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ANALYSIS OF ELECTRONIC PACKAGE COOLING IN AN AIRCRAFT USING CFD

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Abstract. *A problem of interest of the aeronautical industry is the positioning of electronic equipments in racks and the associated ventilation system project to guarantee the equipment operational conditions. The relevance of the proper operation of electronic equipments increases considerably when high economical costs, performance reduction and safety are involved. The appropriate operational conditions of the electronic components happen when the working temperature of the equipment installed in the rack is inside a safety project temperature band. Therefore, the analysis and modelling of heat transfer processes for aircraft rack design becomes mandatory. This paper presents a parametric study considering volumetric and superficial heat generation in electronic equipment within racks in an aircraft. Simulations were performed using the commercial CFD Fluent code and results were compared to experimental data. Simulation results considering volumetric heat generation show higher temperatures of the electronic equipment, with lower velocities in the racks, than those with superficial heat generation.*

Keywords: *Aircraft, electronic package, CFD.*

1. Introduction

Electronic components usually produce more heat than they can dissipate in a natural way. Additionally, with the fast advancement of technology, the trend of electronic packaging tends to minimize components size with high heat generation levels. An insufficient removal of heat can result in temperatures above the equipment operational limits. In these circumstances, the performance, the life and the reliability of the equipment are extremely reduced.

To solve the thermal dissipation problem of electronic components, one method is the enhancement of heat conduction: using a special package material with high thermal conductivity or attaching a fin on the electronic component. Another method is the use of a cooling water channel to lower the temperature. But, the most economical and efficient method consists on the use of a fan to cool the surface of the component (Yang and Fu, 2001).

Bejan and Ledezma (1996) describes several fundamental trade offs that govern the optimization of cooling techniques for heat generating electric devices. Five basic cooling configurations were optimized. It is shown that in forced air cooling above room temperature the fan power requirement is minimum when the heat transfer contact area is optimized to a value that is of the order of one hundred times the flow cross-sectional area.

Das and Pecht (1996) provide a perspective of temperature and related thermal information as an input parameter which can impact the avionics life cycle, as a means to discuss the role of heat transfer in avionics equipment improvement. The influence of temperature on performance and reliability is examined.

Gromoll (1998) analyzes the performance of three micro cooling systems for high density packaging: a micro-heat pipe, a thermosyphon circuit with forced air convection and direct air cooling. Cooling performances were achieved that were otherwise only possible with liquid cooling.

Lee and Vafai (1999) present a comparative investigation of jet impingement and microchannel cooling for high heat flux applications. The thermal performance of each technology evaluated at the respective optimal condition was compared. In another paper, Vafai and Zhu (1999) present the thermal performance and the temperature distribution for a two-layered micro-channel heat sink with counter current flow arrangement for cooling of electronic components; and Fedorov and Viskanta (2000) developed a three-dimensional model to investigate flow and conjugate heat transfer also in a microchannel heat sink for electronic packaging applications.

Due to the rapid development in computational fluid dynamics (CFD) technology, comprehensive airflow analysis has become manageable. Bessaih and Kadja (2000) present a numerical simulation of conjugate, turbulent natural convection air cooling of three heated ceramic components mounted on a vertical adiabatic channel. A two-dimensional, conjugate heat transfer model and the standard k-ε model were used to obtain the dynamic and thermal fields. The SIMPLER algorithm described by Patankar (1981) was used to determine the pressure from the continuity equation.

Rodgers, Eveloy and Davies (2003a) used commercial CFD software, Flotherm, for the thermal analysis of a component-printed circuit board (PCB) heat transfer in forced convection. Experimental measurements are reported and numerical predictive accuracy is assessed (Rodgers, Eveloy and Davies, 2003b).

Eveloy, Rodgers and Hashmi (2004) provide a perspective on the current capabilities of CFD as a design tool to predict component operational temperature in electronic systems. A systematic assessment of predictive accuracy is presented for PCB-mounted component heat transfer using Flotherm code.

Icoz and Jaluria (2004) present a methodology for the design and optimization of cooling systems for electronic equipment. In this approach, inputs from both experimentation and numerical modeling are to be used concurrently to obtain an acceptable or optimal design. The preceding governing equations are solved using the finite volume method for primitive variables on a nonuniform staggered grid. Pressure calculations are done using a scheme similar to the SIMPLER algorithm (Patankar, 1981).

