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Microchannel System Simulation for a Neutron Production Target to be used in the Boron Neutron Capture Therapy

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Abstract

Microchannel system simulations have been performed in this work based on the thermal resistance model developed by Phillips (1988) and followed by Zhimin (1997). As it appears in the bibliography [Phillips (1988), Zhimin et al. (1997), Ludewigt et al. (1997), Tuckerman et al. (1981), Knight et al. (1992)], this system should be capable of removing 1 kW/cm^2, for this reason it is being taken into account for the design of a neutron production target with accelerators.
Based on the previously mentioned model, two codes were written in MATLAB programming platform, that solve the equations of such model in an iterative way under constant pressure between the microchannel ends and constant pumping power, respectively. The codes calculate the total thermal resistance (and its components) of the microchannel system for a given geometry, in addition to the hydrodynamic variables and the system adimentional numbers (Re and Nu).
The codes report total thermal resistance values and adimentional numbers comparable to those reported in different publications [Phillips (1988), Zhimin et al. (1997), Ludewigt et al. (1997)], discrepancies lower than 10%have been found. For this reason it is being considered to design a heat sink that drains the energy depositedby the accelerator's incident beam.

Keywords: Microchannels, BNCT, Simulations.

1. Introduction

This work is part of the development of the Accelerator Bases Boron Neutron Capture Therapy (AB-BNCT), currently ongoing at the National Atomic Energy Commission. Within this project, one of the challenges is to design and build a target, capable of producing thermal neutrons from a high current incident beam (30mA) to irradiate previously boron doped tumors.

This target withstands high mechanical and thermal solicitations and radiation damage. One of the requirements of this designis that it must drain the energy that is deposited by an accelerator beam on a neutron production target. Because the energy density to be drained is approximately 1kW/cm^{2}therefore it is necessary to use a highly effective cooling technique. For this purpose we have chosen to use the microchannel technique, such technique was proposed for the first time by Tuckerman &Pace.

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The possibility to drain power densities in the order of 1 kW/cm^2 performing channels in the micrometric scale was studied in that work.

In a later work, Phillips (1988) developed a thermal resistance model to evaluate the performance of a microchannel heat sink, both turbulent and laminar fluid flow regime, reporting that it is possible to obtain thermal resistances lower than 0.1 C/W/cm^2.

In this work, two codes were written using the thermal resistance model for constant pumping power and pressure drop at the microchannel ends, both under turbulent flow. An exhaustive comparison was made using the geometries and flow conditions reported in the literature [Phillips (1988), Ludewigt et al. (1997),Knight et al. (1992)], discrepancies lower than 10% were found in every studied case. Later the codes were used to evaluate the proposed target designs.

2. Thermal resistance model

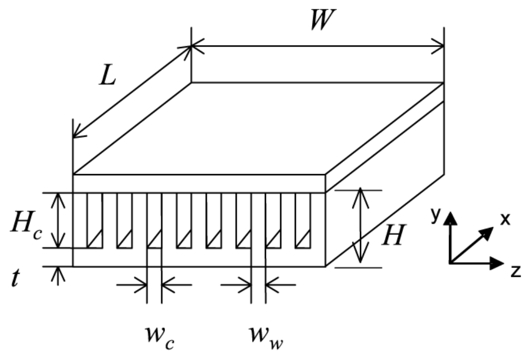

Fig. 1. Microchannel system scheme.

In Fig 1. A geometry scheme of a microchannel system can be seen, where L, W and H are the length, height, and total width of the heat sink respectively, t is the base width, Wc, Ww and Hc are the channel width, the walls width and the channels height respectively. The heat Q is injected by the underside of the heatsink and then drained by the circulating fluid inside the microchannels. According to the model, the top (or upper side) is made of an insulating material.

The total thermal resistance model of the system is given by the equation:

$$R_{tot} = \frac{T_{\sup,\max} - T_{in}}{Q} \tag{1}$$

Where Rtot is the system total thermal resistance, Tsup,max and Tin are the maximum temperature on the surface where the heat is injected and fluid inlet temperature respectively and Q is the injected heat.

