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Approximate and CFD energy performance analyses of industrial heating by water strip modules Part 1

Analisi approssimata e CFD delle prestazioni energetiche di sistemi di riscaldamento industriale con termostrisce radianti Parte 1

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Abstract

The objective of this study is the theoretical evaluation of the energy performance and comfort conditions of an industrial environment heated by a radiant strip heating system fed with hot water, which operates in steady state conditions, by varying the operating conditions. The analysis is carried out in two successive steps: firstly, the convective coefficients and the heating power (both convective and radiant part) of the heating system are evaluated by simulating heat exchange conditions similar to real operations, in comparison with the nominal data according to EN14037 standard. To carry out this preliminary assessment, different references in the scientific literature are considered concerning experimental measurements and/or numerical simulations of similar applications (low or medium temperature radiant heating systems in large rooms). In the second step of this study (the second part of this article), the analysis is deepened by CFD simulations to confirm the preliminary assessments..

Keywords:

- Radiant system
- Water strips
- Industrial heating
- Energy efficiency
- CFD

Sommario

L'obiettivo dello studio è la valutazione teorica delle prestazioni energetiche e delle condizioni di comfort all'interno di un ambiente industriale prodotte da un sistema di riscaldamento a strisce radianti alimentate ad acqua calda che opera in condizioni di regime stazionario al variare delle condizioni al contorno. L'analisi viene svolta in due fasi successive: in una prima fase, vengono quantificati i coefficienti convettivi e le rese (quota convettiva e radiante) del sistema di riscaldamento simulando situazioni di scambio termico vicine alla realtà, a confronto con i dati nominali secondo norma EN14037. Per svolgere questa valutazione preliminare ci si è basati su fonti reperite in letteratura scientifica inerenti misure sperimentali e/o simulazioni numeriche di applicazioni simili (sistemi di riscaldamento di tipo radiante a bassa o media temperatura in ambienti di grandi dimensioni e/o volumetrie). In una seconda fase dello studio, oggetto della seconda parte di questo articolo, l'analisi viene approfondita mediante simulazioni CFD al fine di confermare le valutazioni preliminari.

Parole chiave:

- Sistema radiante;
- Termostrisce radianti;
- Riscaldamento industriale;
- Efficienza energetica;
- CFD

NOMENCLATURE

Symbol	*Meaning*	*Unit*
A	Area	m²
D	Diameter	m
K	Constant of the characteristic equation of the water strip	$W K^{-1}$
L	Length	m
q	Thermal power	kW
T	Temperature	°C
U	Thermal transmittance	$W m^{-2} K^{-1}$
w	velocity	$m s^{-1}$
α	Convective coefficient	$W m^{-2} K^{-1}$
Δ	Difference	–
θ	Outdoor temperature	°C

Subscripts	*Meaning*
a	air
avg	average
back	back side of the water strip
conv	convective
e	external
f	fluid
front	front side of the water strip
i	internal
k	generic internal surface
m	mean
p	plate
r	radiant
tot	total
zone	thermal zone

Introduction

The scope of the calculus procedure described hereafter is to evaluate how the thermal power produced by hot water strips varies as the operating boundary conditions vary with respect to the ideal ones according to EN14037 standard that describes the test method and the test installation for determining the thermal output of pre-fabricated ceiling mounted radiant panels ([1]). The main idea is to verify how much the yield of the heating system increases in real operation conditions with respect to the standard conditions. These are here briefly reported:

- the booth for testing ceiling mounted radiant panels has 4 m x 4 m x 3 m inside dimensions, and air infiltration has to be limited at all;
- all six surrounding surfaces have an emissivity of minimum 0.9, and they are chilled to keep the difference between them and the average temperature of all six surfaces not higher than 0.5 K;
- air and all six surrounding surfaces temperatures are fixed at 20 °C.

The analysis is carried out in steady state conditions. The authors refer to a commercially available water strip model which is assumed to be used in zone 1 of the shed of Figure 1 ([2]). The boundary conditions that have been varied are:

- convective heat transfer coefficient between the heating water strips and the internal air, to simulate possible situations during the real operation of the heating system:
  - door opening on a side wall;
  - skylight opening on the roof;
  - operation of a ventilation system able to renew the air with a rate of 0.5 vol h^{-1};
- presence or absence of anticonvective flashing.

