Design of an Optimized Graphite Electrode Cooling System by using Numerical Simulations

Ivan Mihálik[1*], Tomáš Brestovič[2], Marián Lázár[3], Šimon Hudák[4]

Department of Power Engineering, Faculty of Mechanical Engineering, Technical University of Košice, Slovakia
*Corresponding Author

Received: 03 November 2022/ Revised: 11 November 2022/ Accepted: 19 November 2022/ Published: 30-11-2022

Abstract— *The article deals with the design of optimization of graphite electrode cooling in a plasma reactor. By means of experimentally obtained data during the operation of the plasma reactor, it analyzes the current state of electrode cooling. Based on the measured temperatures at the inlet and outlet of the reactor and on the basis of the calculated flow rate of the cooling medium, it proposes a suitable solution for the optimization of the cooling system in the plasma reactor providing the necessary cooling power by analytical calculations with the use of software support. The proposed solution works on the principle of a water-air type heat exchanger. The verification of this proposal is carried out by numerical simulations using the Ansys CFX software.*

***Keywords*— CFD simulation, cooling, heat exchanger, plasma reactor.**

I. INTRODUCTION

Sufficient cooling is necessary to ensure the long service life of the electrode in the plasma reactor and its proper functioning. The high temperatures reached during operation can cause a high rate of electrode erosion. Heat transfer at the cathode occurs by radiation, convection, conduction and the influence of electron interaction. The cathode is also heated during operation by the electric current that passes through it. As the cathode is one of the elements with the highest temperature in the plasma torch, practically everything that is in contact with it serves as a thermal cooler.

The original cooling system works on the principle of an open cooling circuit. Thus, fresh water is supplied to the cooler using a water service line from the building in which the reactor is located. After heating, the water is then drained into the sewer pipe. Such a cooling system is highly inefficient and results in extremely high consumption of water. For this reason, it is necessary to change the system to a cooling system with a closed water circuit.

II. CURRENT STATE OF THE COOLING SYSTEM

The water cooler is located on the lid of the plasma reactor. It consists of 2 mm welded steel sheets. The dimensions of the water cooler are 200 x 230 mm with a total height of 44 mm. In addition to the water circuit, cooling in the plasma reactor is also provided by a fan. It is located on the frame of the plasma reactor and blows the air around the part of the hollow electrode protruding above the cooler. At the same time, the nitrogen supplied to its upper part serves to cool the electrode. The cooling system is shown in Fig. 1.

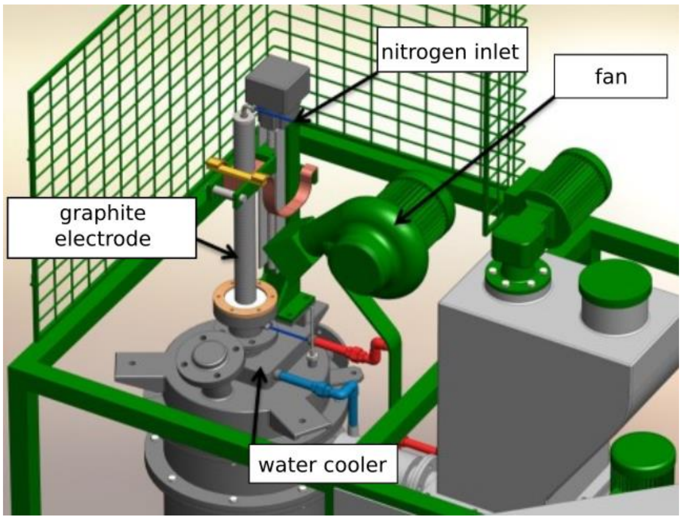


FIGURE 1: Current graphite electrode cooling system

2.1 Measurement of flow and temperatures at the inlet and outlet of the water cooler

To calculate the heat output of the cooler, it was necessary, among other things, to determine the volume flow of the cooling medium and the temperature difference at the inlet and outlet of the medium from the cooler. The required data were obtained by experimental measurement.

The water at the inlet of the reactor was kept at a constant value of 12 °C throughout the duration of the experiment. The temperature of the water at the outlet of the cooler depended on the rising temperature of the reactor. The highest temperature in the plasma reactor was recorded after 132 minutes, when it reached a value of 1 399 °C. The highest measured water value at the outlet of the cooler was 25 °C.

