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HEAT TRANSFER CONSIDERATIONS FOR DC BUSBARS SIZING

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Abstract

The main DC busbars connecting the rectifiers to the potrooms or connecting the potrooms of a potline typically consist of several naturally cooled parallel aluminum bars. For economic optimization, busbar sizing is usually based on the minimum aluminum mass at the maximum allowable temperature. An adequate representation of the system's heat dissipation is therefore required for the efficient and effective design of both the busbars (preventing excessive costs or overheating) and expansion joints.

To accomplish this, detailed Computational Fluid Dynamics (CFD) simulations are used to evaluate the heat losses from a tunnel enclosed group of busbars to the surroundings. In addition, a simplified calculation methodology based on semi-empirical convection correlations and analytical radiation view factors is proposed for the fast evaluation of different design parameters.

The approach is discussed in the context of potential industrial applications, for example increasing the line amperage of existing potlines, to illustrate the methodology's value.

Introduction

In a potline, the DC current is passed from one cell to another by means of busbar systems, typically naturally cooled and arranged in a parallel fashion. To reduce cost, sizing of the bars is usually based on the minimum bar cross-section at the maximum allowable temperature. The cooling of busbars is a non-linear heat-generation, conduction and convection problem and adequate heat dissipation is a primary design consideration. With the amperage creep typical in the smelting industry, the existing busbars tend to experience increased surface temperatures and, consequently, increased thermal stresses. Higher compression levels in the flexible expansion joints are likely to be expected as a result. Past experience has shown that busbars submitted to poor cooling conditions may experience catastrophic failure (melting) and that the expansion joint flexibles of such conductors may undergo severe plastic deformation.

Two main groups of busbar systems can be identified: pot-to-pot busbars and liaison busbars. While the first group is always submitted to the draft of the potroom's chimney effect, the latter may be enclosed inside tunnels and, eventually, be cooled by a natural convection flow generated by the temperature difference between the busbars themselves and the ambient air. Different approaches have been proposed in order to study the cooling of aluminum busbars carrying DC current, such as: determination of the maximum allowable current using analytical models and empirical correlations, the modeling of the 2D heat conduction problem submitted to different convection and radiation boundary conditions, and the 3D modeling of the flow that cools the busbars itself.

The objective of this article is to propose a simplified methodology for the analysis of different parameters (such as busbar geometry and spacing, ambient conditions and current density) on the heat transfer problem of busbars in enclosed tunnels. The proposed 2D approach is based on semi-empirical convection correlations and analytical radiation view factors. A test case is analyzed with a detailed 3D CFD model and its results are used as a basis for comparison with the 2D model outputs. Measurements of in-plant busbar temperatures (not reported here) support the methodology

Description of the Busbar Electro-Thermo-Mechanical Phenomena

Figure 1 shows the internal heat generation and the cooling mechanisms acting over the innermost component of a group of identical busbars enclosed inside a tunnel. Note that the tunnel walls are assumed to be submitted to a sufficient heat extraction rate in order to be able to maintain a uniform ambient temperature T_∞. The considered bar and its immediate neighbours are assumed to be at thermal equilibrium. Furthermore, the current is assumed to be evenly distributed between the conductors.

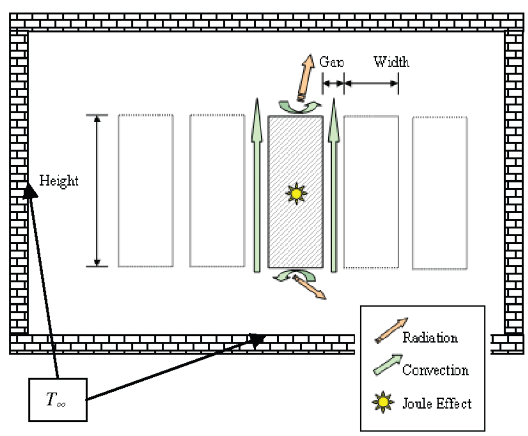

Figure 1 – Heat internal generation and cooling mechanisms for the inner component of a tunnel encapsulated busbar group.

