



Energy efficiency enhancement and waste heat recovery in industrial processes by means of the heat pipe technology: Case of the ceramic industry



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ARTICLE INFO

Article history:

Received 30 December 2017

Received in revised form

25 May 2018

Accepted 8 June 2018

Available online 11 June 2018

Keywords:

Heat pipes

Theoretical approach

Numerical model

Heat transfer

Energy efficiency

Emissions reduction

ABSTRACT

This paper investigates the application of heat-pipe based heat exchanger for improving the energy efficiency of industrial processes. In particular, the case of the ceramic industry is addressed and the potential heat recovery and reduction of fuel consumption is determined. A theoretical model is constructed based on the established, proven performance characteristics of heat-pipe technologies and the performance of the ceramic process are calculated using numerical simulation. The results of the kiln numerical model are then combined to the theoretical model of the heat-pipe based heat exchanger and the heat recovery potential is evaluated as well as the reduction of fuel consumption. The combined theoretical and numerical approach demonstrates that the application of the heat pipes based heat exchanger to the cooling stack of the ceramic kiln enables to recover more than 863 MWh of thermal energy that can be used for heating up the hot air stream of the pre-kiln dryer. Thus, approximately 110,600 Sm³ per year of natural gas can be saved from the burners powering the dryer and the emission of 164 tonnes per year of carbon dioxide can be avoided. Additionally, the avoided cost due to the fuel consumption reduction amounts to more than 22,000 Euro per year. These figures support the application of the heat pipes based heat recovery to the ceramic process from the viewpoint of the improvement the energy efficiency and environmental impact and also of the economic investment.

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1. Introduction

The use of heat pipe heat exchangers to recover waste heat from industrial exhaust streams has been investigated and applied in many sectors such as: steel, food and transportation. Heat pipes are defined as a passive heat transfer technology capable of transferring large amount of heat with no moving parts. A heat pipe is composed of a sealed tube with working fluid acting as the heat carrier. The working fluid is condensed at the top section of the tube and evaporating at the bottom, this will create a two phase heat transfer cycle [1]. Heat pipe heat exchangers are composed of rows of heat pipes exposed to an exhaust stream at the bottom and a cooling fluid (Air, water, etc ...) on the condenser section. Heat pipe units have been tested in different conditions (temperatures,

cooling fluid, evaporator fluid, working fluid). En Tian developed a heat pipe heat exchanger to recover the heat in a high temperature and challenging exhaust to transfer the heat to a clean air stream. The top part of the heat pipe is composed of a tube bundle with the air going through tube while the bottom part is composed of finned pipe. The system was able to save up to 15% of natural gas without any blockage on the exhaust side [2]. Jouhara and Meskimmon investigated the application of heat pipe heat exchanger technology in data centre cooling system. The amount of heat produce by computing can be substantial. This work developed a heat pipe heat exchanger to cool down data centre chamber using none pure air from the outside. The heat exchanger had a potential of 75% of energy savings by utilising heat pipe technology [3]. A.R.Lukitobudi designed a heat pipe based heat exchanger to recover heat from a bakery oven to recover hot air. The effectiveness of the designed heat pipe heat exchanger was 65% and was able to recover between 20 kW and 35 kW [4].

Many investigations have been conducted regarding the waste

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heat recovery from the cooling stage of ceramic kilns. Most of the innovative techniques have been summarized by Refs. [5] and [6] (see Fig. 1). The main waste heat recovery technique developed was the recovery of excess heat from the kiln. The hot air recovered from the cooling zone of the kiln is used to warm up the air for the drying stage. Due to the corrosion issue, standard heat exchangers are used to transfer the heat from the flue gas of the cooling zone to clean hot air which will be then sent to the dryer. The excess heat is also used to feed the burner with hot air as well as the central heating of the plant.

Cogeneration/combined heat and power plants have been developed and applied to ceramic manufacturing (see Fig. 2). The heat generated by the cogeneration gas engine is fed into a mixing chamber with the kiln flue gas. If the hot air temperatures is not sufficient, a gas burner is used. The radiation heating caused by the kiln go through a heat exchanger heated by the cooling water of the cogeneration gas engine. This hot air is then fed into the mixing chamber [7].

Organic Rankine cycle have also been developed and applied in the tiles ceramics manufacturing industry (see Fig. 3). An ORC system has been developed by Peris et al. [8]. The aim was to

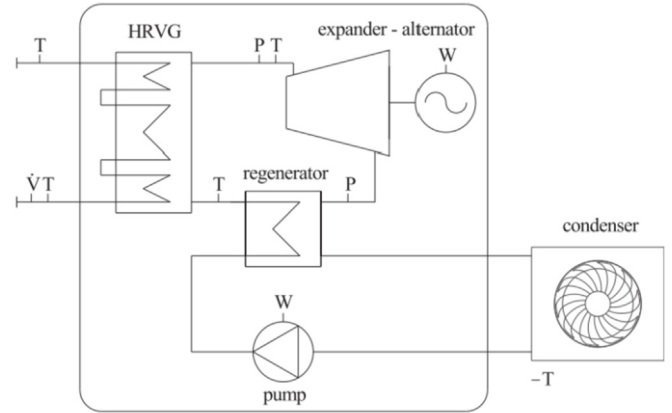


Fig. 3. Schematic of organic Rankine cycle.

recover the heat generated during the indirect cooling section of the kiln to produce electricity. The thermal power recovered from the clean air ranged from 128.19 KW to 179.87Kw. The maximum electrical power output ranged between 21 kW and 18.51 kW. The maximum efficiency of the ORC system was 12.47%. The production of electricity using ORC was not sustainable as the efficiency was low compared to the current system which send the hot air recovered in the indirect cooling stage to the dryers and burners. Also, the payback of the system was 4.7 years.

Luwa engineering have investigated the use of a heat exchanger to recover the flue gas from the firing exhaust stack to feed hot air into the drying stage and the spray dryer on a roller earth kiln [9] (see Fig. 4).

A heat exchanger was installed in the exhaust stack, the heat sink used was water. The hot water was then supplied at a temperature of 99.2 °C to the 2 spray dryers. The heat recovered for the two spray dryers was 1093 kW. As the exhaust of the firing stage is highly corrosive and has high amount of particulates, the use of a classic shell and tube heat exchanger is not ideal. The maintenance cost of such heat exchanger in highly corrosive environment is substantial as the shell and tube will have cold spot located close to the tube plate. This cold spot will condense the exhaust and corrode the plate.

