29th International Conference
ENGINEERING MECHANICS 2023
Milovy, Czech Republic, May 9 – 11, 2023

THE EFFECT OF VARIOUS POSITIONS OF THE STRAIGHT FIN HEAT SINK ON THE ELECTRICAL SCHWITCHBOARD COOLING

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Abstract: *This article is devoted to the numerical solution of simplified mathematical-physical model of the temperature field within and outside of real switchboard controlled by free convection. CFD is performed using OpenFOAM software utilizing several solvers (e.g. chtMultiRegionSimpleFoam) modelling the heat transfer and velocity field in the solid object surrounded by fluid. These solvers are based on the finite volume method. The process of numerical simulations is designed so that it was possible to work efficiently with multiple configurations and switchboards locations, including final processing and results evaluation. The field of temperature for three different cooler positions is assessed and the most optimal configuration is searched for.*

Keywords: Electrical switchboard, OpenFOAM, cooler, temperature field.

1. Introduction

Electrical switchboard (or distribution board) is a device that can distribute the electricity from one or more power supplies to other smaller panels or secondary electrical circuits. It consists of a frame (not present always), a covering, a door and some panels inside connected by busbars or wires. Each switchboard is designed to fulfill the ingress protection (IP rating – dust and water). Switchboards are widely used in industry or in commercial buildings, telecommunication facilities, oil and gas plants, data centers, health care, other buildings and etc. Many manufacturers offering a variety of solutions are established.

This device can provide protection for individual circuits using different fuses or circuit breakers. Normally, main switch button is present for the option of disconnecting from the power supply. This device commonly works with three-phase power supply. There are some requirements for panel board design – dynamic vibration level as well as seismic resistance. Heat resistance is also very important. Passive removal of residual heat is required for the installation of critical infrastructure.

The amount of power going into a switchboard must always be substantially equal to the power going out to the loads (less the losses in internal conductors and consumption by internal devices such as pilot lamps, space heaters, or others) - Hadzhiev et al (2014). The cooling of switchboards can be passive or active and the main goal is to maintain operating temperature range during operation (or even after the end). There are also other requirements as the maximal temperature on the covering surface to protect both the rubber parts and human service.

This paper deals with numerical calculation of temperature field of standard switchboard using in data center. Passive cooling is desirable, no ventilation grid is available and we consider three different variants: straight fin heat sink placed a) on the top b) on the side wall and c) without. The numerical simulations were conducted in OpenFOAM software, the validation was performed against experimental data measured on a scaled model using optical measurement method (not the part of this paper).

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ISSN: 1805-8256, doi: 10.21495/em2023-167

2. Formulation of the problem

The switchboard case is considered as a steel case of dimensions $850 \times 598 \times 2012$ mm with 2 mm thick outer cover and internal structure consisting of supporting frame, serving for fixation of various electronic devices. Switchboard is positioned in the middle of $4 \times 4 \times 3$ m room as a approximation of a standalone operating case without influence of surrounding space.

The problem is defined as: Find the maximal thermal load $\dot{Q}$ of a switchboard of given configuration so that maximal temperature on the switchboard surface is $T_{max} = 55$°C with given far-field temperature $T_{\inf} = 22$°C. Thermal load $\dot{Q}$ is divided in each particular device No. 1-5 as follows: $\dot{Q}_1 = 0.25\dot{Q}$, $\dot{Q}_2 = 0.3\dot{Q}$, $\dot{Q}_3 = \dot{Q}_4 = 0.2\dot{Q}$, $\dot{Q}_5 = 0.05\dot{Q}$. Moreover, two variants of thermal sink are considered. First, the side-mounted fin sink of 600×1200 mm. Second, top-mounted fin sink of 400×400 mm. Heat sinks are considered to be in perfect contact with the switchboard (see Fig. 1).