In turn, the total thermal resistance can be expressed as the sum of different thermal resistance contributions generated by the thermal circuit that generates the heat transport in the heat sink. In the following equation, different contributions of the system total thermal resistance are being taken into account.

$$R_{tot} = R_{bulk} + R_{conv} + R_{constr} + R_{cond}, \tag{2}$$

whereRbulk is the thermal resistance between the entrance and the exit of the channel and it is a measure of how the thermal energy is being transported by the fluid inside the channel, Rcond is the thermal resistance due to energy transfer.

To simplify the equations that correspond to the thermal resistance contributions, it isappropriate to define the following magnitudes:

$$\alpha = \frac{Hc}{Wc} \tag{3}$$

$$\beta = \frac{Ww}{Wc}, \tag{4}$$

whereα is the channel aspect ratio and β is the ratio between the channel width and the channel wall width.

Rbulk resistance is defined by the equation:

$$R_{tot} = \frac{T_{sup,max} - T_{in}}{Q} = \frac{1}{\rho f Cpf G}, \tag{5}$$

whereρf is the fluid density, Cpf is the fluid heat capacity and G is the volumetric flow defined by the equation:

$$G = nH_c W_c V, \tag{6}$$

in which n is the number of channels of the heat sink and V is the average speed of the fluid inside the channel. It is appropriate to introduce the hydraulic diameter D given by:

$$D = \frac{4H_c W_c}{2(H_c + W_c)} = \frac{2\alpha}{1+\alpha} W_c, \tag{7}$$

and the Reynolds Number, Re,

$$\mathrm{Re} = \frac{\rho f V D}{\mu f}, \tag{8}$$

whereμf is the fluid viscosity. Substituting equations 6, 7 and 8 in 5 and operating is obtained:

$$R_{bulk} = \frac{2L(1+\beta)}{Cpf \rho f \, \mathrm{Re}(1+\alpha)(WL)} \tag{9}$$

The convective thermal resistance is defined by:

$$R_{conv} = \frac{1}{hA_{eff}}, \tag{10}$$

where h is the convective heat transmission coefficient and Aeff is the total effective area available to the convective heat transmission coefficient.

$$A_{eff} = nL(W_c + 2\eta H_c), \tag{11}$$

on which the upper surface is considered thermally isolated. The fin efficiency ηis given by the equation:

$$\eta = \frac{\tanh(mH_c)}{mH_c}, \tag{12}$$

with m defined as:

$$m = \sqrt{\frac{2h}{k_s W_w}}, \tag{13}$$

where Ks is the heat sink thermal conductivity.

The convective heat transfer coefficient h, previously introduced, is obtained from the definition of the Nusselt Number, Nu, given by:

$$Nu = \frac{hD}{K_f}, \tag{14}$$

In turn, the Nusselt number for turbulent flow (Re>2300) can be calculated using [Gnielinski (1976)]:

$$Nu = 0.012[1+(\frac{D}{L})^{2/3}](\mathrm{Re}^{0.87}-280)\,\mathrm{Pr}^{0.4}, \tag{15}$$

wherePr=μfCpf/Kf is the Prandtl number, which should be valuated between 1.5 and 500for the equation 15 to be valid.
Finally, replacing 11 and 14 in 10 gives:

$$R_{conv} = \frac{2\alpha(1+\beta)}{(1+\alpha)(1+2\alpha\eta)KfNu(WL)}W_c. \tag{16}$$

The thermal resistance Rconstr is obtained from the equation [Phillips (1988)]:

$$R_{constr} = \frac{-(1+\beta)}{\pi Ks(WL)}\ln[\sin(\frac{\pi\beta}{2(1+\beta)})]W_c. \tag{17}$$

Finally, the conduction thermal resistance Rcond, is given by:

$$R_{cond} = \frac{t}{Ks(WL)}. \tag{18}$$

Is interesting to note that because each term of the thermal resistances are divided by the heat sink total area (WL), the total thermal resistance can be redefined by:

$$R'_{cond} = \frac{\Delta T}{\frac{Q}{WL}}. \tag{19}$$

where Q/WL is the energy density injected on the heat sink.