The volumetric flow rate was determined using an Almemo measuring device located in the cooling circuit. Flow values were recorded at 1-minute intervals. Since flow regulation was not necessary during the operation of the reactor, it was maintained at approximately a constant level of $1{,}667 \cdot 10^{-5}\ \mathrm{m^3 \cdot s^{-1}}$ during the duration of the experiment.

2.2 Determination of the thermal performance of the water cooler

The thermal performance of the cooler can be expressed using the equation (1) as the ratio of the change in the amount of heat over a certain time:

$$P_\mathrm{c} = \frac{\mathrm{d}Q}{\mathrm{d}\tau} = \frac{m \cdot c \cdot \Delta t}{\Delta \tau} \quad \mathrm{(W)} \tag{1}$$

Where dQ is the elementary amount of heat (J), dτ - the elementary time change (s), m - the mass of the cooling medium (kg), c - the specific heat capacity of the cooling medium ($\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$), Δt - temperature difference of the cooling medium between the inlet and outlet of the cooler (K).

Then it holds that:

$$P_\mathrm{c} = Q_\mathrm{m} \cdot c \cdot \Delta t \quad \mathrm{(W)} \tag{2}$$

where Q_m is the mass flow rate of the cooling medium ($\mathrm{kg \cdot s^{-1}}$).

The mass flow in equation (2) can be replaced by the volume flow according to equation (3):

$$P_\mathrm{c} = Q_\mathrm{v} \cdot \rho \cdot c \cdot \Delta t \quad \mathrm{(W)} \tag{3}$$

where ρ is the density of the cooling medium ($\mathrm{kg \cdot m^{-3}}$).

For the calculation, the mean values of the specific heat capacity and density were considered for the temperature interval t_1 = 12 °C and t_2 = 25 °C. Average specific heat capacity of water $\bar{c}$ = 4183,96 $\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$ and average density of water $\bar{\rho}$ = 998,45 $\mathrm{kg \cdot m^{-3}}$. Then for the thermal performance P:

$$P = Q_\mathrm{v} \cdot \bar{\rho} \cdot \bar{c}\,(t_2 - t_1) \quad \mathrm{(W)}$$

$$P = 1{,}667 \cdot 10^{-5} \cdot 998{,}45 \cdot 4183{,}96\ (25 - 12) = 905{,}07\ \mathrm{W} \tag{4}$$

For the new optimized design of the cooling system, a slightly increased cooling power at the level of $P_c = 1$ kW was considered. The reason for this power increase is to ensure the reliable operation of the reactor with a sufficient power reserve.

III. DETERMINATION OF THE OPTIMIZED COOLING SYSTEM PARAMETERS

The optimized system uses a closed cooling water circuit. Water circuit pipes are cooled by air flow through the fan. It is therefore a design of a heat exchanger based on the principle of water - air.

3.1 Determination of the thermal conductivity coefficient "kL"

The thermal conductivity coefficient depends on the type of heat carrier and on the type of the flow. Its increase can generally be achieved by accelerating the flow of the heat carrier. Its value is also influenced by the layout of the tubes and the air flow around them, as the tubes can be arranged in a row or alternately. The optimization of these parameters was achieved using the SPT-NK software, which uses criterion equations for calculation. This software allows to find an optimal solution without the need for analytical calculations.

The thermal conductivity coefficient k_L for the cylindrical wall can be calculated using the equation (5):

$$k_L = \left(\frac{1}{\alpha_1 \cdot D_1 \cdot \pi} + k_c + \frac{1}{\alpha_2 \cdot D_2 \cdot \pi}\right)^{-1}\ (\mathrm{W} \cdot \mathrm{m}^{-1} \cdot \mathrm{K}^{-1}) \tag{5}$$

The middle term kc of the equation (5) expresses heat transfer by conduction through the pipe wall and has very little influence on the overall thermal conductivity coefficient. It is therefore possible to neglect it.

