Impact of a Natural Gas Cooler Design on the Cooling Performance

Mária Čarnogurská[1,*], *Miroslav* Příhoda[2], and *Branislav* Žatkovič[1]

[1]Technical University of Košice, Faculty of Mechanical Engineering, Department of Power Engineering, 042 00 Košice, Slovak Republic
[2]VŠB - Technical University of Ostrava, Faculty of Materials Science and Technology, Department of Thermal Engineering, 17. listopadu 2172/15, 708 33 Ostrava-Poruba, Czech Republic

Abstract. This article describes a methodology for the identification of cooling performance of a natural gas cooler relative to the shape of its heat-transfer surface and presents the outputs of numerical solutions for four different shapes of heat-transfer surfaces in coolers designated as C_A, C_B, C_C and C_D. Calculations were carried out for a cooler with a single row of tubes, and for coolers with two through six rows of tubes that were positioned above one another with an alternating arrangement. In all of the surface shapes, the boundary conditions were respected in order to facilitate the identification of the shape of the heat-transfer surface which is the most appropriate for achieving maximum cooling performance. Out of these four shapes, the best results were observed with the heat-transfer surface of the cooler designated as C_A. The cooling performance of a 1 m long tube with such a surface was 1,650 W.

Keywords: *cooler; finned heat-transfer surface; natural gas; performance*

1 Introduction

Determination of cooling performance of natural gas coolers by applying an analytical method is a rather difficult and time-consuming task. The complexity consists primarily in the determination of individual parameters in the equations that characterise heat transmission or transfer through heat-transfer surfaces of various shapes. An optimal way how to identify the cooling performance is to apply numerical methods with clearly defined boundary conditions.

This article presents results of the investigation into intensity of natural gas cooling with four different types of heat-transfer surfaces of the coolers used at a compressor station (CS) of a gas transit pipeline. The output of the investigation was the evaluation of individual cooling surfaces in terms of cooling intensity with forced convection of the cooling air.

[*] Corresponding author: maria.carnogurska@tuke.sk

2 Problem analysis

A shape of an external heat-transfer surface of a natural gas cooler is determined by the finned tubes, i.e., the fins with circular cross-sections that are installed along the entire length of the cooler tubes. These fins may be of various thicknesses s_r (Fig. 1), various outer diameters d_r, and various pitches b. The total cooling performance of such a device greatly depends on the overall heat transfer coefficient (k) that expresses the heat transfer from the gas through the cooler tube wall into the cooling air. If this overall heat transfer coefficient may be regarded as a constant parameter across the entire heat-transfer surface, the equation for calculating the cooling performance P is as follows [1–3]:

$$P = k \cdot S \cdot \overline{\Delta t} \quad \text{(W)} \tag{1}$$

wherein k is the overall heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$); S is the external surface area of the tubes, which was identified as a sum of the surface areas of the fins S_f and of the smooth sections of the tubes between the fins S_b (m^2); and $\overline{\Delta t}$ is the mean temperature difference between the heat-transfer media (K).

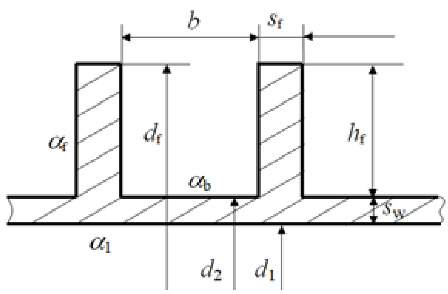

Fig. 1. Geometry of a tube and a fin.