Finally, the system total thermal resistance is given by the sum of the equations 9,16, 17 and 18. And multiplying each term by the heat sink total area (WL) is independent of the sink area.

To obtain the average speed of the fluid one has:

$$\Delta P = 4f\frac{L}{D}(\frac{\rho V^2}{2}), \tag{20}$$

in the case of constant pressure drop in the microchannel. Where f is the friction factor for turbulent flow given by the ecuation [Phillips (1988)],

$$f = A(\mathrm{Re}_{eq})^{B}$$
$$A = 0.00929 + \frac{1.01612}{\frac{x}{D}}$$
$$B = -0.268 - \frac{0.31930}{\frac{x}{D}} \qquad (21)$$
$$\mathrm{Re}_{eq} = \frac{\rho f V Deq}{\mu f}$$
$$Deq = [\frac{2}{3} + \frac{11}{24\alpha}(2 - \frac{1}{\alpha})]D.$$

For the case in which the pumping power (Pow) is considered constant, one has the equation,

$$Pow\Delta PG, \qquad (22)$$

where the volumetric flow G is given by equation 6.

Once the fluid properties and the problem geometry are fixed, MATLAB codes find the fluid average speed V by using equations 20 and 22 accordingly. Once the average speed V is known, Re, Nu and F are calculated to finally obtain the total thermal resistance of the system.

3. Numerical validation of the thermal resistance model

Geometrical values, fluid properties and hydrodynamic conditions from different systems studied in the literature [Phillips (1988), Ludewigt et al. (1997), Knight et al. (1992)] were used to calculate the thermal resistance (and its contributions), the Reynolds and Nusselt numbers in order to perform a numerical validation of the written code.

Regarding the published data by Phillips (1988), we reconstructed the geometry corresponding to Fig. 1 with WC= 300 μm, α= 4, β= 1 and V= 11.59 m/s.

The values found for this case are as follow: Rtot= 0.0822 K/(W/cm2), Re= 6230 and Nu= 48.52 which represents a discrepancy lower than 7%, 1% and 14% respectively.

The discrepancy within the reported values to the Nusselt number is because in this work Nu was calculated in the worst case scenario inside the channels (x=L in equation 15), while in the literature [Phillips (1988)], Nu was calculated to infinity (that is, by letting the length of the channel to infinity), which is a more conservative hypothesis (although not realistic) and it impacts on the total thermal resistance in a way to provide an upper bond.

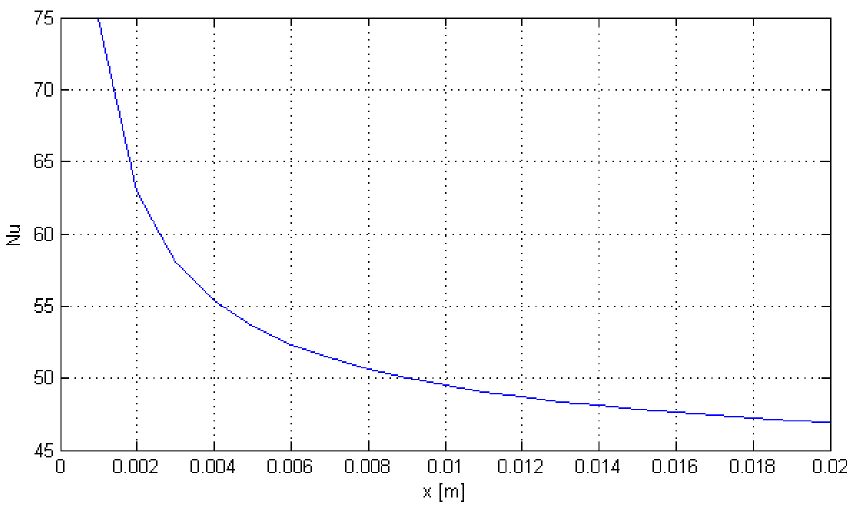

Fig. 2. Nusselt Number (Nu) as a function of the channel position (x).