The design process of new ceramic kilns as well as of retro-fit modifications to existing plants is usually carried out by means of theoretical analysis; such as in Ref. [10]. Where a theoretical formulation for the prediction of the kiln operating characteristics has been proposed under steady state conditions and many simplifications to the physical phenomena had to be made in order to

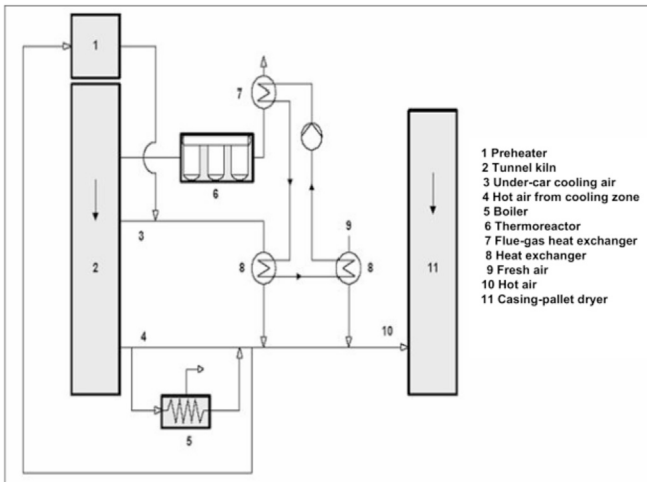


Fig. 1. Schematic view of a combined heat recycling system [5].

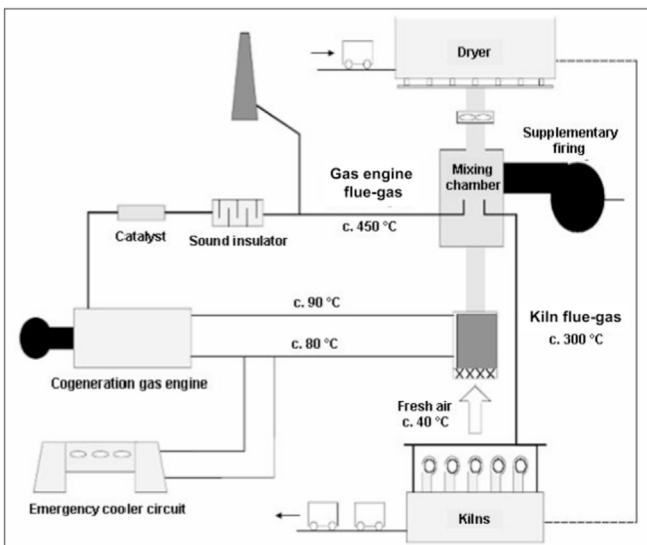


Fig. 2. Cogeneration/combined heat and power plant.

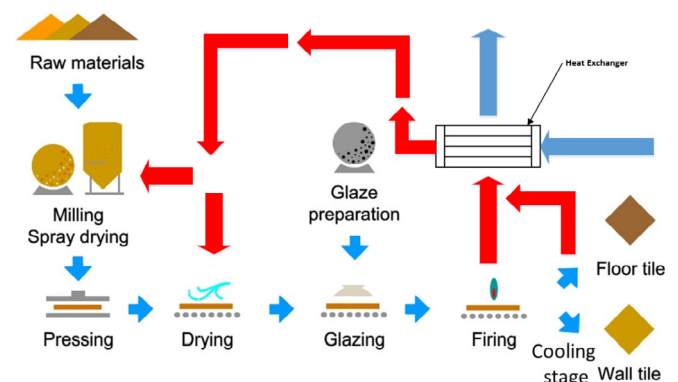


Fig. 4. Luwa heat exchanger system.

close the mathematical model. Mezquita et al. [11] adopted a similar theoretical approach for the calculation of the heat transfer between the hot air flow and the kiln walls. Nevertheless, the design process of ceramic kilns can be greatly improved by employing numerical tool for the evaluation of different solutions and validate the effects before actually realizing the kiln or modifying an existing plant [12]. Furthermore, numerical analysis is extensively adopted for the simulation of complex systems under time dependent operating conditions. In Ref. [13] a new cogeneration system based on the aluminium combustion with water has been investigated and both the numerical analysis and the experiments have been employed for the development of the solid fuel burner. The 0D/1D approach has also been adopted in the food processing industry for the analysis of the fruits and vegetables drying in order to estimate the optimal parameter for the process operations [14]. In Ref. [15] a parametric analysis for different real thermal configurations of thermoelectric generators for HVAC was carried out using a one-dimensional steady-state approach for the heat recovery calculation. Similarly, in Ref. [16] the lumped approach was used to optimize the working cycle of novel concept of cooperation system and the size of the system's components were designed by using the numerical simulation of the entire plant. In Ref. [17] the lumped numerical approach was adopted for evaluating the combined use of the concept of active PV façade and solar cooling and the energy savings for buildings cooling in hot climate were estimated. The adoption of system modelling proved to be fundamental when addressing the effects of a single components on the operation of an entire system; in Ref. [18] the effects on the fuel consumption of the energy recovered by different types of heat exchangers of the energy efficiency of a coffee roasting plant were predicted and the optimal configuration was assessed.

This paper investigates the application of Heat-Pipe based Heat Exchanger (HPHE) to the ceramic process and its effects on the energy efficiency and environmental impact. Implementing heat pipes in heat exchangers adds more advantages and features in comparison to the conventional heat exchangers. The most important feature of HPHE is that the hot stream is fully isolated from the cold stream. The heat is transported from the hot stream to the cold stream passively through the heat pipes. If a few heat pipes are failed, the HPHE can continue functioning without any disruption of the process and the failed heat pipes can be replaced through the scheduled maintenance plan. HPHE have greater heat transfer capability and less heat transfer area than conventional

systems which results in compact units for the same heat quantity and lower cost. In addition, heat pipe based heat exchanger have great life span which can be up-to 20 years and the payback period is usually less than two years. Heat pipe heat exchanger which comprise gravity assisted heat pipes transport the heat in upwards direction where the hot stream and the evaporator section is located at lower level than the condenser section of the heat pipe and the cold stream. The heat transfer through the HPHE starts at very low temperature difference between the hot stream and the cold stream. A schematic drawing of the HPHE is illustrated in Fig. 5 (a). Fig. 5 (b) represents the electrical analogy of HPHE modelling where each heat pipe is considered as an electrical resistance which transports energy between two points that have different voltages. This analogy is used in heat pipe heat exchanger modelling [19] and was used to design the heat pipe presented in this paper.

