# PREDICTION AND COMPARISON OF SIZE OF THE COPPER AND ALUMINUM BUS DUCT SYSTEM BASED ON AMPACITY AND TEMPERATURE VARIATIONS USING MATLAB

by

***THIRUMURUGAVEERAKUMAR S*[a*], *SAKTHIVEL M*[b], *and SHARMILA DEVE V*[c]**

[a] Department of Mechanical Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India
[b] Department of Mechanical Engineering, INFO Institute of Engineering, Coimbatore, Tamil Nadu, India
[c] Department of Electrical and Electronics Engineering, Kumaraguru College of Technology, Coimbatore, Tamil Nadu, India

Original scientific paper
https://doi.org/10.2298/TSCI160407153T

*The main objective of this paper is to propose an algorithm to predict and compare the sizes of the bus bar with materials like copper and aluminum by considering the allowable ampacity and allowable temperature rise with natural and forced convection cooling arrangement. Theoretical analysis is carried out with modified size of the copper bus bar using MATLAB, to analyze the ampacity and temperature variation under the natural and forced convection mode. The algebraic equation developed from thermal model is solved using MATLAB for the determination of the allowable temperature rise and ampacity of rectangular-section bus bars of copper and aluminum and also for different sizes of bus bar. An algorithm has been developed for the analysis. Experimental observations of temperature variation in copper bus bar with standard size under natural and forced cooling mode are validated with the algebraic equation developed from thermal model is solved using MATLAB. It is concluded that bus bar dimensions are compared for the materials copper and aluminum to predict the suitable equivalent dimensions for the same ampacity level and within the allowable temperature rise to reduce the cost of panel.*

*Key words*: *ampacity, temperature rise, air insulated bus bar, heat transfer, MATLAB*

## Introduction

The ampacity of a conductor is limited by the continuously permissible temperature of the system which depends on the properties of the conductor material. Magnetic field from the eddy current interacts with the original electromagnetic field resulting in the reduction of current flow in the center of the conductor. These current tend to flow near the surface of the conductor which is referred as skin region.

The inductance of a conductor varies with the depth of the conductor due to the skin effect. This inductance is further affected by the presence of another current-carrying conductor

* Corresponding author, e-mail: tmvkumar1978@gmail.com

in the vicinity. A short-circuit results in excessive current and also causes excessive heat in the current-carrying conductors. The conductor with high current density will increase the resistance and temperature of the conductor. Power losses are caused by both the ampacity and by the induced eddy current in the phases. An attempt is being made on bus bar arrangement with forced convection cooling to improve heat dissipation with reduced skin and proximity effect. The convection cooling arrangement will be provided to modify and optimize the size of the bus bar configuration.

A 3-D eddy current field model is used to calculate the eddy current losses in an air insulated bus duct system. The temperature rises are evaluated using the coupled fluid field and the thermal field model [1]. A compact bus duct system governing equations are used to determine the temperature rises [2]. Power losses of a bus bar are calculated by the magnetic field analysis and those values are used as the input data for the thermal analysis to predict the temperature [3]. In fig. 2, the magnetic fields in three phases of bus conductor are symmetrical, but distribution of current density in three phases of conductor is unevenly distributed due to skin and proximity effects. Hence current is strongly influenced by the bus bar geometry and in every case current is unevenly distributed in the sub-conductors of every main conductor [4]. The transient capabilities of the bus bar are illustrated by calculating the variations in the bus bar temperature. Temperature variation in the bus bar is used to estimate the thermal time constants. An analytical expression for the time constant of the bus bar is developed [5]. Thermal time constant has predicted for both natural and forced convection mode. From the study, power loss due to heat generation is reduced by 45 % [6]. The analytical algorithm has been implemented in a computer program code that along the allowable ampacity can compute the bus bar temperature and their corresponding power losses [7]. A thermal model is used to calculate both the steady-state and transient temperature variation of the bus bar. A computer program has been formulated to calculate the temperature variation in the bus bar for any time-varying current [8]. The temperature rise of the extra-high voltage gas insulated switchgear bus bar is predicted with magneto thermal finite element analysis. Bus bar power losses are calculated by magnetic field analysis and that data are used as the input to predict the temperature rise [9]. A hybrid finite element – boundary element formulation applied to the analysis of a four conductor device under AC supply and presence of shields. Impedance matrix and correct magnetic field are estimated by developing a model and compared with the experimental outcomes [10].