3.1.1 Heat transfer coefficient on the inside of the pipe α1

It was decided to create a construction of a copper cooler consisting of copper pipes with dimensions of 18x1 mm, while the total length of the pipes was defined as 23 m for the calculation of the heat transfer coefficient inside the pipe. The water flow was provided by a circulating pump, and the speed of the fluid flow through the pipe reached a value of 0,083 $\mathrm{m \cdot s^{-1}}$. The temperature at the inlet to the heat exchanger t_1 was defined as 50 °C and the wall temperature $t_S = 26$ °C. The temperature at the outlet of the exchanger t_2 was calculated using equation (6):

$$t_2 = t_1 - \frac{P_c}{Q_V \cdot \bar{\rho} \cdot \bar{c}}\ (°\mathrm{C})$$

$$t_2 = 50 - \frac{1000}{1{,}667 \cdot 10^{-5} \cdot 998{,}45 \cdot 4183{,}96} = 35{,}64\ °\mathrm{C} \tag{6}$$

3.1.2 Heat transfer coefficient on the outside of the pipe α2

When calculating the heat transfer coefficient on the outside of the pipe, an alternating layout of pipes was considered. Unlike pipes placed in a row, such a layout can ensure a higher value of the heat transfer coef-ficient. The horizontal distance between the axes of the pipes was defined by the value $s_1 = 0{,}025$ m, and the vertical distance $s_2 = 0{,}05253$ m.

At air flow rate $Q_A = 0{,}8334\ \mathrm{m^3 \cdot s^{-1}}$, the flow velocity was 2,5 $\mathrm{m \cdot s^{-1}}$. The temperature $t_3 = 25$ °C was determined by measuring the ambient air temperature. Air density $\rho_A = 1{,}185\ \mathrm{kg \cdot m^{-3}}$ and specific heat capacity of air $c_A = 1\,013\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$ was determined from the tables at temperature $t_1 = 25$ °C and atmospheric pressure $p = 101\,325$ Pa. The temperature of the pipe wall was defined by the mean temperature of the water in the pipe as $t_S = 404$ °C. Air temperature t_4 after removing heat from the tubes was determined from equation (7):

$$\Delta t = \frac{P_c}{Q_A \cdot \rho_A \cdot c_A}\ (°\mathrm{C})$$

$$t_4 = t_3 + \frac{P_c}{Q_A \cdot \rho_A \cdot c_A}\ (°\mathrm{C})$$

$$t_4 = 25 + \frac{1000}{0{,}8334 \cdot 1{,}185 \cdot 1013} \cong 26\ °\mathrm{C} \tag{7}$$

The values of the heat transfer coefficients obtained by the SPT-NK software by using the input parameters are shown in the Table 1.

TABLE 1
VALUES OF THE HEAT TRANSFER COEFFICIENTS

	$(\mathrm{W \cdot m^{-2} \cdot K^{-1}})$
α_1 - the inside of the pipe	146,17
α_2 - the outside of the pipe	91,28

after substituting into equation (5), the coefficient k_L for the cylindrical wall is equal to:

$$k_\mathrm{L} = \left(\frac{1}{146{,}17 \cdot 0{,}016 \cdot \pi} + \frac{1}{91{,}28 \cdot 0{,}018 \cdot \pi}\right)^{-1} = 3{,}0318\ \mathrm{W \cdot m^{-1} \cdot K^{-1}} \tag{8}$$

3.2 Determination of the mean logarithmic temperature difference "($\overline{\Delta t}$)"

The equation (8) for calculating the mean logarithmic difference has the form:

$$\Delta t_\mathrm{LMTD} = \frac{\Delta t_1 - \Delta t_2}{\ln\frac{\Delta t_1}{\Delta t_2}}\ \ (°\mathrm{C}) \tag{9}$$

while it is based on the course of temperatures in the counterflow in Fig. 2.