In order to calculate the potential energy recovered from the ceramic process by means of the HPs technology, a theoretical approach for the calculation of the thermal and fluid-dynamics characteristics of the heat exchanger is proposed. The approach is based on the established, proven performance characteristics of heat-pipe technologies and is used to determine the thermal power that can be recovered from the hot source and the relating pressure drop on both the evaporator and condenser sides. Furthermore, a lumped and distributed parameter numerical model of the entire ceramic kiln is constructed. The model is able to simulate the dynamic behaviour of the kiln under actual operating conditions by accounting for the different components that control the functioning of the kiln, such as the fans, the throttle valves, the burners and the PLC control strategy. Thus, the temperature distribution along the kiln's axis is predicted and the mass flow rate through the different kiln's modules is calculated. The results of the kiln numerical model are then combined to the theoretical model of the heat-pipe based heat exchanger and the heat recovery potential is evaluated. The calculations proved that it is possible to recover 863 MWh of thermal energy and use it for heating up the hot air stream of the pre-kiln dryer. Therefore, the annual fuel consumption saving from the burners powering the dryer amounts to 110,600 Sm³, which correspond to 164 tonnes per year of avoided carbon dioxide emissions and to more than 22,000 Euro per year of avoided fuel cost. Thus, the application of the heat pipes based heat recovery to the ceramic process proves to be beneficial from an energy efficiency and environmental impact point of view and in terms of the economic investment.

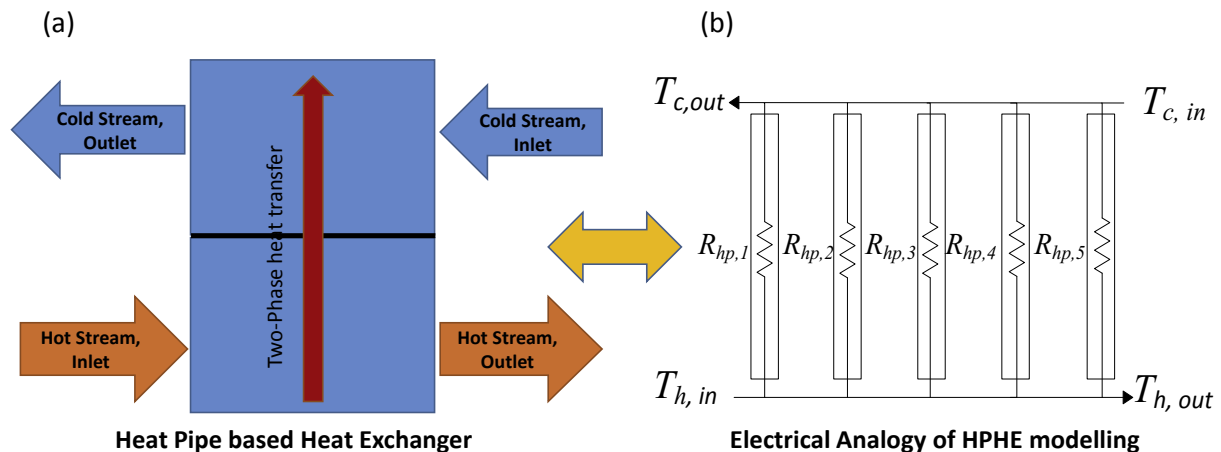


Fig. 5. (a) Schematic of HPHE concept, (b) Electrical analogy of HPHE.

2. Materials and methods

2.1. Theoretical analysis of the heat pipes based heat exchanger

The amount of thermal heat recovered by the HPHE can be determined by the following equation as reported in the literature [19]:

$$Q = UA\Delta T_{LM} \quad (1)$$

Where Q is the HPHE heat recovery (W), U is the overall heat transfer coefficient of the HPHE, A_t is the total heat transfer area of the HPHE, and ΔT_{LM} is the logarithmic mean temperature difference between the hot stream and the cold stream.

The thermal conductance (UA) of the HPHE can be calculated through the following equation:

$$UA = R_t \quad (2)$$

Where R_t is the total thermal resistance of the HPHE.

The total thermal resistance R_t is calculated through the thermal network analogy reported in Ref. [20] as follows:

$$R_t = \frac{R_{hp}}{n} \quad (3)$$

where R_{hp} is the total thermal resistance of one heat pipe and n is the total number of heat pipes in the HPHE.

The thermal performance of a heat pipe heat exchanger can be evaluated by its thermal effectiveness. The thermal effectiveness (ϵ) is the ratio of the actual heat transfer rate to the maximum thermal heat transfer rate of a heat exchanger. It is expressed through the following equation [21]:

$$\epsilon = \frac{Q_{actual}}{Q_{th}} = \frac{\dot{m}_{cold\ air} \cdot C_p \cdot (T_{cold\ air, out} - T_{cold\ air, in})}{C_{min} \cdot (T_{hot\ air, in} - T_{cold\ air, in})} \quad (4)$$

Where the C_{min} is the heat capacity rate (W/°C) which is calculated as follow:

$$C_{min} = \dot{m}_{cold\ air} \cdot C_p \quad (5)$$

Since the mass flow rate of the cold air is less than the mass flow rate of the hot air in this system.

The hydraulic drag caused by finned tubes in tube bundles can be calculated by the following equations in Ref. [22].

$$Eu = 67.6 Re^{-0.7} X_l^* - 0.55 X_t^* - 0.5 F^{0.5} \text{ for } 10^2 \leq Re \leq 10^3 \quad (6)$$

$$Eu = 3.2 Re^{-0.25} X_l^* - 0.55 X_t^* - 0.5 F^{0.5} \text{ for } 10^3 \leq Re \leq 10^5 \quad (7)$$

$$Eu = 0.18 X_l^* - 0.55 X_t^* - 0.5 F^{0.5} \text{ for } 10^5 \leq Re \leq 1.4 \times 10^6 \quad (8)$$

Equation (3) is valid for $1.5 \leq F \leq 16$, $1.13 \leq X_t^* \leq 2$ and $1.06 \leq X_l^* \leq 2$.

Equations (4) and (5) are valid for $1.9 \leq F \leq 16$, $1.6 \leq X_t^* \leq 4.13$ and $1.2 \leq X_l^* \leq 2.35$.

$$X_l^* = \frac{X_l}{d_o}, X_t^* = \frac{X_t}{d_o} \quad (9)$$

Where F is a factor expressing the ratio of extended surface area to the bare base-tube area of a finned tube bundle. F can be calculated from

$$F = \frac{1}{P} \left[\frac{2e_f}{l} \left(1 + \frac{e_f}{d_o} \right) + \frac{\delta_f}{l} \left(1 + \frac{2e_f}{d_o} \right) + 1 \right] \quad (10)$$

where

$$l = (P - \delta_f) \quad (11)$$

Reynolds number is calculated through the following equation:

$$Re = \frac{\rho u_{max} d_o}{\mu}$$

Where u_{max} is the maximum velocity which occurs in the minimum flow area between the tubes.

