously, the advantage of the CRT plant is greater when compared with CRF with older boiler.

TABLE 3 reports the annual value of the specific primary energy consumption and the thermal efficiency. The innovative condensing radiant tubes system features the best energy performance, allowing a very high primary energy saving (till more than 30 %) with respect to the Air system, and a significantly higher energy saving (till more than 7 %) with respect to the CRF one.

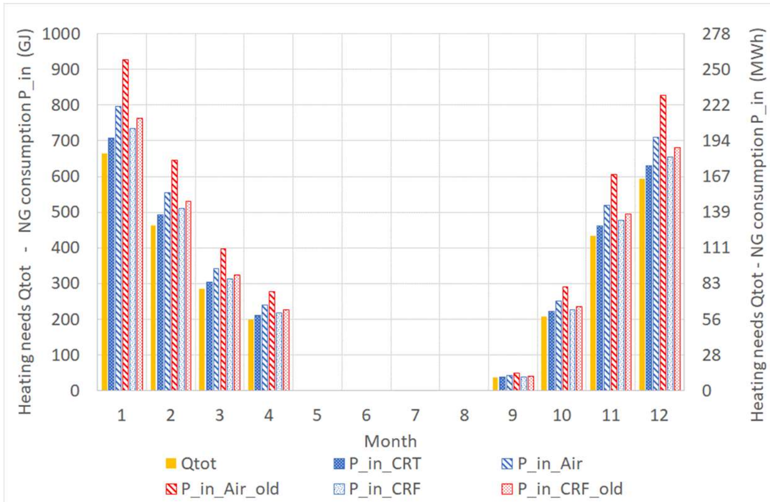


FIGURE 7. Monthly values of heating needs and primary energy consumption (natural gas) for the different heating systems.

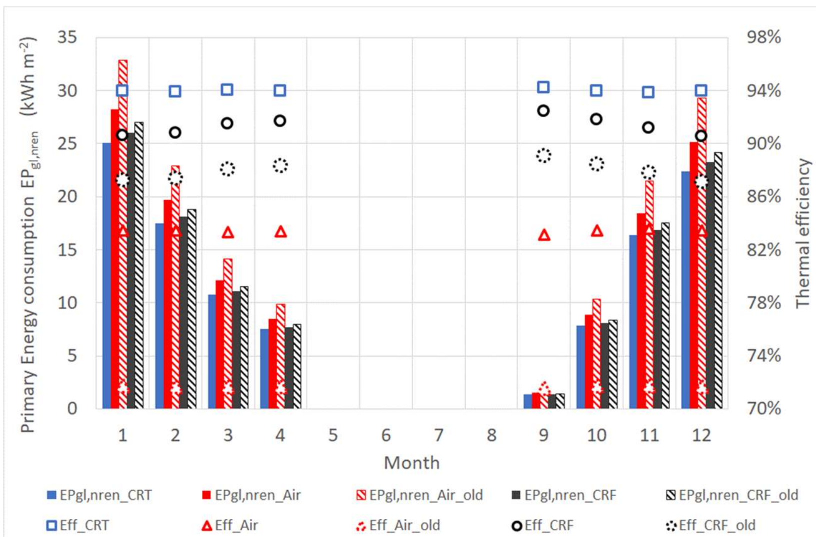


FIGURE 8. Monthly values of specific primary energy consumption and thermal efficiency for the different heating systems.

TABLE 3. Annual values of thermal efficiency, specific primary energy consumption for the different heating systems. Primary energy saving of CRT with respect to the benchmark technologies is reported as well.

Efficiency ($\eta_{th,HHV}$)					$EP_{gl,nren}$ (kWh m^{-2})				
CRT	**Air**	**Air_old**	**CRF**	**CRF_old**	**CRT**	**Air**	**Air_old**	**CRF**	**CRF_old**
93.9%	83.5%	71.7%	91.0%	87.6%	108.9	122.5	142.6	112.4	116.8
Annual primary energy saving of CRT					-	12.5%	30.9%	3.2%	7.2%

As a result of the energy analysis, CRT seems to give a not so higher advantage with respect to CRF. Nevertheless, from the indoor thermal comfort conditions point of view, the two systems perform differently: CRT system allows view factors around one between the heating tubes and persons on the floor, whereas the presence of production equipment and appliances on the floor reduces the useful area for the heat exchange in the case of CRF system. Therefore, the condensing radiant floor system cannot guarantee the same thermal comfort conditions as the radiant tubes.