As regards the first point, various references are considered, even if none of these specifically concern studies focused on water strips; instead, they refer to heating or cooling systems that are somehow similar to these.

The first studies on the effects of the variation of some parameters on the heat exchange of the surfaces of indoor spaces date to the 1950s. Schutrum et al. [3], for example, experimentally measured the effect of the room size and of the non-uniformity of the temperatures of the heating surfaces on the yield of these surfaces. They obtained a marginal effect for the former; instead, for the latter, they highlighted a yield of the panels, both ceiling and on the floor, equal to that which would have occurred with the entire surface heated to a uniform temperature equal to the weighted average with the heated and unheated surface areas.

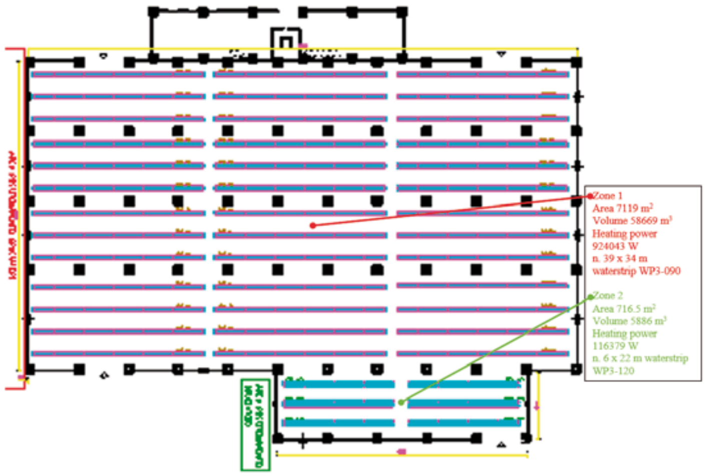


Figure 1 – Factory shed used in the present study (courtesy of Studio di Ingegneria Larovere ing. Marco)

Figura 1 – Capannone industriale utilizzato per il presente studio (si ringrazia lo Studio di Ingegneria Larovere ing. Marco)

Table 1 – Thermal zones of the building

Tabella 1 – Zone termiche dell'edificio

	Thermal zone 1	Thermal zone 2
Floor area (m²)	7119	716.5
Net height (m)	8.24	8.22
Indoor air temp. (°C)	18	18
Net volume (m³)	58669	5886

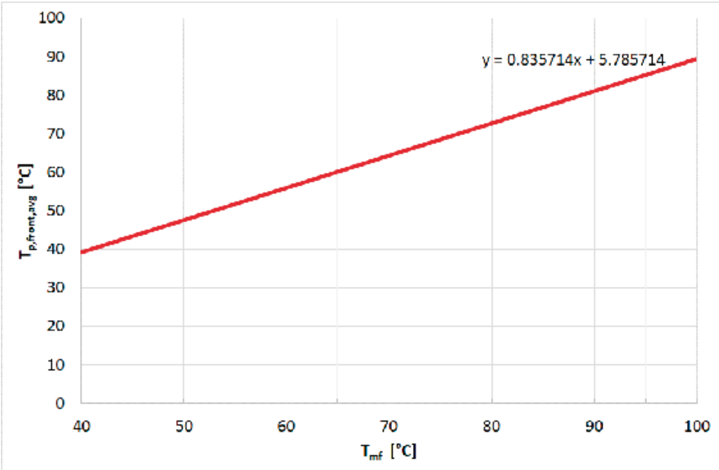


Figure 2 – Front side plate average temperature ($T_{p,front,avg}$) in function of the hot water average temperature (T_{mf})

Figura 2 – Temperatura media della superficie frontale (inferiore) della piastra ($T_{p,front,avg}$) in funzione della temperatura media dell'acqua calda (T_{mf})

Some equations for the natural convection coefficients in the case of radiant ceiling panels have been provided in [4]. More recently, Awbi and Hatton have carried out numerous measurement campaigns to develop equations for the calculation of the heat transfer coefficients for both natural [5] and mixed (natural+forced) [6] convection. In this case, however, the measurements were carried out in a very small room.

In [7], an extensive review on the determination of the convective heat transfer coefficients of surfaces in two and three dimensions was carried out. In this study, the authors highlighted the wide variety of values that can be determined as a function of the different heat transmission configurations (width and position of the heated surfaces, surfaces temperature, etc.).