The pressure drop can be calculated through the following equation:

$$\Delta P = Eu \left(\frac{1}{2} \rho u_{max}^2 \right) N_{rows} c_n \quad (12)$$

where c_n is a correction factor and $c_n = 1$ for $N_{rows} > 6$.

The pressure drop in the expansion and the reducer can be calculated from the following equation

$$\Delta P = \zeta \cdot \frac{1}{2} \rho u^2 \quad (13)$$

where ζ is a loss coefficient related to the dimensions of the expansion and contraction parts. ζ value was obtained from CIBSE guide in Ref. [23].

The values of the pressure loss coefficient are presented in Table 1, and The theoretical results obtained from the correlations are presented in Table 2.

2.2. Numerical modelling of the ceramic kiln

The ceramic kiln simulated in this paper is a real production facility located in the ceramic district in Emilia Romagna – Italy. It is characterized by a production rate of approximately 5000 kg/h of tiles and it is designed for a continuous working load of about 8700 h per year. The entire ceramic kiln includes 56 modules with a length of 2.1 m each and it can be subdivided into the following 5 sections: pre-heating, firing, fast cooling, slow-cooling and outlet section.

The lumped and distributed parameters model of the entire kiln is constructed by means of the LMS Imagine. LAB. AMESim®

Table 1
Pressure loss factor values.

	Evaporator side	Condenser side
Expansion part	$\zeta = 0.755$	$\zeta = 0.48$
Contraction part	$\zeta = 0.08$	$\zeta = 0.043$

Table 2
Theoretical Results of pressure drop.

Pressure Drop Results	
Pressure drop in the core of Evaporator side (Pa)	194
Pressure drop in the condenser side (Pa)	63.7
Pressure Drop in the expansion and contraction (evaporator side) (Pa)	100
Pressure Drop in the expansions (condenser side) (Pa)	8
Total pressure drop (Evaporator Section) (Pa)	294
Total pressure drop (Condenser Section) (Pa)	71.7

software [24]. The numerical approach simulates the compressible flow within the kiln's modules accounting for the thermo-dynamic properties of the considered gas and the semi-perfect gases approach is employed. The gas efflux through an orifice is assumed as one-dimensional and subsonic, while the turbulent nature of the flow is accounted for by using proper coefficients. On any section of the considered system, the physical, mechanical and thermodynamic quantities, such as density, viscosity, pressure, velocity and temperature, can be described by their mean values in space and time on that section. In addition, the thermal library is used for the simulation of the solid parts of the kiln and their influence on the whole heat exchange is taken into account. This approach, the physical properties are more strongly depending on the temperature rather than on pressure, therefore it is more accurate when addressing low-medium pressure applications, in fact the importance of the temperature gradients on the thermodynamic characteristics is more remarkable than the instantaneous pressure variations. Thus, the mass flow of the fluid is calculated as a function of absolute pressure and temperature at the inlet (p_0, T_0), using the following equation:

$$\dot{m} = A_g * C_q * C_m * \frac{p_0}{\sqrt{T_0}} \quad (14)$$

where C_q and C_m are corrective parameters define the effective area utilized to compute the fluid velocity and the mass flow coefficient for a compressible fluid [24].

In the following, the numerical models that simulate each module and component adopted in the real kiln are described in details. The modules that encompass the entire kiln are modelled assembling the sub-models of the different components that are included in the real device, such as the walls, the insulating materials, the rollers, the burners, the valve and the tiles. Particular attention is devoted to modelling the heat exchange between the kiln walls, the hot air flow, the tiles and the rollers. Conduction, convection and radiative heat transfers are evaluated for every different surface and an appropriated calculation method is adopted to compute the value of each thermal exchange between the elements of the kiln quantities and it depends on the temperature difference that is determined between the considered bodies that exchange heat. The thermal energy exchanged by irradiation between the kiln surfaces is a non-negligible component, given the high temperatures of the surfaces, and is defined by the following equation:

$$dh_{rad} = \frac{\sigma * (T_j^4 - T_i^4)}{\frac{1-\epsilon_i}{\epsilon_i A_i} + \frac{1}{A_i F_{ij}} + \frac{1-\epsilon_j}{\epsilon_j A_j}} \quad (15)$$

in which σ is the Stefan-Boltzmann constant, A_i is the emitting area of body i , A_j is the emitting area of body j and F_{ij} is the shape factor with which A_{ij} "sees" A_{ji} .

For each component included in the kiln chambers model, the emitting area and the relative shape factor are have been carefully defined as a function of the real geometries and the respective portions of surface that influence the irradiation, taking into account the relationship:

$$A_i * F_{ij} = A_j * F_{ji} \quad (16)$$

Finally, the leakages between the internal volumes and the external environment have been included in the numerical model. The main contribution of the leakages is given by the space between the rollers and the kiln wall: in the numerical model, this leakage has been defined as a pneumatic connection through which

cold air can enter the kiln chamber leading to a reduction of the overall efficiency.

In order to calculate the temperature difference between the top and bottom surfaces of the tiles and provide an estimate of the temperature gradient inside the ceramic material, the numerical model of the tiles accounts for three layers having the same thickness. The first layer represents the surface of the tile in contact with the rollers and exchanges heat with the rollers by conductive and radiative phenomena. The second one is the core of the tile and exchanges heat only with the adjacent levels by conduction. Finally, the top level simulates the thermal exchanges occurring between the tile's surface and the hot air flow through the module and the burners' flame and the high temperature kiln's walls. Furthermore, the effects of the tiles' motion on the heat transfer is also considered in the modelling. Since it is not possible to define a continuous flow for a solid material, an ad-hoc model is created modelling the tiles as a fluid having the same thermal properties of the real tiles and flowing along the kiln with the real tiles' velocity. Therefore, the effects of the temperature gradient of the tiles in each modules can be correctly predicted.

The standard natural gas-fueled burners are simulated as a thermal power input to the upper and lower chambers of the modules according to the instantaneous fuel consumption determined by the kiln control system as a function of the temperature set point for each module. The same PID controller employed in the real kiln to regulate the temperature in the chambers and thus the natural gas flow rate for the burners is also adopted in the kiln numerical model. Therefore, the total consumption of natural gas is one of the main results of the numerical approach and it is determined as a function of the thermal balance of the kiln chamber to maintain the desired temperature in the selected modules as in the real kiln. The models for different burner's types are also taken into account, such as the recuperative burner type, i.e. the EKO burner.

