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DOI: 10.24425/ame.2025.155868 2025, Vol. 72, No. 3, pp. 455–471

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Improving the fire protection system of the heat exchanger tube sheet

Received 31 July 2024, **Revised** 30 August 2025, **Accepted** 3 September 2025, **Published online** 19 September 2025

Keywords: heat exchanger, CFD, fire protection, tube sheet, refractory

Shell-and-tube heat exchangers are integral to thermal systems in the energy and power sectors. Localised failures at the tube-to-tube sheet connection frequently compromise their operational longevity. Extreme conditions, i.e., high temperatures and pressure, in which the systems typically operate, together with high-temperature gradients, are the primary causes of material degradation and thermomechanical fatigue, which leads to damage that necessitates unplanned shutdowns and costly repairs.

This study compares thermal protection solutions that are applied to mitigate peak temperatures and reduce thermal gradients at the vulnerable tube inlet region. Using steady-state, axisymmetric CFD models, the following configurations were analysed: (1) an unprotected tube sheet, (2) sharp-edge inlet refractory of varying thickness, (3) refractory with a hyperboloid-converged inlet nozzle, (4) refractory with a converged inlet nozzle and orifices of varying narrowing, and (5) refractory with a converged nozzle in combination with ferrules of varying length.

During the course of the analysis, numerous relationships between variables were developed and demonstrated in the paper: the average temperature gradient across the tube sheet as a function of refractory thickness and the corresponding inlet edge temperature, the heat flux distribution downstream of the tube inlet for various thermal protection configurations, and the inlet edge temperature as a function of orifice narrowing.

The findings indicate that refractories of industry-standard thicknesses intensify heat flux density near the tube inlet and locally increase inlet edge temperatures. Introducing a convergent nozzle with a 20% orifice yields the most significant reduction in tube inlet edge temperature of all models analysed. The combined use of ferrules and refractory offers thermal protection for the tube inlet edge and the initial tube length,

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ensuring lower heat flux values further downstream compared to the configuration with a narrowing.

The results and analyses presented in the paper provide a critical insight into the methods of optimising thermal protection applied to extend heat exchanger service life and reduce maintenance interventions.

1. Introduction

Global development and industrialisation drive an increasing demand for energy and products from the chemical and petrochemical industries. Heat exchangers, particularly large tube and shell heat exchanger units, are the crucial components of process lines. According to Markets And Markets [1], the heat exchanger market was worth USD 15.6 b. in 2021 and is expected to grow to USD 19.9 b. by 2026, with a 40% share of shell-and-tube heat exchangers. Although these units have been used for over a century, they still have design flaws that contribute to failures and costly, time-consuming repairs.

There is a great deal of interest among researchers and companies manufacturing heat exchangers regarding their failure rates, for example, Ali et al. [2] and Schwartz [3]. The primary reasons cited by Schwartz [3] for exchanger failures are:

- thermomechanical fatigue,
- erosion,
- corrosion,
- sediment in the pipes.

The failures associated with the first three of these factors occur most frequently near the inlet to the exchanger tubes, and their intensity is related to high temperatures and high-temperature gradients. The temperature of inlet gases distributed on the tube sheet of quench heat boilers can exceed 1 700°C, and that of the heat-receiving medium is often around 300°C.

Thermomechanical fatigue is caused by excessive heating of the tube inlet due to the high heat transfer coefficient in this area. The cyclically alternating high-temperature gradients occurring in this area induce alternating high thermal strain, which can lead to cumulative plastic deformation and, as a result, terminate the material's fatigue life.

The erosion process occurs when a gas containing solid particles is streamlined. Besides of the type of gas and the number and size of particles, the fundamental parameters affecting the intensity of erosion are the temperature of the metal and the turbulence of the flow, both of them causing high shear stresses on the streamlined surface. According to Roy [4] and Zhou [5], erosion intensity increases with the metal temperature when the gas is flowing along a wall. Both excessive heating of the metal and high wall shear stresses occur in the region of the tube inlet.

The occurrence of erosion can accelerate the metal corrosion process due to the damage to its protective layer, according to Surowska [6].

Additionally, due to changes in flow direction and dimensions, overheating around the tubes' inlets is caused by varying heat transfer intensity along the beginning of the tube length, as described by Wisniewski and Wiśniewski [7]. Particularly dangerous are the transients which occur during the service and when the heat exchanger is starting (heating up) and stopping (cooling down). The consequence of varying high-temperature gradients is reduction of the fatigue life of the tube inlet area, leading to sun-like cracks on the inlet edge and cracks in the weldment between the tubes and the tube sheet (Fig. 1), which cause leaks in the system.

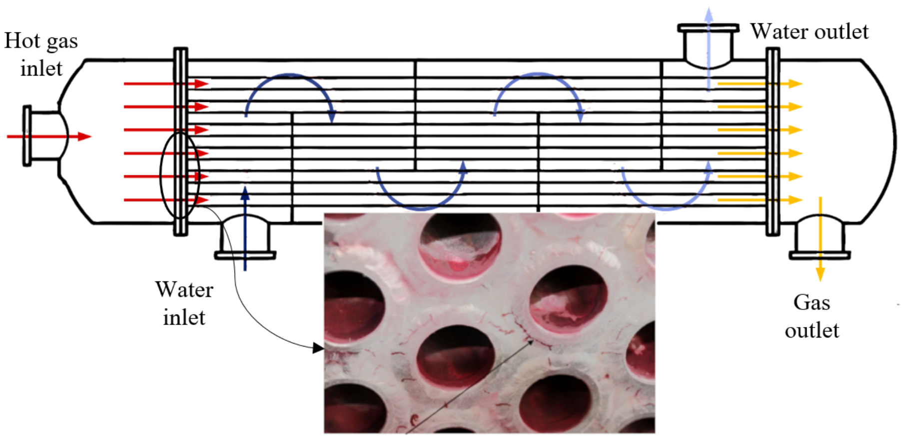

Fig. 1. Schematic of a heat exchanger with the picture of typical cracks at the tube-to-tube sheet weldment