To the best knowledge of the authors, none of the previous works has determined some relationships to quantify the effects of air flow near hot water strip systems hanging from the ceiling. Two works have been carried out in this regard [8] [9], but focused on ceiling panels for cooling. Hence the reason for this work with the purposes listed above.

Methods

Building modelling

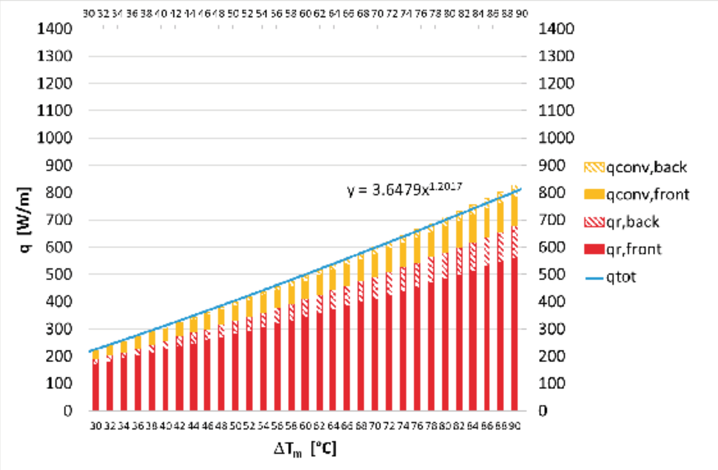
The industrial building (type of use E.8 by the Italian decree DPR 412/93) is located in the province of Cuneo (North-West of Italy), latitude 44°36' N, altitude 404 m a.s.l., 2814 degree days (climatic zone E). The heating period is from 15th September till 30th April. The thermal transmittances expressed in W m^{-2} K^{-1} are: 0.389 for external wall, 0.128 for the floor facing ground, 4.086 for ceiling, 0.208 for ceiling shed, 5.0 for windows.

Table 2 – Equations for the calculation of the convective heat transfer coefficient

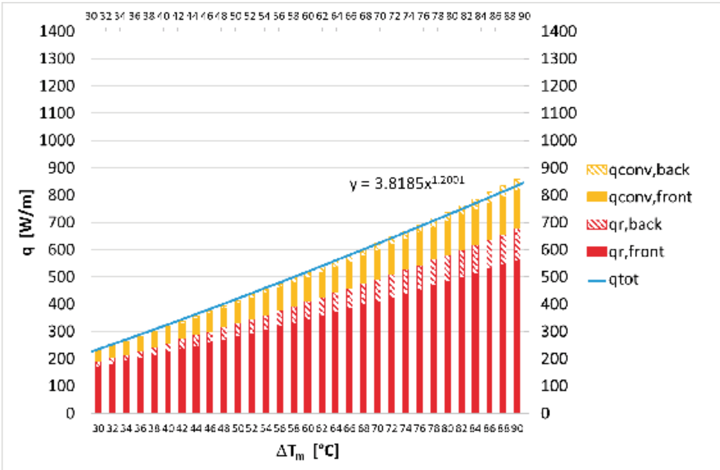
Tabella 2 – Equazioni per il calcolo del coefficiente di scambio termico convettivo

Case nr.	Type of convection	Equation	Reference
1	Natur. Conv. (Front)	$\alpha_{conv,front} = 0.59 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$	[10] (p. 344) [12] (p. 240)
2	Natur. Conv. (Front)	$\alpha_{conv,front} = 0.71 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$	[13] (p. 23)
3	Natur. Conv. (Front)	$\alpha_{conv,front} = 0.87 \cdot ((T_{p,front,avg} - T_{ai})^{0.25})*(4.91/D_e)^{0.25}$	[14] (Eq. (10))
4	Natur. Conv. (Front)	$\alpha_{conv,front} = 1.736 \cdot (T_{p,front,avg} - T_{ai})^{0.16}/D_e^{0.52}$	[1]
5	Forced Conv. (Front)	$\alpha_{conv,front} = (0.0296 \cdot (w \cdot 0.3/1.644e\text{-}5)^{0.8}) \cdot 0.0266/0.3$	[15] (p. 279)
1' – 2' – 3' – 4'	Natur. Conv. Coefficient increase due to outdoor air exchange rate	$\alpha'_{conv,front} = 1.3\ \alpha_{conv,front}$ (based on the increase of 60% @ 1 vol/h, 0 °F)	[3]
-	Natur. Conv. (Back)	$\alpha_{conv,back} = 1.32 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$	[10] (p. 344)