An analysis of such aspects is carried out on two consecutive days of the heating period (e.g. 24th-25th January) with different characteristics in terms of solar radiation and outdoor air temperature (FIGURE 9). FIGURE 10 reports the indoor air temperature and operative temperature for thermal zone 1 for the CRT and Air systems, FIGURE 11 reports the same figures for the CRF system comparing the theoretical case (the whole floor area is available for the heat exchange) with a more realistic case (70 % of the floor is available).

In terms of predicted mean vote (PMV), here not reported for the sake of brevity, all the heating systems allow a value between 0 and 0.4 during the occupation time, that is a predicted percentage of dissatisfied (PPD) between the minimum (5 %) and 8 %. As a matter of fact, these can be considered satisfactory thermal comfort conditions, useful to classify the indoor environment in class B based on ISO 7730 standard. Nevertheless, the heating systems perform differently concerning the operative temperature (dotted red line in FIGURE 10 and FIGURE 11): with the CRT plant such a variable remains more constant and more similar to the air temperature (continuous black lines) during the day with respect to the CRF and, above all, to the Air system. Another advantage of the CRT plant is the greater swiftness of reaching the set-point value at the start of operation (at 6.00 am).

The greater thermal inertia of the condensing radiant floor system has two consequences: a more difficult control capacity of the operative temperature with respect to the radiant tubes solution (above all during the second day when the greater solar radiation thermal gains can disturb the control), and a lower decay of air temperature during the night. This second effect from one side can reduce the time to reach the set-point at the next start-up, but on the other side increases the heat losses across the building all night long.