In the present paper a parametric study considering volumetric and superficial heat generation in electronic equipment within racks in an aircraft is presented. Simulations considering forced and natural convection were performed using the commercial CFD Fluent code (Fluent Inc., 2003) and results were compared to experimental data.

2. Geometry and analysis conditions

The analysis of avionics cooling systems was performed on a package with eleven electronic equipment within racks. There are three exhaust fans, six pressure outlets and a cold air inlet, as it is shown in Fig. 1.

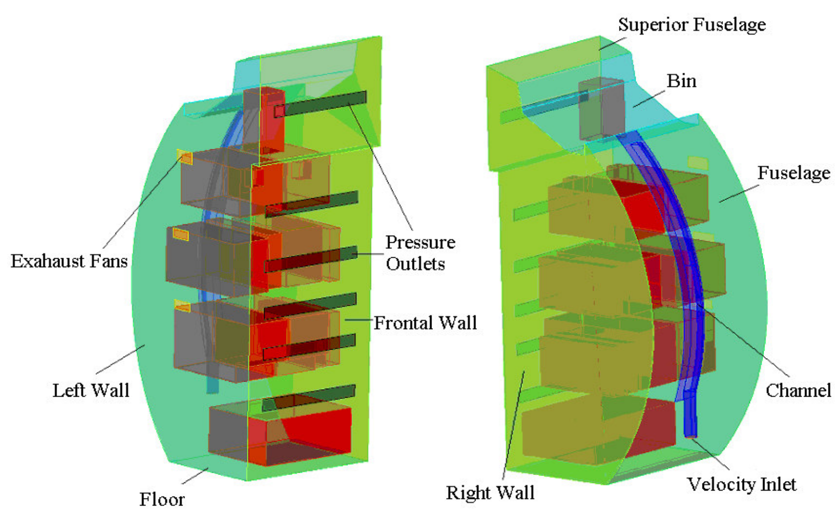


Figure 1. Electronic package within racks in an aircraft.

The avionic system analyzed is similar to that investigated by Pustelnik (2004). There is, however, an important difference in the form of considering the inlet velocity condition in the present work. In Pustelnik (2004) it was considered a velocity inlet condition throw the channel, as it is shown in Fig 2(a). At the present investigation the velocity inlet is considered throw the hole connected to the channel (with the same mass flow), as it is presented in Fig. 2 (b). Additionally, in the present work the analysis of superficial heat generation is also performed.

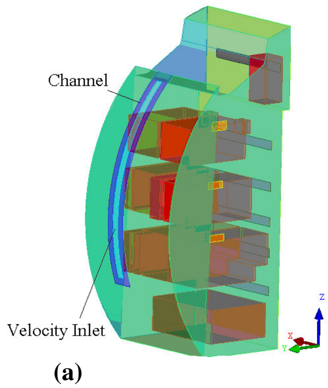


Figure 2. Velocity inlet conditions: **(a)** Pustelnik (2004) **(b)** Present work

To characterize equipment heat transfer, temperatures were measured at the points shown in Fig. 3. The measurements refer to the frontal superficial temperatures of the equipment.

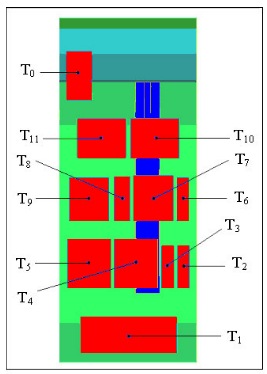


Figure 3. Points of temperature measurement

Temperature values and equipment volumetric heat generation rates (with corresponding superficial heat generation) are shown in Tab. 1.

Table 1. Experimental temperatures and heat generation rates (Pustelnik, 2004).