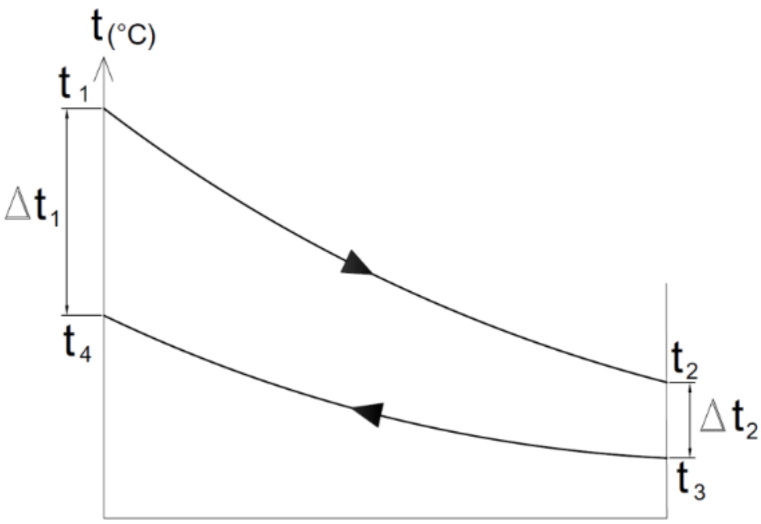


FIGURE 2: Temperature course for counterflow

Then for the temperature differences Δt_1 a Δt_2:

$$\Delta t_1 = t_1 - t_4\ (°\mathrm{C})$$

$$\Delta t_1 = 50 - 26 = 24\ °\mathrm{C} \tag{10}$$

$$\Delta t_2 = t_2 - t_3\ (°\mathrm{C})$$

$$\Delta t_2 = 35{,}64 - 25 = 10{,}64\ °\mathrm{C} \tag{11}$$

However, since it is a cross arrangement of the heat exchanger, a correction of equation (8) using the correction factor ε is necessary to calculate the mean logarithmic temperature difference. Then applies:

$$\overline{\Delta t} = \varepsilon \cdot \Delta t_\mathrm{LMTD}\ \ (°\mathrm{C}) \tag{12}$$

To determine the correction factor ε, it is necessary to know the parameters R and P, for which the following applies:

$$R = \left|\frac{t_3 - t_4}{t_1 - t_2}\right| \tag{13}$$

$$R = \left|\frac{25 - 26}{50 - 35{,}64}\right| = 0{,}07$$

$$P = \left|\frac{t_1 - t_2}{t_1 - t_3}\right|$$

$$P = \left|\frac{50 - 35{,}64}{50 - 25}\right| = 0{,}5744 \tag{14}$$

The correction factor ε can then be determined from the Fig. 3 as $\varepsilon = 0{,}99$.

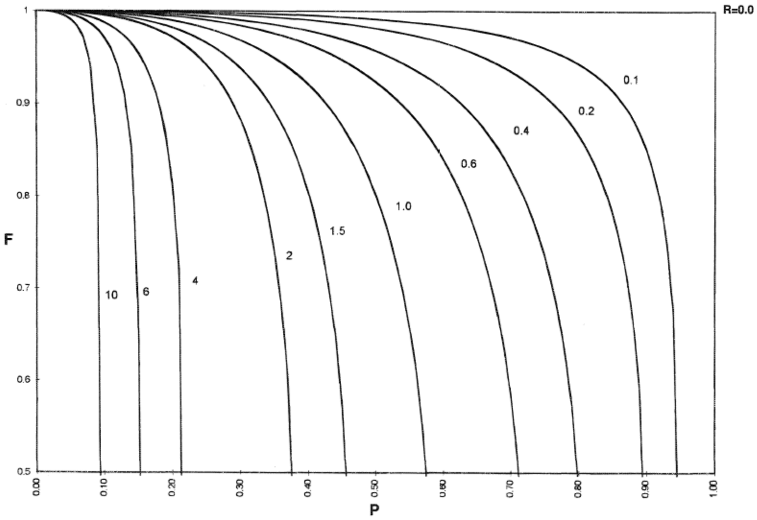


FIGURE 3: Determination of the correction factor

After substituting into equation (11), it holds that:

$$\overline{\Delta t} = 0{,}99 \cdot \frac{24 - 10{,}64}{\ln\frac{24}{10{,}64}} = 16{,}26\ °\mathrm{C}$$

After substituting the thermal conductivity coefficient k_{L} and the mean logarithmic temperature difference $\overline{\Delta t}$ into equation (14) and subsequent adjustment, the following applies:

$$P_{\mathrm{c}} = k \cdot L \cdot \overline{\Delta t} \quad (\mathrm{W}) \tag{15}$$

$$L = \frac{P_{\mathrm{c}}}{k \cdot \overline{\Delta t}} = \frac{1000}{3{,}0318 \cdot 16{,}26} = 20{,}29\ \mathrm{m} \tag{16}$$

Where L is the total required length of copper piping for the design of the heat exchanger with the specified cooling performance (m).

