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TOPOLOGY OPTIMIZATION OF A SIMPLE CONJUGATE FLUID HEAT SYSTEM WITH HEAT SOURCES AND CONVECTIVE BOUNDARY USING ADJOINT METHOD

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Abstract. *In this study, we research the topology optimization of a simple conjugate fluid heat system with heat source and convective boundary. The goal of the optimization process is to make as possible the temperature uniform in a specific region of the domain while constraining the power dissipation across the formed channels. Under the assumption of laminar steady flow, the fluid is modeled by an incompressible Navier-Stokes-Brinkmann equation and the heat transfer is modeled by a convection-diffusion equation, with heat sources and convection boundary conditions. A solver using the Finite Volume Method is built-in open-source CFD package OpenFOAM to solve the governing equations and as an optimization algorithm, we use the method of moving asymptotes (MMA). The continuous adjoint formulation is employed for the sensitivity analysis. The developed tool, allows us to present results in two dimensions with different conjugate fluid heat system configurations and operating conditions. The optimized design shows how the channels are set up, and the fluid passes either adjacent to or near the hottest regions, aiming to achieve a uniform temperature in a specific region of the domain as specified by the objective function. The different cases show the performance of the method by applying it to the optimal design with little computational effort.*

Keywords: *Topology optimization, continuous adjoint method, heat exchanger, OpenFOAM.*

1. INTRODUCTION

Heat exchangers are widely used as fluid cooling devices for removing high heat flux. In this respect, an appropriate design of cooling channels may increase the efficiency of the systems and could guarantee the safeness of the products. Currently the CFD-based Adjoint method remains as a suitable approach for optimization in thermal-fluid dynamics. In topology optimization the entire domain is used to compute whether a fluid cell is favourable or counterproductive for the flow, such that, a desired cost function is minimized. Topology optimization has its roots in structural design optimization and in last decades compact Matlab implementation codes for compliance minimization of statically loaded structures were presented (Sigmund, 2001; Andreassen *et al.*, 2011), also using isoparametric polygonal elements (Talischi *et al.*, 2012) and in three-dimensional domains (Liu and Tovar, 2014). Fanni *et al.* (2013) performed a comparison between different topology optimization algorithms, such as Method of Moving Asymptotes (MMA), Sequential Quadratic Programming (SQP), Optimality Criteria (OC), and Hybrid Cellular Automata (HCA).The Method of Moving Asymptotes lets good control of convergence, stability and speed of the optimization process. For heat conduction systems, topology optimization has been found to provide unconventional tree-like optimal structures of high conductivity material for efficient heat transfer (Subramaniam *et al.*, 2018), e.g. in housing of electrical projector units (Menge *et al.*, 2018). In fluid flow problems, Borrvall and Petersson (2003) introduced the concept of topology optimization for Stokes flow by adding a Brinkman penalization sink term into the momentum equation, with the objective of minimize the dissipated power in the fluid. Then, it was first extended to the Navier-Stokes equations by Gersborg-Hansen *et al.* (2005). Later, Othmer (2008) derived the adjoint equations and the boundary conditions, on the basis of a continuous adjoint formulation, demonstrating the versatility of the approach with respect to changes in the objective function. Since then, several studies have been developed e.g. parallel optimization framework in C++ using MMA (Aage and Lazarov, 2013), implementation of the level set method for the representation and the evolution of surfaces (interface between solid and fluid) (Karpouzas and Villiers, 2014), design of radial flow machine rotors operating on laminar regime (Luís Fernando, 2016), design of channels for non-Newtonian fluids (Kian, 2017), developments for unsteady (Skaar, 2017) and turbulence flows (Dilgen *et al.*, 2018). The main advantage of topology optimization is that the design modifications are accomplished without changing the mesh, saving a lot of computational effort. On the other hand an inherent feature of topology optimization is the ragged surface of the resulting geometries. Hence, starting with topology optimization and then switching to shape optimization is in many cases the most efficient way to solve optimizations problem (Ruberto, 2017). In recent research, Karpouzas (2019) combined shape and topology optimisation methods in a way that retains the strengths of both methods.

