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# Thermal Hydraulic Analysis of the IAEA 10MW Benchmark Reactor under Normal Operating Condition

Hamed Djalal

***Abstract*—**The aim of this paper is to perform a thermal-hydraulic analysis of the IAEA 10 MW benchmark reactor solving analytically and numerically, by mean of the finite volume method, respectively the steady state and transient forced convection in rectangular narrow channel between two parallel MTR-type fuel plates, imposed under a cosine shape heat flux. A comparison between both solutions is presented to determine the minimal coolant velocity which can ensure a safe reactor core cooling, where the cladding temperature should not reach a specific safety limit 90 °C. For this purpose, a computer program is developed to determine the principal parameter related to the nuclear core safety, such as the temperature distribution in the fuel plate and in the coolant (light water) as a function of the inlet coolant velocity. Finally, a good agreement is noticed between the both analytical and numerical solutions, where the obtained results are displayed graphically.

***Keywords*—**Forced convection, friction factor pressure drop thermal hydraulic analysis, vertical heated rectangular channel.

## I. Introduction

To ensure the safe reactor operation and the integrity of the fuel plate during the normal operation of the nuclear reactor, the forced convection presents the most efficient cooling mode to evacuate the heat generated in the reactor core. In return, it requires an accurate prediction of the coolant inlet velocity which is considered the most important parameter on which all the heat removal processes from the nuclear core depend.

In the present study, both regimes of steady state and transient downward forced convection was considered, to perform a thermal hydraulic analysis of the 10 MW IAEA benchmark reactor. Analytical and numerical solutions by mean of the finite volume method are used for solving the both regimes of forced convection, in a narrow rectangular channel, imposed under a cosine shape heat flux and during normal operating conditions. For this framework, a computer program is developed and used to predict the temperature distribution in the coolant and in the fuel plate as a function of the inlet cooling velocity, while the obtained results from the solution of the both considered regimes are presented and favorably compared.

Hamed. Djalal is with the Nuclear Research Center of Draria, Algiers, Algeria (e-mail: homjal@ yahoo.fr).

## II. The Steady State Forced Convection Governing Equations

For the case of one-dimensional monophasic steady state downward fluid flow, the continuity, the momentum and the energy equations are respectively expressed by [7]:

$$\frac{d(\rho v)}{dz} = 0 \tag{1}$$

$$\frac{dP}{dz} = -\frac{f}{D_h}\frac{\rho v^2}{2} - \rho g - \frac{d(\rho\, v^2)}{dz} \tag{2}$$

$$\dot{m}\, cp\, \frac{dT}{dz} = q''\, P_h \tag{3}$$

where, $\rho(\mathrm{kg/m^3})$ , $v(m/s)$ and $P\,(Pa)$ are respectively the coolant density, velocity and pressure, while $f$ is the friction factor. $\dot{m}(k_g/s)$ and $P_h(m)$ are the coolant flow rate and the heated perimeter, $q''(w/m^2)$ is the surface heat flux transferred to the coolant by the nuclear fuel which is assumed to have a cosines form [1].

$$q''(z) = q_{max}\,.\,cos\left(\frac{\pi\,.z}{2.l_p}\right) \tag{4}$$

where, $l_p(m)$ is the half extrapolated length.

### *A. The Analytical Coolant Temperature*

After integrating (3) and also by taking into account (4), then we can express the variation of the coolant temperature as:

$$T(z) = Te + \frac{2\,P_h\;l_p}{\dot{m}\;cp\;\pi}\;\; q_{max}\left[sin\left(\frac{\pi\,.z}{2.l_p}\right) + \;sin\left(\frac{\pi\,.l}{4.l_p}\right)\;\right] \tag{5}$$

### *B. The Analytical Cladding Temperature*

The outer surface clad temperature is calculated by using the Newton law and taking into account (4). This leads to

$$T_{cl}(z) = \frac{q_{max}}{h} cos\left(\frac{\pi\, z}{2\; l_p}\right) + T(z) \tag{6}$$

where $h(w/m^2c^0)$ is the convective heat transfer coefficient is calculated by using the correlation of Dittus and Boelter [1], as:

$$\mathrm{N_u} = \frac{h\,D_h}{K_l} = \;0.0243\;\mathrm{Re}^{0.8}\;Pr^{0.4} \tag{7}$$

where, $K_l(w/m\ °C)$ is the thermal conductivity of the coolant.