Because of the above phenomena, adequate protection of the tube sheet from excessive heating is required. The main thermal protection solutions are refractory – fire-resistant wall composition of bricks and cement, and ceramic inserts – ferrules prefab out of ceramic, mounted in the inlet of the tubes, often with ceramic paper coating. To prevent some of the negative effects of the temperature differences in the tube sheet and the surrounding area, the tubes are reamed by rolling to the tube sheet holes, and the weldments are moved to the cold side on the heat-receiving side, patented by Hong [8]. Another solution is application of double tube sheets with internal cooling, patented by Birk [9]. Compensation in the shell may be used to solve the adverse effects of the temperature difference between the shell and tubes. However, these solutions are often insufficient or difficult to apply for technological reasons.

There is no detailed information in the available literature on the effectiveness of specific design solutions in protecting the tube sheet against heating. Few numerical analyses of flow, fatigue, or erosion phenomena around the inlet and the tubes of shell-tube exchangers can be found. Farrahi et al. [10] presented a flow-mechanical analysis of the effect of plugging individual tubes of a shell-tube heat exchanger on low-cycle fatigue of the structure. Patil with Anand [11] presented an FEM analysis of an entire shell-tube heat exchanger, focusing on using procedures available in the ASME standard. Gao et al. [12] presented a numerical tube sheet erosion process study. These analyses are a selection from among a dozen or so similar publications, which are mostly aimed at showing the possibility of modelling certain phenomena rather than presenting a quantitative result or relationships between specific design solutions and the quality of thermal protection of tube sheets. A quantitative result of a CFD flow analysis of a tube inlet issue using a ferrule was presented by Porter et al. [13]. Interesting results were also presented by Schulz [14], who carried out analytical calculations of the heat transfer intensity immediately downstream of the ferrule. He proposed an effective design improvement in the form of an altered ferrule cross-section in response to the frequent failure caused by excessive tube heating immediately downstream of the ferrule.

2. Purpose and scope of the work

This article aims to fill a gap in the literature providing an in-depth study of the influence of different design solutions and their parameters on the thermal protection of the tube sheet against excessive heating. The analysis considers the system in its steady state, as it is assumed that any improvements will hold merit in the transient state as well. The focus is on the section of the exchanger where damage occurs most often – the tube sheet inlet area, the most exposed to high-temperature gradients part of the device. The expected outcome is the set of recommendations for the thermal protection design of the shell-and-tube heat exchangers.

To obtain accurate and comparable results, axisymmetric models of a single tube inlet under steady-state conditions with different heat protection configurations (Fig. 2) were created. Several different design solutions were considered, i.e.:

- tube sheet without thermal protection (Fig. 2a),
- refractory with sharp edge inlet and variable thickness (from 20% to 600% of the tube diameter – Fig. 2b),
- refractory with a hyperboloid converged inlet nozzle (Fig. 2c),
- refractory with a hyperboloid-converged inlet nozzle with an orifice (narrowing from 5% to 20% – Fig. 2d),
- refractory with a hyperboloid converged inlet nozzle and a ferrule (length from 120% to 200% of tube diameter – Fig. 2e).

For each solution, the following parameters were analysed: the maximum temperature in the metal, temperature distribution and the pressure needed to induce the desired flow. Graphs of the above quantities as functions of varying geometrical parameters were created. The advantages and disadvantages of each solution were presented, the results were discussed, and recommendations were made to protect the tube sheets from excessive heating.

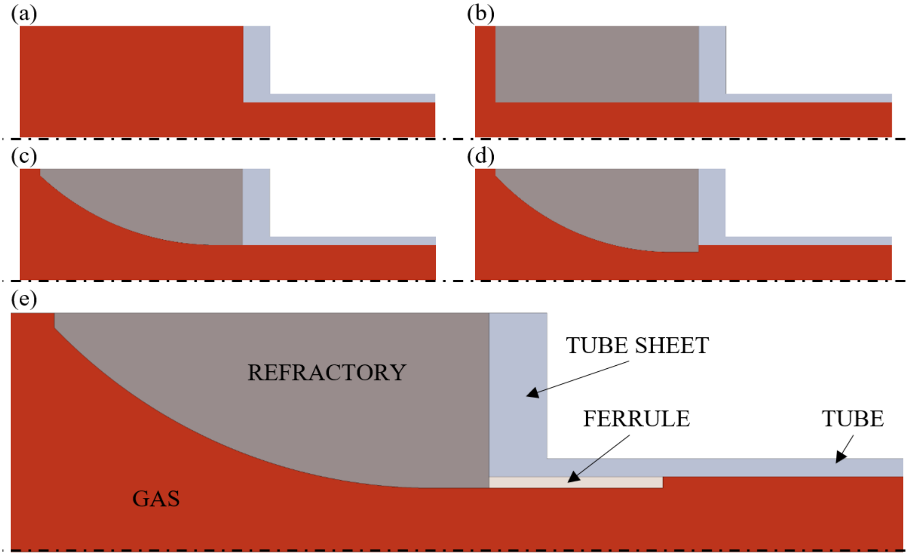

Fig. 2. Design domain (gas-red, metal-blue, refractory-grey) adopted for different design solutions: (a) no use of tube sheet protection, (b) use of refractory, (c) use of refractory with convergent inlet nozzle, (d) use of refractory with convergent inlet nozzle and sharp-edged orifice, (e) use of refractory with convergent inlet nozzle, ferrule and ceramic paper

3. Numerical model

The computational model was based on the quench heat boiler's (QHB) geometry, a process gas cooler, and a typical heat exchanger used in olefin installations at many petrochemical plants. The unit has a cooling capacity of 7 MW, is positioned vertically, and has a diameter of 1 m and a length of 12 m in the shell section. The process gases are a mixture of hydrocarbons and steam with a temperature of 847°C and a pressure of 2 bar. They are distributed through a tube sheet to 68 tubes with the inner diameter of 52mm and the wall thickness of 6.2 mm. The heat-absorbing medium on the shell side is boiling water at pressure p = 12 MPa with the boiling point of T = 329°C. The physical properties of the gas and materials used are given in Table 1. It is assumed that the tube sheets and the tubes are made of the same steel.