## Bus bar arrangement in the power house

The observations have been noted down from the electric power facility of Sangeeth Textiles Mill, Coimbatore, India which has the capacity of 2500 A rating and 1500 kVA transformer substation. In the transformer, the voltage is stepped down to 440 V for distribution to points of consumption.

These arrangements are done in an enclosed chamber made of steel with minimum amount of ventilation with natural convection heat dissipation. The bus bar is made of copper. In this textile mill, there are eight sections consisting of two Spinners (SSB 1 and 2), two auto cones (SSB 3 and 4) and two drawers (SSB 5 and 6), each one of guarding and lighting (SSB 7 and 8). The arrangement of bus bar and power distribution to different sections of the mill is shown in fig. 1. Figure 2 shows the magnetic vector potential (MVP) and total current density distribution in low voltage arrangement with multiple sub-conductors per main conductor [4]. The electricity is transmitted through the main bus bar and distributed to different sections of the mill as per for the load requirement. The current in the main bus bar depends on the number of sections operated. Current fluctuation occurs in the main bus bar when some sections are in-

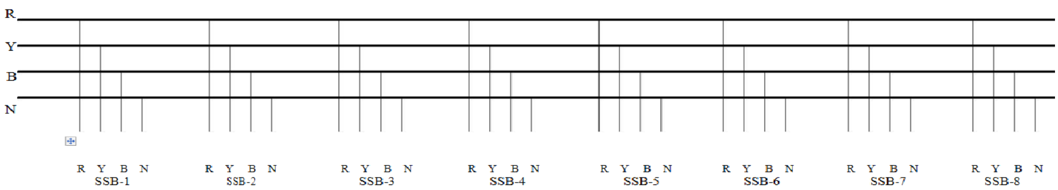


**Figure 1. Bus bar arrangement in the electric power facility**

operative. Bus bar with a rectangular cross-section is being used in the panel board. The sizes of the main bus bars are 100 × 6 mm with 3 sub bus bar per main bus bar in each runs and with a single run of 100 × 6 mm for the neutral. It passes horizontally along the length of the panel board. Experiments are conducted under conditions of natural convection and under conditions of forced convection by keeping the air-flow parallel and perpendicular to the bus bar. Experiments are conducted for different velocities.

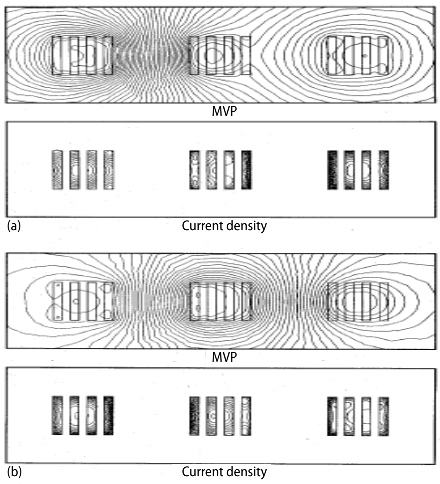


**Figure 2. The MVP and total current density distribution in low voltage arrangement with multiple sub-conductors per main conductor (a) at $\omega t = 0°$ and (b) at $\omega t = 120°$**

## Thermal model of the bus bar

The energy balance equation for the bus bar is written in [5] and [6]:

$$\rho C_p V \frac{\mathrm{d}T}{\mathrm{d}t} = I^2 R(t) - hA_s (T - T_\infty) - \\ - \varepsilon\sigma A_s (T^4 - T_\infty^4) \tag{1}$$

Equation (1) is simplified:

$$\frac{\mathrm{d}T}{\mathrm{d}t} + \left[\frac{hA_s \varepsilon\sigma A_s (T+T_\infty)(T^2+T_\infty^2)}{\rho C_p V}\right](T) = \left[\frac{I^2 R(t)}{\rho C_p V}\right] + \left[\frac{hA_s \varepsilon\sigma A_s (T+T_\infty)(T^2+T_\infty^2)}{\rho C_p V}\right](T_\infty) \tag{2}$$