Busbars are internally heated by the Joule Effect, Equation (1):

$$q'''_{gen} = \left(\frac{\rho}{A^2}\right) I^2 \qquad (1)$$

Where: q'''_{gen} is the generated heat per unit volume, [W/m^3]; $\rho = f(T)$ is the temperature-dependant material's electrical resistivity, [Ω.m]; A is the busbar's cross-section, [m^2]; and I is the intensity of the current flowing inside the considered busbar, [A].

The internally generated heat is assumed to be conducted in the steady state regime from the busbar interior to its surface by the Fourier's Law, Equation (2):

$$q'''_{gen} + \nabla \cdot \left(\underline{k}_{bus} \nabla T\right) = 0 \quad (2)$$

Where: $\underline{k}_{bus}=f(T)$ is the temperature dependant material's thermal conductivity tensor, [W/m.°C]; T is the busbar temperature, [°C] and $\nabla \cdot$ is the divergence operator.

The busbars are cooled by two main mechanisms: convection and radiation, as shown by Equation (3).

$$-\left(\underline{k}_{bus} \nabla T\right)^T \vec{n} = h_{eq}\left(T_S - T_\infty\right) \quad (3)$$

Where: $\vec{n}$ is the outward normal vector to the busbar's boundaries; $h_{eq}=h_{conv}+h_{rad}$ is the equivalent convection (h_{conv}) and radiation (h_{rad}) heat transfer coefficient, [W/m^2]; T_s is the busbar surface temperature, [°C]; and T_∞ is the ambient air temperature, [°C].

For the simplified calculation methodology, it is assumed that adjacent busbars are in thermal equilibrium such that no heat is exchanged by radiation between them, *i.e.*, heat is lost by radiation to the ambient only. The linearization of the radiation heat transfer coefficient h_{rad} is shown in Equation (4).

$$h_{rad} = \frac{F\sigma\varepsilon\left(T_{S,K}^4 - T_{\infty,K}^4\right)}{\left(T_S - T_\infty\right)} \quad (4)$$

Where: F is the view factor from the surface to the ambient; the Stefan-Boltzmann constant is σ=5.669E-8 W/m^2.K^4; ε is the material's emissivity; $T_{s,K}$ is the busbar's surface absolute temperature, [K]; $T_{\infty,K}$ is the ambient air absolute temperature, [K].

Finally, the thermal expansion experienced by a busbar is described by Equation (5).

$$\Delta L = L_0 + \chi_{th}\left(T_S - T_0\right) \quad (5)$$

Where: ΔL is the thermal expansion, [m]; L_0 is the busbar length measured at a reference temperature, [m]; χ_{th} is the temperature-dependant material's thermal expansion coefficient, [°C^{-1}]; and T_0 is the reference temperature, [°C]. T_0=20°C was assumed in this work.

Tunnel Encapsulated Busbar Group Test Problem

A test problem is proposed in order to study the impact of some parameters regarding the tunnel enclosed busbar group electro-thermo-mechanical phenomena, such as the ambient conditions and the current density.

The busbar's emissivity is assumed to be ε=0.2 and temperature-dependant electrical resistivity and thermal conductivity are considered for pure aluminum. The current flowing through the group of bars is assumed to be 300 kA. The problem parameters can be seen in Table 1 and its geometry in Figure 2. Note that the openings located in both tunnel ends allow the internal air to be exchanged with the exterior ambient by means of natural convection.