Table 1. Material properties of process gas and structural components

	Density [kg/m^3]	Cp J/kgK]	Thermal conductivity [W/mK]	Viscosity [kg/ms]
gas	$f(T) \approx 0.65$	$f(T) \approx 2100$	$f(T) \approx 0.0927$	$f(T) \approx 3.31\text{e} - 5$
steel	7190	490	34	–
refractory	2320	810	0.3	–

To assess the thermal protection of various insulation solutions, the simplified axisymmetric model (no flow or heat flux on the sides) was created to represent the area around the inlet to a single tube at the centre of the tube sheet and on the first 1 m of the tube length (Fig. 3). The model's maximum radius (width) corresponds to half the distance between adjacent exchanger tubes (which are in hexagonal layout in real device). A constant mass flow rate of Q = 0.009 kg/s in the axial direction was set at the inlet corresponding to the average flow per tube, and a pressure of 2 bar was set at the outlet.

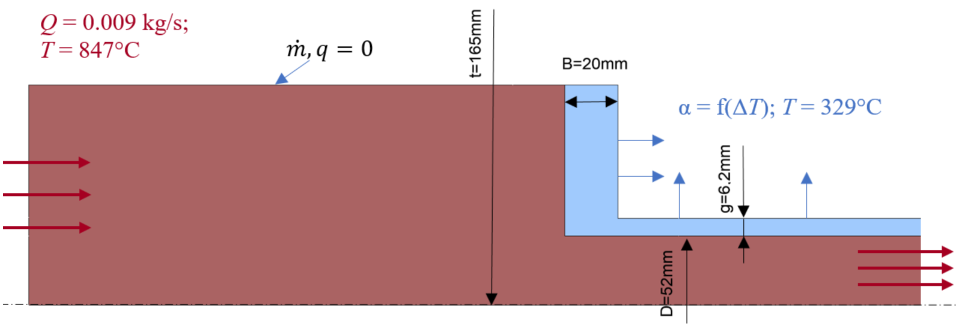

Fig. 3. Sketch of the part of the computational domain with base dimensions (t – tube spacing). The flow inlet is 0.5 m to the left from the tube sheet, and the outlet is 1 m to the right. No heat or mass flux on the top barrier of the domain

Heat transfer between the flow and the structure part was calculated via a software based on the energy equation and the turbulence model's near-wall solution. Thermal convection and conduction were taken into account.

The convective heat transfer condition was set on the shell side surfaces, representing the boiling process. Bulk film temperatures (boiling water temperature T = 329°C) and convective heat transfer coefficients were set according to the enhanced Mostinski correlation proposed by Baki and Sahel [16]:

$$h = 0.009 p_c^{0.69} q^{2/3} Fp \quad \left[\frac{\mathrm{kW}}{\mathrm{m^2K}}\right]$$

and

$$Fp = 1.8\left(\frac{p}{p_c}\right)^{0.17} + 4\left(\frac{p}{p_c}\right)^{4} + 10\left(\frac{p}{p_c}\right)^{10},$$

where p_c is critical pressure [bar], p is operating pressure [bar], and q is the heat flux $\left[\frac{\mathrm{kW}}{\mathrm{m^2K}}\right]$.

Flow and heat transfer calculations were carried out using the Fluent 2024 R2 software. Steady-state analysis was performed for each case, considering continuity, momentum and energy conservation equations. With Reynolds number Re = 6600, the flow was considered turbulent. The 2-parameter k-ω SST turbulence model was used. Abdollahpour et al. [15] compared this model with the experimental data for 2D backwards-facing step flow with Re = 9000, and the results were in good agreement.

A CFD grid was created for each model, with densification in the boundary layer area and the areas of rapid changes in flow shape (Fig. 4). 110 000–280 000 elements were used. The discretisation grids were checked for the sensitivity of the results; the difference in results (maximum temperature, pressure and velocity) between the cases when the grids were refined or coarsened by 25% was less than 1%. The grids were considered sufficiently accurate ($0.1 < y+ < 0.7$ at all walls), and the results were deemed grid-independent.

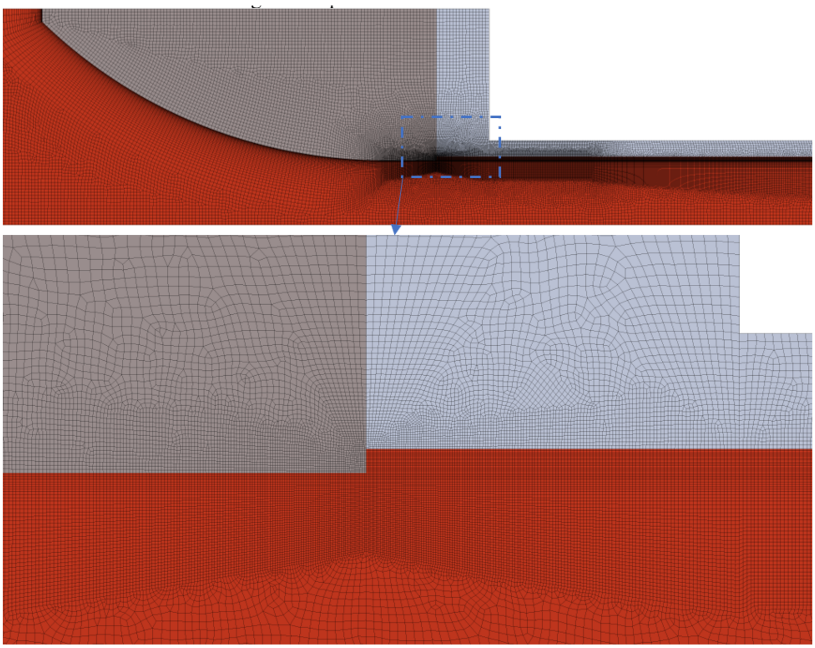

Fig. 4. Schematic of the domain discretisation around the tube inlet. Case with a refractory and sharp-edged orifice

