New Design Solutions for Thermal Insulation Systems for High-Temperature Furnaces

B. Valentini*, A. Plankensteiner*, C. Grohs*

* PLANSEE SE, 6600 Reutte, Austria

Abstract

The reduction of energy consumption of high-temperature furnaces such as for refractory metals sintering and Sapphire single-crystal growth becomes an important engineering issue and primarily addresses the minimization of heat transfer from the hot core of the furnace to the much cooler environment or the actively cooled furnace containment. Usually, thermal insulation systems for high-temperature furnaces either consist of a refractory lining or a series of radiative insulation sheets. The first thermal insulation system is characterized by heat transfer due to thermal conductance, described by Fourier’s law, and the second one by radiative heat transfer, based on the Stefan-Boltzmann law, with convective and advective heat transfer being neglected in both cases. Since both laws have a different dependence on the absolute temperature numerical analyses as presented here show that combining both thermal insulation systems in a unique manner, depending on the used bulk and particulate materials, leads to an optimized and highly energy-efficient novel insulation design. In case of a cylindrical insulation system of hypothetically infinite length, i.e., no heat transfer in other directions than the radial one, the thermal power loss due to heat transfer through the novel insulation system is predicted to be at least 25% less compared to a standard radiation shield system.

Keywords

High-temperature furnace, thermal insulation, refractory metal, refractory ceramic, radiation shield, refractory lining, finite element analysis

Introduction

Refractory metal components are used in a wide range of industries and applications for high-temperature furnaces and thermal-processing plants. For instance, molybdenum and tungsten play a key-role in the design of heating elements and thermal insulation systems for, e.g., high-temperature sintering furnaces and single crystal growth furnaces, respectively, which is primarily due to their high melting point, low vapor pressure, and highest material purity together with outstanding mechanical properties at highest temperatures. This way, furnace operators are allowed to run their furnace systems at higher temperatures for longer times under high-vacuum conditions or under protective atmospheres compared to furnaces driven by material alternatives such as super alloys, graphite or refractory ceramics. Nevertheless, continuously growing demands in environmental compatibility, total cost of ownership, and flexible manufacturing strategies enforce improved furnace designs showing minimized energy consumption, maximized life-time of components and easy-to-handle periodic maintenance procedures, and a kit-like design, respectively. Generally, the so-called hot zone of the high-temperature furnace comprises heating elements and thermal insulation systems with both of them being under increased crucial evaluation in the respective industries with regard to the implication of potential changes in their geometrical and material design onto the overall furnace behavior. The present contribution addresses the design problem of thermal insulation systems for high-temperature furnaces in particular whereas the design problem of heating systems is addressed elsewhere [1].

Principals about Thermal Insulation Systems

For demonstration purposes suppose that for a high-temperature furnace a cylindrical process chamber of finite axial length is heated by a heating system which covers the cylindrical part of the perimeter of the process chamber and itself is confined by a cylindrical thermal insulation system the latter comprising a cylindrical as well as a planar top and bottom insulation system, see Fig. 1. Assume that the heating system is enclosing the component or material to be processed and is designed for being operated at a required temperature at which heat is transferred to the material or component to be processed via thermal radiation and, in presence of processing gases, by convection. Since the heating system transfers thermal energy not only into the inward direction of the process chamber but also to the outward one thermal insulation systems are necessary being capable of retaining as much as possible the thermal energy within the processing chamber. In high-temperature furnace construction two major design concepts are successfully proofed over decades and are used complementary in a large variety of industries: i) radiation shield systems based on refractory metals, and ii) refractory linings based e.g. on graphite or refractory ceramics, both of which being described in the following. It is worth noting that the thermal behavior of these systems is under present consideration whereas other characteristics such as mechanical integrity in a long-term view, suitability to oxidizing and non-oxidizing atmospheres, etc. will only be discussed where applicable for gaining a general understanding.

