Simplified Modeling of Electrical Cabinets

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Calculating temperature inside an electrical cabinet by the use of e.g. VDI or ASHREA standards for sizing HVAC systems is not adequate because they are intended for: (i) sizing economically optimal HVAC systems, (ii) specific climate, (iii) average environment wind conditions, and (iv) typical building constructions. Electrical cabinets should secure functioning of electrical equipment in extreme weather conditions with as economical design as possible. This paper aims to present a relatively simple procedure to model temperature inside electrical cabinets and to analyze different cabinet constructions depending on the weather conditions. The model principle is based on seven dependent energy balances: on each side of a cabinet and of the airflow through it. Two different wall constructions as well as forced and natural ventilation of a cabinet were analyzed. The goal is to avoid using air conditioning systems in electrical cabinets and to use only if necessary fans or electrical heaters. Insulation of walls is suitable for colder continental climates whereas the use of walls consisting of two metal sheets, with air circulating freely between them, is suitable for hotter continental climates for the equipment with heat dissipation inside cabinets of up to 120 W/m^3.

Keywords: electrical cabinets, heat transfer, natural convection, forced convection, wall construction

1. INTRODUCTION

An electrical enclosure is a cabinet for electrical or electronic equipment to mount switches, knobs, and displays and to prevent electrical shock to equipment users and protect the contents from the environment [1]. For the proper functioning and duration of electrical and electronic equipment inside an outdoor electrical cabinet, it is very important to keep the inside temperature in a required temperature range. This range is limited by minimal and maximal (peak) operating temperatures specified by the producer. For the majority of electrical equipment, these temperatures lies in the range from -10 to 50 °C. Exposing the equipment to higher and lower temperatures than these affects their functioning and lifetime. Depending on the environment conditions and heat dissipation inside an electrical cabinet, the manufacturers require precise answers what kind of wall construction to use, whether to use fans and/or electrical heaters or even air conditioners to maintain the inside temperature in a required temperature range. For these reasons, it is very important to predict the temperature inside an electrical cabinet in a simple and reliable manner. Calculating temperature inside an electrical cabinet by the use of e.g. VDI or ASHREA standards for sizing HVAC systems is not adequate because they are intended for: (i) sizing economically optimal HVAC systems, (ii) specific climate and (iii) average environment wind conditions, and (iv) typical building constructions.

This paper aims to present a relatively simple procedure to model temperature inside electrical cabinets and to analyze different cabinet constructions in summer and winter weather conditions. The intention was to develop as simple as possible model that can be implemented to different constructions of electrical cabinets.

Fig. 1 shows the modeled electrical cabinet, which is intended to store electrical appliances in open space. The cabinet is 1.2 x 1.4 x 0.7 m and has a steady heat gain from electrical appliances of 140 W. The cabinet stands on metal support, with the height 100 mm above the ground. For air intake, this metal support has in total 60 openings, 30 on the front and 30 on the back side, each opening with the dimensions 60 x 3 mm (see Fig. 1 and 2). After passing through these openings, the air enters into the cabinet from the bottom (see Fig. 1 and 2). At the top, there is double metal sheet that forms an "attic" with the variable height from 50 to 70 mm (see Fig. 2).

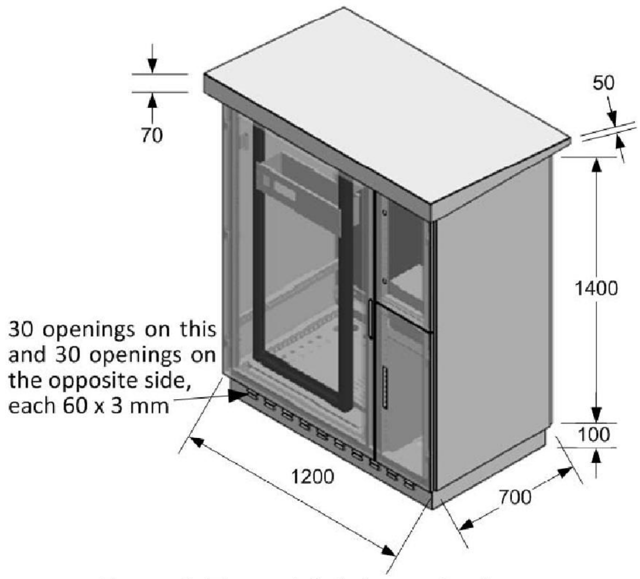

Figure 1: The modeled electrical cabinet.

Two different wall constructions are analyzed. The first one is composed of an outer metal sheet 1.55 mm thick, 20 mm K-FLEX ST insulation, and an inner metal sheet 0.55 mm thick. The proposed insulation has thermal conductivity 0.034 W/mK at -20°C, 0.036 W/mK at 0°C, and 0.040 at 40°C [2], see Fig. 2. The other common insulating materials that could be implemented have the thermal conductivity in the same range, see e.g. [3,4]. The

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second analyzed wall construction is composed of two metal sheets, both 1.55 mm thick, at the distance 20 mm between them, see Fig. 2. In this space air flow is created by natural draft. To construct this kind of wall with forced convection is also possible but the manufacturing of the cabinet would be more difficult and expensive comparing with the analyzed case.