Finally, the flow rate within the kiln is calculated as a function of the operating point of the different fans employed in the real kiln. The working points of the fans are determined according to the operating maps provided by the manufacturers. As in the real kiln, the fans are regulated by varying the rotational speed and the mass flow rate is determined as a function of the resulting pressure drop. The flows are then controlled by including in the numerical models the valves employed in the real system. Their effective areas is determined according to the valve opening position and it can be varied to simulate different operating conditions. More extensive description of the approach employed in the analysis can be found in Ref. [12].

Fig. 6 depicts the numerical layout of the analysed kiln. In particular it is referring to the firing section and the cooling one. The two sections are separated by a fireproof wall that partitions

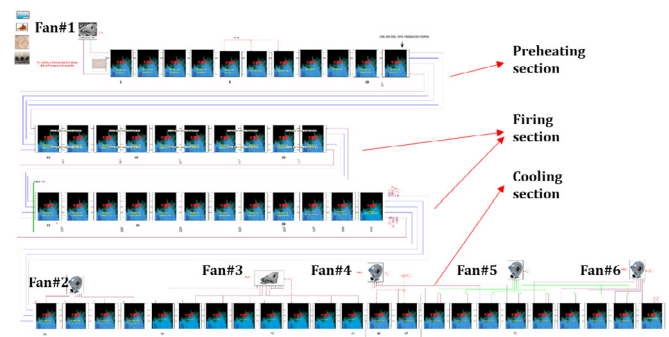


Fig. 6. Layout of the numerical model of the reference ceramic kiln.

the flow between the sections. The gas flow in the kiln is due to the combustion products from the gas burners in the firing sections, while in the cooling modules fresh air is injected in order to cool down the tiles following a prescribed temperature profile. Therefore, in the firing modules the gases flow in the opposite direction with respect to the tile motion, while gases and tiles travel in the same direction.

Fig. 6 outlines the 56 modules as adopted and connected in the real kiln. Each module includes several sub components to model the different parts and physical phenomena that take place in each of them, such as the fans, the burners, the valves. Therefore, the thermo-pneumatic and thermal libraries and the signal library are also adopted in addition to the pneumatic library in order to model accurately the heat transfer, and the kiln control strategy. Thus, the fan, the throttle valves and the burners are regulated using the real operating parameters. This library is also employed to set the necessary boundary conditions, such as the external temperature and pressure in the ducts' model, the equivalence ratio for the burners, the initial temperature of the walls and the kiln chambers. Insulating materials are also accounted for in the analysis as well as the leakages between the internal volume of the kiln and the ambient. Since the pressure within the kiln is always lower than the atmospheric value, the leakages contribute to the gas flow by adding a cold air stream entering the kiln and decreasing its efficiency.

In order to evaluate the kiln fuel consumption particular care has been devoted to the modelling of the burners. Each burner is operated as in the real kiln in order to control the temperature in the module following the temperature set point according to desired operations. Therefore, the natural gas consumption is predicted by the simulation as a function of the thermal balance of the chamber as in the real kiln.

The prediction capabilities of the kiln numerical model are validated by comparing the calculated results against the experimental measurements carried out on the real ceramic facility during regular production operations. The tiles produced during the tests had dimensions of 1.2×1.2 m and height equal to 9 mm and the kiln is run with a production rate of approximately 5000 kg/h. The gas temperature in each module of the firing section is measured with a PT 100 thermocouple, while in the cooling section the gas temperature is usually not measured. In addition, measurements using DATAPAQ[®] instrument were carried out in order to have a redundant measure of the air in the upper chamber of the kiln's modules and an estimate of the temperature of the tiles' surface. The instrument is a thermally insulated box equipped with a data logger connected to 8 thermocouples and it travels along the kiln on a tile's top and measures the temperature of the upstream tiles and air flow. The temperature values are measured with an accuracy of ± 0.3 °C.

Figs. 7 and 8 show the temperature profile of the gases within the kiln upper and lower chambers and the comparison between the measured and calculated values of the temperature, the set point temperature controlled by the process is also plotted. The data are divided by a reference temperature value in order to have non-dimensional quantity. The agreement between the numerical results and the experimental measurements is satisfactory and the trend of the temperature distribution of the hot air flow along the kiln length is correctly predicted. The proposed numerical approach proved also to predict accurately the tiles' temperature along the kiln axis, see Fig. 9. Fig. 10 shows the percentage error of the predicted mass flow through the kiln's fans with respect to the measured values. The errors are within an acceptable range for the lumped and distributed parameter modelling and it demonstrates that the flows in the different sections of the kiln are predicted accurately. This result combined with the temperature distribution

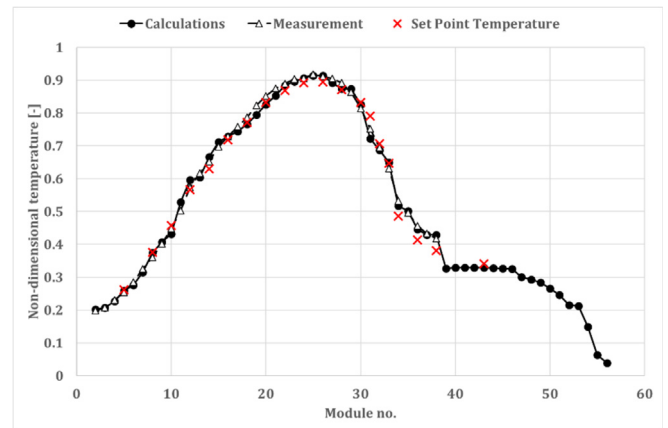


Fig. 7. Non-dimensional temperature profile of the gases along the kiln axis: comparison between the measured and calculated values of the temperature in the kiln upper chambers.

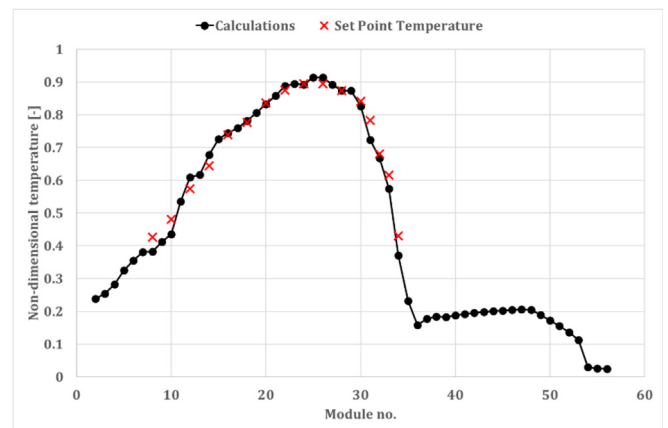


Fig. 8. Non-dimensional temperature profile of the gases along the kiln axis: comparison between the measured and calculated values of the temperature in the kiln lower chambers.