Design Principals of Thermal Insulation Systems

Radiation shield systems consist of a number of typically eight to twelve thin sheets of considerably less than $1\ [mm]$ in thickness - made of molybdenum and/or tungsten - and are arranged in a serial manner this way transferring heat from the hot process chamber to the cold (usually actively cooled) containment of the furnace (cf. Fig. 1) by heat conduction within the metallic sheets as well as thermal radiation in between facing surfaces of those sheets. In the presence of processing gases additional heat is transferred by conduction in the gas as well as convection. For furnaces with heating elements driven at temperatures of $2000\ [°C]$ or above a typical radiation shield system is designed for temperatures at the outermost sheet to be realized at $1000\ [°C]$ or below. By assuming a classical design of a radiation shield system the resulting insulation effect is evaluated to be sufficiently good and goes along with a number of advantages such as low thermal mass, i.e. fast behavior in the thermally transient regime, and a light-weight design however at the disadvantage of life-time limitations with high-temperature creep being induced by gravitation and magnetic force actions (the latter applies to heating systems driven by alternating current only). As a consequence these types of insulation systems represent the first choice for applications where the furnace has to be operated at temperatures considerably above $2000\ [°C]$

under protective atmosphere or vacuum in a temporally cyclic manner, i.e., designated for batch processing.

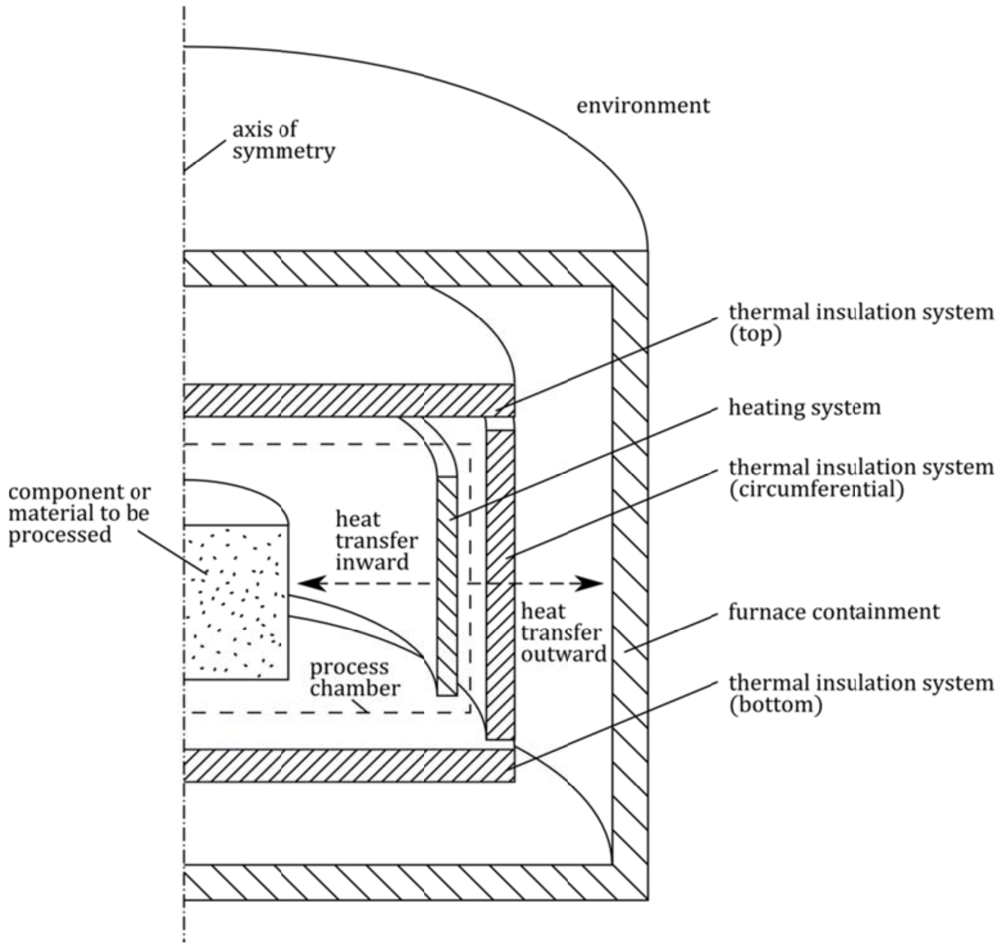

Figure 1: Schematical design of a high-temperature furnace of cylindrical shape.