In coupled thermal-fluid systems, complex channels, fractals, ribs, and series of fluid splitters are examples of the structures present in the final geometries, as a result it still is a complex subject of study (Pietropaoli *et al.*, 2019). With the introduction of additive manufacturing technique, relatively complex geometries can be produced and the possibility to increase structural complexity and efficiency of mechanical components is expected to have a relevant impact in coolant system technology. In this sense, a lot of researches have been developed in this topic, e.g. for application in micro channel devices at low Reynolds number (Dede, 2009; Koga, 2010; Yoon and Koo, 2019), in automotive exahust systems (Hinterberger and Olesen, 2011), in turbines to minimize stagnation pressure dissipation while maximizing heat transfer between fluid and solid region (Pietropaoli *et al.*, 2019), in forced convection flow inside cooling channels (Goeke and Wünsch, 2017), in natural convection heat transfer process inside rectangular cavity (Ruberto, 2017) and in unsteady flows (Kavvadias *et al.*, 2015). Subramaniam *et al.* (2019) worked with multi-objective function combining pressure drop reduction and thermal power maximization in incompressible flows at low to moderate Reynolds numbers. The two objectives were combined linearly using weighted sum method and several optimal designs were generated for different combination of these weighting factors. Yu *et al.* (2020) researched thermal-fluid-structural problems to design a three-dimensional heat sink with load-carrying capability in parallel solver, moreover the choice of Darcy number was investigated because it was observed that even if the Darcy number is low enough to punish the velocity close to zero, it cannot eliminate the thermal convection term in the solid area. Ultimately, Chen *et al.* (2020) implemented topology optimization to design highly conductive fins in a multi-tube thermochemical heat storage unit.

The latest work in topology optimization of thermal fluid systems, considers the heat source in the whole domain. The design of heat exchangers using topology optimization by considering heat sources acting in specific regions of the domain is a novel contribution from this work.

2. RESEARCH METHOD

In this case study we are going to consider two heat transfer systems. The first system with internal heat sources and the second with heat transfer by convection in the walls.The sketch of geometry domain including the definitions of boundaries and dimensions details for both cases are shown in Fig. 1 and Fig. 2 respectively. Let Ω denote the flow domain for two study cases, Γ denote its boundary. The inflow and outflow part of the boundaries are denoted by Γ_i and Γ_o, respectively, Γ_ω is the wall boundary and Γ_{sym} is the symmetry boundary.

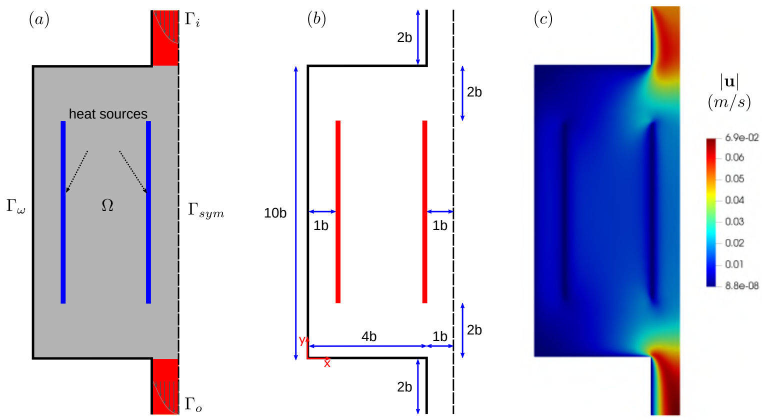

Figure 1. **Heat line sources case. Sketch of geometry domain including the definitions of (a) boundaries and (b) dimensions details. (c) Initial velocity field with $\gamma = 0.4$ uniform in the whole domain.**

To start, let's simplify the flow model without major losses of understanding by making the following basic assumptions:

- Steady state simulation, at low Reynolds numbers, which means flows in laminar regime, commonly found in micro-devices and enclosed environments, and where convection and diffusion are the main transport mechanisms.
- The fluid is Newtonian, which constitutes the rate of shearing stress of the fluid is linearly related to the angular deformation.
- The fluid is incompressible such that the fluid density is constant.
- Constant physical properties.