Equipment	**Experimental temperature**		**Heat generation**	
	ºC	**K**	**W/m^3**	**W/m^2**
T_0	-	-	0	0
T_1	22,7	295,7	4,58	0,16
T_2	31,2	304,2	1524,20	31,17
T_3	31,2	304,2	1586,68	32,38
T_4	22,2	295,2	374,38	16,83
T_5	22,2	295,2	176,64	7,93
T_6	25,0	298,0	89,59	1,83
T_7	25,0	298,0	53,79	2,19
T_8	29,8	302,8	1920,84	39,20
T_9	39,4	312,4	1431,03	57,14
T_{10}	21,7	294,7	765,77	27,11
T_{11}	21,7	294,7	513,92	18,20

3. Mathematical equations

The following equations (Eq. (1) to Eq.(6)) are used to solve the airflow inside the rack. Equation (1) is the continuity equation.

$$\frac{\partial \rho}{\partial t} + div(\rho \vec{u}) = 0 \tag{1}$$

Equation (2) is the momentum equation in cartesian coordinates.

$$\frac{\partial}{\partial t}(\rho u) + div(\rho \vec{u}\vec{u}) = -grad\ p + div(\tau_{eff}) + \rho \vec{g} + \vec{F} \tag{2}$$

Equation (3) represents the effective tensor stress.

$$\tau_{eff} = (\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\rho k \delta_{ij} \tag{3}$$

Equation (4) represents the energy equation, where k_{eff} is the effective conductivity and Φ is the viscous work

$$\frac{\partial \rho e}{\partial t} + div(\rho e \vec{u}) = -p\, div(\vec{u}) + div(k_{eff}\, grad\, T) + \Phi + S_e \tag{4}$$

Equations (5) and (6) are used to solve the turbulence in the flow.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k \tag{5}$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon}{k} + S_\varepsilon \tag{6}$$

4. Numerical simulation

A three-dimensional, turbulent, incompressible, steady state flow of a Newtonian fluid with constant properties was considered. To solve the Navier-Stokes equations the commercial CFD Fluent code (Fluent Inc., 2003), with standard k-ε turbulent model, was used. Simulations were performed considering forced and natural convection with volumetric and superficial heat generation. The Boussinesq model was considered for the natural convection cases.

4.1 Boundary conditions

In the present analysis the boundary conditions were similar to those adopted by Pustelnik (2004). The air inlet condition was modeled as a velocity inlet condition with 9.89 m/s, 10% of turbulent intensity and 0.024 m of hydraulic diameter. The air supply temperature was maintained in 283K. For the exhaust fans was adopted a pressure difference of –0.48 Pa and the outlets were modeled as pressure outlets. The inlet velocity of 9.89 m/s was adopted as the initial simulation condition.

All the walls were considered with constant temperatures. The right, left and front walls, the superior fuselage, the fuselage and the floor were considered with 298K. For the bin and the channel was adopted a temperature of 282 K.

4.2 Grid generation

At first a study of mesh independence was accomplished. After the comparison of the numeric residues, mass conservation and a convergence criterion based on the number of iterations, it was chosen a mesh with 299489 elements. Details of the grid generation are shown in Fig. 4.