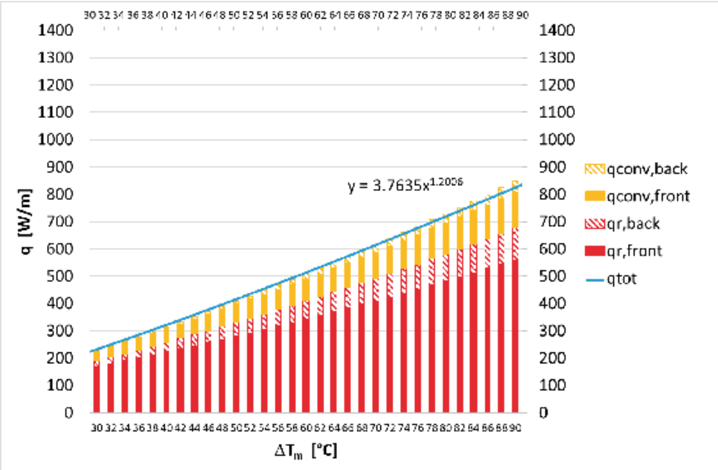
D_e = equivalent diameter = 4 · area/perimeter (m)
w = air velocity (m/s)



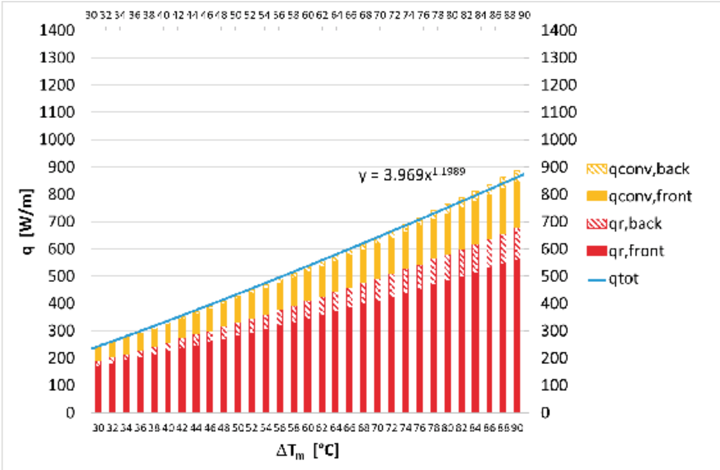
Case 1 – Nat. Conv. – $\alpha_{conv,front} = 0.59 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$