The system of governing equations in steady state form are given by:

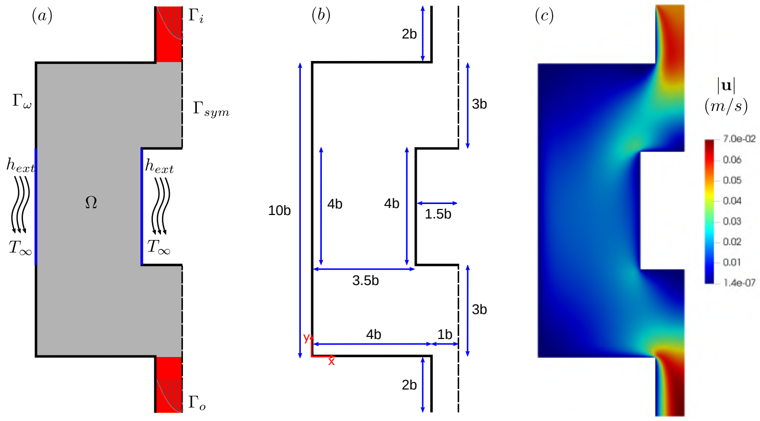

Figure 2. **Convective heat flux case. Sketch of geometry domain including the definitions of (a) boundaries and (b) dimensions details.(c) Initial velocity field with $\gamma = 0.4$ uniform in the whole domain.**

$$-\nabla \cdot (2\nu \mathrm{D}(\boldsymbol{u})) + (\boldsymbol{u} \cdot \nabla)\boldsymbol{u} + \frac{\nabla p}{\rho} - \alpha \boldsymbol{u} = 0. \tag{1}$$

$$\nabla \cdot \boldsymbol{u} = 0. \tag{2}$$

$$-\nabla \cdot (D\nabla T) + \boldsymbol{u} \cdot \nabla T - f_{\Omega_a} = 0. \tag{3}$$

where $\boldsymbol{u}, p$ and T stand for velocity, pressure and temperature, state variables respectively. f_{Ω_a} stands for temperature dependent sources defined in the sub domain Ω_a, $\nu = \mu/\rho$ is the kinematic viscosity, μ is the dynamic viscosity (sometimes also called the absolute viscosity), ρ is the density, $D = k/(\rho Cp)$ is the diffusion coefficient, where k and Cp are the conductivity and specific heat respectively, α is the inverse of the local permeability (Brinkman penalization approach (Khadra *et al.*, 2000)), which allows to differentiate low or high permeability areas. In the fluid region, α is equal to zero, and it signifies that no artificial friction force is added. In the solid region, α is equal to the relatively large value of the order $O(10^5)$ to set the velocity to near zero. The rate of strain tensor is denoted by $\mathrm{D}(\boldsymbol{u}) = 1/2(\nabla \boldsymbol{u} + \nabla^{\mathsf{T}} \boldsymbol{u})$.

For the case of Fig. 1, we utilise the following set of boundary conditions:

$$\boldsymbol{u}|_{\Gamma_i} = \boldsymbol{u}_i, \qquad \boldsymbol{n} \cdot \nabla p|_{\Gamma_i} = 0 \quad \text{and} \quad T|_{\Gamma_i} = T_i. \tag{4}$$

$$\boldsymbol{u}|_{\Gamma_\omega} = 0, \qquad \boldsymbol{n} \cdot \nabla p|_{\Gamma_\omega} = 0 \quad \text{and} \quad \boldsymbol{n} \cdot D\nabla T|_{\Gamma_\omega} = 0. \tag{5}$$

$$\nu(\nabla \boldsymbol{u}) \cdot \boldsymbol{n} - \frac{1}{\rho} p\, \boldsymbol{n}|_{\Gamma_o} = 0 \quad \text{and} \quad \boldsymbol{n} \cdot D\nabla T|_{\Gamma_o} = 0. \tag{6}$$

For the case of Fig. 2, we consider the convective boundary condition on part of blue wall:

$$\boldsymbol{n} \cdot k\nabla T|_{\Gamma_\omega} = h_{ext}(T - T_\infty). \tag{7}$$

where $\boldsymbol{n}$ is the outward unit normal, T_∞ is the external temperature and h_{ext} is the convective heat transfer coefficient.

