Numerical Analysis of a Cold Plate for FM Radio Power Amplifiers

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Abstract. The results of a numerical investigation of heat and fluid flow in a liquid cold plate for FM radio power amplifiers are presented. The objective is to verify, by using a commercial CFD code, the performance of a blister cold plate designed to dissipate the heat generated by a known set of electronic components, in order to limit their maximum temperature during operations. Since in a blister cold plate mainly the cover is thermally active, the cold plate is simplified and lightened by using plastics in the base plate. A 3-D conjugate CFD approach, where thermal and fluid flow analyses are combined, is followed. Several design options for the cold plate are examined and the validity of the full 3-D CFD approach in the dimensioning of the cooling systems of electronic equipment is demonstrated.

1. Introduction

In the proper design of an electronic device it is necessary to ensure an adequate cooling of its active and passive components. Extensive reviews of the available techniques of industrial interest for electronics cooling are those of Çengel [1] and Kraus et al. [2]. Air cooling is the simplest and most common technique. It has several positive aspects, such as low costs and simplicity of implementation; on the other hand, it may show difficulties in dissipating high heat fluxes and often is the source of unacceptable levels of noise [3].

The active electronic components currently used can generate heat fluxes of up to 3 MW/m²; for these high specific powers air cooling may be insufficient [3]. In this case, a cold plate is a practical solution frequently used (Incropera [4]). A cold plate consists of a plate, usually in aluminum, on which are fixed some electronic components and within which one or more hydraulic circuits are obtained for their cooling with water flowing in forced circulation. The design solutions are essentially two: "press-fitted" and "blistered". In a "press-fit" cold plate, the cooling circuits are manufactured with copper tubes pressed in special channels machined on the bottom surface of the plate. This technology is simple and cheap. However, the thermal resistance between tubes and plate could reduce the heat transfer. In a "blister" cold plate, the cooling channels are machined directly on the plate, which is then closed by a thin wall of high thermal conductivity (commonly aluminum). This technology allows better performance and greater design flexibility. On the other hand, the production costs are higher and therefore suitable for dedicated applications.

In the design of a cold plate, the main steps are: determination of number and layout of the cooling

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coils, diameter of the pipes and mass flow of refrigerant in the channels. In a preliminary stage, simplified one-dimensional calculations can be used. In the final design, such an approach seems to be not appropriate for the multidimensional effects of heat transfer, always present in these systems. These make the 3-D numerical approach the most appropriate strategy. Also, in a 3-D CFD approach, the analysis can be simplified, for instance by setting appropriate values of the heat transfer coefficient on the water side of the channels [5]. However, in this case, some flow details are lost.

In this paper, a 3-D conjugate CFD approach, where thermal and fluid flow analyses are combined, is preferred. The objective is to verify, by using a commercial CFD code, the performance of a blister cold plate designed to dissipate the heat generated by a known set of electronic components, in order to limit their maximum temperature during operations. Since in previous experiences [5], it was found that in a blister cold-plate mainly the cover is thermally active, it has been tried to simplify and lighten the cold-plate by using plastic in the base plate. The parameters taken into account in the discussion of the results are the thermal conductivity of the plate and the mass flow rate of refrigerant.

2. Problem statement
Figure 1 is a schematic representation of the cold plate of interest, consisting of a plastic base and of an aluminum cover housing the components to be cooled. In the base plate (21x302x372 mm) are machined cooling channels of square section (10x10 mm). The dimensions of the aluminum plate are 8x302x372 mm. The refrigerant is water. On the top of the cover plate are placed ten electronic components dissipating a known thermal power. In the model, these components are represented as blocks of copper (24.5x9.5x3 mm).

For this cold plate, a large number of simulations were performed to analyze the effect of different parameters. One of the tests is considered as the Reference test case. The thermophysical properties of the materials used in the Reference test case are reported in Table 1. Other significant parameters are:
- electric power dissipated by the electronic components: $P_{el} = 1465.17$ W;
- mass flow rate of the cooling water: $Q_m = 0.457$ kg/s;
- inlet temperature of the cooling water: $T_{in,water} = 25.8$ °C;
- ambient temperature: $T_{\infty} = 22.7$ °C;
- heat flow dissipated by the then electronic components and simulated as an uniform heat flux on their surface: $q = 629.5$ kW/m$^2$;
- thermal coupling between the cold plate and the environment, and modeled with a boundary condition of third type: $h_\infty = 10$ W/m$^2$K;
- to the contact thermal resistance due to the coupling between the cover plate and the electronic components, enhanced by a layer of thermal grease, was imposed a value from literature for metal-metal coupling (Moran et al.[7]): $RTC = 0.7 \times 10^{-5}$ m$^2$K/W.