At the other hand, refractory linings typically comprise a brick construction which may partially be supplemented with fillings of particulate materials. Compared to metallic radiation shield systems as described above the heat transfer is primarily governed by a single heat transfer mechanism, i.e., the conduction of the refractory ceramic. Alumina (Al_2O_3) is the most commonly used material with maximum operational temperatures up to about $1650\ [°C]$. For temperatures up to about $2000\ [°C]$, e.g., stabilized Zirconium oxide (ZrO_2) is used. It is worth noting that these materials typically show some macro- or micro-scale closed- or open-cell-porosity and, this way, the apparent thermal conductivity of those refractory ceramics always is composed by the thermal conductivity of the fully dense refractory ceramic and – in the presence of a gaseous atmosphere – the one of the gas entrapped within the pores and/or cavities of the ceramic material. If fillings of particulate materials are used then heat transfer mechanisms in between the individual particles are such as in between individual thermal radiation shields being described above and, therefore, additionally contribute to the total heat transfer. In comparison to radiation shields, refractory linings are designed for larger temperature gradients within the insulation zone showing a considerably better insulation effect, however, along with much larger wall thicknesses and thermally inertial behavior. This way, refractory linings are typically used for furnaces which are operated in a temporally continuous way under oxidizing or carburizing atmospheres with no active cooling of the furnace containment.

Mathematical Description of Heat Transfer in Thermal Insulation Systems

In furnaces under vacuum conditions or an atmosphere with low gas pressure (at or close to ambient atmospheric pressure) heat transfer in the thermal insulation system is primarily composed of thermal conductance and thermal radiation. For this reason the mathematical description of convective and advective heat transfer is neglected in the following. Mathematically, heat transfer by thermal conductance is described by Fourier's law which is written for the general, three-dimensional case, as

$$q_{k,cond}(T) = -\lambda_k \frac{dT}{dx_k}, \quad k = 1,2,3. \tag{1}$$

In (1) $q_{k,cond}$ $[W/m^2]$ denotes the heat flux, λ_k $[W/(m \cdot K)]$ the thermal conductivity and x_k $[m]$ the coordinate in the k-th direction, respectively. The temperature in the Kelvin scale T $[K]$ (absolute temperature) is related to the temperature in the Celsius scale θ $[°C]$ by $T = \theta - \theta_0$ with the temperature at absolute zero defined as $\theta_0 = -273.15$ $[°C]$. For computing the heat flux through an infinite, thin and homogeneous layer $q_{1\to2,cond}$ $[W/m^2]$ of thickness d $[m]$ Fourier's law (1) can be approximated by the scalar equation

$$q_{1\to2,cond}(T_1, T_2) \cong \lambda_m(T_1, T_2) \frac{T_1 - T_2}{d} \tag{2}$$

where T_1 and T_2 are the temperatures at the faces of the layer and the mean thermal conductivity of the layer λ_m $[W/(m \cdot K)]$ is given as

$$\lambda_m(T_1, T_2) = \lambda\left(\frac{T_1 - T_2}{2}\right). \tag{3}$$

For increasing thickness of the layer (2) becomes more and more inaccurate because of the assumed constant and exclusively on the mean temperature dependent, thermal conductivity of the layer in (3). Hence, using (2) instead of (1) is only feasible if either the gradient of the temperature within the layer is not too pronounced or the thermal conductivity varies only slightly with temperature.