2.1 Continuous adjoint method

Let's assume that the objective functional is defined as $\mathcal{J} = \mathcal{J}(\boldsymbol{u}, p, T, \gamma)$, and the system of governing equations by $\mathbf{R}(\boldsymbol{u}, p, T, \gamma) = 0$, where γ represents the design variable, i.e. the material distribution. The adjoint equations are derived by introducing a Lagrange function $\mathcal{L}$ and reformulating the cost function as:

$$\mathcal{L}(\boldsymbol{u}, p, T, \gamma, \boldsymbol{v}, q, \varphi) = \mathcal{J}(\boldsymbol{u}, p, T, \gamma) - \int_{\Omega} (\boldsymbol{v}, q, \varphi)^{\mathsf{T}} \boldsymbol{R}(\boldsymbol{u}, p, T, \gamma) \, \mathrm{d}\Omega. \tag{8}$$

where $\boldsymbol{v}$, q and φ are Lagrange multipliers that are used to enforce the residuals $\boldsymbol{R}$ (the constraints equations). Through the introduction of Lagrange multipliers, the constrained optimization problem is converted into an unconstrained problem:

find $\boldsymbol{u}, p, T, \gamma, \boldsymbol{v}, q$ and φ satisfying $\mathbf{R}(\boldsymbol{u}, p, T, \gamma) = 0$, such that the Lagrangian functional $\mathcal{L}$ given by Eq. 8 is rendered stationary.

In this new formulation, each argument of the Lagrangian functional is considered to be an independent variable (only subject to the constraints $\mathbf{R}(\boldsymbol{u}, p, T, \gamma) = 0$ and its boundaries) so that each variable may be varied independently.

The total variation of $\mathcal{L}$, disregarding geometry deformation results:

$$\delta\mathcal{L} = \left[\delta_{\gamma}\mathcal{J} - \int_{\Omega} (\boldsymbol{v}, q, \varphi)^{\mathsf{T}} \, \delta_{\gamma}\boldsymbol{R} \, \mathrm{d}\Omega\right] + \left[\delta_{\boldsymbol{u}}\mathcal{J} + \delta_{p}\mathcal{J} + \delta_{T}\mathcal{J} - \int_{\Omega} (\boldsymbol{v}, q, \varphi)^{\mathsf{T}} (\delta_{\boldsymbol{u}}\boldsymbol{R} + \delta_{p}\boldsymbol{R} + \delta_{T}\boldsymbol{R}) \, \mathrm{d}\Omega\right]. \tag{9}$$

We want to choose the adjoints $\boldsymbol{v}$, q and φ so that the second term in brackets in Eq. (9) vanishes. So, we obtain the stationary adjoint equations on the general form:

$$\delta_{\boldsymbol{u}}\mathcal{J} + \delta_{p}\mathcal{J} + \delta_{T}\mathcal{J} - \int_{\Omega} (\boldsymbol{v}, q, \varphi)^{\mathsf{T}} (\delta_{\boldsymbol{u}}\boldsymbol{R} + \delta_{p}\boldsymbol{R} + \delta_{T}\boldsymbol{R}) \, \mathrm{d}\Omega = 0. \tag{10}$$

Considering the decomposition of the functional $\mathcal{J}$ into interior and boundary parts, renders:

$$\mathcal{J} = \int_{\Omega} \mathcal{J}_{\Omega} \, \mathrm{d}\Omega + \int_{\Gamma} \mathcal{J}_{\Gamma} \, \mathrm{d}\Gamma. \tag{11}$$

Then, after elaborating further using integration by parts (see Othmer (2008) for details), we obtain the system of adjoint equations,

$$-\nabla \cdot (2\nu \mathrm{D}(\boldsymbol{v})) - \nabla\boldsymbol{v} \cdot \boldsymbol{u} - (\boldsymbol{u} \cdot \nabla)\boldsymbol{v} + \frac{\nabla q}{\rho} - \varphi\nabla T - \alpha\boldsymbol{v} = \frac{\partial \mathcal{J}_{\Omega}}{\partial \boldsymbol{u}}. \tag{12}$$

$$(-\nabla \cdot \boldsymbol{v}) = \frac{\partial \mathcal{J}_{\Omega}}{\partial p}. \tag{13}$$

$$-\nabla \cdot (D\nabla\varphi) - \nabla \cdot (\boldsymbol{u}\varphi) - \frac{\partial f_{\Omega_a}}{\partial T}\varphi = \frac{\partial \mathcal{J}_{\Omega}}{\partial T}. \tag{14}$$

with the adjoint boundary conditions:

$$\boldsymbol{v}_t|_{\Gamma_i - \Gamma_\omega} = 0. \tag{15}$$

$$\boldsymbol{v}_{\boldsymbol{n}}|_{\Gamma_i - \Gamma_\omega} = \frac{\partial \mathcal{J}_{\Gamma}}{\partial p}. \tag{16}$$

$$\boldsymbol{n} \cdot \nabla q|_{\Gamma_i - \Gamma_\omega} = 0. \tag{17}$$

$$\nu(\boldsymbol{n} \cdot \nabla)\boldsymbol{v} + \boldsymbol{v}(\boldsymbol{u} \cdot \boldsymbol{n}) + \boldsymbol{n}(\boldsymbol{v} \cdot \boldsymbol{u}) + \varphi T\boldsymbol{n} - \frac{q\boldsymbol{n}}{\rho}\Big|_{\Gamma_o} = \frac{\partial \mathcal{J}_{\Gamma}}{\partial \boldsymbol{u}}. \tag{18}$$

$$\boldsymbol{n} \cdot D\nabla\varphi + \boldsymbol{u} \cdot \boldsymbol{n}\varphi|_{\Gamma_o} = \frac{\partial \mathcal{J}_\Gamma}{\partial T}. \tag{19}$$

For convective heat flux case the adjoint boundary results:

$$\boldsymbol{n} \cdot k\nabla\varphi|_{\Gamma_\omega} = h_{ext}\varphi. \tag{20}$$

The first set of terms in brackets of equation 9 remains. It is usually called the optimality condition.

$$\delta_\gamma \mathcal{L} = \delta_\gamma \mathcal{J} - \int_\Omega (\boldsymbol{v}, q, \varphi)^\mathsf{T}\, \delta_\gamma \boldsymbol{R}\, \mathrm{d}\Omega. \tag{21}$$

Equation (21) is an expression for the gradient of the functional in terms of the solution of the system of adjoint Eq. (12) and Eq. (20). The optimality condition states, that the adjoints $(\boldsymbol{v}, q, \varphi)$ reach their optimal values, when the Lagrange functional gradient is zero.

2.2 The Objective Function

We consider the functional of error:

$$\mathcal{J} = \frac{1}{2}\int_\Omega (T - T_{ref})^2\, \mathrm{d}\Omega. \tag{22}$$

where T_{ref} is a reference temperature. Following the work of Yu *et al.* (2020), in order to prevent an unrealistic design, the power dissipation of the fluid device is considered as a constraint.

$$P = -\int_\Gamma \Big(p + \frac{1}{2}|\boldsymbol{u}|^2\Big)\boldsymbol{u} \cdot \boldsymbol{n}\, \mathrm{d}\Gamma. \tag{23}$$

where $\boldsymbol{n}$ is the unit vector normal to the boundary Γ. Then, we compute the required derivatives needed by the adjoint system, as:

$$\frac{\partial \mathcal{J}_\Omega}{\partial T} = (T - T_{ref}). \tag{24}$$

$$\frac{\partial P_\Gamma}{\partial \boldsymbol{u}} = -\Big[\Big(p + \frac{1}{2}|\boldsymbol{u}|^2\Big)\boldsymbol{n} - (\boldsymbol{u} \cdot \boldsymbol{n})\boldsymbol{u}\Big]. \tag{25}$$

$$\frac{\partial P_\Gamma}{\partial p} = -\boldsymbol{u} \cdot \boldsymbol{n}. \tag{26}$$

3. TOPOLOGY OPTIMIZATION

The material distribution method is a common approach in topology optimization. The strategy is to assign each element volume of the discretized domain an individual pseudo-density values $\gamma \in [0, 1]$. These pseudo-density values may be associated with a material physical parameter by a generic increasing monotone and continuously differentiable function of γ. In this work we use the Rational Approximation of Material Properties (RAMP)-type (Stolpe and Svanberg, 2001). In the momentum conservation equation, α is related to the pseudo-density field by the following interpolation:

$$\alpha = \alpha_{max}\frac{q(1-\gamma)}{q+\gamma},\ \gamma \in [0, 1]. \tag{27}$$

In the energy conservation equation, the thermal diffusivity is interpolated from the solid (D_s) to that of the fluid (D_f).

$$D = D_f + (D_s - D_f)\frac{q(1-\gamma)}{q+\gamma},\ \gamma \in [0, 1]. \tag{28}$$

where q is the penalty parameter which governs the shape of the function. Due to non-convexity, as stated by Borrvall and Petersson (2003), when a high value of q is used there might be local optimal solution problems. Thus, the optimization is first solved with a small value of $q = 0.01$ to obtain a better initial guess for the problem and then the penalty parameter value is increased in order to achieve a discrete solution.