### A. The Analytical Fuel Temperature

There are many methods to calculate the center fuel temperature. The simplest method is by using the electric analogy (Ohm law), so we can write

$$T_f(z) = T_{cl} + q_{\max} . \cos\left(\frac{\pi . z}{2.l_p}\right)\left(\frac{e_f}{k_f} + \frac{e_{cl}}{k_{cl}}\right) \quad (8)$$

where $e_f\ (m)$ and $e_{cl}(m)$ are respectively the half fuel and the cladding thicknesses, while $k_f(w/m\ °C)$ is the thermal conductivity of the nuclear fuel.

### B. The Analytical Coolant Pressure Drop

By integrating (2) between the inlet of channel and a position (z) along the channel, the total pressure drop is obtained as follows

$$P(z) - P_{in} = -\frac{f}{D_h}\frac{\rho(z)v(z)^2}{2}(z) - \rho(z)gz - [\rho(z)\, v(z)^2 - (\rho\, v^2)_{in}] \quad (9)$$

The first, second, and the third terms in the right hand side represent respectively the pressure drop by friction, gravity and acceleration.

### C. The Analytical Coolant Velocity

The coolant velocity is determined analytically from the continuity equation, where the coolant density is a function of the coolant temperature as:

$$\rho_{i+1}v_{i+1} = \rho_i v_i \quad (10)$$

## III. Transient Forced Convection Governing Equations

For the case of one-dimensional monophasic transient and downward forced convection, the continuity, the momentum and the energy equations are respectively expressed by the set of equations represented below [7].

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho v}{\partial z} = 0 \quad (11)$$

$$\rho\frac{\partial v}{\partial t} + \rho v\frac{\partial v}{\partial z} = \rho g - \frac{\partial p}{\partial z} - \frac{1}{2}f\frac{\rho v^2}{D_h} \quad (12)$$

$$\rho c_p\frac{\partial T}{\partial t} + \rho v c_p\frac{\partial T}{\partial z} = \frac{q'' p_h}{A} \quad (13)$$

To calculate the temperature distribution in the fuel and in the cladding, the heat transfer equation is used.

$$\rho c_p\frac{\partial T}{\partial t} = \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) \quad (14)$$

### A. The Numerical Coolant Temperature

For the temporal and axial coolant temperature distribution along the channel active length, the energy equation (13) is discretized by the finite volume method over a control volume ΔV=A dz dt, then the following algebraic equation is obtained

$$a_p T_p^{n+1} = a_W\, T_W + a_E\, T_E + b \quad (15)$$

where,

$$a_p = \frac{\rho_p c_{pp} A\Delta z}{\Delta t} + a_W + a_E$$
$$a_W = max[\,\dot{m}_w c_{pw}, 0]$$
$$a_E = max[\,-\dot{m}_e c_{pe}, 0]$$
$$b = q'' p_h \Delta z + \frac{\rho_p c_{pp} A\Delta z}{\Delta t} T_p^n$$

### B. The Numerical Cladding Temperature

The transient outer surface clad temperature is obtained throughout a combined operation, between the discretized heat equation (14) over the considered control volume with (6), then the following algebraic equation is obtained

$$a_P\, T_p^n + a_E T_E^n + a_W T_W^n = b \quad (16)$$

with,

$$a_p = \left(1 - \frac{(k_{cle} + k_{clw})\Delta t}{\rho_{cl} c_{cl} \Delta z^2}\right)$$
$$a_E = \frac{k_{cle}\Delta t}{\rho_{cl} c_{cl}\Delta z^2} \quad a_W = \frac{k_{clw}\Delta t}{\rho_{cl} c_{cl}\Delta z^2}$$
$$b = \frac{q''}{h} + T_{coolant}^{n+1}$$