In the paper, the natural and forced draft through the cabinet are analyzed. In addition, submodels for heat transfer modelling of different parts of the cabinet are presented for the summer and the winter weather conditions.

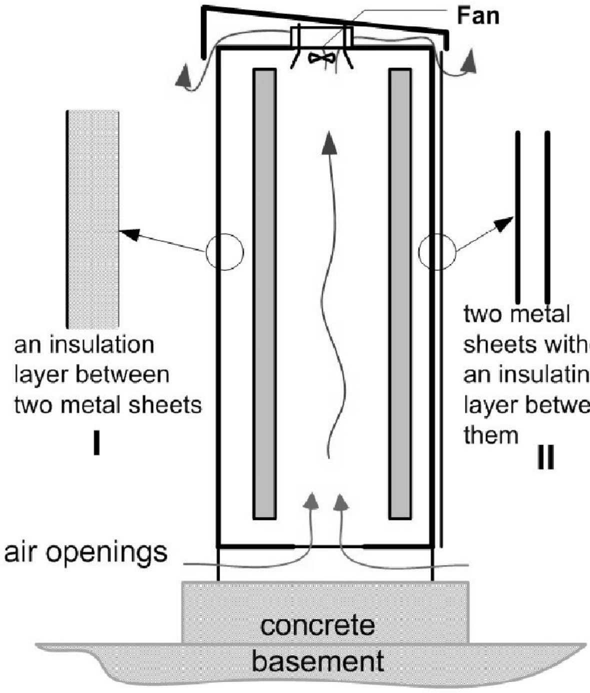

Figure 2: Cross section of the modeled cabinet. Two types of outer walls are examined. The first type of the wall I consists of two metal sheets with a 20 mm insulating layer between them. The second type II consists of two metal sheets that form 20 mm thick air passage between them.

2. MODELLING

Heat flow rate through the cabinet enclosure equals heat transfer rate of the air flowing through the cabinet:

$$\sum_{i=1}^{4} \dot{Q}_{wall,i} + \dot{Q}_{top} + \dot{Q}_{bottom} = \dot{m}_{air} c_{p,air} (t_{out} - t_o), \quad (1)$$

where $\dot{Q}$ in W are heat flow rates through the different parts of the enclosure, $\dot{m}_{air}$ is the mass flow rate of air in kg/s, $c_{p,air}$ is the specific heat capacity of air at constant pressure in kJ/kgK [5], t_o, t_{out} are in °C the temperatures of air at the inlet (environment temperature) and outlet of the cabinet respectively. As can be seen in (1) there is no accumulation of heat in the cabinet. This assumption is made because the mass of an electrical cabinet is relatively small and the majority of parts are made from metals, which have low heat capacity.

2.1. Heat transfer through the side walls

The heat gain or loss through the insulated wall, see Fig. 3 (a), is calculated by

$$\dot{Q}_{wall,i} = kA_i \Delta t_i \quad (2)$$

where k is the overall heat transfer coefficient W/m^{2}K, A is the area of the wall m^2, and Δt_i is the temperature difference. The overall heat transfer coefficient is

$$k = \frac{1}{\frac{1}{\alpha_{in}} + \sum_i \frac{\delta_i}{\lambda_i} + \frac{1}{\alpha_{out}}}. \quad (3)$$

In (3) α is the heat transfer coefficient in W/m^{2}K, subscripts in and out denote inner and outer side of the wall, respectively. λ is the thermal conductivity in W/mK, and δ is the thickness of a wall layer in m. The heat transfer coefficient for the average wind conditions at the outer side of the wall α_{out} are recommended to be 25 W/m^{2}K [3] and 17 W/m^{2}K [6] during the winter and summer period, respectively. The value for the winter period is used here whereas for the summer period, still weather is assumed and the heat transfer coefficient is calculated according to the model [7] for free convection near a vertical surface. These assumptions are on the safe side because wintry conditions accompanied with very cold weather lead to a very low temperature inside an electric cabinet. Oppositely, hot and still weather could cause overheating of a cabinet.

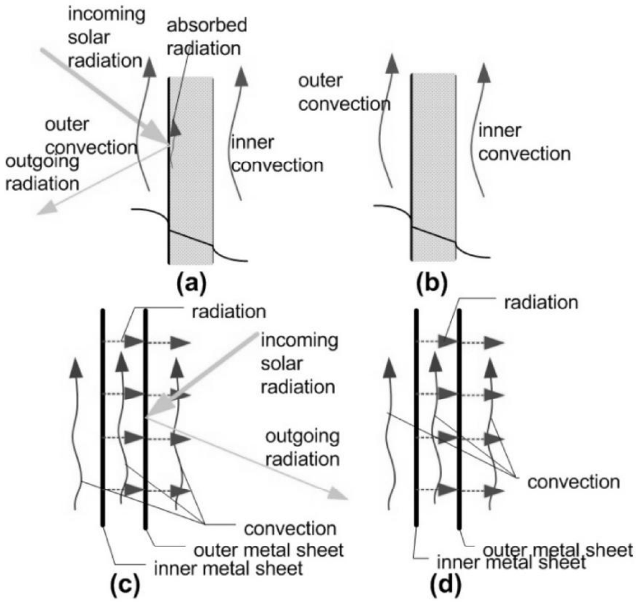

Figure 3: Heat transfer through the analyzed cabinet wall constructions: (a) insulated wall exposed to the Sun light, (b) insulated wall in the shade, (c) non-insulated wall with an air layer exposed to the Sun light, (d) non-insulated wall with an air layer in the shade.