As it can be seen in Fig. 2, the Nusselt Number shows a monotone decreasing behavior as a function of x (position inside of the channel), and Rconv (equation 16) a monotone decreasing behavior as a function of the Nusselt Number, as it can be seen in Fig. 3, giving sustenance to the previously developed.

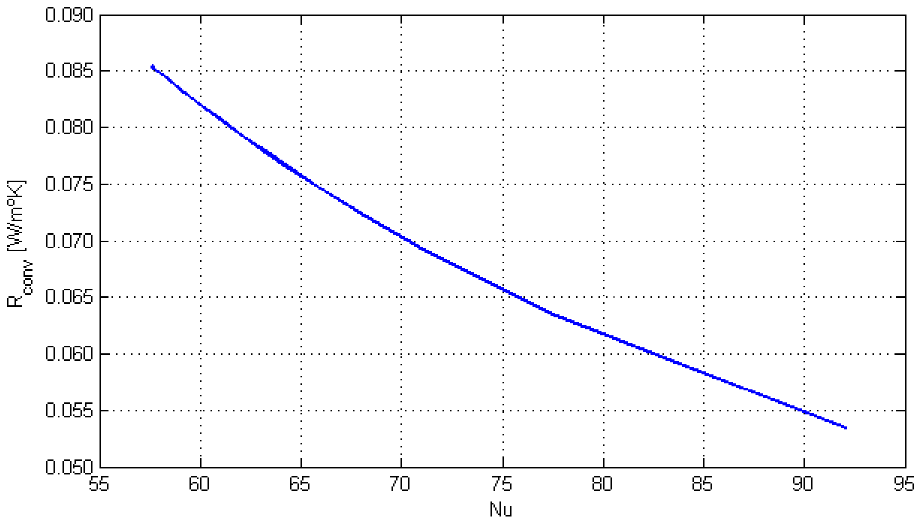

Fig. 3. Convective thermal resistance (Rconv) as a function of Nusselt Number (Nu).

Only some of the geometries developed on Phillips (1988) work were used. All of them where reconstructed and calculated with the codes written in this work, discrepancies between reported and calculated were, in all cases, in the order of the above mentioned.

Regarding geometries and hydrodynamic conditions published by Ludewigt et al. (1997) a temperature variation between the entrant fluid and the target area in the order of 100K was calculated by using the written codes. Such value is consistent with that reported by the simulations made in the cited work.

Also, we reconstructed the geometry and the hydrodynamic conditions published by R. Knight et al.(1992) with Hc=365 µm, Wc=377 µm, W=L= 1cm, G=59,2 cm3/s. In that work a total thermal resistance Rtot = 0.056 K/(W/cm2)and a Reynolds Number Re = 8459 is reported. By means of the use of the codes developed in this work, we obtained a total thermal resistance Rtot = 0.0579 k/(W/cm2)and a Reynolds Number Re = 8247 which represent a discrepancy between the reported values and the calculated values of 4% and 2.5% respectively. The Nusselt Number in the cited work is 85.6, while that obtained from the calculation of the model presented here is 6.39, and represents a discrepancy of 38%, which is due to the fact that the correlations used in both papers are different. Even so, the values of total thermal resistance that were obtained are comparable.

Analysis of the geometry proposed for the Neutron Production Target

Many different geometrical microchannel configurations were evaluated by using the code written and presented in this paper, to be employed in a prototype of the Neutron Production Target, whose basic outline is present in Fig. 4.