The topology optimization optimization of the heat exchanger can be formulated as follows:

Find γ
Minimize $\mathcal{J}$
Subject to:

- $\int_\Omega \gamma / |\Omega| \leq \beta$.
- $P < \bar{P}$.
- $0 < \gamma < 1$.
- Equation (1) and Eq. (3)

where β is the volume fraction occupied by the fluid and $\bar{P}$ is the reference power dissipation.

4. NUMERICAL METHOD

Choosing an efficient numerical technique and reliable optimization algorithm is very important to ensure a stable convergence to an optimum solution. In this respect, the optimization problem is solved the Method of Moving Asymptotes (MMA).

Simulations were performed on a PC with the following specifications: CPU Intel Core3 M 370 2.40GHz, RAM DDR3 2.8GB 1066MHz. The implementation was running in OpenFOAM, adopting the Finite Volume Method. The domain was discretized in 76800 cell volumes,each simulation, on average, took around 6 hours to reach the steady state of the cycle.

5. RESULTS

The results are obtained by adopting the parameters and physical properties presented in Tab. 1 and Tab. 2.

Table 1. **Parameters of design.**

Parameter	Value
Inlet velocity $\boldsymbol{u}_i$ (ms^{-1})	(0 , -0.05)
Inlet Temperature T_i (K)	300
Reference Temperature T_{ref} (K)	300
External Temperature T_∞ (K)	350
Convective heat transfer coefficient (h_{ext})	1000
Characteristic length scale 2b (m)	0.002
Reynolds number (Re)	100
Length of line sources (m)	0.006

Table 2. **Material properties of aluminium and water.**

Material properties	Aluminium	Water
Thermal conductivity, k $(Wm^{-1}K^{-1})$	237	0.61
Density, ρ (kgm^{-3})	2700	1000
Specific heat, Cp $(Jkg^{-1}K^{-1})$	880	4180
Inverse of permeability, α $(kgm^{-3}s^{-1})$	2×10^{-5}	-
Kinematic viscosity, ν (m^2s^{-1})	-	1×10^{-6}

All the optimized designs, have a reference power dissipation constraint of $\bar{P} = 1.58 \times 10^{-7}$ and the volume of fluid fraction of $\beta = 0.4$.

We first present a special case of temperature control, where we will make the two heat sources vary linearly with temperature, like: $f_{\Omega_a} = 10T - 2000$. Furthermore, our error objective function will be located only in the region occupied by the source and we will seek to reach a constant temperature equal to $315\,K$.

Figure 3 shows the optimized results, where we see that the fluid circulating in the duct forms a kind of encapsulation to ensure that the region occupied by the two sources is kept at a uniform temperature equal to $315\,K$.

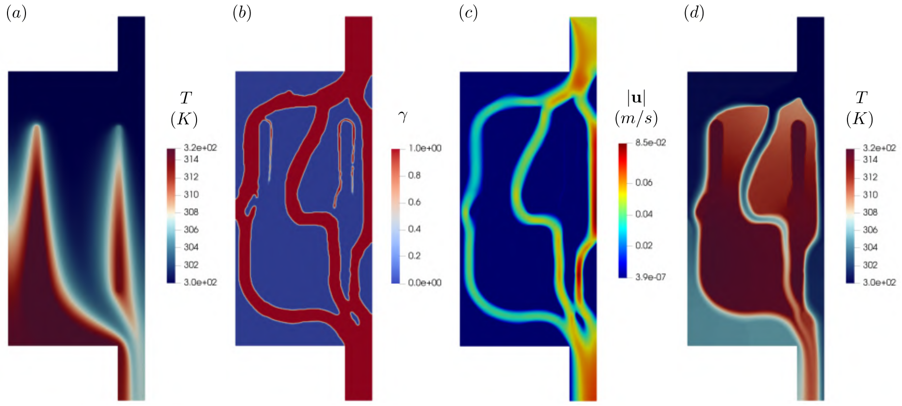

Figure 3. **Topology optimization of thermal-fluid flow system with two non uniform heat line sources. Initial (a) primal temperature field with $\gamma = 0.4$ uniform in all the domain. Optimal results, (b) pseudo-density, (c) primal velocity and (d) temperature fields.**