To calculate natural convection at the outer wall surface, the average dimensionless heat transfer coefficient ([8] as cited in [7]) for laminar and turbulent flows near a vertical surface in the range from Ra = 10^{-1} to Ra = 10^{12} is defined by ([9] as cited in [7])

$$Nu = \{0.825 + 0.387[Ra f_1(Pr)]^{1/6}\}^2 \quad (4)$$

The function f1(Pr) allows for the effect of the Prandtl number in the range 0.001 < Pr < ∞ ([10] as cited in [7]):

$$f_1(Pr) = \left[1 + \left(\frac{0.492}{Pr}\right)^{9/16}\right]^{-16/9} \quad (5)$$

In (4) and (5), Pr, Nu, Ra, Gr are the Prandtl, Nusselt, Rayleigh Ra=GrPr, and Grashof number, respectively. The equations to calculate these numbers can be found e.g. in [11,12]. The characteristic length to calculate Nu, Ra, and Gr number is the height of the surface. The physical properties of air necessary to calculate these dimensionless numbers are taken at the average temperature $\frac{1}{2}(t_{wall} + t_0)$, which means that the iterations are needed to solve the system of equations. Two important assumptions regarding the temperatures are:

- an outer wall has a constant surface temperature. As the outer walls are made of metal sheet this assumption is in accord with reality, and
- the inner enclosure of a cabinet is at a constant temperature equal to the temperature of electrical equipment inside it. This temperature is the temperature of the cabinet. This assumption is made because the electrical equipment, which is mostly made of metal, is in a physical contact with the inner metal sheet of a metal cabinet.

To calculate heat gain or loss from the walls in the shade, for each wall (2)-(5) are used. The radiant heat gain or loss from these walls is neglected.

For the walls exposed to the Sun light, a sol-air temperature is calculated and then the same procedure using (2)-(5) implemented (see Fig. 3 (a) and 3 (b)). Sol-air t_{sa} temperature is a variable used to calculate cooling load of a building and determine the total heat gain through exterior surfaces [13]. It is the temperature, which under conditions of no direct solar radiation and no air motion, would cause the same heat transfer into a house as that caused by the interplay of all existing atmospheric conditions [14]. This temperature takes into account solar radiative flux and infrared exchanges from the sky and is calculated by [6]

$$t_{sa} = t_o + \frac{aI}{\alpha_s} - \frac{\varepsilon \Delta R}{\alpha_s}. \qquad (6)$$

In (6) a is the degree of absorption, which is according to the Kirchhoff's law equal to the degree of emission. For a shiny iron sheet with zinc surface the degree of emission is 0.228 [15].

In (6) I is the global solar irradiance in W/m^2. The data on solar irradiance depending on local time, latitude, and longitude can be found in [6,16]. For the 20th of July, 15^{00} local time, the global solar irradiance on the horizontal surface is 634.3 W/m^2 and on the vertical surface is 595.1 W. α_s is the heat transfer coefficient calculated by (4) and (5).

In the last term on the right side of (6), ε is the degree of emissivity and ΔR in W/m^2 accounts for the infrared radiation due to difference between the external air temperature and the apparent sky temperature and radiation that the surface emits as a black body at the environment temperature [6]. In this paper this term is neglected as it is usually neglected for vertical surfaces in determining cooling load [6].

Heat transfer for the wall made of double metal sheet (see Fig. 3 (c) and (d)) with an air layer is composed of radiant and convective heat gain or loss and is determined by the use of a system of equations. For this kind of wall exposed to the Sun light two energy balance equations (7) and (13) and the equation for the natural draft through the space between metal sheets (14) are used. The energy balances are for the outer surface (8) and airflow through the space between these surfaces (14). Solar irradiance to the outer surface (see Fig. 3 (c)) is used to heat air on both sides of the wall and by radiation transferred to the inner surface:

$$\varepsilon I A = \alpha_{out} A(t_{wall} - t_o) + \alpha_{in} A(t_{wall} - t_a) + \frac{\sigma}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} A(T_{wall} - T) \qquad (7)$$

t_o, t_{wall}, t_a in °C are the temperatures of the environment, the wall and the air flowing through the layer between two metal sheets, respectively. T_{wall}, T are absolute temperatures in K of the wall and inner metal sheet, respectively, and A is the surface area of the wall in m^2. As it is already mentioned and explained, the temperature of the inner metal sheet is equal to the cabinet temperature. α_{out} in W/m^{2}K is the heat transfer coefficient at the outer side of the wall determined by (4) and (5), which means that the subsystems (7)-(9) should be solved by iteration too. Still weather during the summer period is also assumed here. The third term on the right-hand side of (7) is the expression for the net thermal energy transferred by thermal radiation from the hotter to the colder surface for parallel plates of equal size. In this term $\varepsilon_1 = \varepsilon_2 = 0.228$ and $\sigma = 5.6704 \cdot 10^{-8}$ W/m^{2}K^4 is Stefan-Boltzmann constant.