Overall heat transfer coefficient k for a clean (without any sediments) heat-transfer surface of a finned tube is calculated using the following equation [2]:

$$\frac{1}{k} = \left(\frac{1}{\alpha_1} + \frac{s_w}{\lambda_w}\right) \cdot \frac{S}{S_1} + \frac{1}{\alpha_2 \cdot f(\eta_f)} \quad (W \cdot m^{-2} \cdot K^{-1}) \tag{2}$$

wherein s_w is the tube wall thickness (m); λ_w is the thermal conductivity coefficient of the tube material ($W \cdot m^{-1} \cdot K^{-1}$); S_1 is the internal surface area of the tubes that corresponds to a diameter d_1 (m^2); α_1 is the heat transfer coefficient for the inner side of the tubes ($W \cdot m^{-2} \cdot K^{-1}$); α_2 is the heat transfer coefficient for the heat transfer from the external finned surface into the surrounding environment ($W \cdot m^{-2} \cdot K^{-1}$); and the $f(\eta_f)$ formula is a function that depends on, inter alia, the fin efficiency η_f.

3 Thermal balance of the cooler

In the gas cooling process, a certain portion of the heat is not conducted directly into the cooling air but into the cooler structure itself. These losses are expressed by the loss coefficient η_z. The losses identified in the investigated coolers represented 0.5 – 3 %; i.e., the value of the η_z coefficient ranged from 0.995 to 0.97. The real cooling performance P_2 may then be calculated as follows [1]:

$$P_2 = P_1 \cdot \eta_z \qquad \text{(W)} \tag{3}$$

or

$$Q_{m1} \cdot c_1 \cdot (t_{1,1} - t_{1,2}) \cdot \eta_z = Q_{m2} \cdot c_2 \cdot (t_{2,2} - t_{2,1}) \qquad \text{(W)} \tag{4}$$

The $Q_{m1} \cdot c_1$ product contained in equation (4) represents the performance capacity of the gas C_1, while the $Q_{m2} \cdot c_2$ product represents the performance capacity of the air C_2.

The mean temperature difference $\overline{\Delta t}$ contained in equation (1) for cross-flows of heat-transfer media was calculated using the following equation [1]:

$$\overline{\Delta t} = \psi \cdot \frac{\Delta t' - \Delta t''}{\ln \dfrac{\Delta t'}{\Delta t''}} \qquad \text{(K)} \tag{5}$$

wherein $\Delta t' = t_{1,1} - t_{2,2}$ and $\Delta t'' = t_{1,2} - t_{2,1}$.

The first figure in the temperature index represents the heat transfer medium (1 – gas; 2 – air) while the second figure after a comma means either the entry (1) or the exit (2). The correction coefficient ψ is expressed using the P and R criteria as follows:

$$\psi = \frac{\ln A}{n \cdot (1 - R) \cdot \ln\left(1 + \dfrac{1}{R} \cdot \ln \dfrac{R - 1}{R \cdot A^{1/n} - 1}\right)} \qquad (1) \tag{6}$$

wherein $A = \dfrac{1 - P}{1 - R \cdot P}$ (1), $P = \dfrac{t_{1,2} - t_{1,1}}{t_{2,1} - t_{1,1}}$ (1), $R = \dfrac{t_{2,1} - t_{2,2}}{t_{1,2} - t_{1,1}}$ (1), n is the number of heat exchanger runs, as shown in Fig. 2.

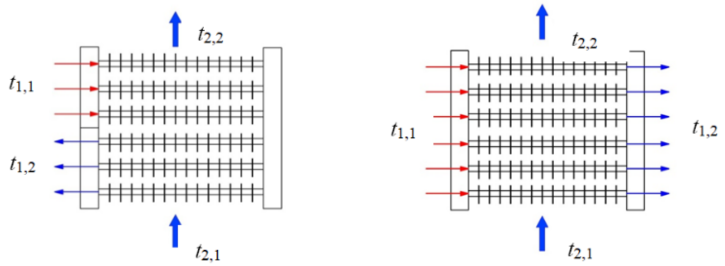

Fig. 2. A scheme of natural gas coolers with the media cross-flow.

