Research of the influence of intensification of heat transfer on distribution of temperature in the active core of the gas cooled nuclear reactor of the «GT-MHR» project

V S Kuzevanov and S K Podgorny
National Research University «MPEI» Volzhskiy Branch, Volzhskiy 404110, Lenina 69, Russia

Email: kuzevanov@vfmei.ru, serkonpod@gmail.com

Abstract. The maximum wall temperature of a cooling channel of a nuclear reactor is one of the factors that affects directly of the safety and reliability of the nuclear reactor. In this paper suggested an equation, which allows calculating the maximum wall temperature of the cooling channel of the nuclear reactor with heat transfer enhancer installed, without enormous calculations.

1. Introduction
«GT-MHR» project [1] includes vessel high temperature graphite moderate nuclear reactor with thermal neutron specter and helium as the heat carrier. The «GT-MHR» project devoted to solve the problem of usage of energy of nuclear fuel to fulfill energy requirement in industry and increase efficiency of the power generation.

The active core consists of an assembly of hexagonal graphite fuel elements, which are stacked in the core to form columns. The active core height is 7.93 m. The heat carrier flows through the cooling channels of two diameters: 12.7 mm and 15.88 mm.

A local heat transfer enhancement allows achieving basic requirement in limiting maximum wall temperature in the most stressed zone of the active core.

Nowadays calculation methods allow modelling heat transfer in the active core of the nuclear reactor and to determine all properties of the nuclear reactor. In the same time, those methods requires high level computing performance and a lot of time. Also using those methods are not necessary at the preliminary calculations stage.

The goal of this paper is to obtain an equation, which allows calculating the maximum wall temperature by the minimum information about the cooling channel and without high level computing performance.

2. Equation obtaining
Accepted assumptions:
1. Heat generation rate distribution on the height and the radius of the active core with heat transfer enhancement does not change.
2. Heat transfer enhancement does not affect on the inlet pressure in the active core.
3. For the current Reynolds number (Re) range, a Darcy friction factor (ξ) is known or can be calculated at known heat transfer enhancer construction.
Determine upper index «\(Z\)» for properties of a cross section of the cooling channel with temperature of the heat carrier \(T_{h.c}^Z\) at \(Z\) coordinate. Denote \(\theta_{h.c}^Z = T_{h.c}^Z - T_{in}\), where lower index «\(h.c\)>> related to heat carrier; index «\(in\)>> related to inlet to the active core; \(T\) – average temperature of the heat carrier at the cross section of the cooling channel.

Then, a heat balance equation for an arbitrary cooling channel in steady state

\[
Q^Z = G \cdot c_p \cdot \theta_{h.c}^Z,
\]

(1)

where \(Q^Z = \int_{\frac{H_0}{2}}^Z q_i(Z) dz, \) W; \(Z\) – coordinate of the cross-section of the cooling channel, m; \(H_0\) – the active core height, m; \(q_i\) – linear heat flux, W∙m\(^{-1}\); \(G\) – mass flow, \(\) kg∙s\(^{-1}\); \(c_p\) – specific isobaric heat capacity, \(\) J/kg°C.

From the equation (1) follows relation

\[
\frac{dG}{G} = \frac{d\theta_{h.c}^Z}{\theta_{h.c}^Z}.
\]

(2)

Heat transfer equation for the cross-section for \(Z\) coordinate represented in following form

\[
\alpha \cdot \left(\theta_w^Z - \theta_{h.c}^Z\right) = \frac{q_i}{\Pi_{th}},
\]

(3)

where \(\alpha\) – heat transfer coefficient, \(\frac{W}{m^2°C}\); \(\theta_w^Z = T_w^Z - T_{in}\); index «\(w\)>> related to the surface of the cooling channel; \(\Pi_{th}\) - heated perimeter, m.

According to relation suggested by B Petukhov [2], a Nusselt number (Nu) is in proportion to product \(\xi \cdot Re\) or

\[
\alpha = a \cdot \xi \cdot G,
\]

(4)

where \(a = \frac{c_p \cdot c_i}{8 \cdot f \cdot \left(k + \xi (Pr)\right)}\); \(f\) – surface square of the cross-section of the cooling channel, m\(^2\).

For a gas heat carrier the Prandtl (Pr) number is close to one and almost a constant and in the same time coefficient \(k\) at high Re values is also close to one and by neglecting temperature effects \(c_i\), let us make an assumption that \(a \equiv const\).