α_{in} in W/m^{2}K is the heat transfer coefficient at the inner side of the wall. As heated vertical channels act as chimneys, the model for free convection in internal flows [17] is used to described this kind of convection for vertical channels:

$$Nu_s = Nu_s(Gr_s \cdot Pr) \qquad (8)$$

$$Nu_s = \frac{\alpha s}{\lambda} \qquad (9)$$

$$Gr_s^* = \frac{g\beta(T_w - T_A)s^3}{\upsilon^2} \cdot \frac{s}{h} \qquad (10)$$

$$Pr = \frac{\nu}{a} \qquad (11)$$

In the above equations, T_w and T_A are the wall and the air temperatures in K, respectively. β is the coefficient of volume expansion. The characteristic length s from which Nu and Gr are calculated is the width of the channel s=d in m. The reference temperature for the properties is the average between the wall and the air temperature. The Nusselt number, which is used to calculate the heat transfer coefficient in the duct, is calculated by:

$$Nu_s = 0.69(Gr_s^* Pr)^{1/4}. \qquad (12)$$

The energy balance for the air flow through the space between two metal sheets (see Fig. 3 (c) and (d)) is

$$\alpha_{in} A(t_{wall} - t_a) + \alpha_{in,2} A(t - t_a) = \dot{m}_{air,c} c_{p,air}(t_{out,c} - t_o) \qquad (13)$$

In (13) subscript c means channel, o at the environment temperature and *out* at the exit of the channel. The channel is open towards atmosphere at the top and at the bottom. t in °C is the temperature of the inner metal sheet, which is assumed to be equal to the temperature of the cabinet. $\alpha_{in,2}$ is obtained in the same manner as α_{in} but has a different value due to different temperature difference at the inner side of the wall.

Mass flow rate of air in the channel must be equal to the mass flow rate of air cause by natural draft in the channel

$$\dot{m}_{air,c} = \rho_o \frac{\pi d_h^2}{4}\left[\frac{2g(\rho_o-\rho_{out})h}{\frac{\lambda l\rho_{out}}{d_h}+\sum\zeta\rho_{out}}\right]^{1/2}, \qquad (14)$$

where ρ in kg/m^3 is the air density at the entrance o, and at the exit *out* of the channel, λ is the friction coefficient, l in m is the length of the channel (height in this case), d_h in m is the hydraulic diameter, $\sum\zeta$ is the summarized minor loss coefficient. Finally the heat loss or gain from this wall is

$$\dot{Q}_{wall} = \alpha_{in}A(t-t_a) + \frac{\sigma}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}A(T_{wall}-T) \qquad (15)$$

(15) is also applied to calculate heat transfer from the non-insulated wall that stands in the shade, which is shown in Fig. 3 (d). The global solar irradiance in this case is zero, and two assumptions are made: (i) the outer metal sheet is at the ambient temperature $t_{wall} = t_o$, and (ii) the inner metal sheet is at the cabinet temperature. The characteristic length s from which Nu and Gr are calculated is the half of the channel width s=d/2 in (8)-(11) and instead (12), the following equation is used to calculate the Nusselt number

$$Nu_s = 0.61(Gr_s^* Pr)^{1/4}. \qquad (16)$$

For this wall, the same procedure with the same assumptions, which is used for the wall in the shade is used to calculate the heat loss from the cabinet during the winter period, because in the winter critical weather conditions for the equipment operation are during the night.

2.2. Heat transfer at the bottom of the cabinet

A mechanism of the heat transfer at the bottom of the cabinet in the summer period is shown in Fig. 4. The critical period is when the front or backside of the cabinet is exposed to the solar irradiance, see Fig. 1. In that case, a part of solar irradiance εIA_b, where A_b is the net area of the bottom exposed to the Sun light, hits the metal part. All the absorbed heat is transferred to the circulating air:

$$\varepsilon IA_b = \dot{m}_{air}c_{p,air}(t_{in,c} - t_o) \text{ where,} \qquad (17)$$

$t_{in,c}$ in °C is the temperature of air entering into the cabinet from below.

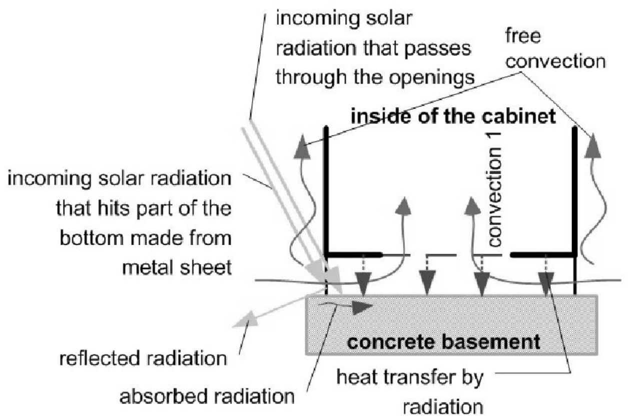

Figure 4: Heat transfer at the bottom of the cabinet. Convection 1 is forced convection when a fan is used to circulate air through the cabinet, otherwise it is free convection. Equal airflows from both sides are assumed.