Calculations were carried out for diverse geometric configurations with varying parameters, as described in Section 2. The reference configuration was assumed to be the model without the thermal protection applied. Since the critical factors for the heat exchanger's lifetime are corrosion, erosion and thermomechanical fatigue failure at the tube-to-tube sheet connection, minimisation of the following parameters was considered as an evaluation criterion for the fire protection system:

- the maximum value of the metal temperature gradient,
- temperature at the inlet edge of the tube,
- average gradient through the tube sheet (Fig. 3).

The pressure drop between the inlet and outlet was calculated to check and keep the “flow cost” and heating efficiency at an acceptable level.

The primary goal of thermal protection is to minimise the temperature difference between the surfaces of the tube sheet, i.e., protect the tube sheet from overheating. Limiting the temperature value and its maximum gradients at the inlet edge to the tube directly impacts the component’s service life extension by reducing the corrosion and erosion intensity, as well as the thermomechanical fatigue process in the start cycles.

4. Results and discussion

The graphs of the results are made for the whole parameter span, but the maps of velocity and temperature distribution are shown only for some chosen configurations: (a) no thermal protection, (b) refractory 75 mm thick with sharp edge inlet, (c) refractory 150 mm thick with sharp edge inlet, (d) refractory 150 mm thick with convergent nozzle, (e) refractory 150 mm thick with convergent nozzle and orifice, (f) refractory 150 mm thick with convergent nozzle and ferrule 4mm thick and 60 mm long.

The results of the gas velocity distribution for the different fire protection configurations are shown in Fig. 5. In the case of sharp edge tube inlet (configurations a, b, c), the fluid accelerates near the inlet, flows around the face of the tube sheet, and then it enters the tube, in which there is a recirculation region of about one

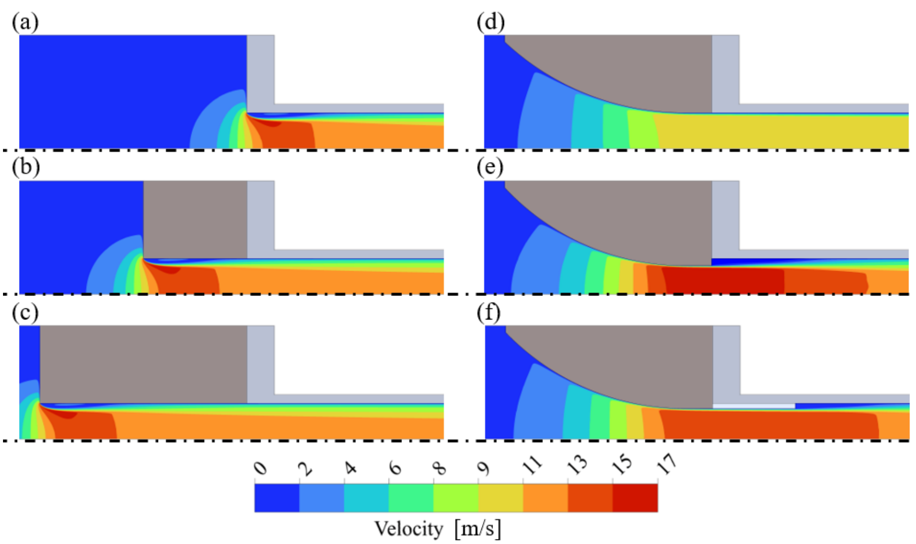

Fig. 5. Gas velocity [m/s] distribution showing the effect of the thermal protection configuration on the flow separation zone near the tube inlet: (a) no thermal protection, (b), (c) refractory with sharp edge inlet, (d) refractory with convergent nozzle, (e) refractory with convergent nozzle and orifice, (f) refractory with convergent nozzle and ferrule (white coloured)

and a half tube diameters formed at the inlet edge (Fig. 5a-c). Near it, the maximum flow velocity of 16 m/s occurs. The overpressure on the inlet to the model is p_0 = 38.5 Pa, which will be taken as the reference value.

The gas velocity distribution is more uniform when using a refractory with an inlet convergent nozzle (configurations d, e, f) under the given flow conditions. For smooth refractory tube transition (no narrowing), no flow separation occurs, and the maximum flow velocity is as low as 10 m/s (Fig. 5d). Pressure drop is 20% smaller relative to the reference configuration (no fireproof).

The use of a refractory with a convergent nozzle with an orifice leads to the flow separation behind the orifice edge (Fig. 5e). With a narrowing of 20%, the maximum velocity is 17 m/s, and the pressure drop is 6% higher compared to the reference value.

The velocity distribution when using a convergent nozzle inlet and the ceramic insert-ferrule (Fig. 5f) shows a smooth acceleration at the inlet to the convergent nozzle; the maximum velocity is reached at the minimum cross-section and is 14 m/s. The flow separation area is formed behind the back edge of the insert.

The presence of metal structure in the recirculation zone lowers the gas temperature, creating a low-temperature gas area (Fig. 6a, e, f). This also leads to a quick formation of the thermal boundary layer along the entire tube length.