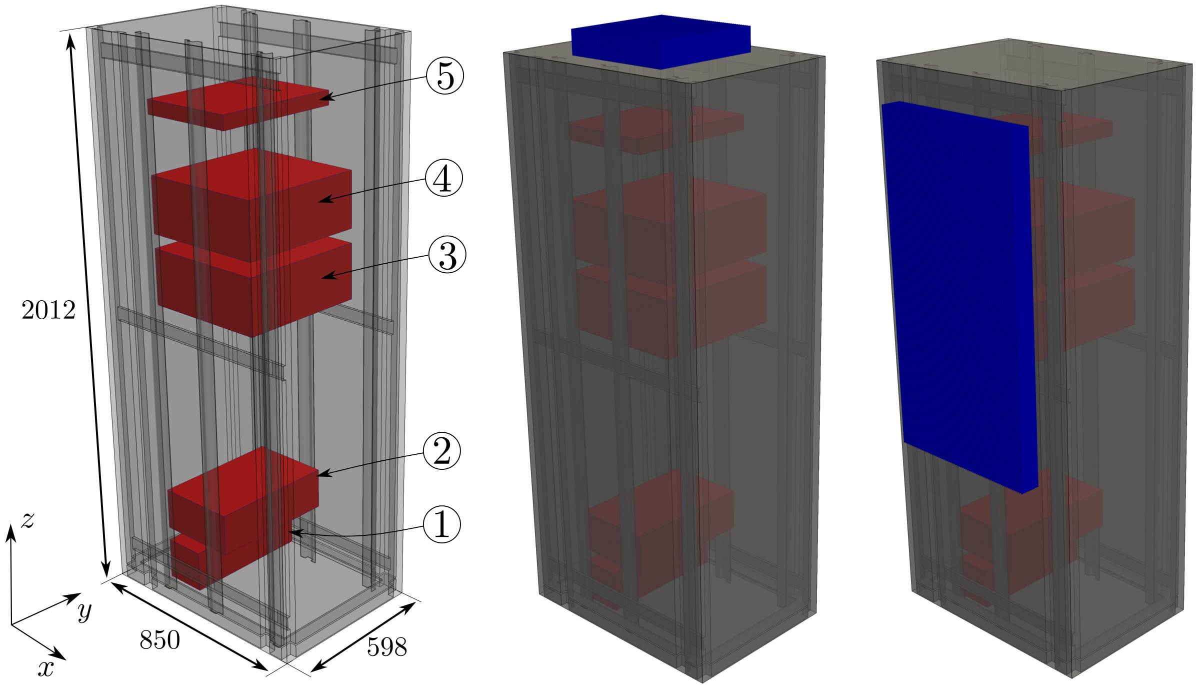

Fig. 1: Left: Geometry of the switchboard with five devices; dimensions in millimeters. Middle: Switchboard equipped with top-mounted heat fin sink. Right: Switchboard equipped with side-mounted heat fin sink.

3. Mathematical model

3.1. Fluid

The motion of a compressible gas in fluid regions is described by the following system of equations expressing conservation of mass, momentum, and energy:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j u_i) = -\frac{\partial p_{rgh}}{\partial x_i} - \frac{\partial(\rho g_j x_j)}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\tau_{ij}^v + \tau_{ij}^t\right) \tag{2}$$

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial(\rho u_j h)}{\partial x_j} + \frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho u_j k)}{\partial x_j} = -\frac{\partial(q_i + q_i^t)}{\partial x_i} + \frac{\partial p}{\partial t} - \rho g_j u_j + \frac{\partial(\tau_{ij} u_i)}{\partial x_j} \tag{3}$$

where ρ is density, u_i is velocity, p is total pressure, $p_{rgh} = p - \rho g_j x_j$ is dynamic pressure, g_j is acceleration due to the gravity, $\tau_{ij}^v = \nu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$ is viscous stress tensor, $\tau_{ij}^t = \nu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$ is turbulent stress tensor, ν is molecular viscosity and ν_t is turbulent viscosity, $h = e + \frac{p}{\rho} = c_p T$ is enthalpy with $k = \frac{1}{2}u_j u_j$

being kinetic energy, e internal energy, c_p specific heat at constant pressure and T temperature. Heat transfer by molecular and turbulent diffusion is q_i and q_i^t, respectively.

3.2. Solid

Heat transfer in solid region is given by Fourier's law:

$$\frac{\partial(\rho h)}{\partial t} = \frac{\partial}{\partial x_j}\left(\alpha\frac{\partial h}{\partial x_j}\right) + \dot{Q} \tag{4}$$

with $\alpha = \frac{\kappa}{c_p}$ being thermal diffusivity and κ thermal conductivity. The volumetric sources of heat are denoted as $\dot{Q}$.

3.3. Interface

Coupling conditions between solid and fluid regions:

$$T_f = T_s \tag{5}$$

$$\kappa_f\frac{\partial T_f}{\partial n_i} = -\kappa_s\frac{\partial T_s}{\partial n_i} \tag{6}$$

where T_f resp. T_s is coupled temperature at boundary of fluid resp. solid region and similarly κ_f resp. κ_s. Term $\frac{\partial}{\partial n_i}$ represents partial derivative with respect to i-th component of outer normal at the solid-fluid interface.

3.4. Turbulence

The effects of the turbulence are modeled using the turbulent viscosity approach within the RANS framework employing standard two-equation k-ω-SST model. The above system of equations is then closed by equation of state for ideal gas, $\frac{p}{\rho} = rT$.

4. Numerical simulations

The numerical solutions were obtained with a second-order, finite-volume method using multi-region steady-state solver *chtMultiRegionSimpleFoam* shipped with the OpenFOAM-v2112 package.