This assumption (17) is made because there are 30 openings in the bottom part of the cabinet, so there is a relatively large area for the heat transfer between metal sheet and the incoming air. The solar irradiance that passes through the openings $A_{openings}$, which are modeled as black bodies, heats the concrete basement beneath the cabinet. This thermal radiation heats the concrete beneath the cabinet, which is assumed to be at 40°C. This is on the safe side because in practice this temperature is during summer lower than the ambient temperature. The heat transfer by radiation from the metal sheet to the concrete beneath the cabinet is

$$\dot{Q}_{bottom} = \frac{\sigma}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}A_{bottom}(T - T_{concrete}), \qquad (18)$$

where ε_1 and $\varepsilon_2$are the degrees of emissivity for the shiny iron sheet with zinc surface and concrete, respectively.

During the winter weather there is no need for forced circulation of air through the cabinet. In that case As there is no need for forced circulation of air through the cabinet during winter, the heat transfer at the bottom of the cabinet is composed of the radiation and convection

$$\dot{Q}_{bottom} = \dot{Q}_{bottom,convection} + \dot{Q}_{bottom,radiation}. \qquad (19)$$

Heat transfer rate by radiation is calculated by (18) assuming that the temperature of the concrete beneath the cabinet is -3°C. This assumption is made according to the recommendations in [3].

Convective heat transfer is due to natural convection in the bottom space

$$\dot{Q}_{bottom,convection} = \alpha_b A_b(t - t_o). \qquad (20)$$

The temperature of air in that space is assumed to be equal to the environment temperature t_o, and the temperature of bottom part of the cabinet is assumed to be equal to the cabinet temperature. To calculate heat transfer coefficient, the model for free convection over horizontal surface [7] is used. For turbulent flow i.e. $Raf_2(Pr) > 7\cdot 10^4$ the Nusselet number, which is used to calculate heat transfer coefficient is calculated by [7]

$$Nu = 0.15[Raf_2(Pr)]^{1/3}. \qquad (21)$$

The function $f_2(Pr)$ defines the effect of the Prandtl number over the entire range 0<Pr<∞ and is given by [7]

$$f_2(Pr) = \left[1 + \left(\frac{0.322}{Pr}\right)^{11/20}\right]^{-20/11}. \qquad (22)$$

The characteristic length for calculating Ra and Nu numbers is l=ab/2(a+b) for rectangular surfaces. The physical properties of air are taken at the average temperature.

2.3. Heat transfer at the top of the cabinet

Heat transfer at the top of the cabinet is composed of radiation and convection

$$\dot{Q}_{top} = \dot{Q}_{top,convection} + \dot{Q}_{top,radiation}. \qquad (23)$$

The model of the heat transfer at the top of the cabinet is shown in Fig. 5. Absorbed solar irradiance is transferred from the top metal sheet by natural convection to the outside air, by forced convection to the inside air and by radiative heat transfer to the lower metal sheet:

$$\varepsilon I_t A_t = \alpha_{t,o}A_t(t_{sa,t} - t_o) + \alpha_{t,iu}A_t\left(t_{sa,t} - \frac{t_{a,in}+t_{out}}{2}\right) + \frac{\sigma}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}A_t(T_{sa,t} - T). \quad (24)$$

Air that flows in the attic above the cabinet exchanges heat by forced convection with the upper and the lower metal sheet

$$\dot{m}_{air}c_{p,air}\left(t_{out}-t_{a,in}\right)=\alpha_{t,iu}A_t\left(t_{sa,t}-\frac{t_{a,in}+t_{out}}{2}\right)+\alpha_{t,il}A_t\left(t-\frac{t_{a,in}+t_{out}}{2}\right) \qquad (25)$$

The subscripts t, o, i, sa, u, l in (24) and (25) mean top, outside, inside, sol-air, upper, and lower, respectively. $t_{a,in}$ and t_{out} in °C are the inlet and the outlet air temperature in the space at the top of the cabinet, respectively. In other words, $t_{a,in}$ is the temperature at the exit of the fan (see Fig. 5). It was assumed that the lower metal sheet is at the temperature T of the cabinet, because air flows through openings in that surface. It is also assumed that the airflow is divided into two equal streams.

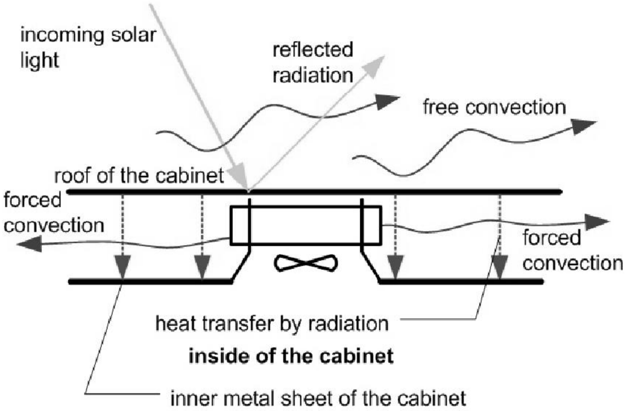

Figure 5: Heat transfer at the top of the cabinet.