The total heat flux by thermal radiation between the infinite, parallel faces of a layer, filled with no gas or a non-absorbing, non-emitting and non-radiating gas, is derived from the Stefan-Boltzmann law and Kirchhoff's law. The Stefan-Boltzmann law

$$e_b(T) = \sigma T^4 \tag{4}$$

defines the total flux of energy radiation from a black body e_b $[W/m^2]$ which depends only on the Stefan-Boltzmann constant $\sigma = 5.6704 \cdot 10^{-8}$ $[W/(m^2 \cdot K^4)]$ and the absolute temperature to the fourth power. The radiative behavior of the components of a furnace is not identical to those of a black body, i.e., the total flux of energy emitted from real construction parts is usually significantly lower than those emitted from a black body: $e_{gb}(T) \leq e_b(T)$. However, their radiative thermal behavior can be described in a sufficiently accurate way by the behavior of a so-called diffuse, gray body. A body is called diffuse if its emittance (absorbance) does not dependent on the angle at which the radiation is emitted (absorbed). In addition it is called gray if its emittance (absorbance) is independent of the wavelength of the emitted (absorbed) thermal radiation. Therefore, with the aid of Kirchhoff's law for a diffuse, gray body

$$\varepsilon(T) = \alpha(T) \tag{5}$$

defining the equality of the emittance $0 < \varepsilon \leq 1$ $[-]$ and absorbance α $[-]$, the heat flux emitted from a diffuse, gray body e_{gb} $[W/m^2]$ can be written as

$$e_{gb}(T) = \varepsilon(T)e_b(T) = \varepsilon(T)\sigma T^4. \tag{6}$$

Considering the geometrical configuration of the two diffuse, gray, infinite, parallel faces of the layer the heat flux $q_{1\to 2,rad}$ $[W/m^2]$ is finally derived as (see e.g. [2])

$$q_{1\to 2,rad}(T_1, T_2) = \sigma \frac{(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \tag{7}$$

where the emittances of the two faces are assumed to be independent of the temperatures at the faces, i.e., $\varepsilon_1 = const.$ and $\varepsilon_2 = const.$.

Hybrid Thermal Insulation Systems

In conventional thermal insulation systems heat transfer in a refractory lining takes place primarily due to thermal conductance while heat transfer in a radiation shield system is dominated by thermal radiation. The main difference between these two kind of heat transfer mechanisms is their dependence on the absolute temperature, i.e., according to Fourier’s law (1) thermal conductance depends on the difference of the absolute temperatures to the first power and based on the Stefan-Boltzmann law (4) thermal radiation depends on the difference of both absolute temperatures to the fourth power. For typical values of the thermal conductivity of high-temperature insulation materials of $\lambda \approx [0.1, 2.0]$ $[W/(m \cdot K)]$ and thickness of the refractory lining of $d \approx [40, 100]$ $[mm]$ the coefficient λ_m/d in (2) is in the range of about 1 to 50 $[W/(m^2 \cdot K)]$. In contrast, for typical values of emissivities of $\varepsilon_1, \varepsilon_2 \approx [0.2, 0.5]$ the coefficient $\sigma/(1/\varepsilon_1 + 1/\varepsilon_2 - 1)$ in (7) is approximately $6 \cdot 10^{-9}$ to $19 \cdot 10^{-9}$ $[W/(m^2 \cdot K)]$ resulting in a difference of up to eleven orders of magnitude between the coefficients in (2) and (7). Hence, if heat transfer occurs at low absolute temperatures heat flux by thermal radiation is (much) smaller compared to that by thermal conductance. At high temperatures the exact opposite becomes true, in particular if the difference between the absolute temperatures is sufficiently pronounced. Consequently, by combining a refractory lining and a radiation shield system within one thermal insulation system, denoted in the following as “hybrid thermal insulation system”, it is possible to design a novel, even more energy-efficient thermal insulation system. In general, the hybrid thermal insulation system can be composed of a few refractory linings and a few radiative insulation sheets but usually it is feasible to use only one refractory lining and one system of radiative insulation sheets. Thus, there are two possibilities for arranging these two parts but the logical and more promising variant with respect to the minimization of the heat flux is placing the refractory lining part in the zone with the highest temperatures, i.e., in the innermost region of the hybrid thermal insulation system, and the radiation shield system part in the zone with the lowest temperature. A comparison of conventional insulation systems with hybrid insulation systems will be presented in the next section and can also be found, e.g., in [3] and [4].