With considering Nunner equation [3] for gases, let us make an assumption that relation (4) is correct not only for smooth channels it is also correct for channels with long heat transfer enhancers. From the relations (3) and (4) follows relation

\[
\xi \cdot G \cdot \left(\theta_w^Z - \theta_{h.c}^Z\right) = const.
\]

(5)

With considering (2) and after differentiation relation (5) obtaining following differential equation
\[
\frac{dG}{G} + \left(1 - \frac{\theta_{h,c}^2}{\theta_{w}^2}\right) \cdot \frac{d\xi}{\xi} + \frac{d\theta_{w}^2}{\theta_{w}^2} = 0 .
\]  

(6)

Notice that differential \(d\xi\) related only with change properties of the surface of the cooling channel.

During increasing the mass flow thorough, the cooling channel \(\theta_{h,c}\) and \(\theta_{w}\) are decreasing. At the same time during decreasing, the mass flow through the cooling channel \(\theta_{h,c}\) and \(\theta_{w}\) are increasing. According to this statement \(\frac{\theta_{h,c}}{\theta_{w}}\) is a constant.

Then equation (6) has a simple solution

\[
G \cdot \theta_{w}^2 \cdot \xi^{\sigma} = \text{const} ,
\]

where \(\sigma = 1 - \frac{\theta_{h,c}^2}{\theta_{w}^2}\).

Let index «0» be related to the smooth channels. Deploying the heat transfer enhancer is changing state of the cooling channel, and if condition (7) is performed, then a following equation can be obtained

\[
\theta_{w}^2 = \theta_{w,0}^2 \cdot \frac{G_{0}}{G} \left(\frac{\xi_{0}}{\xi}\right)^{\sigma_{0}} ,
\]

(8)

where \(\sigma = 1 - \frac{\theta_{h,c,0}^2}{\theta_{w,0}^2}\); \(G\) and \(\xi\) – properties of the cooling channel after the heat transfer enhancer deployment.

Notice that the Darcy friction factors \(\xi\) and \(\xi_{0}\) represent average properties of smooth channel and the channel after the heat transfer enhancer deployment.

During the calculation of the wall temperature, acceptable comparison of the wall temperatures between smooth and not smooth channels possible only with equal mass flow. Then with \(G_{0} = G\) equation (8) obtaining form

\[
\theta_{w}^2 - \theta_{h,c}^2 = \left(\theta_{w,0}^2 - \theta_{h,c}^2\right) \frac{\xi_{0}}{\xi} ,
\]

(9)

which follows from relation (5).

Both relations (8) and (9) means, that length of the heat transfer enhancer and length of the cooling channel are equal to height of the active core. With considering equal of the mass flows, relation \(\frac{\xi_{0}}{\xi}\)

can replaced by relation \(\frac{\Delta P_{0}}{\Delta P}\) where \(\Delta P = P_{in} - P_{out}\), where index “out” related to heat carrier at the outlet of the active core. Obviously, this type of the heat transfer enhancer is cannot be optimal.

Considering that the long heat transfer enhancer, which has length \(l_{i}\), deployed on a section of the smooth cooling channel with maximum wall temperature. For the channel with heat transfer enhancer, which also has length \(l_{i}\), deployed from equation (9) after convert and in form for \(Z_{\text{max}}\) coordinate
\[ \tilde{\vartheta}_w^Z(Z_{\text{max}}) = \frac{1 + \gamma \cdot \frac{\delta T_{h,c}}{\tilde{\vartheta}_{w,0}(Z_{\text{max}})}}{1 + \gamma} \]  

(10)

where \( \gamma = \frac{H_0}{l_i} \left( \frac{\Delta P}{\Delta P_0} - 1 \right) \); \( \tilde{\vartheta}_w(Z_{\text{max}}) = T_w(Z_{\text{max}}) - T_{\text{int}}^{\text{in}}, \ \delta T_{h,c} = T_{h,c}(Z_{\text{max}}) - T_{\text{int}}^{\text{in}}, \ T_{\text{int}}^{\text{in}} - \) heat carrier temperature at the inlet of the section with deployed heat transfer enhancer at coordinate \( Z_{\text{int}} \), m.;

\[ T_{\text{int}}^{\text{in}} = T_{\text{in}} + \int_{Z_{\text{int}}}^{Z_w} q_i(Z) \text{d}Z. \]

For current active core the maximum temperature of the wall of the cooling channel situates in 8 fuel element. According to this statement, CFD simulation performed for following options of deployment and length of the heat transfer enhancers \( l_i = 0.793 \cdot m \):
- in fuel element №8: \( m = 1 \);
- in fuel element №7 and №8: \( m = 2 \);
- in fuel element №7, №7 and №9: \( m = 3 \);
- in all fuel elements: \( m = 10 \).