IV. NUMERICAL SIMULATION OF AIRFLOW AND HEAT TRANSFER OF AN OPTIMIZED COOLING SYSTEM

The heat exchanger model must be designed in such a way that the air flow velocity reaches a value of at least 2,5 $\mathrm{m \cdot s^{-1}}$ in every place of the cooler. A total of 50 tubes were used, each 400 mm long. They are located inside the exchanger chamber in a pipe with a cross-section of 400 x 355 mm and a total length of 300 mm. 49 standardized arcs were used to connect the individual pipes to each other. Since these arcs are not located inside the exchanger chamber, they ensure heat transfer only through natural convection. For this reason, they were not included in the total required pipe length. The model of the designed exchanger chamber in Fig. 4 was created using SolidWorks CAD software.

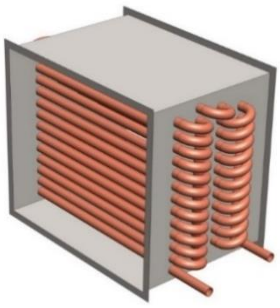
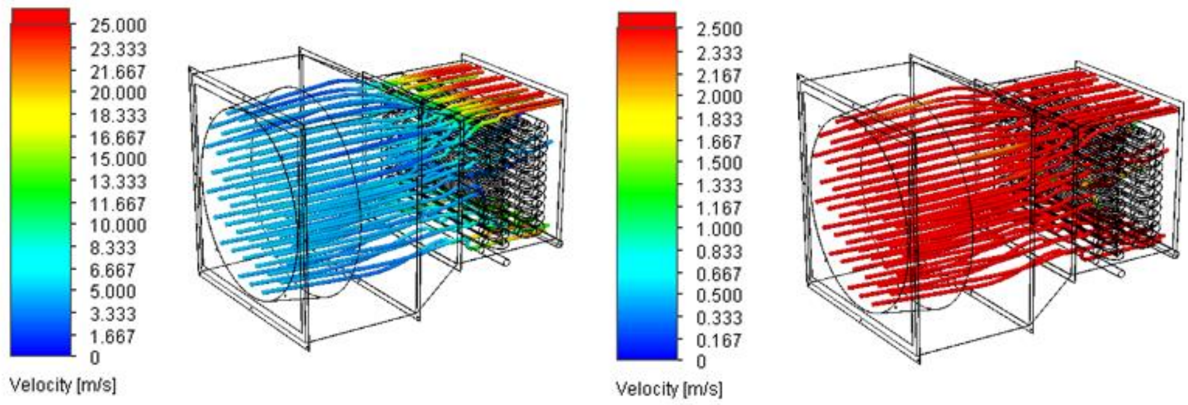


FIGURE 4: Exchanger chamber

Before the simulation of heat transfer, it was first necessary to verify on the given model whether the estimated air flow Q_A = 0,8334 $m^3 \cdot s^{-1}$ is satisfactory for the given layout of pipes. The FloXpress add-on in the SolidWorks software was used to verify this assumption. The result of the simulation are graphic contours in Figure 5 a) depicting the air speed. In Fig. 5 b) the scale is bounded by the value 2,5 $m \cdot s^{-1}$.



a) Airflow – unbounded scale **b) Airflow – scale bounded by 2,5**

FIGURE 5: Airflow velocity in the heat exchanger

It is clear from the Figure 5 b) that the air reaches the required air flow speed in almost every place of the exchanger. The air flow and the layout of the tubes can therefore be considered satisfactory.

The Ansys CFX program was used for heat transfer simulation. At the inlet to the exchanger, a temperature of 50 °C and a water flow rate of 0,01667 $kg \cdot s^{-1}$ were defined, while laminar flow was considered. A relative pressure of 0 Pa was defined at the outlet of the exchanger. The heat transfer coefficient on the surface of the tubes in the inner part of the exchanger for forced convection has the value α_2 = 91,28 $W \cdot m^{-2} \cdot K^{-1}$. The heat transfer coefficient on the surface of the arcs in the outer part of the exchanger for natural convection was defined as a function by using the SPT-VK software.