$T_{sa,t}$ in K and $t_{sa,t}$ in °C is the sol-air temperature at the upper metal sheet. This temperature is calculated neglecting infrared exchanges from the sky by

$$t_{sa_t}=t_o+\frac{aI_t}{\alpha_{t,o}}. \qquad (26)$$

The heat transfer coefficient above the top cover of the cabinet $\alpha_{t,o}$ in W/m^{2}K is calculated by the model for free convection over horizontal surface [7] given by (21) and (22).

The heat transfer coefficients in W/m^{2}K inside the attic, at the upper metal sheet $\alpha_{t,iu}$, and at the lower metal sheet $\alpha_{t,il}$, are both calculated by the model for forced convection presented in [18]. The procedure consists of calculating the Reynolds number first

$$Re=\frac{wd_e}{\nu}, \qquad (27)$$

where w is the velocity of air, ν kinematic viscosity, and $d_e=4A/O$ equivalent diameter. As all the flows that are analyzed in this paper cause forced convection in the the transition region $2300 \le Re \le 10^4$, the model of Gnielinski [18] is used. The model interpolates the regime between permanent laminar and turbulent flow. The equation is as follows

$$Nu=(1-\gamma)Nu_{lam,2300}+\gamma Nu_{tub,10^4} \qquad (28)$$

where γ is given by

$$\gamma=\frac{Re-2300}{10^4-2300}, \text{ and } 0\le\gamma\le 1. \qquad (29)$$

In (26) $Nu_{lam,2300}$ is the Nusselt number at Re=2300 obtained by (30)-(32), whereas $Nu_{tub,10^4}$ is the Nusselt number at Re=10^4 obtained by (33).

For constant wall temperature and laminar flow

$$Nu_{m,T,2300}=\{49.371+(Nu_{m,T,2,2300}-0.7)^3+Nu^3_{m,T,3,2300}\}^{1/3} \text{ where} \qquad (30)$$

$$Nu_{m,T,2,2300}=1.615(2300Pr\frac{d_i}{l})^{1/3} \text{ and} \qquad (31)$$

$$Nu_{m,T,3,2300}=\left(\frac{2}{1+22Pr}\right)^{1/6}(2300Pr\frac{d_i}{l})^{1/2}. \qquad (32)$$

For constant wall temperature or constant heat flux and turbulent flow

$$Nu_{m,10^4}=\frac{(\frac{0.0308}{8})\cdot 10^4 Pr}{1+12.7\sqrt{\frac{0.0308}{8}}(Pr^{\frac{2}{3}}-1)}\left[1+(\frac{d_i}{l})^{2/3}\right]. \qquad (33)$$

The range of validity for this model is $2300 \le Re \le 10^4$, $0.6 \le Pr \le 1000$ and $d_i/l \le 1$.

The Prandtl number of gases depends very little on temperature [18]. The effect on heat transfer exerted by variation of air properties is taken into account by [18]

$$Nu=Nu_m(\frac{T_{air}}{T_W})^{0.45}. \qquad (34)$$

T_{air} and T_W are the average air and wall temperatures, respectively.

The model used to calculate heat loss at the top during the winter period is similar to the one used at the bottom of the cabinet and is composed of convective and radiative heat losses:

$$\dot{Q}_{bottom}=\dot{Q}_{bottom,convection}+\dot{Q}_{bottom,radiation}. \qquad (35)$$

Three main assumptions are: (i) there is no airflow in the attic, (ii) the temperature of the lower metal sheet is equal to the cabinet temperature, and (iii) the temperature of the upper metal sheet is equal to the environment temperature. These assumptions allow the radiative heat loss to be calculated by the equations for radiative heat transfer between parallel plates of equal size (see the third term on the right-hand side of (24)). The convective heat loss is calculated by the model used to described natural convection over horizontal plates given by (21) and (22) [7].

2.4. Heat transfer inside the cabinet

During the winter period only the cabinet enclosure transfers heat with the surroundings whereas during summer there is airflow through the cabinet. The flow causes forced or natural convection inside the cabinet depending whether natural or forced circulation of air is created. If there is the forced circulation of air through the cabinet, the air receives heat by forced convection

$$\dot{m}_{air}c_{p,air}\left(t_{a,in}-t_{in,c}\right)=\alpha_{in}A_{in}\left(t-\left(\frac{t_{a,in}+t_{in,c}}{2}\right)\right), \qquad (36)$$

where α_{in} in W/m^{2}K is the heat transfer coefficient inside the cabinet, A_{in} in m^2 is the area for heat transfer inside the cabinet, and all other properties are explained in (1), (17), and (25). α_{in} is in this paper calculated by the use of the model for forced convection [18], which is explained in (27)-(34).