### C. The Numerical Fuel Temperature

Also by the same way, the transient fuel temperature is carried out by a combined operation, between the discretized heat equation (14) by the finite volume method and over the considered control volume, with (8), then we got

$$a_P\, T_p^n + a_E T_E^n + a_W T_W^n = b \quad (17)$$

with,

$$a_p = \left(1 - \frac{(k_{fe} + k_{fw})\Delta t}{\rho_f c_f \Delta z^2}\right)$$
$$a_E = \frac{k_{fe}\Delta t}{\rho_f c_f \Delta z^2}$$
$$a_W = \frac{k_{fw}\Delta t}{\rho_f c_f \Delta z^2}$$
$$b = q''\left(\frac{e_f}{k_f} + \frac{e_{cl}}{k_{cl}}\right) + T_{cl}^n$$

### D. The Numerical Coolant Pressure Drop

The coolant pressure drop along the channel is obtained throughout the discretization of (12), by the finite volume method, then we got

$$a_P P_P = a_W P_W + b \quad (18)$$

where,

$$a_p = a_W = 1$$
$$b = -\frac{\Delta z}{\Delta t}(\rho_P^{n+1} V_P^{n+1} - \rho_P^n V_P^n) - (\rho_e V_e^2 - \rho_w V_W^2) - \rho g \Delta z - \frac{f_P \Delta z}{2D_h}(\rho_P V_P^2)$$

The issues by using a centered scheme for approximate, the control volume interfaces pressures, is that the numerical outlet coolant pressure is unchanged with the variation of the inlet coolant velocity as the analytical one in Fig. 2. This problem is remedied by using a quick scheme [6], where the control volume interfaces pressures are expressed as follows:

$$P_e = \theta\left[\frac{1}{2}(P_p + P_E)\right] + (1-\theta)\left[\frac{3}{2}P_p + \frac{1}{2}P_W\right]$$

$$P_w = \theta\left[\frac{1}{2}(P_p + P_W)\right] + (1-\theta)\left[\frac{3}{2}P_p + \frac{1}{2}P_E\right]$$

So, taking into account the two previous formulas, the coefficients of (18) are

$$a_p = a_W = 1$$

$$b = \frac{1}{2(\theta-0.5)}\left[-\frac{\Delta z}{\Delta t}(\rho_P^{n+1}V_P^{n+1} - \rho_P^n V_P^n) - (\rho_e V_e^2 - \rho_w V_W^2) - \rho g \Delta z - \frac{f_P}{2D_h}(\rho_P V_P^2)\,\Delta z\right]$$

The main advantage, by using a quick scheme, is that the value of the coefficient (θ) can be chosen in such way to force the numerical outlet coolant pressure to be as close as possible to the analytical one.

*E. The Numerical Coolant Velocity*

The coolant velocity along the channel is determined by the discretization of the continuity equation (11), by the finite volume method over a control volume ΔV=A dz dt. Then, the following algebraic equation is obtained.

$$a_P V_P = a_W V_W + b \tag{19}$$

where,

$$a_p = (3\rho_e - \rho_w)$$
$$a_W = (\rho_e + \rho_w)$$
$$b = -\frac{2\Delta z}{\Delta t}(\rho_P^{n+1} - \rho_P^n)$$

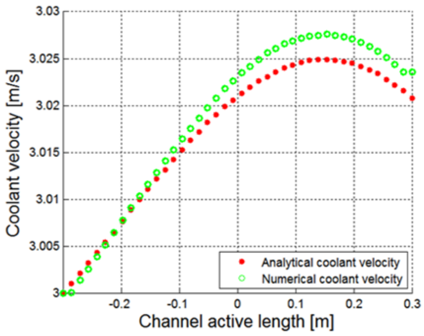


Fig. 1 A comparison between the analytical steady state and transient numerical coolant velocity

In Fig. 1, an example of the comparison between the both analytical and numerical coolant velocity distribution along the hot channel is shown.

## IV. The Friction Factor Correlations

To calculate the friction factor in the rectangular heated channel more accurately, two corrections are introduced; the first one is given by the factor ξ as follows [2]

$$f = \xi\ f_d \tag{20}$$

where, for water $\xi = \left[\frac{\mu_w}{\mu_b}\right]^{0.85}$ and $\mu_w$, $\mu_b$ are respectively the fluid dynamics viscosity for the temperature of the wall and the bulk temperature.

The Darcy friction factor $f_d$ is calculated according to the flow regime of the coolant where the three following cases are considered.