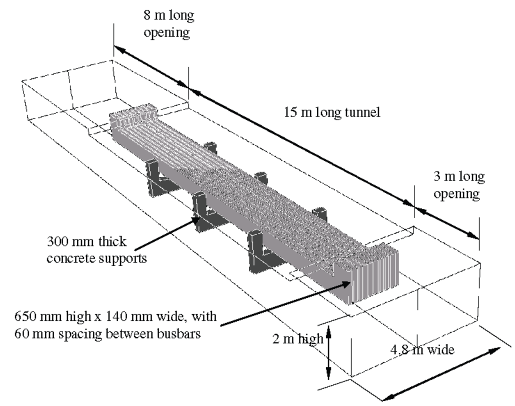

Figure 2 – Tunnel encapsulated group of busbars test problem's geometry.

Table 1 – Test problem changing parameters.

Case #	# of Busbars	Ambient Temperature T_∞ [°C]
I	10	20
II	11	20
III	11	-30

Complex 3D CFD Analysis of the Tunnel Encapsulated Busbar Group Test Problem

A fully described 3D CFD model is built using the commercial package FLUENT in order to analyze the proposed test case. The model geometry is shown in Figure 2, and includes approximately 500,000 hexahedral cells. This model includes the busbars, the concrete support structure and the air in the tunnel.

The 3 m long and 8 m long opening at the ends of the tunnel are treated as zero gauge pressure boundaries that allow cool ambient air to enter the tunnel and hot air to exit the tunnel through the same opening. This can be seen in Figure 6. The air is treated as an incompressible, turbulent ideal gas with temperature dependant density.

The busbar solid volume is included in the model and an energy source is applied to the volume simulating the busbar joule heat release. Surface to surface radiation is accounted for using the Discrete Ordinates (DO) radiation model.

Turbulence is modeled using the RNG κ-ε two equation model with the differential viscosity model enabled to allow for low Reynolds effects. The non-equilibrium wall function was used to resolve near wall effects. Sufficient computational grid was included near the busbar surface to ensure Y+ values were within the 30-300 range necessary for accurate prediction of wall heat transfer.

CFD Results for the Test Problem

Figure 3 to Figure 5 show the temperature profiles obtained with the CFD model for Cases I, II and III, respectively.

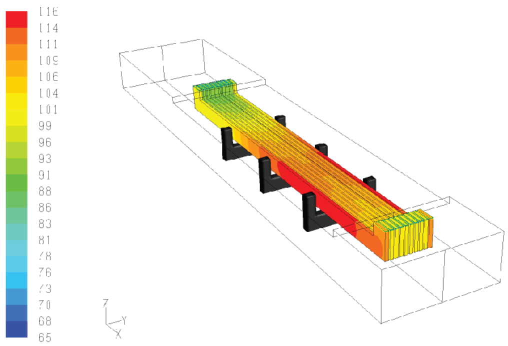

Figure 3 – Busbars temperature [°C] distribution for Case I: 3D CFD model.

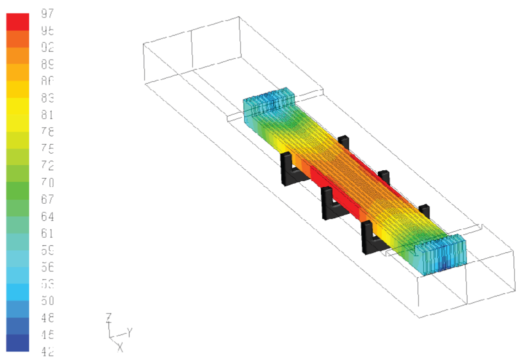

Figure 4 – Busbars temperature [°C] distribution for Case II: 3D CFD model.

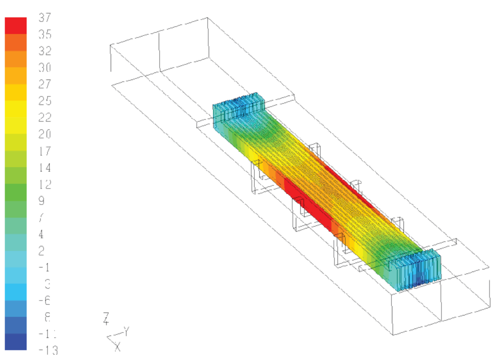

Figure 5 – Busbars temperature [°C] distribution for Case III: 3D CFD model.