If air flow due to natural draft is used in the cabinet, it complicates its construction, because the air flow should be prevented during the winter. (1) and an additional equation for the air flow by natural draft should be solved. This equation is identical with (14), and should be solved overall and for all three sections of the cabinet through witch air flows. In addition, to calculate α_{in} in (36), the model used for natural convection, which is presented in (8)-(12), should be used.

3. RESULTS

For both analyzed cabinet construction it is taken: a constant internal heat dissipation of 140 W, the forced ventilation through the cabinet with the air exchange rate of 120 h^{-1} or 141.12 m^3/h, the area of $A_{in} = 10.08\ m^2$ for the internal heat transfer between the air and the interior of the vertical surface perpendicular to the sunlight, respectively.

The winter weather conditions are defined by: night time, average windy conditions at -20 °C.

the cabinet, and 50% of the cabinet cross section, i.e. 0.5(1.2x0.7) m^2 is free for airflow.

The summer weather conditions are defined by: the date is the 20th of July, daytime at 3 p.m., still weather with the temperature of 40°C. 634.3 W/m^2 and 595.1 W/m^2 are the global solar irradiance on the horizontal and

Results of the analysis for the summer weather conditions and the insulated cabinet are shown in Table 1 and in Fig. 6.

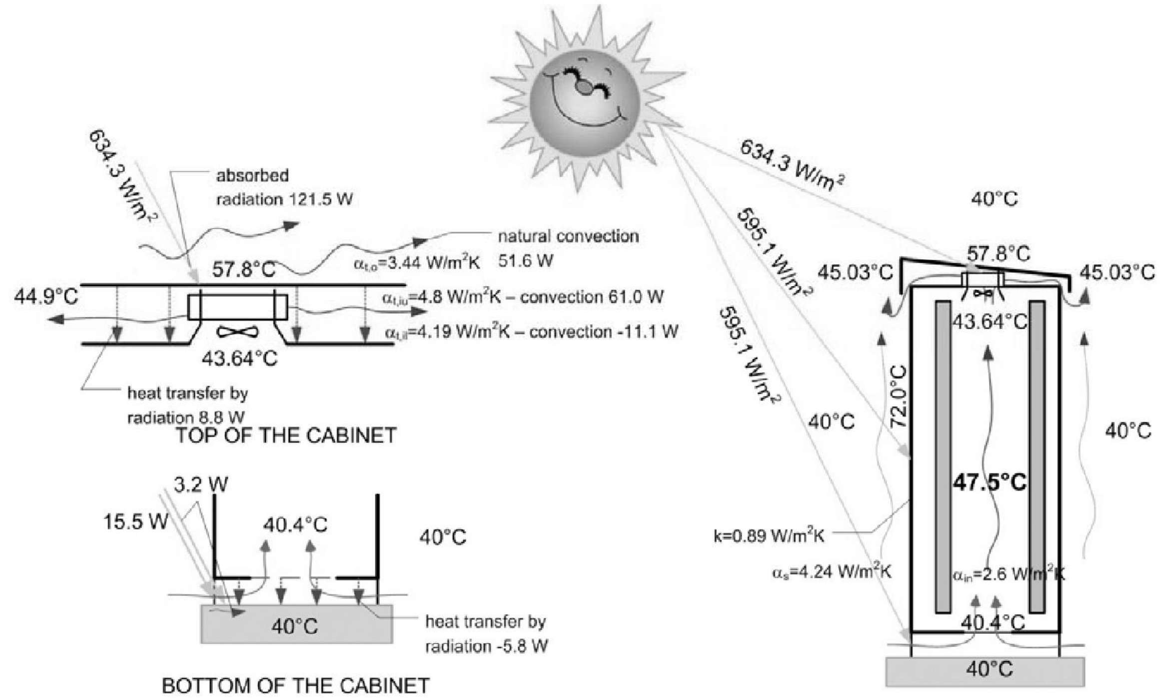

Figure 6: The characteristics of the heat transfer for the insulated cabinet and the summer weather conditions.