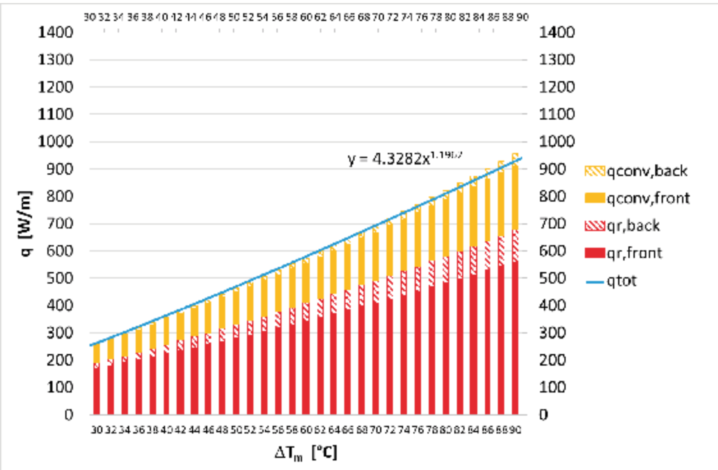
Case 1′



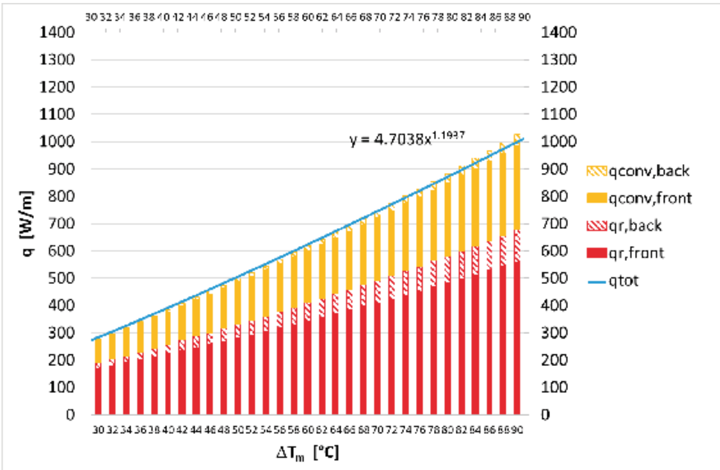
Case 2 – Nat. Conv. – $\alpha_{conv,front} = 0.71 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$



Case 2′



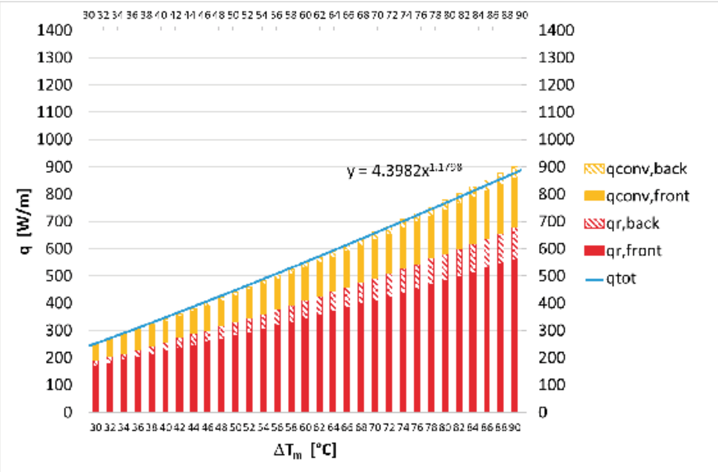
Case 3 – Nat. Conv. – $\alpha_{conv,front} = 0.87 \cdot ((T_{p,front,avg} - T_{ai})^{0.25}) * (4.91/D_e)^{0.25}$



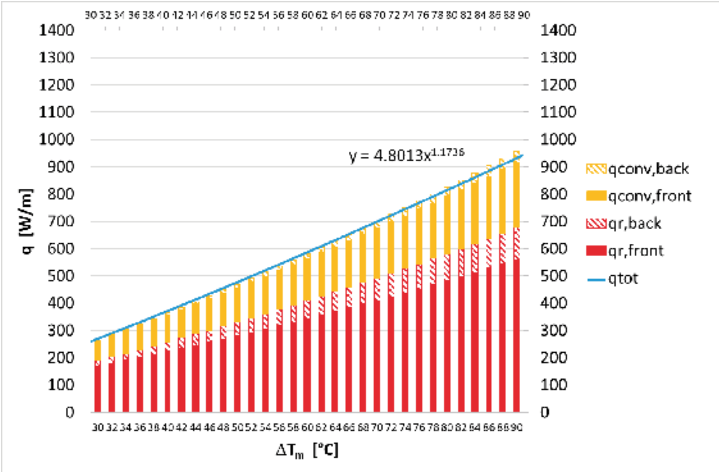
Case 3′

Figure 3a – Thermal power exchanged by the water strip without flashing presence and with thermal insulation for the cases considered in Table II. Heat flux is expressed in watt per meter of linear length of the water strip as it is referred to a strip having 0.9 m width

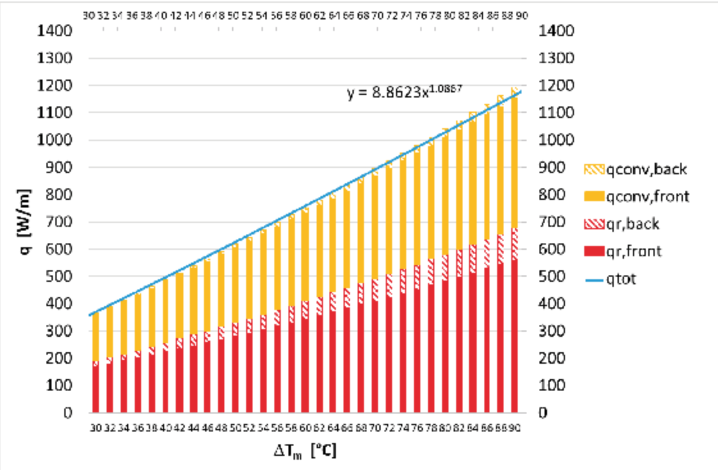
Figura 3 – Potenza termica scambiata dalla termostriscia in assenza di scossalina ed in presenza di isolamento termico per i casi considerati in Tabella II. Il flusso termico viene espresso in watt per metro lineare di lunghezza della termostriscia poichè si riferisce ad una striscia di larghezza pari a 0,9 m