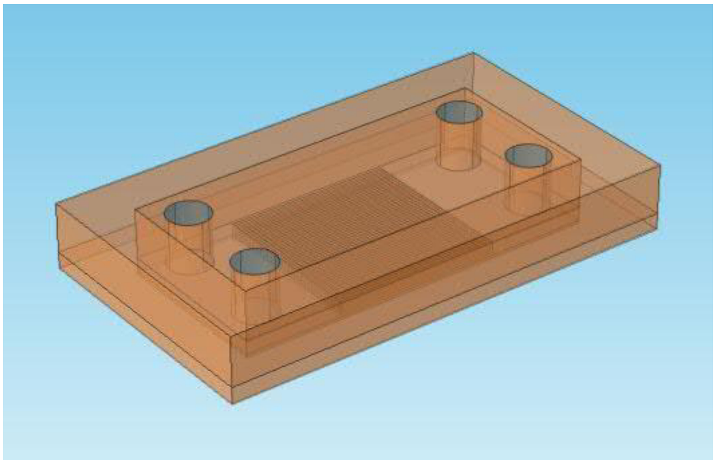

Fig. 4. Proposed validation prototype outline.

From this evaluation, an analysis for candidate heat sink geometry is presented. The chosen geometry is given by the following dimensions: Wc = 500 μm, $\alpha = 4$, $\beta = 1$, t = 1mmL = W = 10cm y Ks = 400, where Ks is the substrate thermal conductivity, in this case, copper. Fluid mechanics in turbulent flow regime and fluid structure interaction simulations were made from this geometry by using COMSOL Multiphysics computer simulation program.

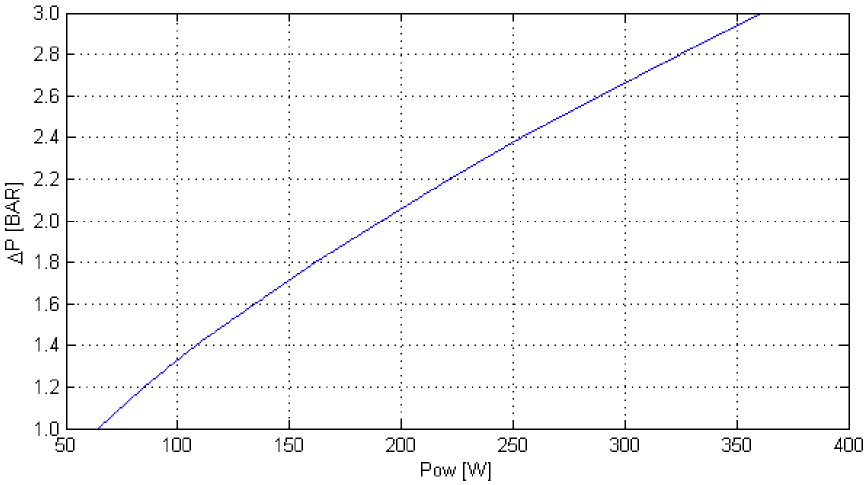

Fig. 5. Microchannel pressure drop as a function of the pump power.

After an analysis of the results, in Figure 5 we can see that with pump power lower than 400W, pressure drops of 3 bar are achieved at the ends of the microchannels. So, if you take into account the entire water circuit,1HP pump pressure would be sufficient to reach pressures of at least 2 bar (depending on the quality of the water pump) at the ends of the microchannels.

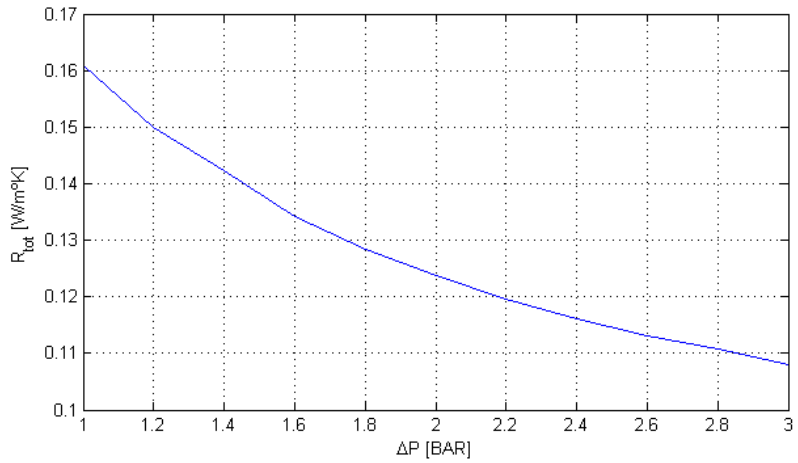

Fig. 6. Microchannel system total thermal resistance as a function of the pressure drop at the ends thereof.