Heat transfer coefficient inside a tube α_1 was calculated using the following equation:

$$Nu = 0.021 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \left(1.27 - 0.27 \cdot \frac{T_w}{T_{1,\text{mean}}}\right) \qquad (1) \tag{7}$$

wherein T_w is the temperature of a tube wall (K) and $T_{1,\text{mean}}$ is the mean value of the gas temperature (K). A characteristic length in Nu and Re criteria in equation (7) is the inner diameter of the tube d_1.

The calculation of heat transfer coefficient on the external side of a bundle of finned tubes was made using the following equation:

$$Nu = C \cdot Re^{0.6} \cdot K_{\mathrm{f}}^{-0.15} \cdot \mathrm{Pr}^{1/3} \qquad (1) \qquad (8)$$

The value of the constant parameter *C,* relative to the tube arrangement and a number of tube rows in the bundle, ranged from 0.20 to 0.38.

Coefficient K_{f} was identified using the following equation:

$$K_{\mathrm{f}} = \frac{S}{S_2} \qquad (1) \qquad (9)$$

wherein S_2 is the external surface area of the smooth tubes (m^2).

Circular fins are subject to the following equation:

$$K_{\mathrm{f}} = 1 + 2 \cdot \frac{h_{\mathrm{f}} \cdot (h_{\mathrm{f}} + d_2 + s_{\mathrm{f}})}{(b + s_{\mathrm{f}}) \cdot d_2} \qquad (1) \qquad (10)$$

A characteristic length in *Nu* and *Re* criteria in equation (8) is the outer diameter of the tube d_2.

4 Numerical calculations

Identification of a cooler's performance by applying an analytical procedure does not facilitate an adequate comparison of heat-transfer surfaces in terms of heat removal intensity since different coolers have different designs of their heat-transfer surface (tube diameter, fin diameter, fin thickness and fin pitch). The key parameters that affect the cooling performance of a cooler during its utilisation are as follows:

- Gas flow rate (an amount of the cooled natural gas);
- Gas temperature at the entry into the cooler;
- Air flow rate (determined by a number of fans and their performance);
- Shape and size of the heat-transfer surface;
- Tube arrangement in the cooler's block (above one another, alternating);
- Cooler blocks arrangement (cross-flow, multi-cross flow).

4.1 Characteristics of the heat-transfer surface of the examined coolers

The heat-transfer surface of the investigated natural gas coolers was determined by the parameters listed in Table 1 and by the geometry presented in Fig. 1. Numerical calculations were made separately for the case of forced convection with a single row of tubes in the cooler, and then for two through six rows. In a simulation, natural gas with a mean temperature of 65 °C was flowing through the cooler tubes. At this temperature, a numerical solution was made for the cooling intensity of finned tubes in all types of cooling surfaces.

The numerical solution was made with a symmetrical section of one tube fin in each row of tubes. The input boundary conditions were a gas pressure of 7.45 MPa and a gas temperature of 65 °C, while the heat-transfer surface was assumed to be clean.

Table 1. Typical dimensions of natural gas coolers at the CS.

Parameter		C_A	C_B	C_C	C_D	Unit
d_1	Inner diameter of the tube	0.025	0.02058	0.02	0.025	m
d_2	Outer diameter of the tube	0.03	0.0254	0.025	0.03	m
s_1	Tube spacing	0.065	0.065	0.065	0.065	m
s_w	Tube wall thickness	0.0025	0.00241	0.0025	0.0025	m
d_f	Fin diameter	0.058	0.0568	0.055	0.057	m
s_f	Fin thickness	0.00065	0.0002	0.0007	0.0007	m
b	Fin pitch	0.0025	0.0026	0.0023	0.0026	m
n	Number of tubes in the cooler	1,323	4,704	2,115	1,134	ks

4.2 Physical properties of the used media

The numerical solution was made with two different gases—natural gas and air. The properties of the gas with a temperature of 65°C and the air with a temperature of 20°C are listed in Table 2. The air flow was in the transit section; that is why the Shear-Stress Transport (SST) turbulence model was selected for the numerical calculation.