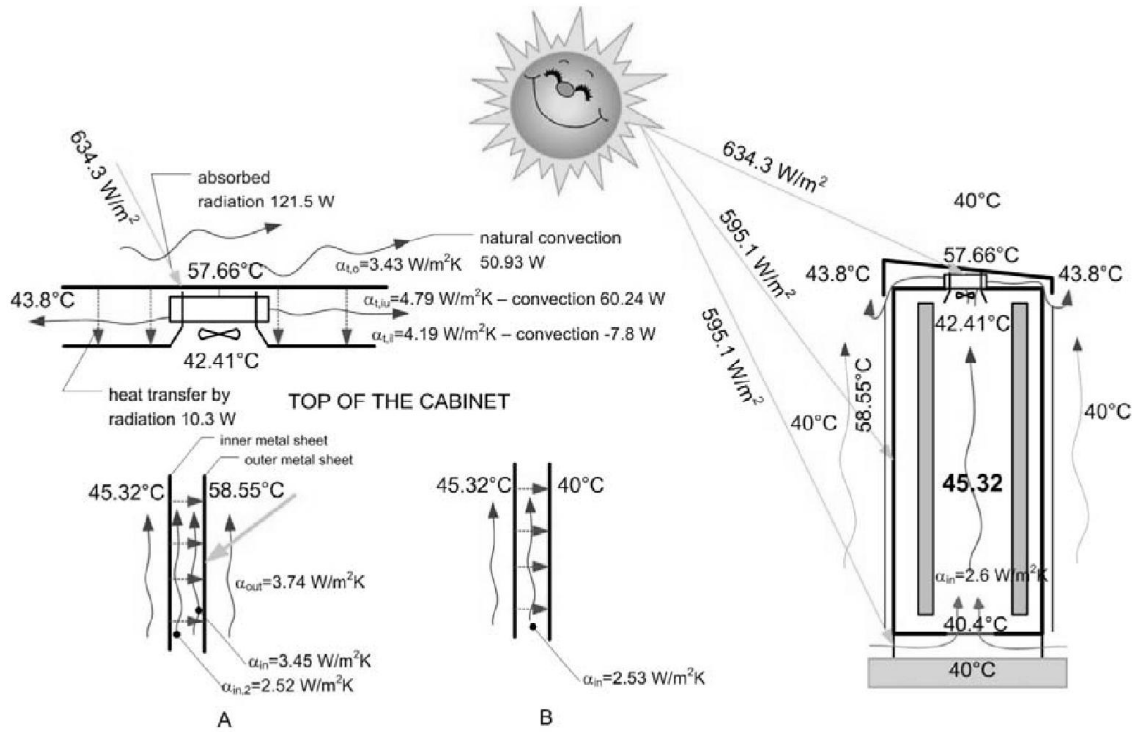

Figure 7: The characteristics of heat transfer for the non- insulated cabinet and the summer weather conditions. A- wall exposed to the Sun light, B – wall in the shade

Table 1: The energy balance for the insulated cabinet for the summer weather conditions. + is for heat gains, - for heat losses of the cabinet

Side of the cabinet	Area m^2	Radiation W	Convection W	Total W
Top	0.84	8.8	-11.1	-2.4
Side exposed to the Sun light	1.68	-	36.7	36.7
Rear side	1.68	-	-11.2	-11.2
Lateral side 1	0.98	-	-6.6	-6.6
Lateral side 2	0.98	-	-6.6	-6.6
Bottom	0.84	-5.8	0	-5.8
Heat gains of air inside the cabinet (W)				-144.1
Heat gains from thermal appliences				140,0

Applying the described procedure for the winter weather conditions and the insulated cabinet gives the temperature in the cabinet -2.8°C.

For the non-insulated cabinet the results for the summer weather conditions are shown in Table 2. and Fig. 7. Results shown in Fig. 6 for the bottom part of the insulated cabinet are identical for the non-insolated cabinet because the constructions of these parts of the cabinets are identical.

Table 2: The energy balance for the non-insulated cabinet for the summer weather conditions.

Side of the cabinet	Area m^2	Radiation W	Convection W	Total W
Top	0.84	10.3	-7.8	2.5
Side exposed to the Sun light	1.68	22.3	-3.8	18.5
Rear side	1.68	-8.2	-22.6	-30.8
Lateral side 1	0.98	-4.8	-13.2	-18
Lateral side 2	0.98	-4.8	-13.2	-18
Bottom	0.84	-4.1	0	-4.1
Heat gains of air inside the cabinet (W)				-90.2
Heat gains from thermal appliences				140,0

Applying the described procedure for the winter weather conditions and the non-insulated cabinet gives the temperature inside the cabinet -13.8°C.

4. CONCLUSIONS

This analysis is carried out for the specific heat gain inside the cabinet of 119 W/m^3, which means that the following conclusions apply for the cabinets with similar specific heat gains. Generally, to avoid using an air conditioning system in an electrical cabinet in the continental climate, air circulation should be promoted during the summer period, but during the winter period it should be prevented. In addition, to prevent absorption of solar radiation, the outer surface of an electrical cabinet should have as low as possible the degree of absorption.

Electrical cabinets with insulated walls are suitable for cold weather. The insulation prevents giving off heat during hot weather. Depending on the heat dissipation of the containing electrical equipment, this kind of cabinets are more likely to require forced ventilation during summer periods whereas during winter period they should need only proper air tightness. Insulated walls should be used when the containing electrical equipment has lower levels of heat dissipation, less than 120 W/m^3.

On the other hand, the cabinets, which have walls consisting of two metal sheets with ambient air freely circulating between them would for the heat dissipation level of 120 W/m^3, inside the cabinet require only natural ventilation during summer period, whereas during winter, except air tightness, they would require additional electrical heaters. This wall construction is suitable for hot conditions and higher levels of heat dissipation inside the cabinets, larger than 120 W/m^3. To release heat more easily, this construction could be improved with forced air circulation between two metal sheets, which would require a bit complicated manufacturing of the cabinets.

The weak point of this paper is the lack of experimental verification. The reason is that the company for which we developed this modeling procedure has not yet produced the designed electrical cabinet. Nevertheless, the modeling procedure uses verified heat transfer models as well as all the assumptions were made on the safe side.

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