Next, in the configurations to follow, we seek to make the entire domain uniform as much as possible until the specified reference temperature is reached. Then, we see that the topological configuration of the fluid occupy the regions of higher temperature. Figure 4 shows the result of the topological optimization of a thermal-fluid system with two uniform sources $f_{\Omega_a} = 500\,W/m^3$. We can see a decrease in the maximum temperature and a more homogeneous temperature field in relation to the initial one.

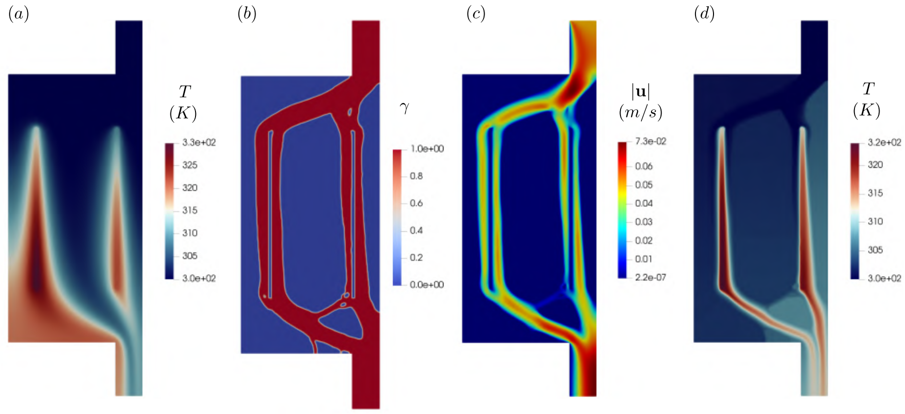

Figure 4. **Topology optimization of thermal-fluid flow system with two uniform heat line sources, case 1. Initial (a) primal temperature field with $\gamma = 0.4$ uniform in all the domain. Optimal results, (b) pseudo-density, (c) primal velocity and (d) temperature fields.**

The solid material considered in the previous example is Aluminum, which has a high thermal conductivity and which contributes to the transfer of heat through the walls. Heat exchangers are often lined with insulating material or exposed to air, so we consider the case where the solid material with very low conductivity, here we consider the conductivity of the air to be equal to $0.024\,W/mK$. Figure 5 shows that the result is very similar to the previous one, where a significant drop in temperature is seen in relation to the initial one. The final topology in both cases is similar as well.

Finally we consider the cases where part of the boundary is exposed to convective heat transfer. Figure 6 shows how the fluid occupies the region where convective heat transfer is occurring, adopting a shape such that the fluid accelerates in that region and contributes to heat transfer. Figure 7 shows the case of two borders exposed to heat transfer by convection, result manifested a significant decrease in the maximum temperature. Let us note that the final conduit adopts the form that the fluid passes attached to the walls due to the established objective function, which is to make the temperature uniform throughout the domain equal to $300\,K$.

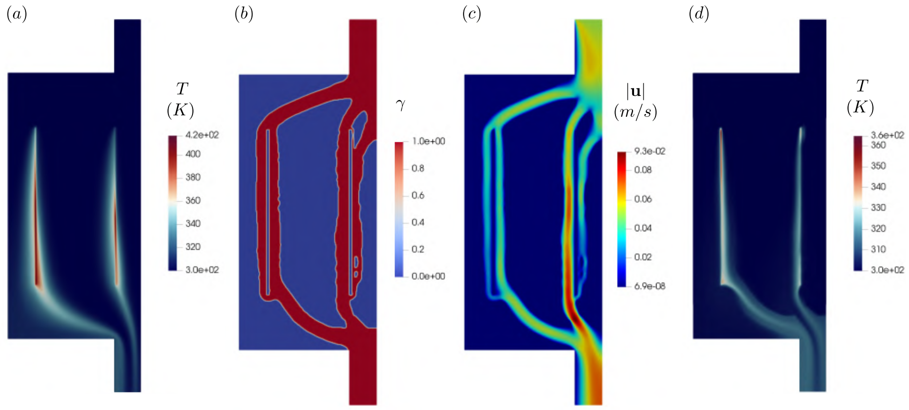

Figure 5. **Topology optimization of thermal-fluid flow system with two uniform heat line sources, case 2. Initial (a) primal temperature field with $\gamma = 0.4$ uniform in all the domain. Optimal results, (b) pseudo-density, (c) primal velocity and (d) temperature fields.**