Variants to the Reference case were analyzed to evaluate the effect of different parameters on the performance of the cold plate. In these variants, the following parameters were changed:
1. thermal conductivity of the base plate: a high value (60 W/mK), characteristic of an enhanced plastic added of a filler, and a low value (0.35 W/mK), characteristic of a standard plastic;
2. mass flow rate: 0.2285, 0.475 and 0.914 kg/s;
3. thermal conductivity of the electronic components: a high value typical of a component with prevailing copper (401 W/mK), and a low value, typical of a prevailing ceramic (30 W/mK).

3. Mathematical model
The numerical investigation is based on the following simplifying assumptions:
- incompressible Newtonian fluid;
- steady state;
- constant properties;
- negligible buoyancy forces;
- turbulent flow and $k-\varepsilon$ model;
- negligible radiation heat transfer.
The equations to be solved are those assuring the conservation of mass, momentum and energy [6]:

\[ \nabla \cdot (\rho \vec{u}) = 0 \]  
\[ \nabla \cdot (\rho \vec{u} \times \vec{u}) = -\nabla p' + \nabla \cdot \left[ \mu_{\text{eff}} \left( \nabla \vec{u} + \nabla \vec{u}^\top \right) \right] \]  
\[ \nabla \cdot (\rho \vec{u} \vec{h}_{\text{ux}}) = \nabla \cdot \left[ \left( \lambda + c \frac{\mu}{\sigma_{\text{H}}} \right) \nabla T \right] \]

**Table 1.** Reference Case: thermophysical properties.

| Material                  | Density (kg/m\(^3\)) | Specific Heat Capacity (J/kgK) | Thermal Conductivity (W/mK) | Viscosity (Pa s) |
|---------------------------|------------------------|--------------------------------|-----------------------------|-----------------|
| Water                     | 997                    | 4181.7                         | 0.6069                      | 8.899 \(10^{-4}\) |
| Plastic (base plate)      | 1100                   | 2010                           | 60                          |                 |
| Aluminium (cover plate)   | 2702                   | 903                            | 237                         |                 |
| Copper (electronic components) | 8933                  | 3.85                           | 401                         |                 |
where:

\[ h_{\text{ec}} = h + \bar{u}^2/2 \]  
\[ p' = p + 2/3 \rho k \]  
\[ \mu_{\text{eff}} = \mu + \mu_i \]

and the turbulent viscosity is calculated with a standard \( k-\varepsilon \) model [6].

The \( k-\varepsilon \) turbulence model is widely used in the literature because it is robust and efficient, and converging in an acceptable time. In this application, wall functions were specified near the walls. The technique of inflation was used with a scaling function such that the coordinate \( y^+ \) of the first grid point was always greater than 11.06:

\[ y^+ = \max\{11.06; y^+\} \]  

An advective High Resolution scheme was chosen for the equations of mass, momentum and energy conservation and for the turbulence equations.

The boundary conditions are:

- Adiabatic on lateral and bottom surfaces of the base plate, lateral surfaces of the cover plate and lateral surfaces of the electronic components;
- Boundary condition of the third kind on the upper surface of the cover plate:
  \[ -\lambda \frac{\partial T}{\partial n} = U_x (T_w - T_x) \]  

- Heat flow imposed on the upper surfaces of the electronic components;
- Mass flow rate and temperature assigned in the input section of the cooling channel;
- Thermal contact resistance in the interface between electronic components and cover plate.

For the generation of the computational grid the code, ANSYS Meshing was used. The grid is hybrid, composed of tetrahedron inside the domains and of five prismatic layers (inflated layers) near the walls, where the wall functions are implemented (Figure 2).

In Table 2, the geometrical properties of the grid are listed, by indicating for each domain the total number of elements (tetrahedrons and prisms), the number of nodes and the maximum size of the elements.

The size of the elements was chosen to have a maximum size of 1.5 mm in the fluid domain. The total thickness of the five layers of inflation at the wall is 1.5 mm, with a growth rate from the wall toward the center of the channel 1.2. In the other remaining solid domains, a maximum size of the elements of 6 mm was chosen.