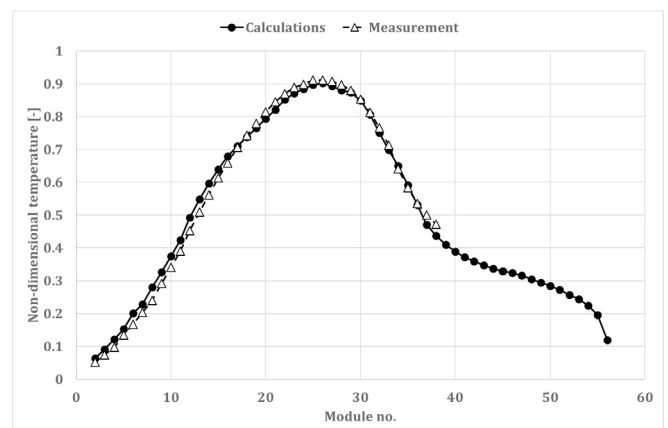


Fig. 9. Non-dimensional temperature profile of the tiles along the kiln axis: comparison between the measured and calculated values.

of the tiles and the air in the upper and lower chambers proves that the calculated thermal behaviour of the kiln agrees closely with the experimental measurement. Therefore, the calculation can be used to extend the analysis of the kiln where

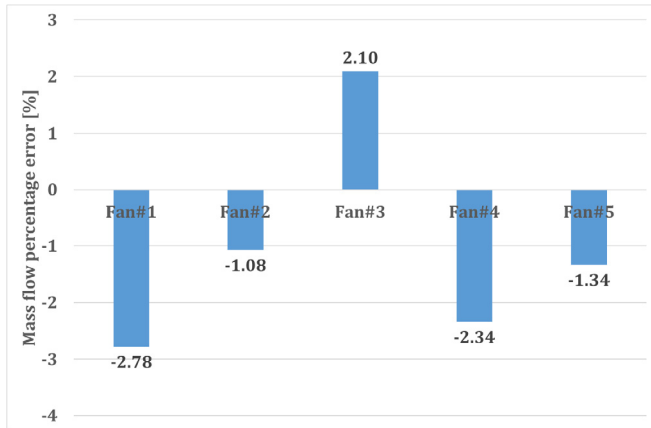


Fig. 10. error of the calculated mass flow rate through the kiln's fans with respect to the measured values.

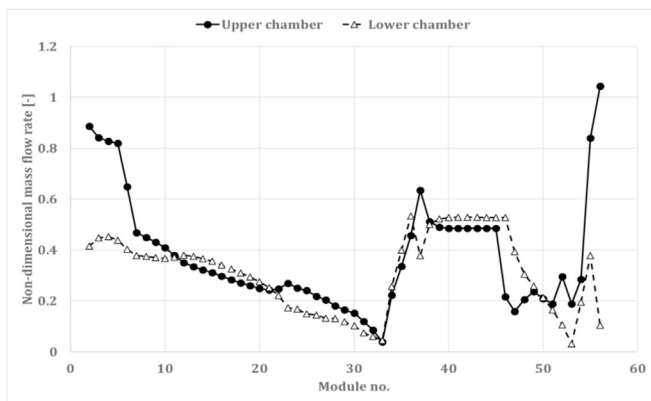


Fig. 11. Non-dimensional air mass flow rate profile along the kiln axis upper and lower chambers.

measurements are not available. In particular, from Fig. 7, it is possible to notice that the measured values stop at module no. 38, in fact during the experiments the equipment failed to measure the remaining modules. Therefore, in the cooling section the temperature values are available only for the first 5 modules. Since the thermal behaviour of the kiln was correctly predicted, it is possible to determine the tiles' temperature through the entire cooling sections also for the modules where no experimental data are available.

A good agreement between the experimental measurements and the calculations proved was also found in terms of natural gas consumption. The predicted fuel rate necessary to power the kiln during regime operation was calculated within a 5% error with respect to the measure kiln fuel consumption.

Once the numerical model is validated against experimental measurements, simulation can be employed to widen the analysis of the kiln operation by investigating quantities that are difficult to be measured. In particular, Fig. 11 shows the non-dimensional air mass flow rate for all the kiln's modules within both the upper and the lower chambers. The calculation of the flow rate in the different sections of the kiln is then used in the design of the heat pipe based heat exchanger. In fact, the flow rate for every fan employed in the system and therefore the flow through each stack is determined and in particular the cooling gas stack at the end of the fast cooling section is investigated and studied as a possible location for the heat recovery by means of the heat pipe system.

3. Results and discussion

3.1. Thermal design of the heat pipes based heat exchanger

A Heat Pipe Heat Exchanger (HPHE) was designed to recover the heat from an exhaust of hot air to heat a fresh supply air. The waste hot air stream accounted for in this analysis is flowing through the cooling stack operated by the Fan#4. The numerical model described in the previous section was used to estimate the flow characteristics through the stack, such the mass flow rate and the temperature of the gases. In particular, the gases temperature was calculated in the section of the plant where the heat pipes based heat exchanger is supposed to be located. Thus, the thermal losses from the kiln module to the exchanger position were taken into account.

The recovered heat is used in the pre-kiln dryer, which operates with a large hot air flow rate at 100 °C approximately. The hot air flow to the dryer is heated up by natural gas burners, thus the recovered heat from cooling stack of the kiln by the heat pipes based heat exchanger enables to decrease the amount of fuel consumed. Since the dryer operates with air, the same fluid was selected as the cold operating fluid for the heat pipes; therefore, it can be directly mixed with the flow from the burners to the dryer. Table 3 presents the thermal parameters of the heat pipes based heat exchanger calculated following the theoretical approach described in the previous section and on the basis of the flow characteristics estimated by means of the numerical model of the entire kiln.

The geometry of the HPHE are presented in Table 4.

The HPHE consists of 200 heat pipes finned with helical fins to increase the heat transfer area. Fig. 12 illustrates three-dimensional design of the HPHE. The 2D design and the dimensions of the heat exchanger are illustrated in Fig. 13. The arrangement of the pipes in the HPHE is presented in Fig. 14 (a) and the fins dimensions are

Table 3
Thermal design parameters.