Equation (2) is similar to the differential equation:

$$\frac{\mathrm{d}T}{\mathrm{d}t} + a(T) = C \tag{3}$$

Solution for the given differential equation is:

$$T_{i+1} = \frac{c}{a}(1 - \mathrm{e}^{-at}) + T_i(\mathrm{e}^{-at}) \tag{4}$$

where

$$a = \left[\frac{hA_s \varepsilon\sigma A_s \left(T+T_\infty\right)\left(T^2+T_\infty{}^2\right)}{\rho C_p V}\right](T)$$

$$C = \left[ \frac{I^2 R(t)}{\rho C_p V} \right] + \left[ \frac{h A_s \varepsilon \sigma A_s \left( T + T_\infty \right) \left( T^2 + {T_\infty}^2 \right)}{\rho C_p V} \right] \left( T_\infty \right)$$

The thermal time constant is the term relating to the geometrical, physical, and thermal properties of the bus bar. The bus bar attains a steady-state temperature after a few units of thermal time constant.

The thermal time constant is:

$$\tau = \frac{\rho C_p \dfrac{V}{A_s}}{h + \varepsilon \sigma \left( T^2 - T_\infty^2 \right) \left( T - T_\infty \right)} \tag{5}$$

Free convection coefficient, $h$, for a vertical plate is calculated by using the following local Nusselt number correlation:

$$\mathrm{Nu}_x = 0.508 \, \mathrm{Pr}^{0.5} \left( 0.952 + \mathrm{Pr} \right)^{-0.25} {\mathrm{Gr}_x}^{0.25}$$

To calculate average convection coefficient, $h$, average Nusselt number correlation:

$$\mathrm{Nu} = \frac{4}{3} \left[ 0.508 \, \mathrm{Pr}^{0.5} \left( 0.952 + \mathrm{Pr} \right)^{-0.25} \mathrm{Gr}^{0.25} \right]$$

Similarly for the forced convection cooling with air-flow perpendicular to the bus bar, the correlation used is:

$$\mathrm{Nu} = 0.205 \, \mathrm{Re}^{0.731} \, \mathrm{Pr}^{1/3} \tag{6}$$

With the air-flow parallel to the bus bar axis, correlation used:

$$\mathrm{Nu} = 0.664 \, \mathrm{Re}^{1/2} \, \mathrm{Pr}^{1/3} \tag{7}$$

These Nusselt number values are used to determine heat transfer coefficient, $h$, in the energy balance equation.

## Analytical algorithm

The algorithm developed carried out in the research work is being given in fig 3. The experimental work was conducted in a textile mill and a suitable thermal model is developed to determine the temperature of the bus bar under natural and forced convection mode. The program has been written in MATLAB for the algebraic equation developed from thermal model.

## Results and discussion

The steady-state temperature in the phase Y is greater than other two phases due to high ampacity. Theoretical analysis has been carried out by considering energy balance equation for the bus bar and the algebraic equation developed from thermal model is solved using MATLAB by implementing the computer program code for determining the temperature rise of rectangular-section bus bars made of copper and aluminum and also for different sizes of bus bar. Experimentally observed steady-state temperatures in the Y phase of a standard size copper bus bar and numerically computed using MATLAB are given in the tab. 1. Numerically calculated temperature variation in the copper bus bar under natural convection mode is validated with experimental observations. Also in tab. 1, steady-state temperatures attained in the copper bus bar are given by considering the forced convection analysis for the air-flow in two directions.