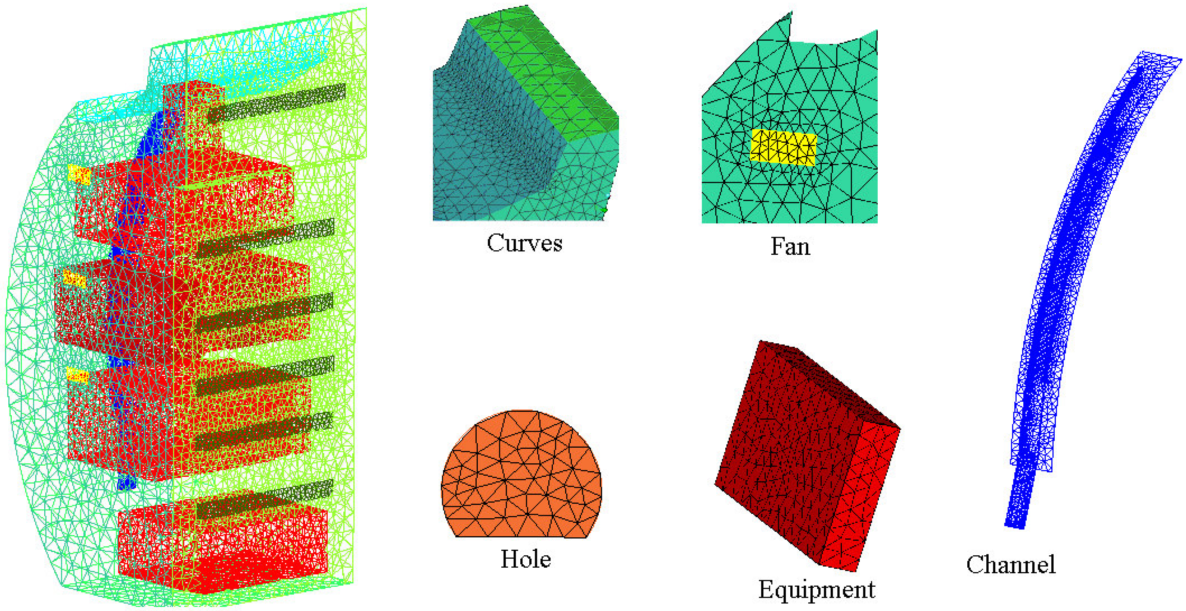


Figure 4. Grid generation details

5. Results

Initially simulations considering forced convection with volumetric and superficial heat generation were performed. Superficial frontal temperatures of the equipment are shown in Fig. 5 and velocities in a middle plane are shown in Fig. 6. Comparison of equipment simulation temperatures and experimental data are shown in Tab. 2.

Soon afterwards simulations were accomplished considering situations of natural convection for the analysis of equipment fail due to superheating. In this case, frontal temperatures of the equipment are shown in Fig. 7 and velocities in a middle plane are shown in Fig. 8. Comparison of equipment simulation temperatures is shown in Tab. 3.

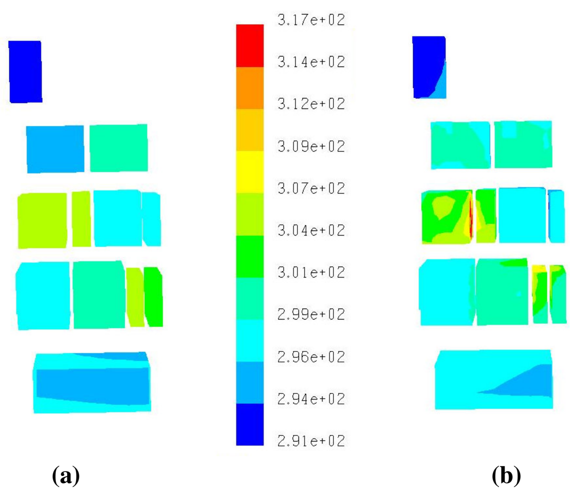
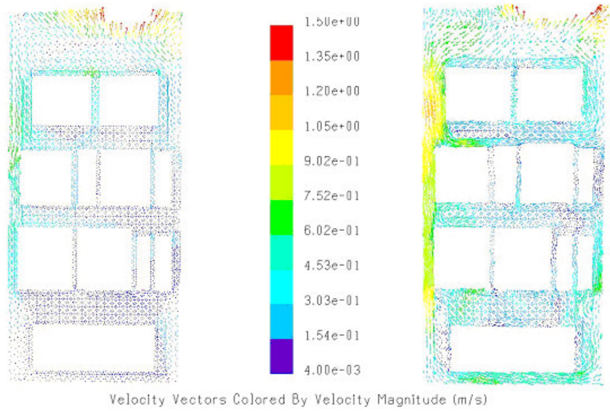


Figure 5. Equipment temperatures (K) considering forced convection.
(a) volumetric heat generation **(b)** superficial heat generation

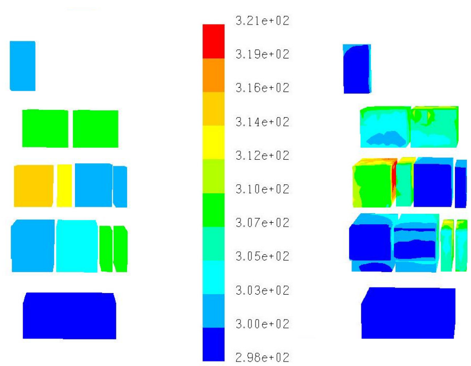


(a) **(b)**

Figure 6. Velocities (m/s) in a middle plane considering forced convection.
(a) volumetric heat generation **(b)** superficial heat generation