Table 2. Physical properties of the gas and air.

Parameter		Value	Unit
$t_{1,1}$	Gas temperature at the entry	75	°C
$t_{1,2}$	Gas temperature at the exit	55	°C
T_{mean}	Mean gas temperature	65	°C
Q_{Vg}	Volume flow rate of the gas	18,982	$m^3 \cdot h^{-1}$
p_g	Gas pressure	7.45	MPa
ρ_g	Gas density	45.96	$kg \cdot m^{-3}$
$c_{p,g}$	Gas heat capacity	2.7053	$kJ \cdot kg^{-1} \cdot K^{-1}$
λ_g	Gas thermal conductivity	0.046075	$W \cdot m^{-1} \cdot K^{-1}$
μ_g	Dynamic viscosity of the gas	$13.424 \cdot 10^{-6}$	Pa·s
λ_a	Air thermal conductivity	0.02515	$W \cdot m^{-1} \cdot K^{-1}$
ν_a	Kinematic viscosity of the air	$1.54 \cdot 10^{-5}$	$m^2 \cdot s^{-1}$
$c_{p,a}$	Air heat capacity	1.005	$kJ \cdot kg^{-1} K^{-1}$
ρ_a	Air density	1.205	$kg \cdot m^{-3}$
v_0	Air velocity at the outlet from the fan	4.7	$m \cdot s^{-1}$

4.3 Calculation area

The modelled area was investigated at a mean temperature of natural gas on the inner side of the tube. The calculation area represented the point on the heat-transfer surface located in the middle of the tube length and the width of the cooling surface. The model of the symmetrical part of one fin and one tube for the cooling surface with a single row of tubes is shown in Fig. 3. For this model of the heat-transfer surface, the cooling performance of all of the coolers was plotted by applying the following command *areaInt(Heat Flux)@Domain Interface1 Side2*. The results are shown in Fig. 4.

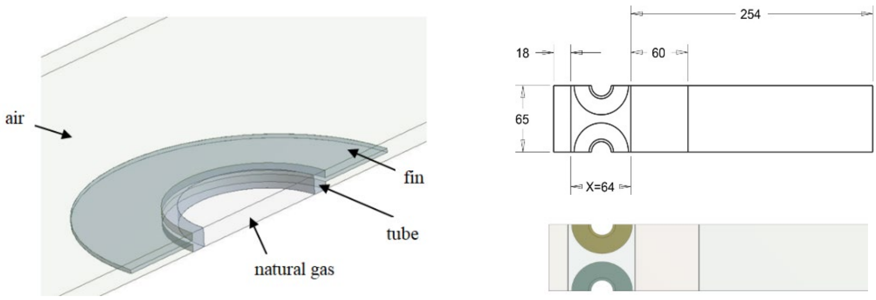

Fig. 3. Model of the symmetry of one tube and one fin – in one-row arrangement of the heat-transfer surface.

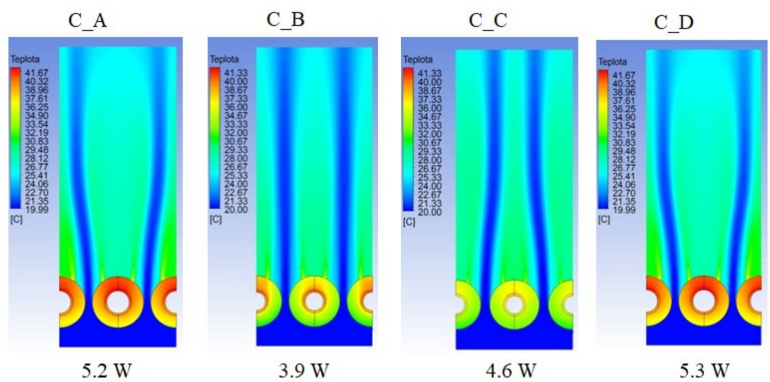

Fig. 4. Cooling performance with one row of tubes (P_{f1}).