*A. The Laminar Fluid Flow*

For laminar flow, the correlation used is valid only for Reynolds number less than 2000, and $K_R$ represents the Reynolds correction for the non-circular channel [2].

$$f_d = \frac{64}{Re\,K_R} \tag{21}$$

where,

$$K_R = \frac{2}{3} + \frac{11}{24}\alpha(1-\alpha) \text{ and } \alpha = \frac{\text{the channel width}}{\text{the channel length}}$$

*B. The Transient Fluid Flow*

For the case of transient fluid flow where the Reynolds number varies between 2000 and 5000, the friction factor without taking into account the Reynolds correction for the non-circular channel, is evaluated by a linear interpolation as follows [3]

$$f_d = fl + \left(\frac{Re - 2000}{3000}\right)(f_t - f_l) \tag{22}$$

$fl$ is the friction factor for laminar flow for Reynolds number equal to 2000. $f_t$ is the friction factor for turbulent flow for Reynolds number equal to 5000.

*C. The Turbulent fluid flow*

For turbulent fluid flow in unheated channel and without taking into account the Reynolds correction for the non-circular channel, the correlation used is valid only for Reynolds number greater than 5000 [4].

$$\frac{2}{\sqrt{f_d}} = 1.7372\, ln\left(\frac{Re}{1.964\, ln(Re) - 3.8215}\right) \tag{23}$$

In Figs. 2 and 3, we showed the distribution of the total pressure drop in the channel with the variation of inlet coolant velocity.

## V. Results and Discussion

In the four figures below, we display the coolant and cladding temperature as a function of the coolant inlet velocity, in order to determine the minimal inlet coolant velocity, which can evacuate properly the nuclear fuel generated heat, always with the respect of the specific safety limit where the cladding temperature should not reach or exceed 90 °C. All the water properties are calculated as a function of pressure and temperature by the polynomial correlations of the work [5].