Comparison of Conventional Insulation Systems with Hybrid Insulation Systems by Numerical Analyses

For comparing the thermal behavior of conventional insulation systems with hybrid insulation systems with regard to their insulation efficiency and temperature distribution within the insulation system numerical simulations are applied. In the presented numerical simulations heat transfer is considered only by thermal conduction and thermal radiation. Thus, convective and advective contributions to the heat transfer are assumed to be sufficiently small this way not influencing the computed results.

Numerical Model

Each of the numerical simulations is based on a one-dimensional numerical model representing only the spatial extension orthogonally through the insulation systems consisting of n infinite, parallel layers of refractory linings and/or radiation shields. The computations are carried out by applying (2), (3) and (7) which are assumed to be sufficiently accurate for describing the thermal behavior of the insulation systems. Therefore, the heat flux across the i-th layer is defined according to (2) and (7), respectively, as

$$q_i = \begin{cases} \lambda_{i,m}\left(\frac{T_i - T_{i+1}}{2}\right) & \textit{thermal conductance,} \\ \sigma \frac{(T_i^4 - T_{i+1}^4)}{\frac{1}{\varepsilon_{i,1}} + \frac{1}{\varepsilon_{i,2}} - 1} & \textit{thermal radiation} \end{cases} \tag{8}$$

where the mean thermal conductance $\lambda_{i,m}$ is derived from (3) as

$$\lambda_{i,m} = \lambda_i\left(\frac{T_i - T_{i+1}}{2}\right). \tag{9}$$

On condition that the heat fluxes across the individual layers have to be equal, i.e.,

$$q_1 = q_2 = \ldots = q_i = \ldots = q_{n-1} = q_n \tag{10}$$

the nonlinear algebraic system of equations

$$R_i = q_i - q_{i-1}, \qquad i = 2, \ldots, n \tag{11}$$

is solved iteratively by means of Newton's method (see, e.g., [5])

$$J_{ij}^{(k)} \Delta T_j^{(k)} = -R_i^{(k)} \tag{12a}$$

$$T_j^{(k+1)} = T_j^{(k)} + \Delta T_j^{(k)}, \qquad k = 0, 1, \ldots \tag{12b}$$

for the unknown absolute temperatures T_j, $j = 2, \ldots, n$. In (11) R_i denotes the vector of residua which becomes zero for the converged solution in (12), hence, fulfilling the primarily stated condition (10). ΔT_j designates the update of the absolute temperatures and the Jacobian matrix J_{ij} is defined as

$$J_{ij}^{(k)} = \frac{\partial R_i^{(k)}}{\partial T_j^{(k)}}. \tag{13}$$

The starting vector of the absolute temperatures for Newton's method (12) is chosen as

$$T_j^{(0)} = T_1 + \frac{j-1}{n}(T_{n+1} - T_1) \tag{13}$$

where the known absolute temperatures T_1 (heating system) and T_{n+1} (furnace containment), respectively, represent the prescribed boundary conditions of the numerical model.

The numerical models are composed of the heating system, the thermal insulation system and the furnace containment. In between the heating system and the thermal insulation system and the thermal insulation system and the furnace containment, respectively, heat is only transferred by radiation, i.e., no thermal conductance takes place between these parts. Four different insulation systems are investigated: two conventional insulation systems and two hybrid insulation systems. The two conventional insulation systems are i) a conventional radiation shield system and ii) a conventional refractory lining. The hybrid insulation systems are iii) an assembly of one refractory lining (inside) and one radiation shield system (outside) and iv) an assembly of one radiation shield system (inside) and one refractory lining (outside). Figure 2 shows schematically the design of the four numerical models. In the respective models the radiation shields are assumed to be very thin compared to the overall thickness of the thermal insulations. Hence, heat conduction within the radiation shields is omitted in the numerical computations. In the numerical models the number of layers of the thermal insulation systems is defined as eight, resulting in $n = 10$ layers, altogether (see Table I). Their thickness is uniformly defined as $d_i = 5\ [mm]$, $i = 1, \dots, n$, resulting in an overall thickness of the thermal insulation systems of $D = \sum_{i=2}^{n-1} d_i = 40\ [mm]$. Furthermore, each layer of the thermal insulation system transfers heat either by thermal conductance with $\lambda_i = 0.2 + 3 \cdot 10^{-4} \cdot (T + \theta_0)\ [W/(m \cdot K)]$ or by thermal radiation with $\varepsilon_{i,1} = \varepsilon_{i,2} = 0.3$. These are typical values for the thermal conductivity of stabilized Zirconium oxide fillings (at least determined experimentally ([6] and [7]) in the range of 0 to $1000\ [°C]$) and the emissivity of radiation shields made of tungsten. In all numerical simulations the temperature of the furnace containment is kept constant at $\theta_{n+1} = \theta_{out} = 100\ [°C]$. The temperature of the heating system is varied in the range of $\theta_1 = \theta_{in} = [100, 2500]\ [°C]$ in steps of $100\ [°C]$. For each applied temperature θ_1 the heat fluxes q_i, $i = 1, \dots, n$, and the temperatures at the faces of each layer θ_i, $i = 2, \dots, n$, are computed for steady state conditions.