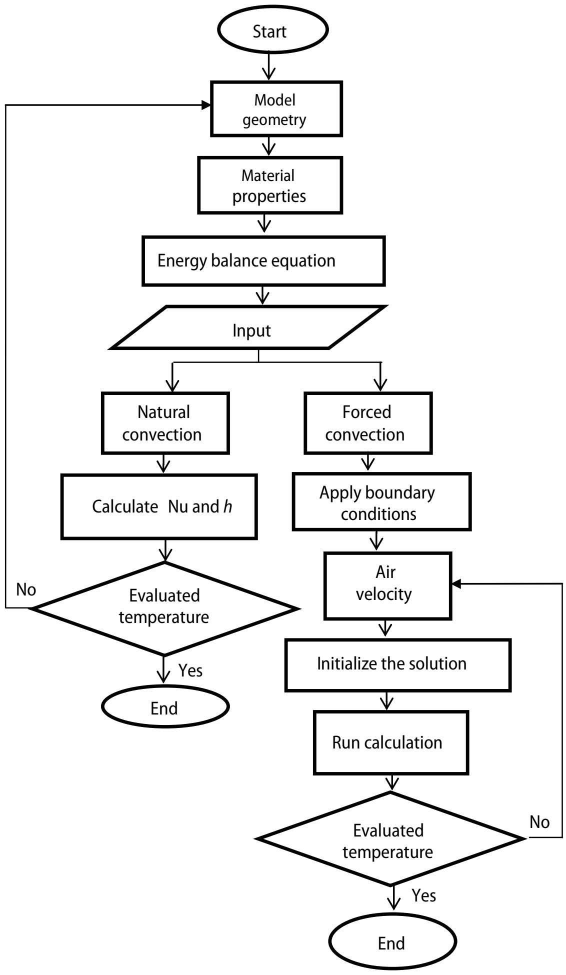


**Figure 3. Algorithm for calculation of heat transfer coefficients due to natural and forced convection from bus bar systems**

**Table 1. Variations in temperature of copper bus bar with the time constant under conditions of natural and forced convection**

| Time constant, $\tau$ | Temperature variation under natural convection in copper bus bar [°C] | | Temperature variation under forced convection in copper bus bar [°C] | |
|---|---|---|---|---|
| | Observed value | Calculated value | Air-flow parallel to bus bar axis | Air-flow perpendicular to bus bar axis |
| $\tau$ | 71 | 60.77 | 38.46 | 35.22 |
| $2\tau$ | 73 | 71.35 | 40.83 | 36.40 |
| $3\tau$ | 76 | 75.25 | 41.71 | 36.84 |
| $4\tau$ | 77 | 76.68 | 42.03 | 37.00 |
| $5\tau$ | 78 | 77.21 | 42.15 | 37.06 |
| $6\tau$ | 78 | 77.40 | 42.19 | 37.08 |
| $7\tau$ | 78 | 77.47 | 42.21 | 37.09 |

Figure 4 shows the temperature variation with the time constants under natural and forced convection mode for the copper bus bar. With natural convection, the temperature of the bus bar steeply increases (60 °C) due to high current flow. Then temperature increases and attains steady-state condition (77 °C) at $4\tau$. Due to forced convection arrangement the temperature increases from initial condition (38 °C) and attains the steady-state condition (42 °C) at same $4\tau$. For the forced convection mode of heat transfer from the bus bar, the temperature rise will be reduced by 45% compared with natural convection mode. When compared to parallel and perpendicular air-flow in copper bus bar, the temperature rise due to perpendicular air-flow is reduced by 12%.

The current carrying capacity of bus bar with aluminum material is lower than the copper material. Due to less cost, aluminum has been chosen for this research work as alternate to copper. Figure 5 shows the temperature variation for aluminum bus bar under natural and forced convection mode of heat transfer. Hence, the algebraic equation developed from thermal model is solved using MATLAB for aluminum bus bar under natural and forced convection mode to determine the temperature variation with time constant which is given in the tab. 2. During natural convection, the bus bar temperature suddenly increases to 78 °C due to high current flow. Then it increases gradually up to steady-state condition (105 °C). When the air with a velocity of 1 m/s flows to the parallel direction of the aluminum bus bar, the temperature in the bus bar increases to 42 °C and this value is lower than the natural convection value due to