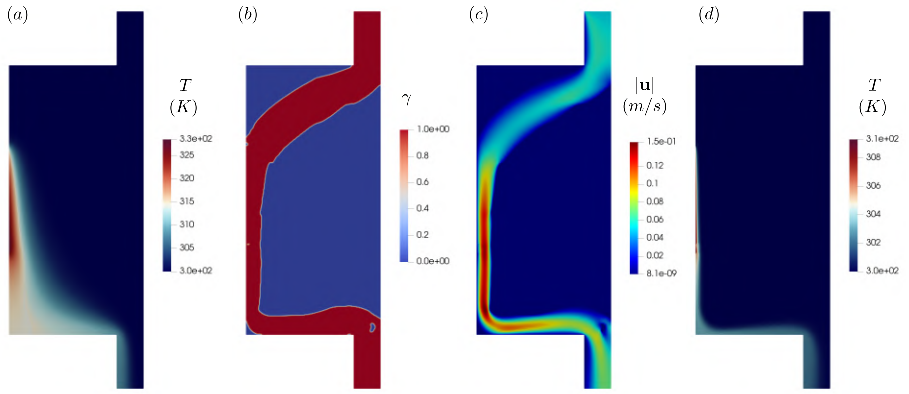

Figure 6. **Topology optimization of thermal-fluid flow system with one convective heat flux. Initial (a) primal temperature field with $\gamma = 0.4$ uniform in all the domain. Optimal results, (b) pseudo-density, (c) primal velocity and (d) temperature fields.**

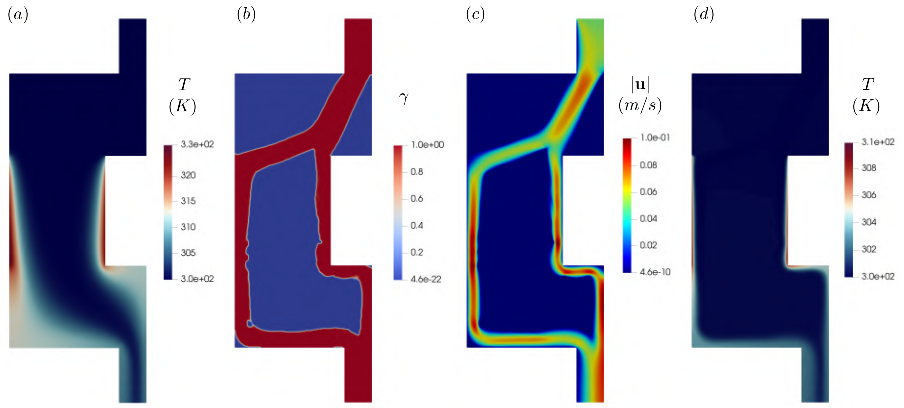

Figure 7. **Topology optimization of thermal-fluid flow system with two convective heat flux. Initial (a) primal temperature field with $\gamma = 0.4$ uniform in all the domain. Optimal results, (b) pseudo-density, (c) primal velocity and (d) temperature fields.**

6. CONCLUSIONS

In this study, we researched a topology optimization of heat exchangers using the continuous adjoint formulation. We considered heat sources acting in specific regions of the domain and convective boundary condition on a partial region of the boundary. It was obtained results in two dimensions with different heat exchanger configurations and operating conditions. The resulting channels were set up, aiming to achieve a uniform temperature in a specific region or in the whole domain as specified by the objective function, with the fluid passing either adjacent to or near the hottest regions, i.e. where the sources were placed.

In future works, new geometric configurations as well as new objective functions should be studied. In order to improve small imperfections of the final topologies obtained, we should investigate on constraint functions.

7. ACKNOWLEDGEMENTS

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