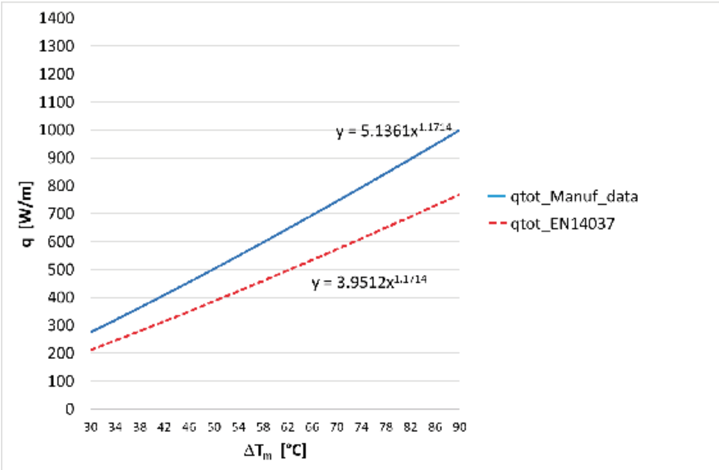
Case 4 – Nat. Conv. – $\alpha_{conv,front} = 1.736 \cdot (T_{p,front,avg} - T_{ai})^{0.16}/D_e^{0.52}$



Case 4′



Case 5 – Forced Conv. (w = 1 m/s) – $\alpha_{conv,front} = (0.0296 \cdot (w \cdot 0.3/1.644e\text{-}5)^{0.8}) \cdot 0.0266/0.3$



Data from the water strip producer [9] [16]

Figure 3b – Thermal power exchanged by the water strip without flashing presence and with thermal insulation for the cases considered in Table II. Heat flux is expressed in watt per meter of linear length of the water strip as it is referred to a strip having 0.9 m width. The last graph reports the performance of the water strip according to the EN14037 standard but without thermal insulation in the collectors [16] and the performance determined by using the formula of case 2′ in the EN14037 operating conditions

Figura 3 – Potenza termica scambiata dalla termostriscia in assenza di scossalina ed in presenza di isolamento termico per i casi considerati in Tabella II. Il flusso termico viene espresso in watt per metro lineare di lunghezza della termostriscia poichè si riferisce ad una striscia di larghezza pari a 0,9 m. L'ultimo grafico riporta le prestazioni della termostriscia secondo norma EN14037 ma senza isolamento termico nei collettori [16] e quelle determinate utilizzando la formula del caso 2' in condizioni operative secondo EN14037

The building is divided into two thermal zones (Figure 1) whose main characteristics are reported in Table I [1]. In this study, we refer to thermal zone 1 only.

Calculus procedure

Here, the calculus procedure to evaluate the total heat power (convective+radiant) exchanged by the water strips considering "real operation conditions" instead of "standard conditions" (i.e., indoor air and surfaces temperature fixed at 20 °C, and no air flows, following the EN14037 standard before mentioned in the Introduction). Heat flux is expressed in watt per meter of linear length of the water strip as it is referred to a strip having 0.9 m width.