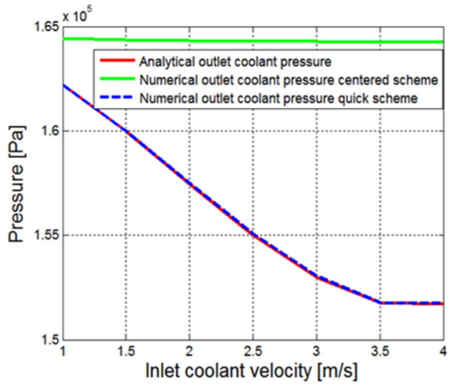


Fig. 2 The outlet coolant pressure drop along the hot channel

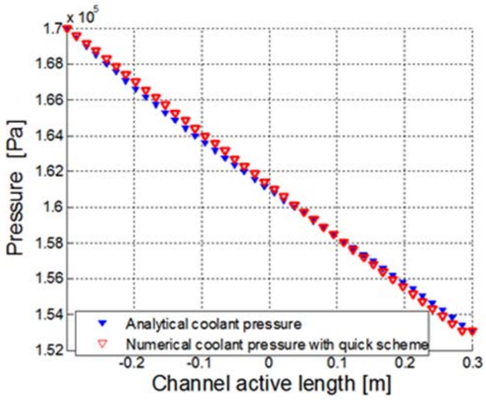


Fig. 3 The coolant pressure drop along the hot channel

### *A. The e Both Regimes of Forced Convection Respectively for $v_{in} = 1 - 3\ (m/s)$*

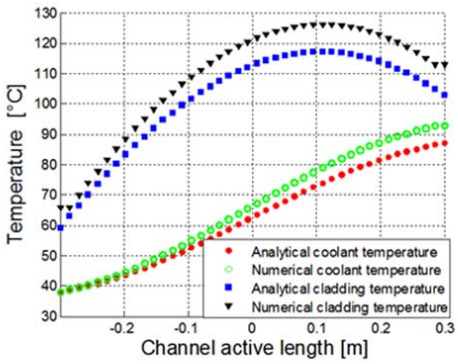


Fig. 4 The coolant and cladding temperatures variations along the hot channel

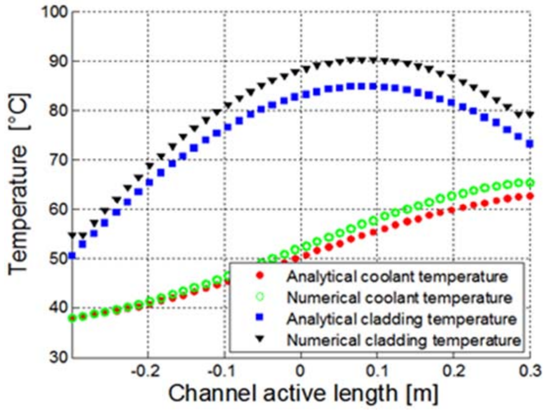


Fig. 5 The coolant and cladding temperatures variations along the hot channel

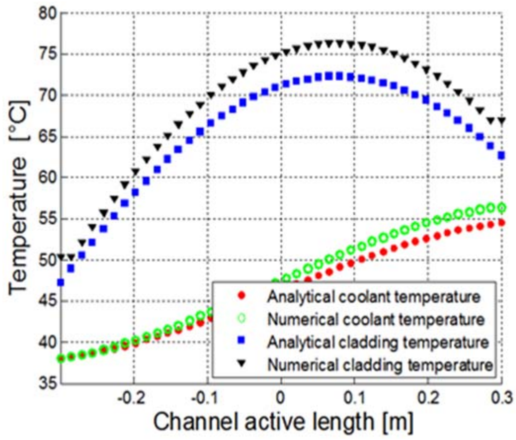


Fig. 6 The coolant and cladding temperatures variations along the hot channel

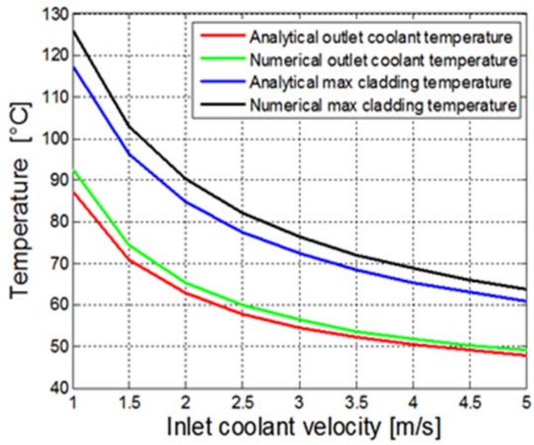


Fig. 7 The outlet coolant and the max cladding temperatures as a function of the inlet coolant velocity

## VI. Conclusions

The main goal of this study is to determine the minimal coolant velocity to evacuate properly the thermal heat generated in the reactor core, without reach a critical state, where the cladding temperature must stay below a specific safety limit 90 °C. For this purpose, a computer program is developed to calculate and determine the coolant and cladding temperatures distributions and the pressure drop in a hot channel of nuclear fuel element. Then, the obtained results of the both solutions are compared to each other for different cases of the inlet coolant velocity.

The interest conclusions are summarized as follows: From the obtained results, we note that the velocity of the coolant must be greater than or equal to 2.5 m/s, to ensure that the cladding temperature does not reach or exceed the considered safety limit.

We noticed also that the analytical steady state and the numerical transient solutions of the forced convection give almost the same results when the coolant and the cladding temperatures difference does not exceed respectively 3 °C and 5 °C.

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## References

[1] H. Elkhatib, "Modeling and simulation of loss of the ultimate heat sink in a typical material testing reactor,'' Annals of nuclear energy, vol. 51, 2012, pp. 156-166.

[2] K. Ardaneh, "An analytical solution for thermal–hydraulic analysis and safety margins in MTR-type research reactors cooled by natural convection," Annals of nuclear energy, vol. 51, 2013, pp. 282-288.

[3] INVAP S.E, "Caudvap V 3.60, MTR PC user's manual", 2012.

[4] T. M. M. A. Elmaaty, "Natural convection cooling for LEU irradiated fuel plates," Annals of nuclear energy, vol. 40, 2012, pp. 116-121.

[5] Pascal Pezzani, "Propriétés Thermodynamiques de l'eau," Techniques de l'Ingénieur, 1992, w120.

[6] B. P. Leonard, ''Ultra-Sharp Nonoscillatory Convection Schemes for High-Speed Steady Multidimensional Flow, ''NASA TM 1-2568 (ICOMP-90-12), NASA Lewis Research Center, 1990.

[7] N. E. Todreas, "Nuclear Systems I," Hemisphere Publishing Corporation, Washington, 1990.