Flow behaviour has a significant impact on heat transfer characteristics. The heat transfer intensity changes strongly along the recirculation zone behind the separation point Fig. 7. In the cases with no protection, at first there is a drop in

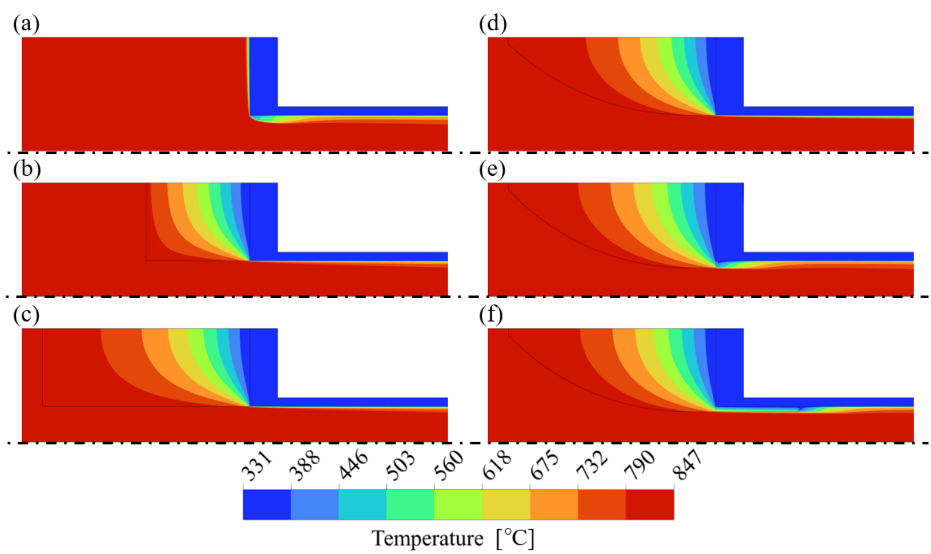

Fig. 6. Temperature distribution [°C] at the tube inlet for different fireproof configurations: (a) no thermal protection, (b), (c) refractory with sharp edge inlet, (d) refractory with convergent nozzle, (e) refractory with convergent nozzle and orifice, (f) refractory with convergent nozzle, ferrule and ceramic paper

the heat flux, then a rapid rise with a maximum around 40 mm from the inlet, followed by a continuous decrease. In most cases, there is a visible singularity of high heat flux at the beginning of the tube (first 0.1–2 mm of the tube length). This phenomenon is natural and can be explained as follows: a rapid axial temperature change caused by the change in material properties of the surrounding components results in high heat transfer values at the connection point. However, the modelling assumptions may amplify these maximum values; the absence of a gap between the refractory and the metal or the presence of sharp edges are among the main factors contributing to such a connection. Interestingly, when the refractory cover is thin (10–25 mm), this problem does not exist. If the refractories are thin enough, their cylindrical surface stays in the low-temperature area of the recirculation zone, Fig. 8; therefore, there are no significant temperature gradients. Thicker refractories have their cylindrical surface long enough for the temperature to get close to the

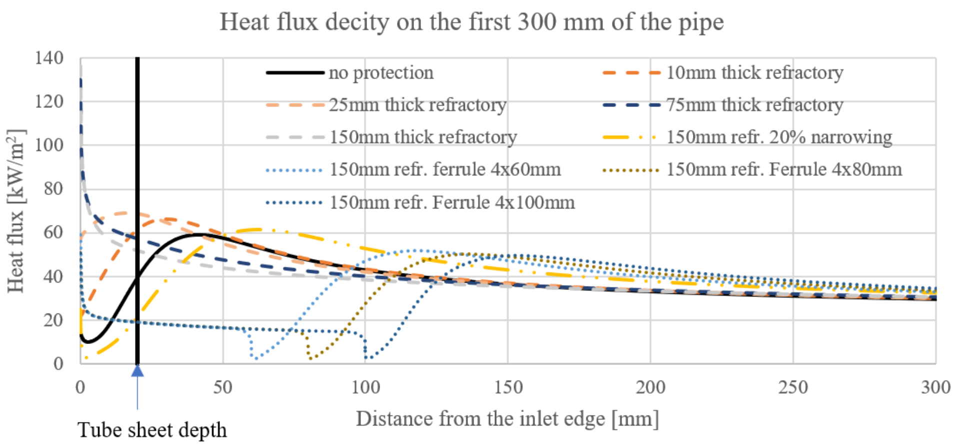

Fig. 7. The heat flux density distribution on the gas side along the tube for different protection solutions

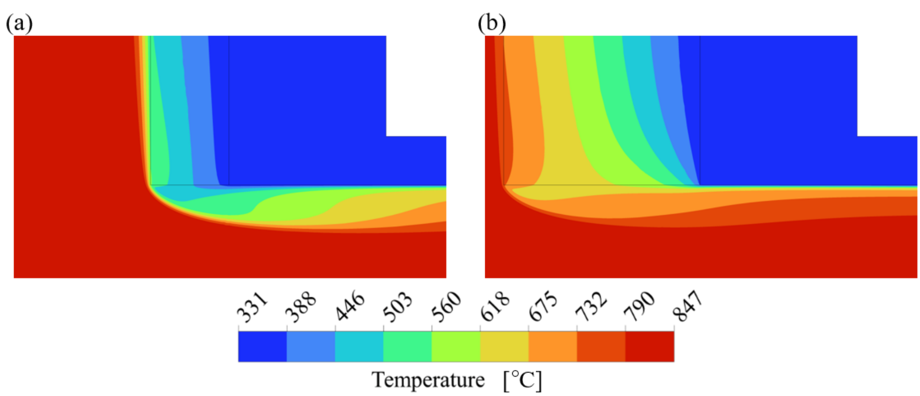

Fig. 8. Temperature distribution near the tube inlet edge: (a) 10 mm refractory, (b) 25 mm refractory

inlet temperature of the gas. In such a case, there occur very high-temperature gradients near the connection point between the refractory and the tube sheet. As a result, high heat flux values on the inlet edge decrease monotonically along the flow direction. Even in the case of the refractory and ferrule protection, there appears a noticeable heat flux peak; it is again caused by the rapid axial temperature change of the ferrule and the refractory near the inlet edge.