The cooling performance for the six-row arrangement of the heat-transfer surface is shown in Fig. 5.

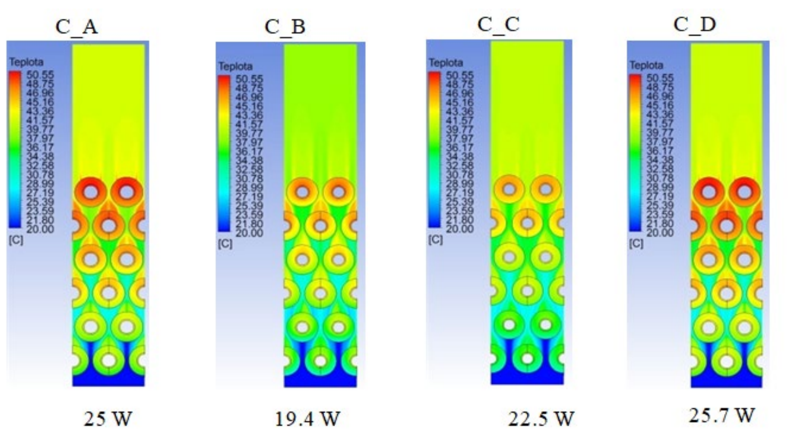

Fig. 5. Cooling performance with six rows of tubes (P_{f6}).

The evaluation of changes in the temperature along the height of the cooling surface that was formed of a single row of a finned tube, up to the six-row arrangement, was made using a section where several fins were located in the middle of the tube length. Thermal fields were plotted using the following command: *areaAve(Temperature)@Domain Interface1 Side2*. For a single row of fins, a 3D image of the thermal field is shown in (Fig. 6). The temperatures on the fins in individual coolers ranged from approximately 30 to 41°C.

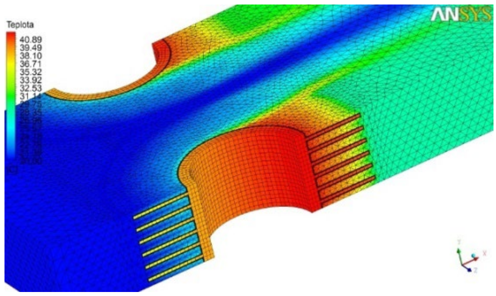

Fig. 6. Thermal field on the heat-transfer surface with a single row of tubes.

The correlations between the cooling performance and the number of rows of tubes in forced convection, for 1 row of tubes (P_f) and for 1 through 6 rows of tubes arranged above one another with an alternating arrangement, are graphically represented in Fig. 7.

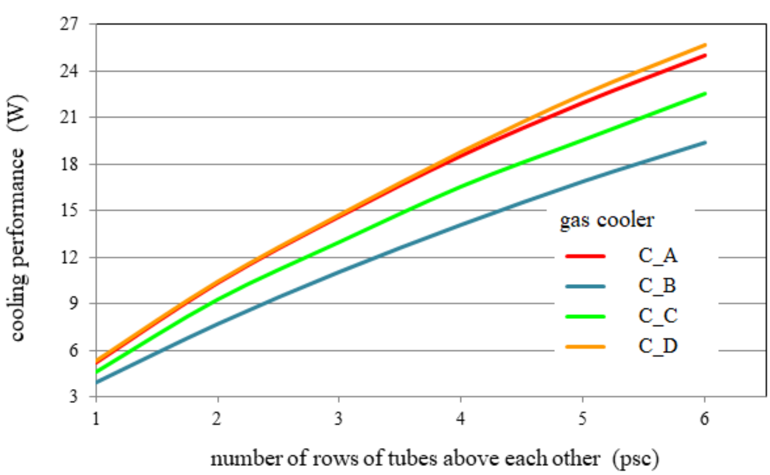

Fig. 7. Cooling performance of individual heat-transfer surfaces (1 fin and 1 through 6 rows of tubes).