1) calculus of the hot water flow average temperature $T_{f,avg}$. It is calculated on the basis of the difference between such temperature and

that of indoor air T_{ai} (ΔT_m) (with T_{ai} = 15 °C) (ΔT_m is considered varying in the 30-90 °C range following the available performance data of the water strip) [10]:

$$T_{f,avg} = \Delta T_m + T_{ai}$$

2) calculus of the lower side (front) water strip (plate) average temperature $T_{p,front,avg}$ in function of the hot water average temperature $T_{f,avg}$ on the basis of literature reference data [11] (Figure 2);
3) calculus of the upper side (back) water strip (plate) average temperature $T_{p,back,avg}$ in function of the presence of thermal insulation and on the basis of literature reference data [11]:

$$T_{p,back,avg} = IF\left(THERMAL\ INSULATION = YES; 0.37; 0.52\right) \cdot T_{p,front,avg}$$

4) calculus of the mean radiant temperature of the thermal zone 1 (T_{mr}). It is determined as the weighted average (where the weights are the areas) of the internal surface temperatures T_{sk}, considering that all surfaces have the same emissivity:

$$T_{mr} = \frac{\sum_k A_k \cdot T_{sk}}{\sum_k A_k}$$

T_{mr} is calculated on the basis of the thermal transmittance values U_e and the outdoor design air temperature q; suitable external and internal convective heat transfer coefficients (a_e = 25 W m^{-2} K^{-1} and a_i [12], respectively) are set:

$$\alpha_{ik} \cdot (T_{ai} - T_{sk}) = U_e^* \cdot (T_{sk} - \theta) \rightarrow T_{sk} = \frac{\alpha_{ik} \cdot T_{ai} + U_e^* \cdot \theta}{U_e^* + \alpha_{ik}}$$

where $U_e^* = 1/(1/U_e + 1/\alpha_e)$ is the thermal transmittance taking into account the external convective heat transfer coefficient. On the basis of the data reported in [12], the results are:

$$T_{mr,zone1} = 11.4\ °C \qquad T_{mr,zone2} = 10.6\ °C$$

5) Calculus of the radiant heat flux exchanged by the lower and upper plate parts (L is the plate length):

$$q_{r,front} = 5.67 \cdot 10^{-8} \cdot 0.95 \cdot L \cdot \left(T^4_{p,front,avg} - T^4_{mr,zone1}\right) \quad (W/m)$$

$$q_{r,back} = 5.67 \cdot 10^{-8} \cdot 0.95 \cdot L \cdot \left(T^4_{p,back,avg} - T^4_{mr,zone1}\right) \quad (W/m)$$

6) Calculus of the convective heat transfer coefficient between indoor air and the lower part (front) ($a_{conv,front}$) and the upper part (back) ($a_{conv,back}$) of the water strip on the basis of the reference reported in Table II. Case n. 1 and n. 4 represent the heat convection between the water strip and the indoor air without induced air flow (i.e., natural heat convection), case n. 1' and 4' represent the same equations considering some induced air flow (i.e., opening a door in a lateral wall, opening a skylight, or activating an outlet turret in the shed roof, activating the ventilation plant with an air change rate of 0.5 vol h^{-1}). In the latter cases, an increase of 30% of the convective coefficient $a_{conv,front}$ is supposed, based on reference [4] where an increase of 60% of the convective coefficient is fixed with an air change rate of 1 vol h^{-1}. Case n. 5 represent the heat convection between the water strip and the indoor air with induced air flow as forced heat convection with air velocity of 1 m s^{-1}.
7) Determination of the convective thermal power exchanged by the lower and upper part of the water strip, with and without flashing presence (flashing is supposed to reduce the convective heat transfer by 25% [11]):

$$q_{conv,front} = IF\left(FLASHING\ PRESENCE = YES; 0.75; 1\right) \cdot \alpha_{conv,front} \cdot L \cdot \cdot \left(T_{p,front,avg} - T_{ai}\right) \quad (W/m)$$

$$q_{conv,back} = IF\left(FLASHING\ PRESENCE = YES; 0.75; 1\right) \cdot \alpha_{conv,back} \cdot L \cdot \cdot \left(T_{p,back,avg} - T_{ai}\right) \quad (W/m)$$