In the case of a 150 mm thick refractory and a 20% narrowing, the recirculation zone right behind the refractory seems to isolate the tube surface well. Further along, there is an area of elevated heat flux, followed by a monotonous drop in the heat flux.

In the case with a refractory and a 4 mm thick ferrule, the heat flux is relatively low along the ferrule length, but not as low as in the case with a refractory and a 20% narrowing. Behind the ferrule, the heat flux drops almost to 0, then rises to the value 16% lower than in the case with the narrowing (for the ferrule 60 mm long), and later decreases monotonically. Ferrules of 80 mm and 100 mm protect a longer distance of the tube, and additionally ensure that the heat flux behind their ends is lower by 3.5% and 4.5%, respectively.

The relations illustrated in Fig. 7 can be well recognised in the distribution of the temperature in the tube sheet and the tube connection shown in Fig. 9. In the reference configuration, the temperature of the front surface of the tube sheet is elevated to $T = 342°\text{C}$, with a temperature at the edge of the tube inlet increased to $T = 364.3°\text{C}$ (Fig. 6a). The use of refractory protects the tube sheet

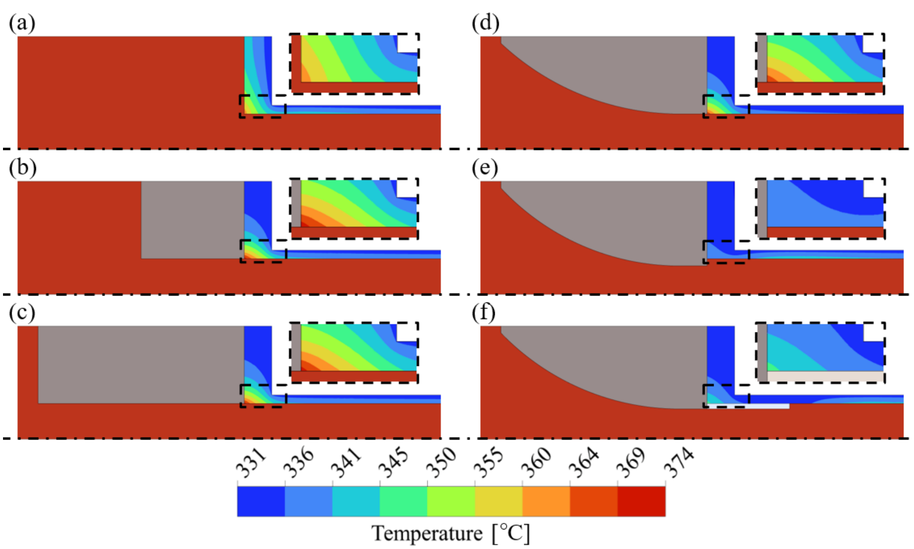

Fig. 9. Metal temperature distribution [°C] at the tube inlet for different fireproof configurations: (a) no thermal protection, (b), (c) refractory with sharp edge inlet, (d) refractory with convergent nozzle, (e) refractory with convergent nozzle and orifice, (f) refractory with convergent nozzle, ferrule and ceramic paper

surface from overheating (Fig. 9e $T = 333.5°C$), but the local temperature of the tube inlet edge rises significantly (Fig. 9e $T = 373.9°C$). In the case of the model with a refractory with a converged nozzle inlet, there is a slight decrease of the inlet edge temperature, $T = 372.8°C$. Introducing the orifice to the system leads to lower temperatures around the inlet area; for 20% narrowing, the inlet edge temperature drops to $T = 340.6°C$. It is the lowest inlet edge temperature obtained for all considered cases, which is consistent with the heat flux distribution shown in Fig. 7, proving that the local recirculation region can be a more effective thermal protection than the ceramic inserts. However, the maximum value of the tube temperature $T = 343.9°C$ is reached around 70 mm from the inlet edge. The most complex heat protection, case f, ensures inlet edge temperature $T = 348°C$. It also shifts higher temperatures, $T = 341.9°C$, further away from the tube edge.

The distribution of the heat flux discussed previously relates to the inlet edge temperature, which for regular refractory (with sharp inlet edge) differs depending on the refractory thickness, as shown in Fig. 10. For refractories thinner than 25 mm, the isolated front face of the tube sheet combined with low heat flux density near the inlet edge contributes to the edge temperature lower than in the case of an unprotected tube sheet ($T = 364.3°C$). When the refractory thickness rises, the edge temperature increases, reaching a maximum of $T = 374°C$ at 75 mm; then it asymptotically decreases to a value of $T = 372°C$. For refractories thicker than 400 mm, no change is visible in the inlet edge temperature.

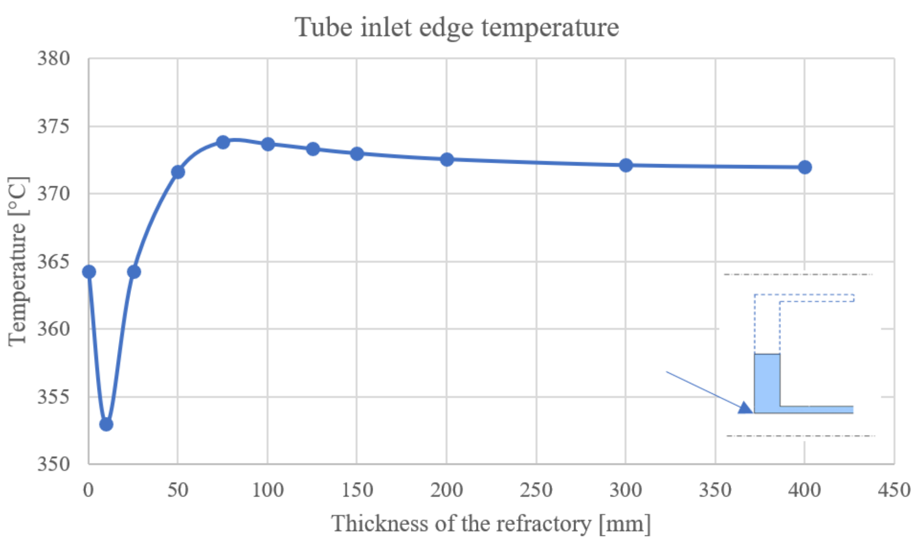

Fig. 10. Temperature at the tube inlet edge as a function of regular refractory thickness (sharp-edged inlet)