The cooling performance of a 1 meter long finned tube for the individual types of the examined heat-transfer surfaces was identified for 1 through 6 rows of tubes in the heat-transfer surface based on the identified cooling performance. This required knowing the fin pitch, which was calculated as follows:

$$t_f = b + s_f \qquad \text{(m)} \tag{11}$$

The t_f values for all four heat-transfer surfaces are listed in Table. 3.

Table 3. Fin pitches for individual coolers.

Fin pitch	C_A	C_B	C_C	C_D	Unit
t_f	$3.15 \cdot 10^{-3}$	$2.8 \cdot 10^{-3}$	$3 \cdot 10^{-3}$	$3.3 \cdot 10^{-3}$	m

For the C_A cooler with the single-row arrangement of the cooling surface (Fig. 4), the cooling performance of a 1 m long tube (P_{l1}) was calculated as follows:

$$P_{l1} = \frac{P_{f1}}{t_f} = \frac{5.2}{3.15 \cdot 10^{-3}} = 1{,}650 \qquad (\mathrm{W \cdot m^{-1}}) \tag{12}$$

Calculated values of cooling performance P_{l1} for the individual types of heat-transfer surfaces applicable to 1 meter of the finned tube are listed in Table 4.

Table 4. Cooling performance of 1m long heat-transfer surface with a single row of tubes.

Parameter	**C_A**	**C_B**	**C_C**	**C_D**	**Unit**
P_{l1}	1.650	1.393	1.533	1.606	$\mathrm{W \cdot m^{-1}}$

5 Conclusion

An analysis of the results of numerical solutions that were made for all four types of heat-transfer surfaces with 1 through 6 rows of tubes has brought the following conclusions:

- Cooling performance increased with a higher number of rows of tubes positioned above one another with an alternating arrangement. With forced convection, the best cooling performance was observed with the heat-transfer surfaces of C_A and C_D coolers. This may be attributed to the dimensions of the tubes and fins and the consequent value of K_f coefficient. The K_f value for the C_A cooler was 14.2 while for the C_D cooler the K_f value amounted to 13.1. With a higher value of this coefficient, a calculation based on equation (8) revealed a lower value of the *Nu* criterion and subsequently a lower value of the heat transfer coefficient α_2. As for C_B and C_C coolers, the values of K_f coefficient represented 19.2 and 17.3, accordingly.
- Dimensions of the tubes and fins affected not only the K_f coefficient but also the fin efficiency, and hence also the value of the $f(\eta_f)$ function.

This paper has been written with the financial support from the VEGA granting agency within the project no. 1/0626/20, VEGA 1/0532/22 and from FMT VŠB-TUO within the project no. SP 2022/13.

References

1. M. Příhoda, M. Čarnogurská, J. Molínek, Tepelný výkon chladiča zemného plynu s čistou a zanesenou teplovýmennou plochou. SLOVGAS **3** (2008).
2. D. B. Spalding, J. Taborek, *Heat Exchanger Theory, Heat Exchanger Design Handbook 1*. Hemisphere Publishing Corporation (1983).
3. *VDI-Wärmeatlas. 10. Auflage*. Berlin, Heidelberg, Springer Verlag (2006).
4. S. Kutateladze, V. M. Borišanskyj, *Příručka sdílení tepla.* Praha, SNTL (1972).
5. M. Rédr, M. Příhoda, *Základy tepelné techniky*. Praha, SNTL (1991).
6. ANSYS, Inc.: ANSYS CFX Tutorials. [online]. [cit.23-04-2014]. Available online: http://www.mecheng.osu.edu/documentation/Fluent14.5/145/cfx_tutr.pdf