The fluid flow and heat transfer characteristics of the cold plate were calculated with the finite volume commercial code ANSYS CFX 12.0. This release allows the simulation of the thermal contact resistances; this is an interesting opportunity when modeling the liquid cooling of electronic devices.

**Table 2. Grid information.**

| Domain                | Nodes | Elements | Max Elem. Size |
|-----------------------|-------|----------|---------------|
| Aluminium             | 128348| 640105   | 20 mm         |
| Fluid                 | 346798| 893001   | 1.5 mm        |
| Plastic               | 19644 | 89526    | 20 mm         |
| Electronic Components | 728   | 1543     | 7 mm          |
| All Domains           | 495518| 1624175  | 6 mm          |
4. Numerical results

4.1. Reference Test Case

In Figures 3, 4, 5 and 6 are shown the results of the calculations for the Reference test case.

Specifically, Figure 3 shows the distribution of velocity inside the fluid domain. The mass flow is not equally distributed in the three coils; in particular near the outlet, on the left in Figure 3, the mass flow is higher, while near the inlet is lower. Near the entrance of the coils is observed the presence of recirculations. The extension of the area occupied by the recirculations shrinks when moving from inlet to outlet. This behavior is justified by the presence of sharp edges at the entrance. The existence of a possible correlation between the lower mass flow rate in the first coil and a higher temperature of the electronic components cooled by this coil is thence expected. In Figure 3, it may be observed that at the outlet the flow shows a pronounced swirl. This can be significant when more plates are connected in series.

In Figure 4 is shown the distribution of temperature on the surface of the aluminum cover of the blister. From Figure 4, it can be seen that the two resistances near the outlet and those near the inlet are the most disadvantaged (in particular, it is evident that there is a larger area interested by a disturbance in the distribution of temperature). This is due not only to the reduced flow, but also to the misalignment of the electronic components relative to the coils; these stressed components are not "strictly" placed on the coil and only a small portion of them is "wetted" by the fluid.

In Figure 5 are shown the temperature patterns on the surface of the ten electronic components. The two components near the inlet on the right show a temperature higher than the others, confirming the less intense heat transfer observed in Figure 4.

For an accurate analysis of the situation, in Table 3 are reported the mean values of temperature on
the upper surface of the components. These values are ordered as the electronic components in Figures 3-6. Despite the coil near the inlet (left) is the most disadvantaged in terms of mass flow rate, the electronic components placed on the internal area of this coil are well cooled. This is due to the placement of the components well over the pipe and for this reason well "wetted" by the fluid.

In Figure 6 is shown the distribution of heat flux at the wall of the pipe, confirming the behavior observed in Figures 3-5.

**Table 3.** Average temperature on the surface of the electronic components.

| Temperature (°C) |
|-----------------|
| 55.64 56.19 56.30 56.70 57.39 |
| 56.02 55.54 56.22 55.79 56.84 |

**Figure 3.** Distribution of velocity inside the fluid domain.

**Figure 4** Temperature distribution on the surface of the base plate.
4.2. Variants to the Reference Test Case

In Figure 7, for the variants to the Reference test case described in Section 2, are compared the results in terms of temperature distribution on the surface of the aluminum plate.

The values assumed by the significant parameters in these five cases are reported in Table 4, together with the values of the bulk temperature of the water at the outlet and the average temperature on the surface of the electronic components. The cases reported in Table 4 differ for a parameter at time. Case (a) is the Reference one; in case (b) the thermal conductivity of the plastic base is low; in case (c) the mass flow rate is half than in (a); in case (d) the mass flow is twice than in (a); in case (e) the thermal conductivity of the electronic components is low.

On the top surface of the cold plate, for a decrease of the thermal conductivity of the plastic base (b), the extent of the area showing high values of temperature grows and the local maximum of temperature increases.

When decreasing the mass flow rate (c), it is observed an increase of the extension of the areas with
a high temperature; also the local maximum of temperature increases; the opposite happens when the mass flow rate is increased (d).

If the thermal conductivity of the base plate decreases ($\lambda_{\text{base}}$ 60 → 0.35 W/m K as from case (a) to case (b)), the maximum local temperature on the plate grows slightly (~1 °C in this case).

In the last case (e), the mean surface temperature of the electronic components is approximately twice that observed for a high thermal conductivity.