The temperature difference across the tube sheet in the reference configuration is $\Delta T = 11°C$ (Fig. 11). The application of a refractory reduces the temperature difference, which monotonically decreases with the rising refractory thickness and reaches $\Delta T = 2.2^{\circ}$C for the refractory thicknesses greater than 125 mm. The temperature difference of 2.2°C appears primarily due to uneven axial heat flux distribution at the beginning of the tube length (Fig. 7). Where this area is well protected, a further drop in the temperature difference can be observed. When a ferrule and refractory protection is applied, ΔT drops to 1.1°C, and when the refractory with an orifice is used, ΔT drops to 0.9°C.

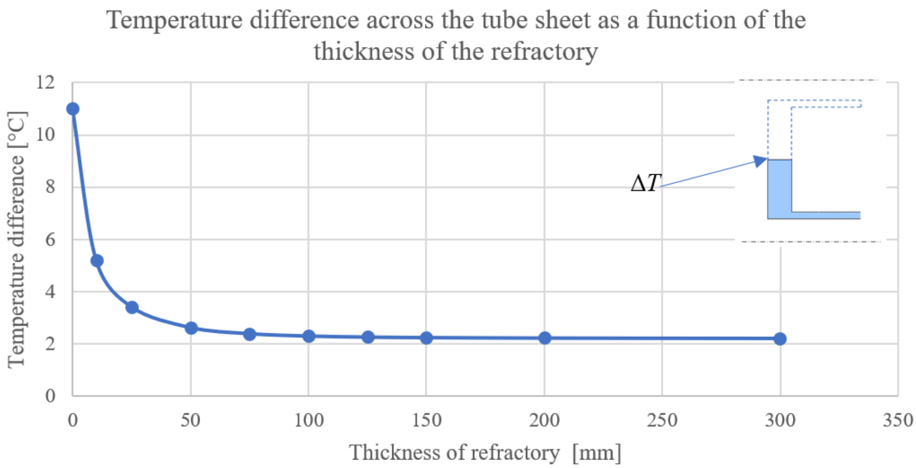

Fig. 11. Metal temperature difference across the tube sheet as a function of regular refractory thickness (sharp-edged inlet)

In the model with a convergent nozzle inlet with narrowing (orifice), the recirculation zone formed behind its edge has a positive effect on protecting the tube inlet area from excessive heating. The refractory cross-section is reduced by the orifice, which lowers the inlet edge temperature but increases the pressure drop required to sustain the desired flow rate. Reducing the refractory orifice diameter by 5% increases the necessary pressure by 3%. Still, it lowers the edge temperature by 21.8°C (around 49% less than the metal temperature difference for the refractory configuration without the narrowing) (Fig. 12). When the orifice diameter is 20% smaller than the tube inside diameter, the inlet edge temperature drops by 32.8°C (75% change), but in turn, the required overpressure on the inlet grows by 20%.

The main conclusions drawn from investigating different types of thermal protection are:

- The refractory reduces the heat transfer to the tube sheet wall and shifts the flow separation point from the tube sheet edge to the refractory inlet edge. This results in more intensive heat transfer in the recirculation zone, which might start at the tube inlet for thicker refractories. When the thickness is below 25 mm, the edge temperature of the refractory-protected tube sheet in the function of refractory thickness drops compared to the unprotected tube

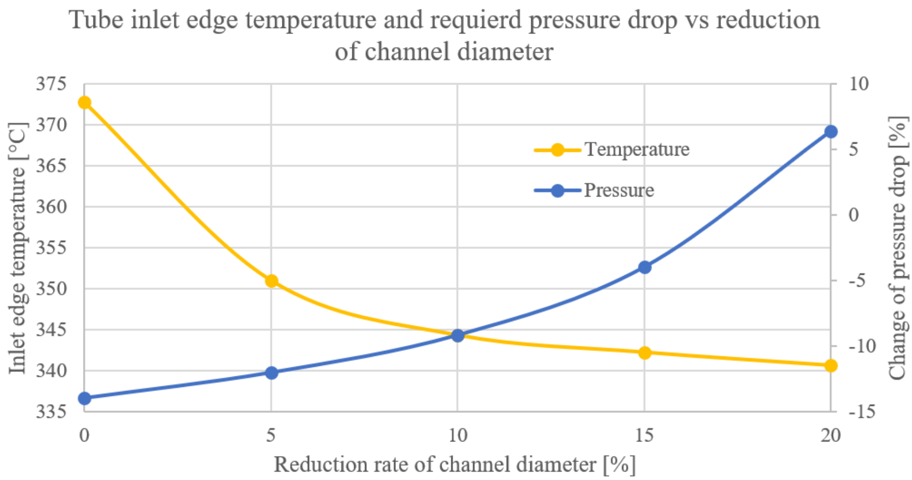

Fig. 12. Maximum temperature of the tube inlet edge [°C] and change of pressure drop [%] required for sustaining the flow rate (relative to reference solution) vs reduction rate of channel diameter by the orifice

sheet, reaching a minimum of 353°C for 10 mm (Fig. 10). The maximum value (T = 378.9°C) occurs for the thickness equal to 75 mm. For a thicker refractory, it asymptotically decreases to T = 372°C with no further change beyond 400 mm (for the analysed structure).