3. CFD simulation

CFD simulation were performed with ANSYS Fluent[4]. CFD simulation includes modelling of a gas flow in the smooth channel and in the channel with heat transfer enhancer for mass flows: \((G) 0.025 \text{ kg} \cdot \text{s}^{-1}, 0.035 \text{ kg} \cdot \text{s}^{-1} \) and \(0.055 \text{ kg} \cdot \text{s}^{-1} \). Heat flux at the wall of the cooling channels has a cosine distribution or piecewise linear with the same thermal power. Heat flux value in the center of the active core for the cosine distribution of heat flux \( q_f^0 = 23564.285 \text{ W} \cdot \text{m}^{-2} \). Heat transfer enhancer geometry and its properties are shown at the figure 1 [5].

![Heat transfer enhancer](image)

Figure 1. Heat transfer enhancer. Geometry properties: \( D = 15.88 \text{ mm}, \frac{t}{D} = \text{var}, \frac{d}{D} = \text{var} \).

Length of the section of the cooling channel with heat transfer enhancer \( = 0.793 \cdot m \), m.

During comparison of the maximum wall temperatures of the cooling channels calculated by (8) and results of the CFD simulation, acceptable geometry of the heat transfer enhancer, which deploys at full length of the cooling channel, were not found.

Comparison of the maximum wall temperatures of the cooling channels calculated by (10) with CFD simulation is shown below on the figures 2-5, where on ordinate axis the maximum wall temperatures of the cooling channels calculated by (10) represented, on abscissa axis results of CFD simulation are represented.
Studied options of the heat transfer enhancers: a) $\frac{t}{D} = 5$, $\frac{d}{D} = 0.9$, $m = 1$; b) $\frac{t}{D} = 0.5$, $\frac{d}{D} = 0.9$, $m = 1$; c) $\frac{t}{D} = 5$, $\frac{d}{D} = 0.93$, $m = 1$; d) $\frac{t}{D} = 5$, $\frac{d}{D} = 0.93$, $m = 2$; e) $\frac{t}{D} = 5$, $\frac{d}{D} = 0.93$, $m = 3$.

Figure 2. Comparison $T_w(Z_{\text{max}})$ values between values calculated by (10) and results of CFD simulation. Option of the heat transfer enhancer a). 1 - $\frac{q_0}{q_0^*} = 1$; 2 - $\frac{q_0}{q_0^*} = 1.2$; 3 – piecewise-linear distribution of the heat flux.

Figure 3. Comparison $T_w(Z_{\text{max}})$ values between values calculated by (10) and results of CFD simulation. Option of the heat transfer enhancer b). 1 - $\frac{q_0}{q_0^*} = 1$; 2 - $\frac{q_0}{q_0^*} = 1.2$; 3 – piecewise-linear distribution of the heat flux.
Figure 4. Comparison $T_w(Z_{\text{max}})$ values between values calculated by (10) and results of CFD simulation. Option of the heat transfer enhancer c). 1 - $\frac{q_0}{q_0'} = 1$; 2 - $\frac{q_0}{q_0'} = 1.2$; 3 – piecewise-linear distribution of the heat flux.

Figure 5. Comparison $T_w(Z_{\text{max}})$ values between values calculated by (10) and results of CFD simulation. 1 – Option of the heat transfer enhancer d); 2 – option of the heat transfer enhancer e).

As shown at figures 2-5, equation (10) provides good precision in calculating the maximum wall temperature of the cooling channel with known mass flow and pressure drop. Maximum wall temperature of the cooling channels, which provides by equation (10) can be used as a basic criteria in developing of the mass flow distribution in active core.

In addition, during calculations acceptable option of the heat transfer enhancer were found (option c), which allow to reduce maximum wall temperature by 20-45 °C.

4. Conclusion
According to basic statements heat and mass transfer the equation, which allows calculating maximum wall temperature of the cooling channel by known mass flow rate and Darcy friction factor were obtained.
Obtained equation takes into account influence of the heat transfer enhancer without calculating heat transfer coefficient. Also obtained equation can be very useful due to minimum information that must be known.

References

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