V. EVALUATION

The simulation results confirmed that the proposed heat exchanger solution is satisfactory. The Fig. 6 shows the temperatures on the surface of the heat exchanger tubes.

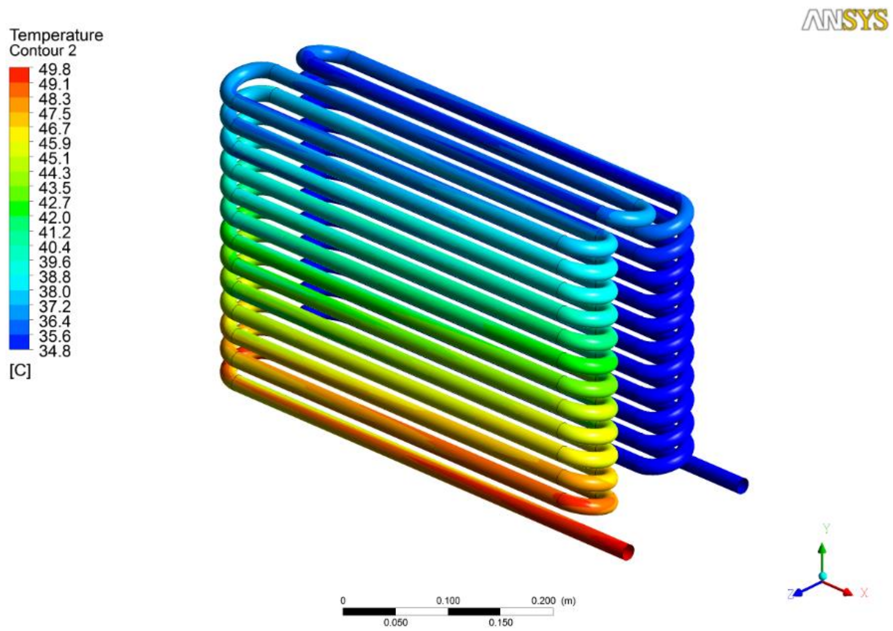


FIGURE 6: Temperature contours of the heat exchanger

The temperature of the water in the pipeline at the outlet of the heat exchanger reaches 30,5 °C. The heat flow removed by the flowing air reached a value of almost 1200 W. The increase in cooling power compared to the proposed 1000 W was probably caused by the neglect of the length of the pipe bends when determining the total length.

VI. CONCLUSION

The resulting design of the air-water exchanger provides a cooling capacity approximately 20% higher than the required 1 kW. Due to the relatively robust construction of this solution, it would therefore be possible to shorten the total length of the exchanger tubes, which would allow more compact dimensions of the device to be achieved. Minimizing the dimensions could also be achieved by using a heat exchanger equipped with different ribbed surfaces.

ACKNOWLEDGEMENTS

This paper was written with the financial support from the VEGA granting agency within the project solutions No. 1/0626/20 and No. 1/0532/22 from the KEGA granting agency within the project solutions No. 012TUKE-4/2022 and with financial support from the granting agency APVV within the Project Solution No. APVV-15-0202, APVV-21-0274 and APVV-20-0205.

REFERENCES

[1] Balaji, C., Srinivasan, B., Gedupudi, S. (2020). Chapter 7 - Heat exchangers. In: Heat Transfer Engineering, p. 199 – 231. ISBN: 978-0-12-818503-2

[2] Zhukov, M. F. – Zasypkin, I. M. (2007). Thermal Plasma Torches. ISBN: 978-1-904602-02-6

[3] Galimore, S. (1998). Operation of a High-Pressure Uncooled Plasma Torch with Hydrocarbon Feedstocks.

[4] Lendhard, R., Kaduchová, K., Jandačka, J. (2014). Numerical simulation of indirectly heated hot water heater. *Advanced Materials Research*, vol. 875-877, p. 1693-1697. https://doi.org/10.4028/www.scientific.net/AMR.875-877.1693.