Computational mesh comprises of cca 2.2 mio, 0.8 mio, and 1.4 mio tetrahedral cells in the internal (fluid), switchboard (solid), and external (fluid) region, respectively.

As a boundary conditions were prescribed: $u_i = 0$, $\frac{\partial p_{rgh}}{\partial n_i} = 0$ for the walls and outer region boundary, with $p = 100$ kPa at the top outer wall. Outer walls temperature was set as $T = 22$°C and bottom of the switchboard was set as $\frac{\partial T}{\partial n_i} = 0$. The turbulent kinetic energy k_t and dissipation of turbulent kinetic energy ω were prescribed by wall functions.

Thermal sources were set via *codedFvOptions* as a volumetric heat sources for given sets of cells within each electronic device, see Fig. 2.

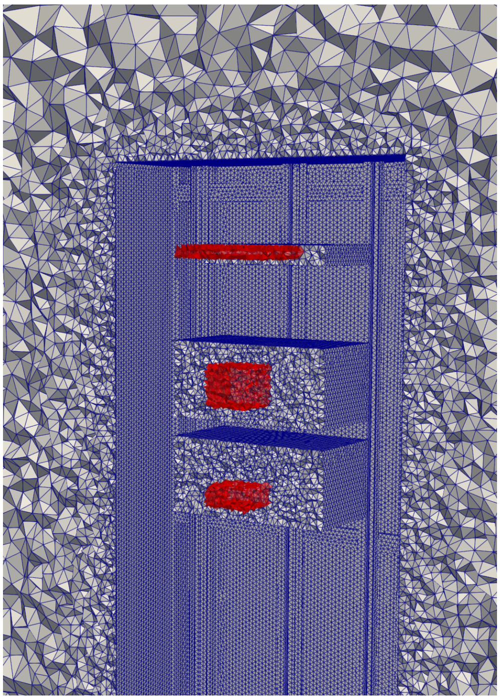

Fig. 2: Detail of a computational mesh in the vicinity of the switchboard; internal fluid region is not displayed. Red cell-sets represents heat sources within the solid region.

5. Results

Tab. 1: Maximal thermal load of the switchboard and the efficiency of the heat sink

	without heat sink	side heat sink	top heat sink
$\dot{Q}$ $[W]$	509.4	593.7	619.3
$\dot{Q}_{sink}$ / $\dot{Q}$ $[\%]$	-	25.1	10.2
$\dot{Q}_{sink}$ / S_{sink} $[Wm^{-2}]$	-	42.5	197.8

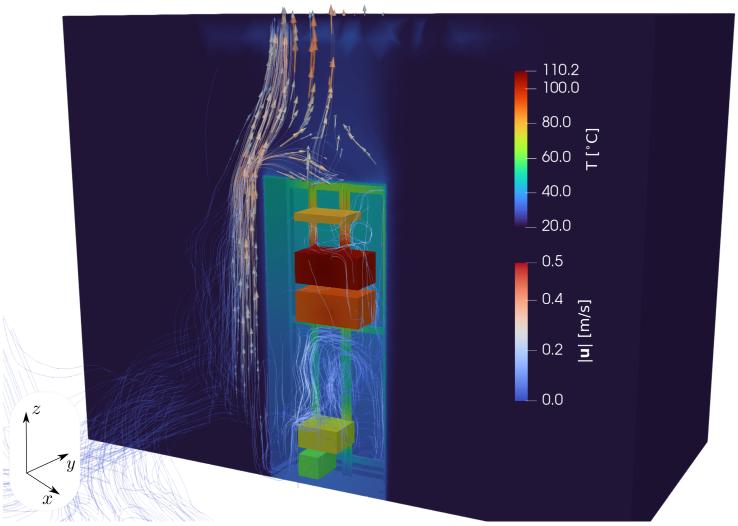

Fig. 3: Temperature field in partial cut of computational domain at $y = 0$ (internal region not displayed); velocity streamlines with tangent velocity vectors in the vicinity of outer surface of the switchboard and inside the switchboard.

6. Conclusions

The boundary conditions (firstly surface temperature) were defined according to the technical norm and maximal possible thermal load of the switchboard was evaluated. The case without passive cooling using fin sink is characterized by maximal thermal load of approx. 510 W. The sink located on the side wall enables the load increase by 16 %. Top-located sink works best despite the fact it has smaller area in comparison to the side-located sink. Then the thermal load can be increased by 21 % compared to the base case. More, the top-located sink is characterized by the best heat flux per unit area (see Table 1). Further calculations will follow with the aim of analyzing the influence of the location of the switchboard in the room and possibly in a row of other switchboards.

Acknowledgments

An institutional support RVO61388998 is gratefully acknowledged.

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