Thermal design parameters	
Hot air mass flow rate	6000 kg/h
Cold air mass flow rate	2640 kg/h
Hot air average specific heat capacity	0.241 kcal/kg·°C
Cold air average specific heat capacity	0.241 kcal/kg·°C
Hot air inlet temperature	204 °C
Hot air outlet temperature	145 °C
Cold air inlet temperature	30 °C
Cold air outlet temperature	164 °C
Recovered heat	99.299 kW
UA	1438.2 W/°C
Thermal effectiveness of the HPHE	77.93%

Table 4
Heat pipe Heat exchanger mechanical geometry.

Heat exchanger Geometry	
Total Number of pipes	200
Number of tubes per row	10
Number of Rows	20
Heat pipe Geometry	
Heat pipe total length	1.518 m
Evaporator side length	0.962 m
Condenser side length	0.531 m
External pipe diameter	0.028 m
Fins geometry	
Fins type	Helical
Fins thickness	1 mm
Fins height	10 mm
Fins pitch	4 mm

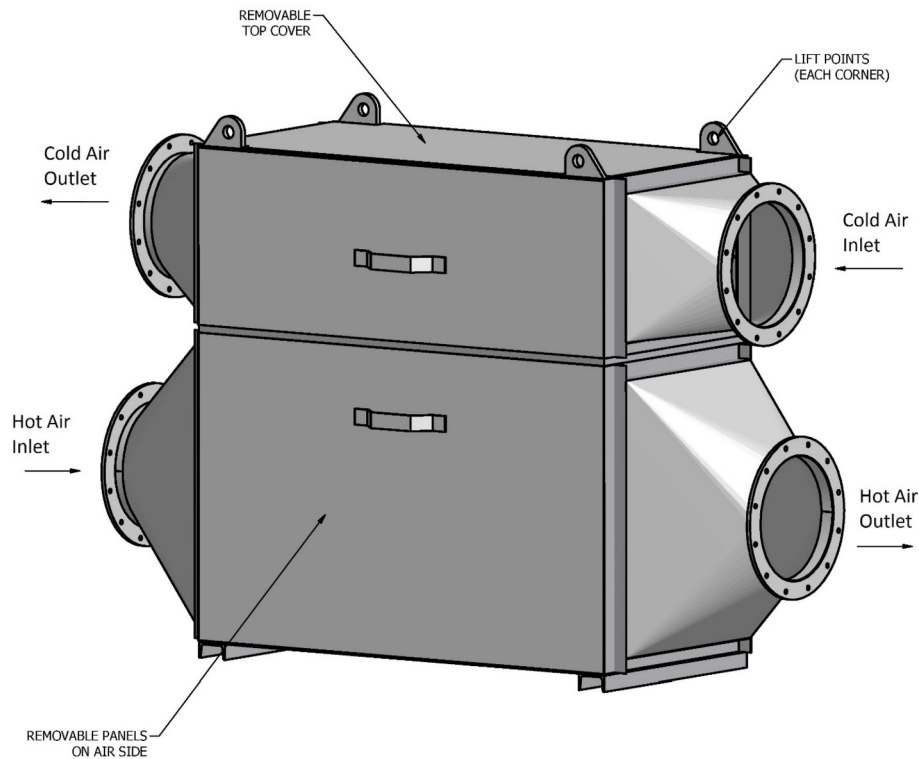


Fig. 12. 3D Design of the Heat Pipe Heat exchanger.

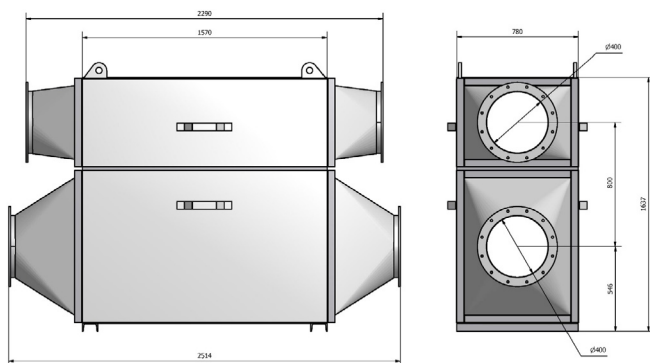


Fig. 13. 2D design and dimensions of the Heat Pipe Heat exchanger.

illustrated in Fig. 14 (b).

3.2. Effects of the heat pipes based heat exchanger on the kiln operation

The effects of the heat pipes based heat exchanger on the kiln are analysed under actual operating conditions by means of the numerical model described in the previous section. The numerical model of the HPs exchanger is developed and added to the layout of the entire kiln, see Fig. 15. The model of the heat exchanger is based on the thermal characteristics calculated by means of the theoretical approach and described in Table 3. In particular, the model reproduces the same global heat transfer coefficient in order to predict the instantaneous thermal power that can be recovered.

The proposed numerical approach for investigating the effects of the heat pipes based heat exchanger on the real kiln operations can be used also to address the possibilities of recovering waste heat in other industrial applications, such as furnaces and boilers.

Table 5 lists the results that can be achieved by employing the heat pipes based heat exchanger to the ceramic process. The numerical model calculated an annual heat recovery of 863 MWh with an additional electric energy consumption of approximately 47 MWh/y. In fact, the increased working load of Fan #4 due to the higher pressure drop requires a larger power input of 5.41 kW. Nevertheless, the thermal energy recovered from the cooling stack and employed to heat a substantial amount of the hot air flow for the pre-kiln dryer avoids the consumption of 110,600 Sm³ per year of natural gas, which corresponds to more than 22,000 Euro saved per year. Additionally, the natural gas saving leads to avoided carbon dioxide emissions equal to 164 tonnes/y. These figures are remarkable and support the application of the heat pipes based heat exchanger in the cooling stack of a ceramic kiln; furthermore, the results are even more promising when considering that in a medium size ceramic plant four kilns of similar size as the studied one are usually employed.

4. Conclusion

This paper investigates the application of heat-pipe based heat exchanger for improving the energy efficiency and the environmental impact of industrial processes. The case of the ceramic industry is studied and the heat recovery from the cooling stack of the kiln using the heat pipes technology is analysed.