It can be seen in Figure 6 that, from a global point of view, the tunnel/busbars arrangement inhibits the fresh air coming from the exterior to reach the vicinity of the central support. Furthermore, this particular geometry leads to a severe reduction in the air flow velocities close to the exterior vertical surfaces of the busbars group, as shown by Figure 7. The resulting flow pattern reduces considerably the cooling of the external busbars (see Figure 3 to Figure 5), making them hotter than the inner components when, intuitively, the opposite is expected. Finally, Table 2 shows the maximum and volume average temperatures obtained with the CFD model for the inner busbars of the group – note that the temperature averaging was performed in the central region of the tunnel (between concrete supports, away from the openings).

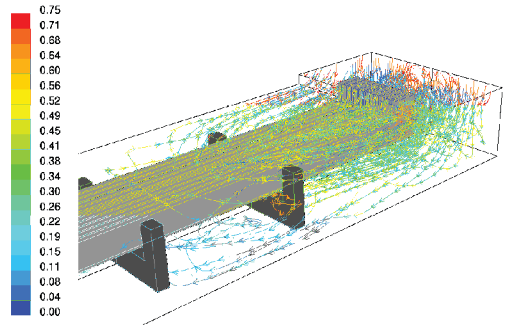

Figure 6 – Air velocity [m/s] profile for Case II.

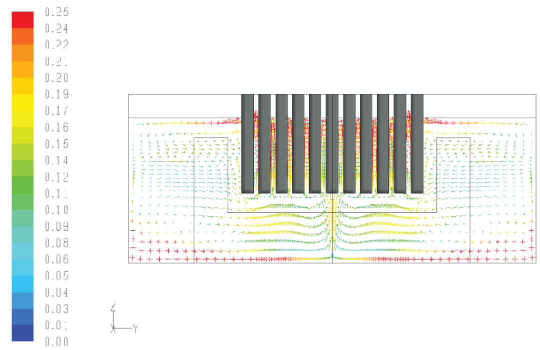

Figure 7 – Air velocity [m/s] profile for Case II: lateral view.

Table 2 – Maximum and volume average temperatures obtained with the 3D CFD model for the inner busbars of the group (in the central region of the tunnel, away from the openings), [°C].

Case #	Maximum Temperature T_{Smax}	Average Temperature T_{Savg}
I	109.0	106.0
II	90.0	87.5
III	29.0	26.0

Proposed Simplified 2D Approach for the Analysis of the Tunnel Encapsulated Busbar Group Test Problem

A simplified 2D non-linear fully-coupled electro-thermal approach is proposed to evaluate the maximum temperature of the centermost component of an evenly shaped tunnel encapsulated group of busbars, which is submitted to a restricted cooling air flow. The model geometry consists of the cross-section of the innermost conductor, as shown in Figure 1.

Internal Heat Generation and Conduction

The Biot number, shown in Equation (6), may be interpreted as the ratio between a body's resistance to internal conduction and its resistance to heat losses to the ambient[1]. The evaluation of the Biot number in a given aluminum busbar's cross section will lead to $Bi << 1$[2], which indicates that the convection and radiation mechanisms are dominant or, in other words, that the solid's cross

[1] As mentioned earlier, these heat transfer mechanisms are modeled by Equations (2) and (3), respectively.

[2] Assuming half of the busbar's height as a characteristic dimension d_c = 0.325 m, an equivalent heat loss coefficient h_{eq} = 10 W/m^2.K, and a busbar's thermal conductivity k_{bus} = 200 W/m.K, the resistances ratio is Bi = 0.01625.

section will experience only a small thermal gradient due to its high capacity to conduct heat. This result allows the evaluation of the Joule Effect – as described in Equation (1) – by means of the cross section's average temperature.

$$Bi = h_{eq} \frac{d_c}{k_{bus}} \qquad (6)$$

Where: Bi is the Biot number.