The total thermal resistance reached at ΔP = 2 bar is 0.124 K/(W/cm2), for which reason at that pressure drop the temperature difference between the entering water and the place with the highest temperature of the heat sink would be 124 K if an energy density of 1 kW/cm2 is considered. Also, as it is shown in Fig. 6, the total thermal resistance of the microchannel system could be reduced to 0.108 K/(W/cm2) at a pressure drop of 3 bar giving a ΔT=108K under the same energy density conditions previously mentioned.

It’s interesting to note, as it is shown in Fig. 7, that throughout the range of pressures previously mentioned, the Reynolds Number is over 2300, thus maintaining the condition of turbulent flow.

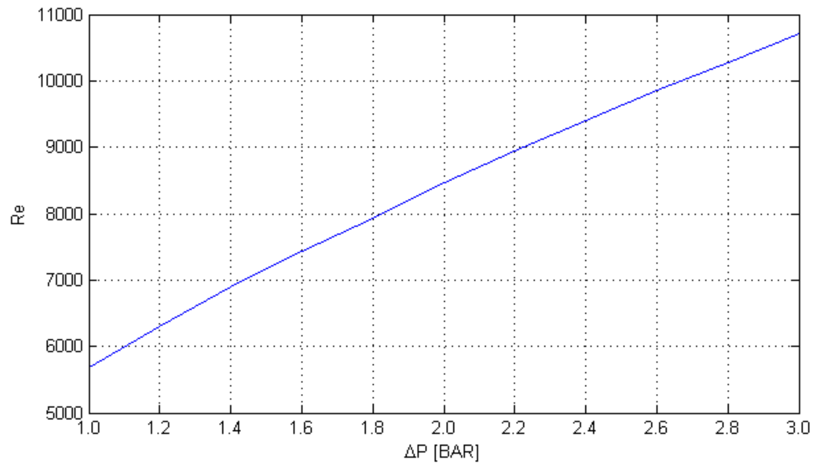

Fig. 7. Reynolds Number as a function of the pressure drop between the ends of the microchannels.

The required flow for such geometrical configuration as a function of the pressure drop between the ends of the microchannels,is shown in Fig. 8, where a 56.45 l/min flow is required at 2 bar. To see if both the microchannels structure and the heat sink base withstand the fluid pressures, fluid structure interaction simulations were made to evaluate the material displacement induced by the circulating fluid. For this, turbulent flow simulations were previously performed to achieve a pressure drop of 2 bar at the ends of the microchannels. As it is shown in Fig. 9, the fluid enters the heat sink and generates a maximum pressure of4.8 bar in the base.

In figure 10, the fluid pressure generates a maximum deformation of 1mm in the inlet base of the fluid without notably affecting the microchannels.

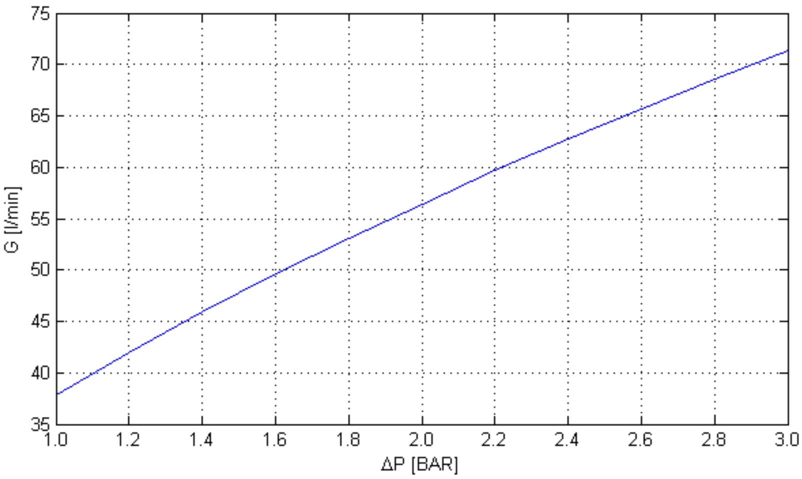

Fig. 8. Fluid flow as a function of the pressure drop at the ends of the microchannels.