- The benefit of protecting the tube sheet with a thick (at least 125 mm) refractory is a lower temperature difference between tube sheet surfaces – ΔT = 2.2°C compared to ΔT = 11°C for an unprotected tube sheet (the reference configuration). Further drop in ΔT can be obtained by decreasing the heat flux from the inner tube wall by decreasing the refractory hole and applying an additional tube surface protection. The model with a convergent nozzle with the inlet diameter reduced by 20% results in ΔT = 1.5°C, and the model with a refractory with a ferrule (20% diameter reduction) results in ΔT = 1.6°C (Fig. 9d, e).
- The use of a convergent nozzle as the inlet leads to smooth acceleration of the fluid and causes that there's no recirculation region (Fig. 5d). It brings the advantage of a 14% lower overpressure relative to the sharp edge inlet required to maintain the desired flow. In this solution, the boundary layer at the tube edge is still thin and undeveloped, which means there are high velocity and temperature gradients. Together with the rapid change of the thermal properties of the refractory-tube sheet connection, this results in a high inlet edge temperature of T = 372.8°C.
- The use of a convergent nozzle with an orifice leads to positive effects of flow separation behind the orifice. A recirculation region with much lower gas temperature is created, resulting in a lower heat flux. The temperature of the tube sheet edge in the function of narrowing drops rapidly from

T = 372.8°C (no narrowing) to T = 351°C (for the narrowing of just 5%) and further to T = 340.6°C (for the narrowing of 20%). The overpressure needed to induce the desired flow is 6% higher for the narrowing of 20% than for the reference configuration. For the 20% narrowing, the maximum value of the temperature on the tube surface is shifted further away; it reaches T = 343.9°C, 70 mm behind the inlet edge.

- The most comprehensive solution – a refractory with a convergent nozzle and ferrules that cover the initial part of the tube surface – results in a relatively low tube sheet edge temperature of T = 348°C (66% lower temperature difference through the wall than for 150 mm regular refractory protection). This solution guarantees the lowest temperature value on the tube surface behind the inlet edge, with an increase in pressure drop of 2% compared to the reference solution. In the authors' opinion, this design offers the best overall protection of the tube sheet and the tube-to-tube sheet connection.

5. Summary

The lifetime of the tube-shell heat exchangers is often determined by corrosion, erosion, and thermomechanical fatigue at hot spots, which might finally lead to cracks that appear at the tube-to-tube sheet connection. Maximum metal temperature levels, high-temperature gradients and temperature change rates at the inlet to the tubes are the primary causes of these processes. This is why the hot tube sheet in tube shell heat exchangers must be protected with a refractory in order to limit its temperature and temperature gradient. Unfortunately, the effectiveness of thermal protection is often insufficient in the area of the tube inlet. This study aimed to analyse and compare the impacts of different tube sheet thermal protections, particularly on the inlet area to the exchanger tube channels.

For this purpose, a steady-state, axisymmetric CFD model of the flow through a single exchanger tube was created using the Ansys Fluent software. The flow and heat transfer phenomena were computed in the waste heat boiler hot tube sheet with the following design solutions: (1) an unprotected tube sheet, (2) sharp-edge inlet refractory of varying thickness, (3) refractory with a hyperboloid-converged inlet nozzle, (4) refractory with a converged inlet nozzle and orifices of varying narrowing, and (5) refractory with a converged nozzle in combination with ferrules of varying length. The effects of the refractory thickness and its inlet shaping, the ferrules and the ceramic inserts on the flow structure, temperature, and pressure needed to induce the desired flow rate were then analysed in detail. Specifically, numerous relationships between variables were developed and demonstrated: the average temperature gradient across the tube sheet as a function of refractory thickness and the corresponding inlet edge temperature, the heat flux distribution downstream of the tube inlet for various thermal protection configurations and the inlet edge temperature as a function of orifice narrowing. The obtained results made it possible to determine the influence of individual design solutions on the temperature distribution in the tube sheet and in the area of the tube inlet.

The following recommendations can be formulated the for the engineers on how to design the hot tube sheet thermal protection that effectively reduces temperature and temperature gradients and thus increases the service life of this sensitive structural element.

- Protecting the tube sheet from heat is necessary, but it is worth considering the consequences of different solutions. A too-thin refractory (i.e., a tube sheet cover) does not sufficiently protect the tube sheet from overheating. However, increasing the refractory thickness significantly raises the local temperature and the heat flux density near the tube inlet.
- The use of a convergent nozzle inlet refractory in the case of axisymmetric flow lowers the required overpressure and slightly reduces the pipe inlet edge's temperature, but it doesn't significantly improve the overall temperature distribution.
- Adding an orifice to the convergent nozzle, a solution simple to apply, improves the previously discussed designs. In the analysed configuration, 20% narrowing provided the lowest temperature on the inlet edge and shifted the higher heat flux area further away from the tube sheet.
- The application of a refractory with a ferrule insert offers the most comprehensive protection. It ensures low inlet edge temperature and the lowest heat flux values behind the tube inlet area.

A study-state model of axisymmetric gas flow was presented. The thermomechanical fatigue appears in cyclically changing temperature conditions (then starting and stopping). Thus, transient analysis would help to verify the conclusions. The real flow distribution is often non-uniform, and the flow at the tube inlet may be non-axial. Distribution of the inlet flow to the tubes requires separate analyses. Non-axial flow near the inlet to the tubes may change the temperature distribution in this area. Nevertheless, using a long convergent nozzle in the refractory helps to improve the flow structure before it enters the tube. We can state then that the above recommendations should help one to design the tube sheet thermal protection and understand the principles of different thermal protection solutions.

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