Radiation View Factors

The radiation view factor from the top and bottom busbars surfaces to the tunnel walls is assumed to be $F_{top/bottom\text{-}tunnel} = 1$. Analogically, the view factor between of an external busbar's vertical surface and the tunnel walls is also assumed to be equal to unity, $F_{external\text{-}tunnel} = 1$.

On the other hand, the view factor between two parallel, identical, infinitely long plates that share the same center line[3] $F_{internal\text{-}internal}$ can be analytically calculated by Equation (7). Finally, the view factor between one of these internal vertical surfaces and the tunnel walls $F_{internal\text{-}tunnel}$ is the considered to be the remainder of the view factor between two adjacent internal surfaces, see Equation (8). Note that $F_{internal\text{-}tunnel}$ is the view factor F to be used with Equation (4)

$$F_{internal-internal} = \frac{\gamma}{2H}\left(\sqrt{\left(\frac{2H}{\gamma}\right)^2 + 4} - 2\right) \qquad (7)$$

$$F = F_{internal-tunnel} = 1 - F_{internal-internal} \qquad (8)$$

Where: $F_{internal\text{-}internal}$ is the view factor between two parallel, infinitely long, identical vertical surfaces sharing the same center line; γ is the gap between the busbars, [m]; and H is the busbar's height, [m]; $F_{internal\text{-}tunnel}$ is the view factor between one infinitely long, internal surface and the tunnel walls.

Natural Convection

Regarding convection heat transfer, it is assumed that the busbar group is not submitted to a forced air flow, thus being cooled by means of natural convection. The dimensionless groups of interest are: Grashof number, Rayleigh number, and the Nusselt number, given by Equations (9) to (11), respectively.

$$Gr_f = \frac{g}{T_{f,K}}\left(T_S - T_\infty\right)\frac{d_c^3}{\nu_f^2} \qquad (9)$$

Where: Gr_f is the Grashof number evaluated at film temperature, [-]; g=9.81 m/s^2 is the acceleration of gravity; $T_{f,K}$ is the absolute film temperature $T_f = (T_S - T_\infty)/2$, [K]; d_c is the characteristic dimension of the problem, [m]; and ν_f is the air's temperature-dependant kinematic viscosity evaluated at film temperature, [m^2/s].

$$Ra_f = Gr_f \frac{\nu_f}{\alpha_f} \qquad (10)$$

Where: Ra_f is the Rayleigh number evaluated at film temperature; and α_f is the air's temperature-dependant thermal diffusivity evaluated at film temperature, [m^2/s].

$$Nu_f = h_{conv} \frac{d_c}{k_{air,f}} \qquad (11)$$

Where: Nu_f is the Nusselt number evaluated at film temperature; and $k_{air,f}$ is the air's temperature-dependant thermal conductivity evaluated at film temperature, [W/m.K].

The correlations describing natural convection are commonly described by means of a function like Equation (12). For the case of busbar's horizontal hot surfaces facing up and down, classical heat transfer correlations [2] are assumed and the applicable coefficients p and q are shown in Table 3.

$$Nu_f = p\left(Ra_f\right)^q \qquad (12)$$

Where: p and q are coefficients obtained empirically.

Regarding the vertical surfaces, a first approach would be to consider correlations obtained for single vertical surfaces immersed in an abundant mass of fresh air, see Table 3. However, Figure 6 suggests, as mentioned before, that the air's natural flow is inhibited by the tunnel geometry, *i.e.*, that there is restricted flow of fresh air entering the tunnel for cooling the bars.

Table 3 – Coefficients to be used with Equation (12): natural cooling of a single vertical surface and hot horizontal surfaces facing up and down.