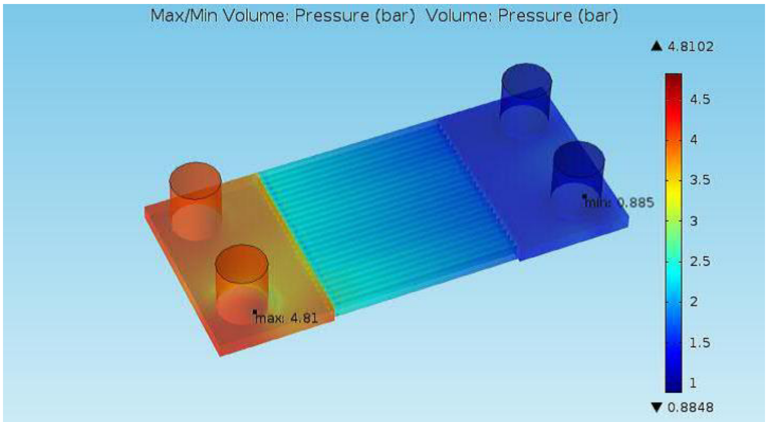

Fig. 9. System fluid pressure.

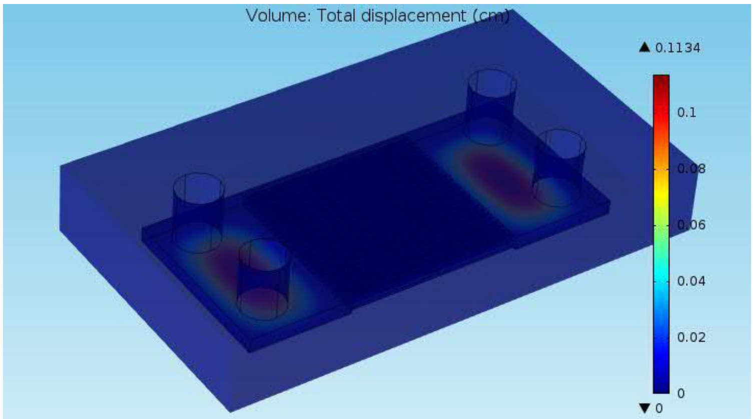

Fig. 10. Total system displacement due to fluid-struture interaction.

This analysis shows us that it would be feasible to build a heat sink by using the microchannel technique that is able to drain the required energy density with a conventional system, namely, a water pump capable of delivering 60 l/min at 3 bar. Besides, by using the simulations made in COMSOL Multiphysics, it can be seenthat the fluid pressure does not cause significant microchannel deformation.

4. Conclusions

In this paper we have presented the thermal resistance model proposed by Phillips (1988) and continued by Zhimin(1997). From that model MATLAB codes were written; such codes calculate the thermal resistance (and its components), adimentional numbers Reynolds and Nusselt and the hydrodynamic variables of the system in turbulent flow regime under conditions of constant pressure drop at the ends of the microchannels and constant pumping power respectively.

Comparisons of systems published in the literature were made[Phillips (1988), Ludewigt et al. (1997), Knight et al. (1992)], and in every case values comparable to published data were found within a discrepancy lower than 7% for the total thermal resistance and discrepancies lower than 4% and 14% for Reynolds and Nusselt, respectively.

Having done a numerical validation with published data, many different microchannel geometrical configurations were evaluated to be employed in a Neutron production target prototype and the analysis of a particular geometry was presented which meets the AB-BNCT Neutron production target thermal requirements (to drain 1 kW/cm2).

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