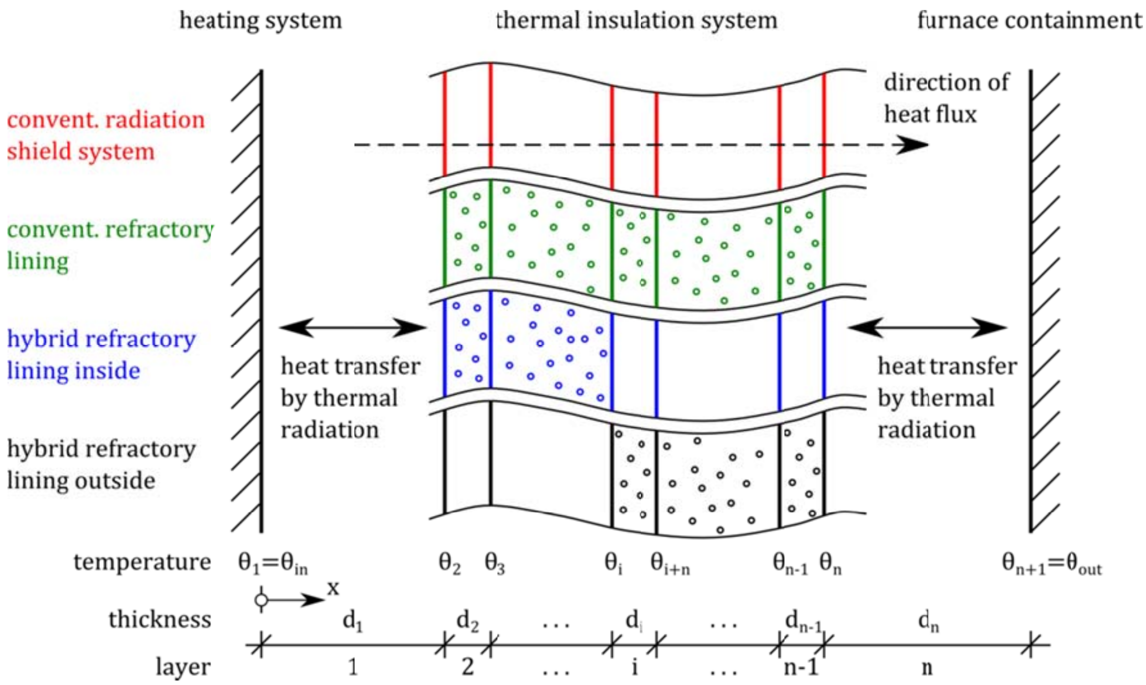

Figure 2: Schematical design of the numerical models.