Surface Orientation	p	q	Characteristic Dimension	Validity Range
Horizontal Down	0.27	0.25	Busbar's thickness t_k	-
Horizontal Up	0.59	0.25		$Ra_f \leq$ 1E9
	0.105	0.33		$Ra_f >$ 1E9
Single Vertical	0.59	0.25	Busbar's height H	1E4 $\leq Ra_f \leq$ 1E9
	0.1	0.33		$Ra_f >$ 1E9

In order to represent this physical situation where there is a restrict flow of the cooling medium, a correlation originally conceived to describe the natural convection in a vertical enclosure [3] was adopted, Equation (13). Note that the considered characteristic dimension is the gap between busbars γ.

[3] This is the general case of adjacent vertical surfaces from two neighboring internal components of group of busbars. Other analytical expressions for other applications can be found in [1].

Geometrical Range: $H/t_k \geq 3.0$ and $3.0 \leq H/\gamma \leq 16.0$
Characteristic Dimension: Gap between busbars γ

$$Nu_f = 1 \qquad 2E4 < Gr_f$$

$$Nu_f = 0.18 Gr_f^{1/4}\left(\frac{H}{\gamma}\right)^{-1/9} \qquad 2E4 \leq Gr_f < 2E5$$

$$Nu_f = 0.065 Gr_f^{1/3}\left(\frac{H}{\gamma}\right)^{-1/9} \qquad 2E5 < Gr_f \leq 1.1E7 \qquad (13)$$

Where: t_k is the busbar's thickness, [m].

2D Model Implementation

The 2D approach described above was implemented using the ANSYS FEA package. Elements described as PLANE55 were used to simulate the in-plane heat conduction. Joule Effect was included as an element based uniform heat source updated each iteration. Since the temperature gradients are expected to be small, the model's convergence is based in an absolute criterion regarding the busbar's average temperature.

Temperature-dependant convection and radiation heat transfer mechanisms were considered as an equivalent 2nd order boundary condition. The considered convection correlations for top and bottom busbar surfaces are described by Equation (12) & Table 3. Their radiation view factor is $F_{top/bottom\text{-}tunnel} = 1$. Regarding the vertical busbar surfaces, both correlations for vertical enclosures, Equation (13), and single surfaces in abundant cooling medias, Equation (12) & Table 3, were tested. The radiation view factor is calculated by Equation (8).

Finally, it must be stressed that this model can be implemented in any standard Finite Element or Finite Difference code.

2D Model Results for the Test Problem

Figure 8 shows the temperature distribution for Case II obtained with the 2D model when considering the vertical enclosure correlation. Note that, although the absolute temperature difference is smaller then 0.5°C, its profile is asymmetric due to different heat transfer rates from top and bottom surfaces.

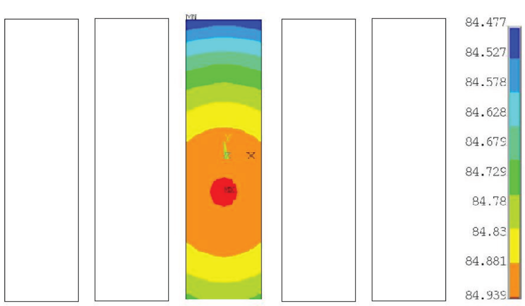

Figure 8 – Busbars temperature [°C] distribution for Case II: vertical enclosure.

Figure 9 shows the comparison between the volume average temperatures obtained for the inner busbars of the group with the CFD model (shown previously in Table 2) and the 2D model results when considering convection correlations for both vertical single surfaces and enclosures.

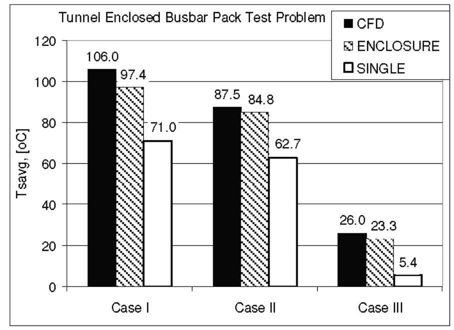

Figure 9 – Average temperatures T_{Savg} for the innermost busbars of the group: 3D CFD model, 2D model considering convection correlations for both vertical single surfaces and enclosures.