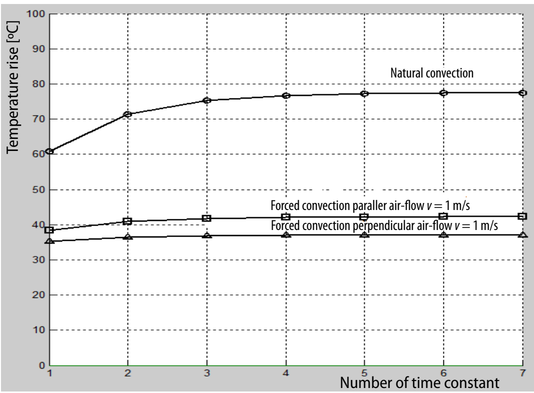


**Figure 4. Temperature variation for copper bus bar under natural and forced convection**

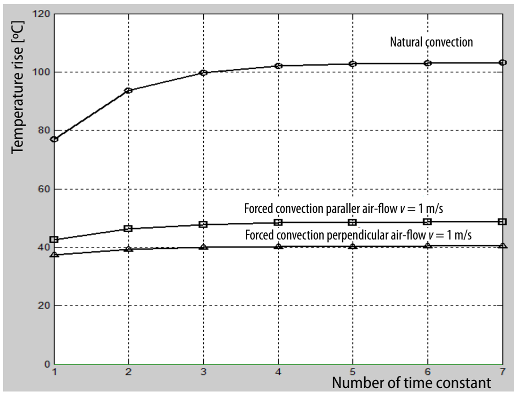


**Figure 5. Temperature variation for aluminum bus bar under natural and forced convection**

**Table 2. Variations in temperature of aluminum bus bar with the time constant under conditions of natural and forced convection**

| Time constant, $\tau$ | Temperature variation under natural convection in aluminum bus bar [°C] | Temperature variation under forced convection in aluminum bus bar [°C] | |
|---|---|---|---|
| | Calculated value | Air-flow parallel to bus bar axis | Air-flow perpendicular to bus bar axis |
| $\tau$ | 77.04 | 42.50 | 37.26 |
| $2\tau$ | 93.62 | 46.36 | 39.20 |
| $3\tau$ | 99.71 | 47.79 | 39.91 |
| $4\tau$ | 101.96 | 48.31 | 40.18 |
| $5\tau$ | 102.78 | 48.50 | 40.27 |
| $6\tau$ | 103.09 | 48.57 | 40.31 |
| $7\tau$ | 103.20 | 48.60 | 40.32 |

the air velocity. The bus bar attained steady-state condition at temperature of 48 °C and at time constant of 4$\tau$. When compared with forced and natural convection mode of heat transfer, the temperature rise in the forced convection mode will be reduced by 48% with natural convection mode. Comparing parallel and perpendicular air-flow in aluminum bus bar, the temperature rise due to perpendicular air-flow is reduced by 17%. Therefore, perpendicular air-flow is preferred for forced convection cooling.

Bus bar degradation mechanisms such as, corrosion and oxidation typically occur around 85 °C. For aluminum bus bar of standard size (100 mm width) with natural convection mode, the temperature rise approaches 100 °C which is closer to the damage condition of the bus bar.

Therefore, forced convection heat dissipation with the air-flow perpendicular to the bus bar arrangement is preferable when aluminum replaces the copper bus bar.

In the aspect of optimizing the size of the bus bar, the algebraic equation developed from thermal model is solved using MATLAB by considering different standard sizes of 35 mm, 50 mm, and 100 mm bus bars of copper and aluminum materials under the forced convection arrangement. Variation in temperature of bus bar with the time constant for different sizes of bus bar for the air-flow in the direction of perpendicular to bus bar are plotted in the figs. 6 and 7 for copper and aluminum bus bars, respectively.

In fig. 6 copper bus bar for 100 mm width, steady-state temperature is reached at 37 °C. If the width is reduced to 50 mm the steady-state temperature increases by 10%. Furthermore, if the width of the bus bar is stepped down to the next standard size (35 mm) the steady-state temperature increases by 18%. Whereas in fig. 7 aluminum bus bar for 100 mm width, steady-state reaches at 40 °C. If the width is reduced to 50 mm the steady-state temperature is increased by17%. Besides, if the width of the bus bar is stepped down next standard size (35mm) the steady-state temperature is increased by 32%.