8) Calculus of the radiant, convective and total thermal power exchanged by the water strip:

$$q_{r,tot} = q_{r,front} + q_{r,back} \quad (W/m)$$

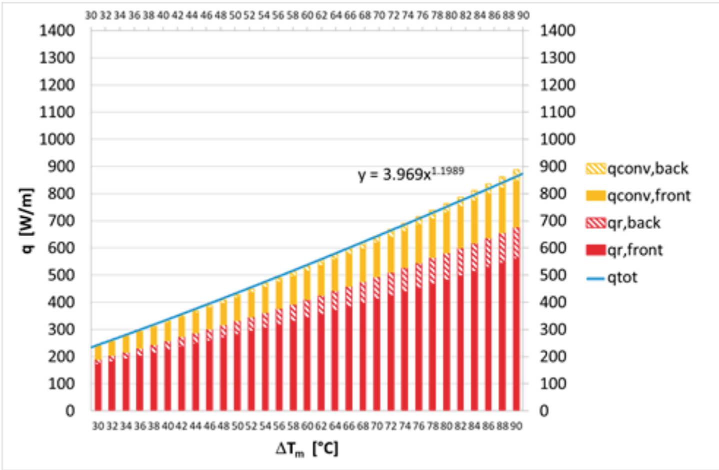


Figure 4 – Thermal power exchanged by the water strip (expressed in watt per meter of linear length of the water strip as it is referred to a strip having 0.9 m width) with no flashing presence and with thermal insulation for the operation conditions of the EN14037 standard

Figura 4 – Potenza termica scambiata dalla termostriscia (espressa in watt per metro lineare di lunghezza per una termostriscia di larghezza pari a 0,9 m) in assenza di scossalina ed in presenza di isolamento termico in condizioni operative secondo EN14037

AiCARR Journal / Vol 69, n. 4, 60 - 67, 2021

$$q_{conv,tot} = q_{conv,front} + q_{conv,back} \quad (\mathrm{W/m})$$

$$q_{tot} = q_{r,tot} + q_{conv,tot} \quad (\mathrm{W/m})$$

Results and discussion

The results are reported in terms of thermal power expressed in watt per meter of linear length exchanged by the water strip for the different cases described in the previous section (Table II). The performances stated by the water strip producer are reported as well (last graph in Figure 3). Some highlights can be deduced by comparing the graphs of Figure 3:

- the equation that approximates in the best way the real performance of the water strip is that of case 2' ($a_{conv,front} = 0.71 \cdot ((T_{p,front,avg} - T_{ai})/D_e)^{0.25}$ [14]) with a multiplying coefficient of 30%. In fact, by comparing the "power" equations reported on the graphs, it can be deduced that the one related to case 2' allows for the best approximation of that of the EN14037 standard to evaluate the thermal performance in function of ΔT_m:

$$q_{tot} = K\left(\Delta T_m\right)^n$$

As a matter of fact, this corresponds to the increase of the convective heat transfer coefficient when the water strip operates in real conditions with respect to the EN14037 test conditions. Furthermore, the effectiveness of the case 2 equation is confirmed by using the equation in EN14037 test conditions (Figure 4): the blue curve (q_{tot}), and so the relative "power" equation, can approximate quite well that one by the water strip producer (last graph in Figure 3).

- the flashing and thermal insulation presence on the upper part of the water strip reduces the thermal power. This is due to a decreasing of the convective heat transfer on both plate sides for the former, and of the convective and radiation heat transfer on the upper side for the latter.

Conclusions

The analysis carried out in the first part of this study, based on the reference literature, allows to estimate the increase of the convective heat transfer coefficient of a water strip heating system in the order of 30% - 40% in normal operating conditions (i.e. the presence of induced air flow near the radiant panels due, for example, to the opening of a door on a side wall, or to the opening of a skylight on the roof, or to the operation of the ventilation system with air change flow of 0.5 vol h^{-1}) compared to the operating conditions of the EN14037 standard.

Considering that the convective heat transfer weighs for a percentage variable between 35% and 45% of the total thermal output of the water strip (depending on the model and the ΔT_m), an increase in the overall yield in the order of at least 10% - 20% compared to the data stated according to the EN14037 standard is surely achievable.

Moreover, two further issues can be considered:

- in real operating conditions, the water strips "see" internal surface temperatures that probably are not uniform and lower than the internal air temperature, which commonly is around 20 °C (i.e., the test conditions according to EN14037);
- in the case of non-insulated water strips, there is a further thermal power exchanged with the environment, as can be seen from the data measured in the test room according to the EN14037 standard (blue curve in the last graph of Figure 3, [17]).

It can be concluded that the increase in the overall yield of the heating strips, compared to the data measured according to the EN14037 standard, can be at least of the order of 30%.

The purpose of the second part of this paper is to verify this conclusion by means of CFD simulations.

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