Discussion

Figure 10 shows the absolute difference for the maximum temperature predicted with the 2D approach for the innermost busbar of the group with respect to the CFD results. The CFD results are consider to provide an accurate representation of the temperature distribution in the busbars based on plant experience.

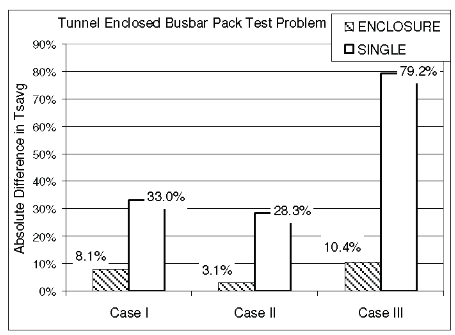

Figure 10 – Absolute difference in the average temperatures T_{Savg} calculated with the proposed 2D approach regarding the CFD results from Table 2.

As expected, the convection correlation for single vertical surfaces immersed in an abundant mass of fresh air cannot realistically represent the physical situation of a group of busbars enclosed in a tunnel where air flow is restricted. The same level of accuracy is expected when the classical, analytical approach[4] [4] to busbar sizing is applied for the analysis of this kind of arrangement, since its integral form of the energy equation is often derived assuming that the conductors are submitted to free air. Consequently, an enclosure will "reduce the busbar heat dissipation due to reduction in (natural) cooling air flow" and, regarding the applicability of the method to this particular application, "an accurate figure can only be obtained by testing".

On the other hand, it can be seen that the 2D model can reasonably reproduce the average temperatures of the innermost busbar of the group when the convection correlation for vertical enclosures is considered – the absolute differences regarding the average temperatures obtained by CFD were smaller than 10%.

[4] Although the temperature rise equations shown in this particular reference were developed for copper conductors, they can be derived for aluminum applications by considering appropriate material properties.

CFD however, allows the determination of hot spots in the system due to localized air flow restrictions. In Hatch's past experience, measured *in situ* maximum temperatures of the innermost component of a given group of busbars[5] enclosed in a tunnel-like structure and submitted to a restricted natural cooling flow were reproduced by the proposed 2Dmethod within 5% of accuracy.

Finally, it must be pointed out that the proposed 2D method is not able to replace CFD modeling due to, of course, its inability to describe 3D effects in both conductors (thermal gradients across length) and cooling media (complex flow patterns). The proposed method is rather to be considered as an evaluation tool of different design parameters based on a worst-case condition. A complete evaluation of the problem is warranted when design factors of safety are being stretched.

Potential Applications of the Proposed 2D Model

The proposed 2D method can be used, for example, to study the influence of the busbar's geometry and spacing. Figure 11 shows that the busbar's temperature tends to decrease with increasing surface-to-volume ratio, as expected. On the other hand, Figure 12 shows that busbar temperature increases with increasing height-to-gap ratio due to the reduction in radiation-to-ambient view factor from the vertical surfaces. All remaining parameters (current density, ambient temperature, surface emissivity) were kept constant, according to Case II – Figure 2 and Table 1.

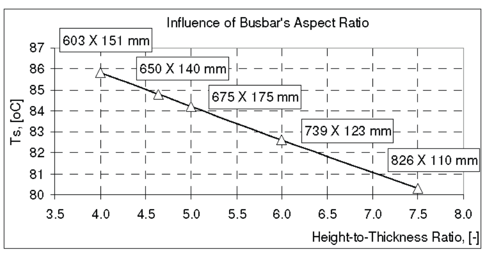

Figure 11 – Impact of cross section's aspect ratio in the maximum temperature of the innermost busbar from Case II.