In terms of average temperature on the electronic components (Table 4), for a low value of this parameter it is good, but not significant, a high thermal conductivity of the base plate and a high mass flow rate. On the opposite, the thermal characteristics of the electronic components is significant.

In Table 5 are shown the results of the pressure drop analysis for the three cases (b), (c) and (d), characterized by the same parameters and different the mass flow rates. In the range of flows of practical interest there is a slight dependence of the losses from the Reynolds number, due mainly to the inlet and outlet sections of the channels.

5. Concluding remarks
The detailed design of a cold plate requires the use of CFD, since its behavior is strongly 3D and the conjugate heat transfer is prevailing. The approach followed in this paper allows useful information to be obtained.

Special attention should be paid to the hydraulic aspects. It is not difficult to obtain unbalanced flows between the coils machined in the cold plate, resulting in a non uniform heat transfer.

### Table 4. Average temperature on the surface of the components and bulk temperature in the outlet.

| Case | Temperature ( °C ) |
|------|---------------------|
| a    | $T_{\text{avg, out}} = 27.39$ |
| $Q_m = 0.457$ kg/s | $\lambda_{\text{base}} = 60$ W/m K |
|       | $\lambda_{\text{electr}} = 401$ W/m K |
| b    | $T_{\text{avg, out}} = 26.56$ |
| $Q_m = 0.457$ kg/s | $\lambda_{\text{base}} = 0.35$ W/m K |
|       | $\lambda_{\text{electr}} = 401$ W/m K |
| c    | $T_{\text{avg, out}} = 27.33$ |
| $Q_m = 0.2215$ kg/s | $\lambda_{\text{base}} = 0.35$ W/m K |
|       | $\lambda_{\text{electr}} = 401$ W/m K |
| d    | $T_{\text{avg, out}} = 26.18$ |
| $Q_m = 0.914$ kg/s | $\lambda_{\text{base}} = 0.35$ W/m K |
|       | $\lambda_{\text{electr}} = 401$ W/m K |
| e    | $T_{\text{avg, out}} = 26.18$ |
| $Q_m = 0.914$ kg/s | $\lambda_{\text{base}} = 0.35$ W/m K |
|       | $\lambda_{\text{electr}} = 30$ W/m K |

### Table 5. Pressure drop analysis

| Case | $Q_m$ ( kg/s ) | Re | $\Delta p$ ( Pa ) | $\beta$ ( - ) |
|------|----------------|----|-----------------|---------------|
| b    | 0.2215         | 6223| 4317.4          | 5.04          |
| c    | 0.457          | 12839| 16664.9         | 4.87          |
| d    | 0.914          | 25677| 65615.3         | 4.79          |
Figure 7. Temperature distribution in the upper surface of the aluminium plate for the five simulations ‘a’, ‘b’, ‘c’, ‘d’, ‘e’ of Table 4.

Sharp-edged geometries of the inlets in the manifolds give rise to secondary recirculations reducing the heat transfer.

The position of the electronic components has a significant influence on the performances. The components eccentric to the channels machined in the cold plate show values of temperature higher than the other. However, the position of the components is a parameter that generally can not be changed by the thermal designer and for this reason the hydraulic circuits have to be adapted to each specific layout.
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Nomenclature

| Symbol | Description                  | Unit          |
|--------|------------------------------|---------------|
| c      | specific heat                | J/(kgK)       |
| h      | specific enthalpy            | J/(kgK)       |
| \( \bar{n} \) | unitary external normal   | m             |
| \( p' \) | modified pressure Eq.(5)   | Pa            |
| P      | power                        | W             |
| q      | heat flow per unit area      | W/m²          |
| \( Q_m \) | mass flow rate              | kg/s          |
| Re     | Reynolds number              | -             |
| RTC    | thermal contact resistance   | m²K/W         |
| T      | temperature                  | °C            |
| \( \bar{u} \) | velocity                    | m/s           |
| U      | heat transfer coefficient    | W/(m²K)       |
| \( y' \) | dimensionless wall distance  | -             |

Subscripts

- avg average
- base base plate
- eff effective
- el electric
- in inlet
- out outlet
- tot total
- w wall
- \( \infty \) ambient

Greek symbols

| Symbol | Description                  |
|--------|------------------------------|
| \( \beta \) | pressure drop coefficient   |
| \( \lambda \) | thermal conductivity        |

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