Table I: Models set-up of the different thermal insulation systems: heat transfer by thermal conductance only (C) and by thermal radiation only (R)

layer i	convent. radiation shield system	convent. refractory lining	hybrid refractory lining inside	hybrid refractory lining outside
1 (inside)	R	R	R	R
2 to 5	R	C	C	R
6 to 9	R	C	R	C
10 (outside)	R	R	R	R

Analysis Results

In Fig. 3 the predicted heat fluxes from the heating system to the furnace containment are plotted over the applied temperatures of the heating system at fixed furnace containment temperature for the four thermal insulation systems. As can be seen in Fig. 3, for all numerical models the predicted heat flux strictly increases with increasing temperature of the heating system. For low to medium temperatures up to about $\theta_{in} = 1250\ [°C]$, the conventional radiation shield system insulates best, i.e., the predicted heat flux is lower than those of the other thermal insulation systems. In contrast, up to this temperature the conventional refractory lining shows the worst thermal insulation behavior of the four systems. The heat flux computed for the two hybrid thermal insulation systems lies within the range of the two conventional ones. In the range of $1250 \leq \theta_{in} < 1800\ [°C]$ the hybrid insulation system with the refractory lining located in the hot zone (inside) is the preferred choice as an energy-efficient insulation system. For higher temperatures $\theta_{in} \geq 1800\ [°C]$ the conventional refractory lining insulates best. Above approximately $\theta_{in} = 1600\ [°C]$ the heat flux through the conventional radiation shielding system exceeds the heat flux of all other thermal insulation systems. In the whole investigated temperature range, the hybrid insulation system with the refractory lining located in the cold zone (outside) never becomes the most energy-efficient one and always is less energy-efficient than the other hybrid thermal insulation system.

The predicted temperatures in the thermal insulation can be seen for selected temperatures $\theta_{in} = [100, 500, 900, 1300, 1700, 2100, 2500]\ [°C]$ in Fig. 4. The coordinate $x\ [mm]$ in Fig. 4 denotes the distance from the heating system in outward direction, cf. Fig. 2. Hence, the heating system is positioned at $x = 0\ [mm]$ and the furnace containment at $x = 50\ [mm]$, respectively. The thermal insulation system is located in the range of $x = [5, 45]\ [mm]$. For low and medium temperatures θ_{in} the temperature gradient within the thermal insulation system is most pronounced in the layers where heat is transferred by thermal radiation. For high temperatures θ_{in} the temperature gradient is steepest in the layers representing the refractory lining part. For fixed temperatures θ_{in} the temperatures at the interior and exterior face of the thermal insulation systems, respectively, do not vary considerably for the four thermal insulation variants. However, within the thermal insulation systems the temperatures differ significantly for the different thermal insulation systems. From the temperature gradient within the thermal insulation system it can be inferred whether the maximum admissible temperature of used material in the respective layer is exceeded or not. Furthermore, the energy consumption required for heating up and cooling down the respective layers can be estimated.