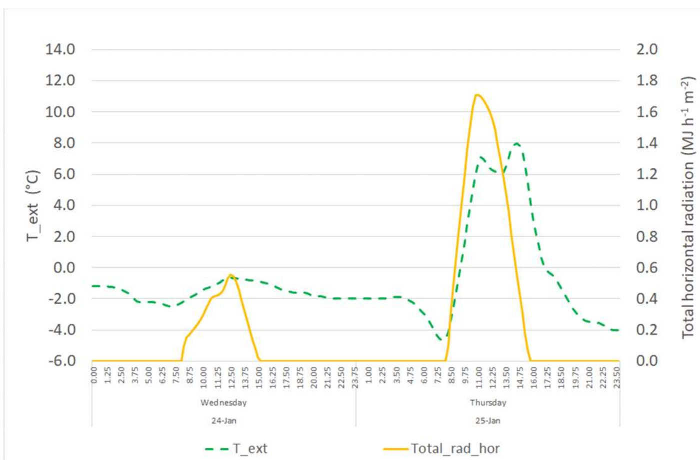
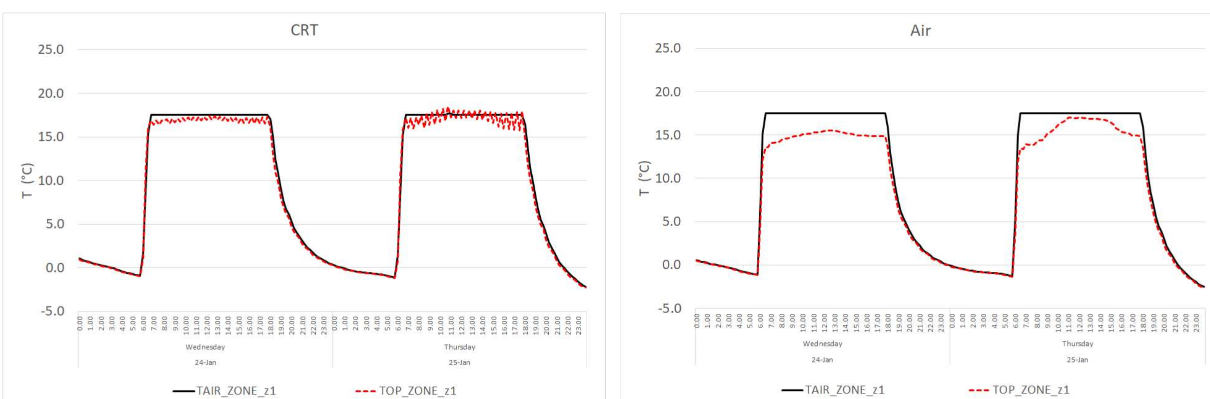
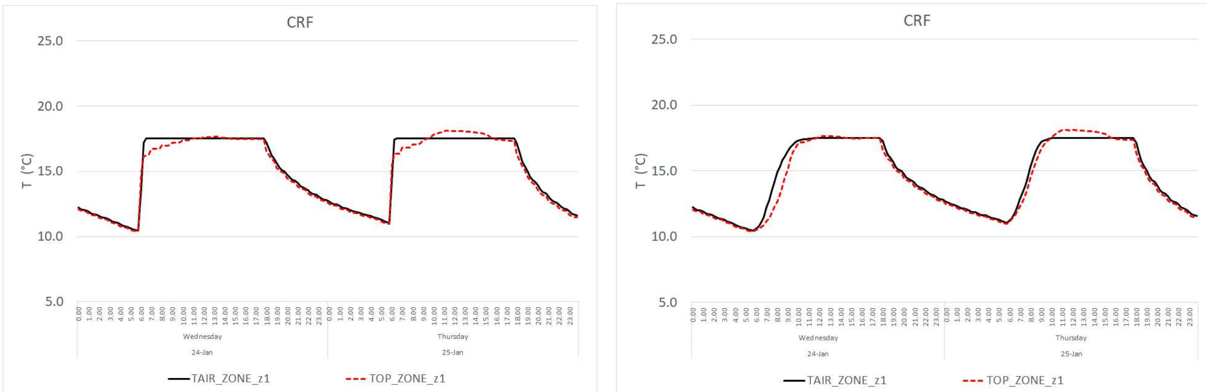


FIGURE 9. Outdoor air temperature and global solar radiation on the horizontal for two consecutive days of the heating period.



(a) (b)

FIGURE 10. Indoor air temperature (TAIR) and operative temperature (TOP) for thermal zone 1: (a) condensing radiant tubes system; (b) air heater system.



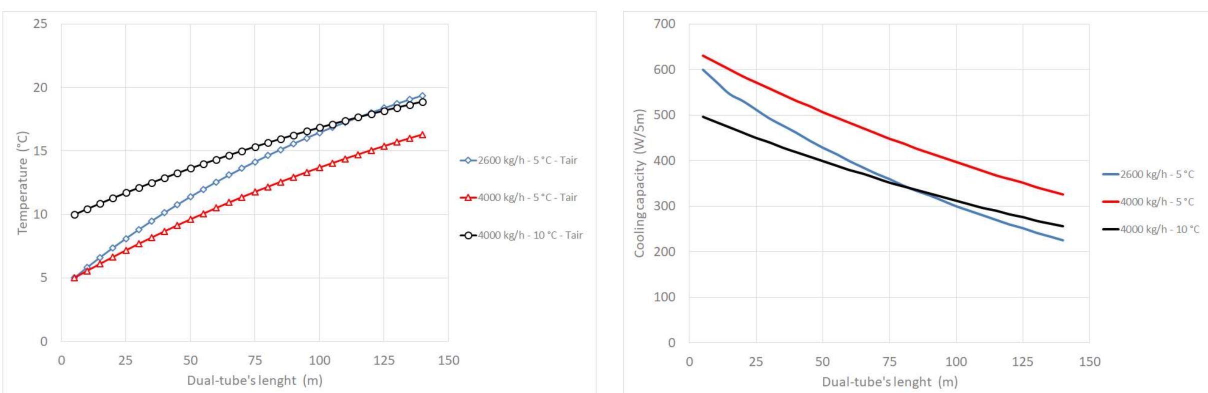
(a) (b)

FIGURE 11. Indoor air temperature (TAIR) and operative temperature (TOP) for thermal zone 1: (a) condensing radiant floor system with radiant area equal to the available floor area; (b) with radiant area less than the available floor area by 30 %.