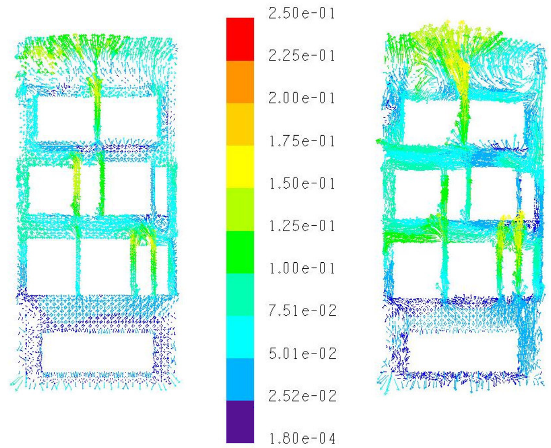
Table 2. Comparison of equipment simulation temperatures, considering volumetric and superficial temperatures with forced convection, and experimental data.

Forced convection					
Equipment	T_{exp} (K)	Fluent			
		T_{vol} (K)	$T_{exp} - T_{vol}$ (K)	T_{sup} (K)	$T_{exp} - T_{sup}$ (K)
T_1	295.7	295.5	-0.2	296.9	1.2
T_2	304.2	303.2	-1.0	300.9	-3.3
T_3	304.2	304.4	0.2	302.2	-2.0
T_4	295.2	300.7	5.5	299.6	4.4
T_5	295.2	297.1	1.9	298.3	3.1
T_6	298.0	297.1	-0.9	298.3	0.3
T_7	298.0	298.4	0.4	298.3	0.3
T_8	302.8	305.5	2.7	303.5	0.7
T_9	312.4	304.4	8.0	304.9	7.5
T_{10}	294.7	299.5	-4.8	299.6	-4.9
T_{11}	294.7	295.9	1.2	299.6	-4.9



(a) **(b)**

Figure 7. Equipment temperatures (K) considering natural convection.
(a) volumetric heat generation **(b)** superficial heat generation



(a) **(b)**

Figure 8. Velocities in a middle plane considering natural convection.
(a) volumetric heat generation **(b)** superficial heat generation

Table 3. Comparison of simulation equipment temperatures considering volumetric and superficial temperatures with natural convection

Equipment	T_{vol} (K)	T_{sup} (K)	$T_{vol} - T_{sup}$ (K)
T_1	298.1	298.6	0.5
T_2	307.7	304.4	-3.3
T_3	308.6	303.3	-5.3
T_4	303.3	299.8	-3.5
T_5	301.6	298.6	-3.0
T_6	300.7	299.8	-0.9
T_7	301.6	299.8	-1.8
T_8	312.9	305.6	-7.3
T_9	314.7	309.1	-5.6
T_{10}	309.4	305.6	-3.8
T_{11}	307.7	303.3	-4.4

Analyzing the results presented in Fig. 5 and Tab. 2 it can be verified that some differences in the simulation temperatures of the equipment occur, but they are not significant, except for the equipment 11 (at the top of the system). Otherwise, the velocities (Fig. 6) are quite different, what explains the temperature differences observed between the volumetric and superficial heat generation condition.

In the case of natural convection it can be more clearly verified that the temperatures of the equipment considering volumetric generation (Fig. 7 and Tab. 3) are higher than those with superficial heat generation. Also with natural convection the velocities (Fig. 8) in the case of volumetric heat generation are lower than those with superficial generation.

6. Conclusions

Differences of equipment temperatures between the volumetric and the superficial heat generation analysis could be verified. It is possible to see lower temperatures in the surfaces of electronic devices with superficial heat generation in comparison with volumetric heat generation. Different local velocities near to the equipments are also verified in the case with superficial generation and are the responsible for the non uniform temperatures on the devices.

If only the flow is on interest, then the superficial generation is the best approach, because it is more conservative. In the other hand, if the concern is with the temperatures on the equipment surfaces, the volumetric generation is a better approach.

7. Acknowledgements

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7. Responsibility notice