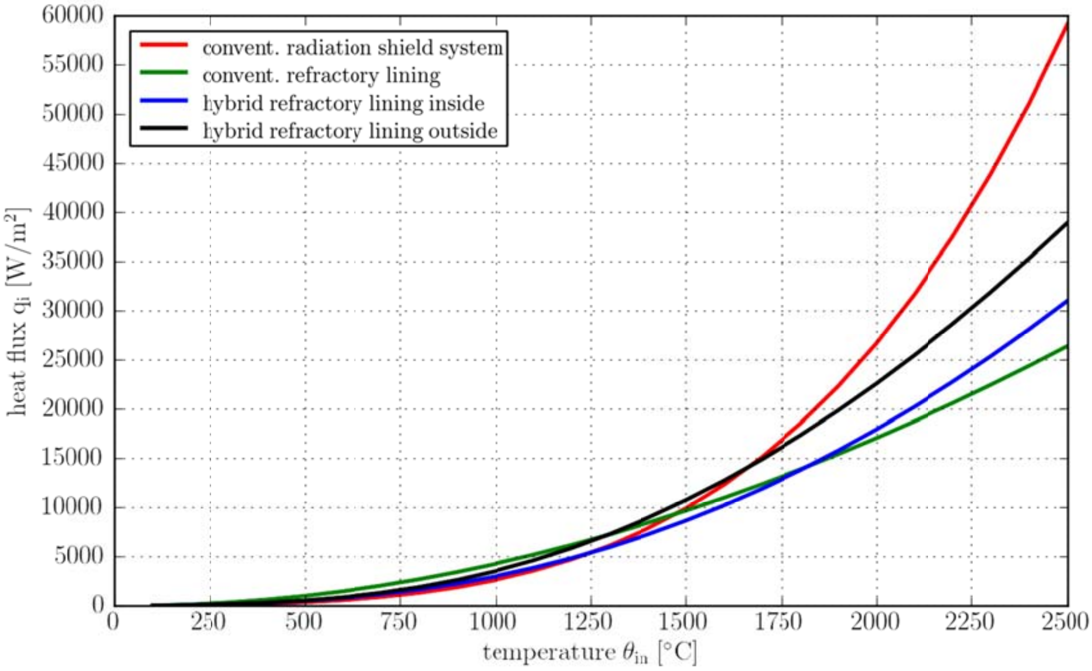

Figure 3: Predicted heat flux of the four thermal insulation systems.

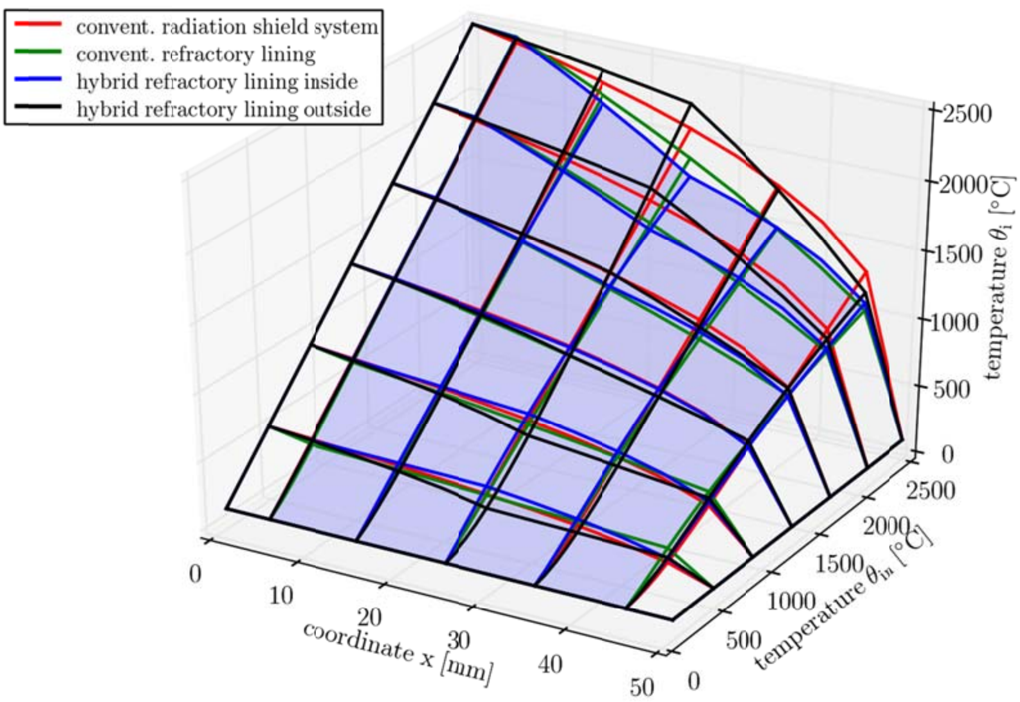

Figure 4: Predicted temperature distribution in the thermal insulation of the four thermal insulation systems for $\theta_{in} = [100, 500, 900, 1300, 1700, 2100, 2500]$ [°C].

Conclusions

As has been shown in this contribution, a novel hybrid thermal insulation system, composed of one refractory lining and one radiation shielding system, can significantly improve the energy-efficiency of high-temperature furnaces with respect to the reduction of the heat flux from the hot interior of the furnace to the cold exterior of the furnace. The improved energy-efficiency is based on positioning the refractory lining part of the hybrid thermal insulation system in the hot zone of the insulation system and the radiation shielding system part in the cold zone of the insulation system. The novel hybrid thermal insulation can be optimized with regard to its energy-efficiency for different kinds of high-temperature furnaces by adjusting the dimensions of the refractory lining and the number of radiation shields, respectively.

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