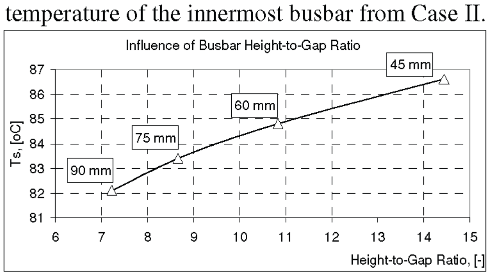

Figure 12 – Impact of busbar gap in the maximum temperature of the innermost busbar from Case II.

The proposed 2D method can also be used to evaluate retrofit alternatives. Consider a hypothetic smelter that has a crossover busbar system with an initial installed length L_0 = 60 m, originally designed for 300 kA, and with a surface emissivity ε = 0.2 during commissioning. Due to the design specifications of the selected electrical insulator, the maximum allowable surface temperature is T_{Smax} = 90°C. The maximum expansion joint design capacity is ΔL_{max} = 0.1 m (at a reference temperature T_0 = 20°C). Consider also that, during the last 10 years, this plant increased its line current up to 380 kA. The busbars surface emissivity has evolved to 0.6 as a result of the material's aging. Assume that all remaining parameters are as per Case II.

Figure 13 and Figure 14 show, respectively, that both T_{Smax} and ΔL_{max} limits are achieved at a line current smaller than 325 kA with ε = 0.2. Even considering ε = 0.6, these limits are achieved at a line current ≈ 350 kA. One possible solution would be the application of a high emissivity painting (ε = 0.99) on the surfaces of the group of bars. It would reduce the temperature and keep both T_S and ΔL within design specifications up to 380 kA.

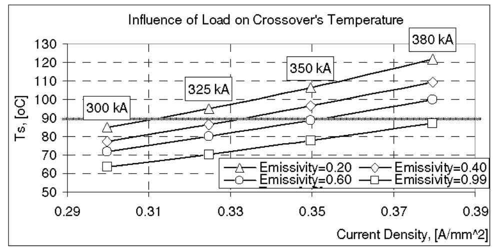

Figure 13 – Impact of load in the maximum temperature of the innermost component from hypothetic crossover busbars.

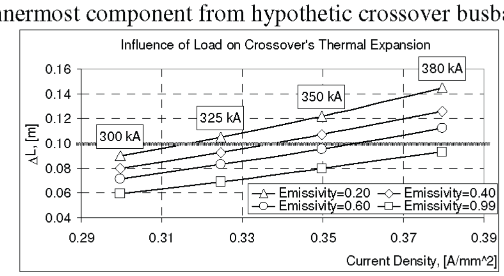

Figure 14 – Impact of load in the thermal expansion of the innermost component from hypothetic crossover busbars.

Conclusion

A 2D method for the evaluation of the heat transfer problem of busbars in enclosed tunnels is proposed. Results compared with a 3D CFD model showed absolute differences smaller then 10% for the volume average busbar temperature in the central tunnel section. The 2D model was able to predict actual temperatures within 5% accuracy. The simplified approach is not computationally intensive and a complete set of analyses can be performed in approximately one hour. It is, thus, intended to be used as a tool for conceptual design based in a worst case condition and can be applied to the retrofit of existing busbar systems submitted to restricted natural cooling flows. Complex CFD modeling can then be used to verify 2D model findings and to study the system's particularities, as warranted.

References

1. http://www.me.utexas.edu/~howell/tablecon.html
2. **J. P. Holman**, "Heat Transfer" - 8th Ed. McGraw-Hill (1997), 696 pages. Eq. (7-25) & Table 7-1.
3. **D. R. Pitts** & **L. Sissom**, "Schaum's Outlines Heat Transfer" - 2nd Ed. McGraw-Hill (1998), 365 pages. Eq. (8.35) to (8.37).
4. http://www.copperinfo.co.uk/busbars/pub22-copper-for-busbars

[5] Load = 0.42 A/mm², height-to-thickness ratio H/t_k = 5.0, height-to-gap ratio H/γ = 16.0, surface emissivity calibrated according to contact thermocouple and pyrometer readings.