First, a theoretical approach for the calculation of the thermal and fluid-dynamics characteristics of the HPs heat exchanger has been proposed based on the established, proven performance characteristics of heat-pipe technologies. In particular, the global heat transfer coefficient has been estimated and the pressure drop in the evaporator and condenser streams has been calculated. Afterward, the lumped and distributed parameter numerical model of the entire kiln has been constructed and validated against experimental measurements. The agreement between the calculations

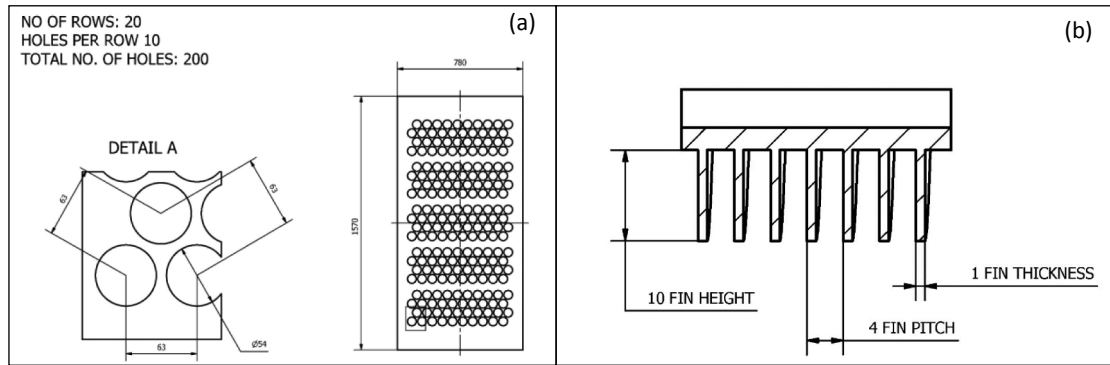


Fig. 14. (a) Heat pipes arrangement, (b) fins dimensions.

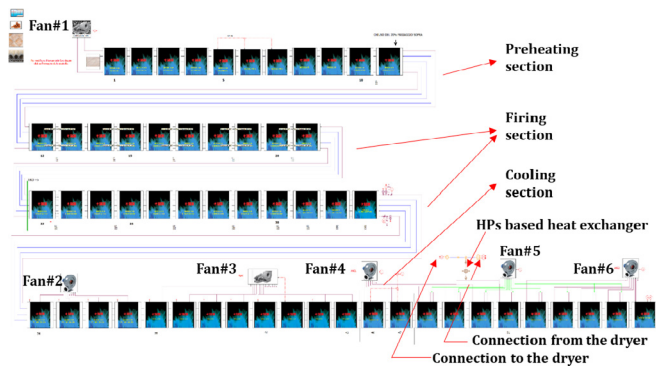


Fig. 15. Layout of the numerical model of the cooling section of the reference ceramic kiln equipped with the heat pipes based heat exchanger in the cooling stack of Fan#4.

Table 5
Effects of the Heat pipe Heat exchanger on the ceramic processes efficiency and cost.

Recovered energy from HPs	863,901	kWh
Additional fan power requirement	5.41	kW
Fuel savings	110,599	Sm ³ /y
Avoided fuel cost	22,120	Euro/y
Avoided CO ₂	164	tonnes/y

and the measurements resulted satisfactory and the kiln operation was simulated under actual working conditions.

Therefore, the proposed numerical approach can be used as a design tool for evaluating the possibilities of exploiting waste heat sources not only in industrial ceramic kilns but also in other industrial applications involving furnaces and boilers.

In fact, by means of the numerical simulation of the kiln, it was possible to estimate the flow rates through the different modules and specifically the hot gas flow through the cooling stack and its temperature were calculated. Thus, the heat pipes based heat exchanger has been designed according to proposed approach and dimensioned on the basis of the simulated fluid-dynamics characteristics of the gas flow.

Finally, the potential energy recovery by means of the heat pipes technology was calculated for the cooling stack and the effects on the ceramic processes efficiency and cost were evaluated. The calculations demonstrated that for one kiln under regular operation it was possible to recover more than 863 MWh in terms of thermal energy and use it for heating up the hot air stream of the pre-kiln dryer. Therefore, approximately 110,600 Sm³ per year of natural gas can be saved from the burners powering the dryer and the emission of 164 tonnes per year of carbon dioxide can be avoided.

In addition, the cost savings due to the fuel consumption reduction amount to more than 22,000 Euro per year, indeed, the calculated payback of the system was about two years. The application of the heat pipes technology for enhancing the energy efficiency and the environmental impact of the ceramic process proved to be advantageous from the viewpoint of the energy consumption and also when assessing the economic investment of the solution.

Acknowledgements

The present work is part of Design for Resource and Energy efficiency in CerAMiC Kilns (DREAM) project. The project has received funding from the European Union's Horizon 2020 research and innovation programme under grant agreement No 723641.

Nomenclature

Symbol	
A	Total heat transfer area of the heat pipe heat exchanger (m ²)
A_g	Geometrical cross section area (m ²)
A_i	Emitting area of body i (m ²)
A_j	Emitting area of body j (m ²)
C_q	Flow coefficient (–)
C_m	Flow coefficient (–)
C_{min}	Heat capacity rate (W/°C)
C_p	Specific heat capacity (W/kg·°C)
c_n	Constant
d_o	External tube diameter (m)
dh_{rad}	Radiative heat transfer per unit time (W)
Eu	Euler number (non-dimensional)
e_f	Fin height (m)
F	Ratio of extended surface area to the bare base-tube area of a finned tube bundle
F_{ij}	Shape factor with which A_{ij} “sees” A_{ji}
l	Minimum width between two fins (m)
\dot{m}	Mass flow rate (kg/s)
N_{rows}	Number of rows
p	Helical fins pitch (m)
P	Pressure (Pa)
p_0	Inlet pressure (Pa)
Q	Heat transfer rate (W)
R_{hp}	Thermal resistance of single heat pipe (°C/W)
R_t	Total thermal resistance of the heat pipe heat exchanger (°C/W)
Re	Reynolds number (non-dimensional)

T	Temperature (°C)
T_0	Inlet temperature (°C)
T_i^4	Temperature of the i-th surface (°C)
T_j	Temperature of the j-th surface (°C)
ΔT_{lm}	Logarithmic mean temperature difference (°C)
U	Overall heat transfer coefficient ($W/m^2 \cdot ^\circ C$)
u	Velocity (m/s^2)
X_t	Transverse pitch of the heat pipes array (m)
X_l	Longitudinal pitch of the heat pipes array (m)

Greek

ϵ_i	Emissivity of the i-th surface (–)
ϵ_j	Emissivity of the j-th surface (–)
ζ	Pressure loss factor
σ	Stefan-Boltzmann constant ($kg\ s^{-3}\ K^{-4}$)
ρ	Density (kg/m^3)
μ	Dynamic viscosity (Pa.s)

Subscripts

th	Theoretical
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