The effect of the partial unavailability of the floor area due to occupation by production equipment and warehouses is shown in FIGURE 11. The increasing of thermal inertia of the heating system in case of reduced available area implies an increasing difficulty to guarantee the indoor thermal comfort condition during the start-up of the plant; moreover, the set-point values are reached with a delay of about two hours. This is also the main cause of the greater primary energy consumption of the CRF plant as previously described, because condensing boiler has to operate at nominal power for longer.

Cooling operation: energy performance

Based on the hypotheses and calculation procedure before described, the cooled air and wall temperature along the tubes are determined. Cooled air enters the tube at 5 °C, after 70 m (the first half) its temperature raises to 13.6 °C, and at the return (140 m) its temperature is 19.4 °C when considering a mass flow rate of 2600 kg h^{-1} (FIGURE 12a). For an indoor air temperature of 30 °C, such values allow a cooling capacity around 10 and 11 kW. More than half of this power (6-7 kW) is exchanged in the supply tube, whereas in the return tube only 4-5 kW can be exchanged due to the higher air temperature (FIGURE 12b).



(a) (b)

FIGURE 12. Temperature of cooled air inside the tubes (a) and cooling capacity for every 5 m of tubes' length (b) varying the air mass flow rate and inlet temperature.

Although roughly, it is interesting to evaluate how these performances vary when the operation conditions of the "cooling tubes" modify. Considering a higher flow rate of cooled air (4000 kg h^{-1} again at 5 °C), the air velocity increases (from 8 to 13 m s^{-1}), so does the internal convective coefficient (37 W m^{-2} K^{-1}), with the consequence of a lower air and wall temperature at the tube's exit (FIGURE 12a). This implies a greater cooling capacity (13 kW), especially in the return tube (FIGURE 12b). At higher mass flow rate the risk of condensation of humidity is greater, so a higher inlet air temperature has to be considered (i.e. 10 °C). In this case, a cooling power similar to the first conditions can be featured (10 kW), with the supply tube more penalized with respect the return (FIGURE 12b).

CONCLUSIONS

The present study is focused on the comparison of energy and thermal comfort performance of three heating systems for an industrial building located in a severe climatic conditions resort. The innovative condensing radiant tubes system allows to satisfy the heating load with a very interesting primary energy saving that can reach 30 % with respect to a traditional air heater system, and 7 % with respect to a radiant floor coupled to a condensing boiler. Nevertheless, in terms of thermal comfort conditions the condensing radiant tubes allow a better performance, especially during the first hours of operation in the morning, as in real application the floor area is not fully available due to the presence of the production equipment and warehouses. Moreover, with the CRT system the operative temperature remains more constant and more similar to the air temperature during the day with respect to the CRF and, above all, to the Air system.

An analysis of the cooling capacity of the radiant tubes plant is performed based on a 180 kW thermal power model, revealing that a cooling effect variable between 200 and 600 W every 5 m of the tubes is possible. This make the radiant tubes system a competitive solution with respect to other solutions (such as the radiant floor system coupled to condensing boiler) for the annual climatization of industrial buildings, with a greater capacity of guarantee indoor thermal comfort conditions.

ACKNOWLEDGMENTS

The Authors would like to kindly thank Eng. Gimmi Fraccaro, Eng. Francesco Cerboni, Eng. Daniele Lazzaron (Officine Termotecniche Fraccaro s.r.l.) for the data provided concerning the heating performance of condensing radiant tubes, and Eng. Marco Larovere (Studio di Ingegneria Larovere ing. Marco) for data concerning the building.

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