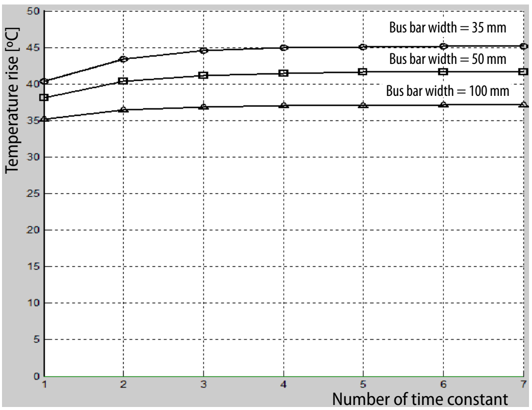


**Figure 6. Temperature variation with time constant for various standard sizes of copper bus bar**

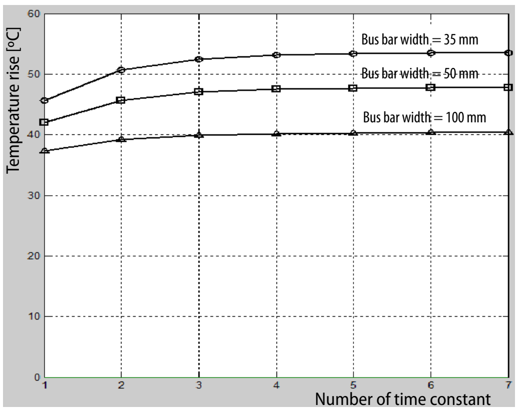


**Figure 7. Temperature variation with time constant for various standard sizes of aluminum bus bar**

Since the steady-state temperature for forced convection (perpendicular air-flow) of aluminum bus bar of 35 mm width is reached to 53 °C which is below that for a standard size copper bus bar of 100 mm width by natural convection (77 °C). Hence aluminum bus bar can also be selected appropriately for effective cost reduction.

Comparing figs. 4-7, difference in temperature variation with time constant for copper or aluminum bus bar under natural convection varies very much. Whereas for bus bars with different sizes under forced convection heat dissipation both copper and aluminum bus bars experience same level of temperature rise and within safe region. Hence it is understood that aluminum bus bar with reduced sizes can be used in place of copper under the forced convection mode.

In figs. 8 and 9, 3-D variation of temperature of copper and aluminum bus bar, respectively, for different air velocities and for different time constants under forced convection mode are shown. The upper positioned graph represents parallel air-flow and the lower positioned graph represents perpendicular air-flow. The temperature range in the aluminum bus bar is higher than the copper bus bar. It is observed that perpendicular air-flow causes high heat dissipation and hence the temperature rise will be minimum compared with the parallel air-flow and the graph shows the effect of variation of air velocity on temperature reduction. Up to air velocity of 1 m/s, the temperature reduction will be more effective than air velocity beyond 1 m/s

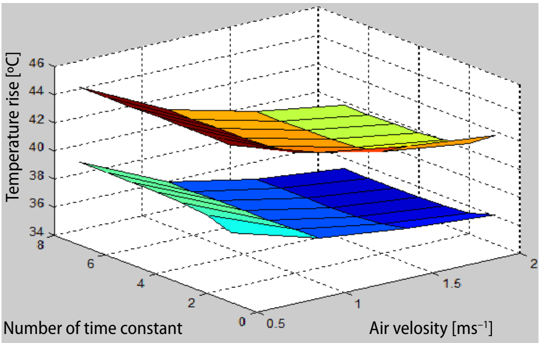


**Figure 8. The 3-D variation of temperature with different air velocities and time constant for copper bus bar**

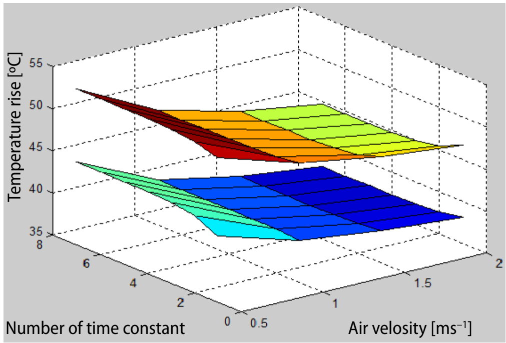


**Figure 9. The 3-D variation of temperature with different air velocities and time constant for aluminum bus bar**

## Conclusions

An algorithm has been developed to perform analysis to determine the temperature variations in the bus bar of different materials of copper and aluminum with different standard size using MATLAB. An experimental observation of temperature variation for standard size copper bus bar is validated with theoretical analysis using MATLAB. Then, with forced convection cooling arrangement, the temperature variation of copper bus bar is observed and it is found that the temperature variation is well under the safety range. Hence a theoretical analysis is carried out with different size of copper and aluminum bus bar with different air velocities to predict the temperature rise which is also under the safety range. It is suggested that by reducing the size of bus bar, cost of panel can be saved. During the course of validation of the analytical algorithm, it is confirmed that due to forced convection (perpendicular air-flow) heat dissipation, response time (time constant) for attaining steady-state condition is improved. Hence, proximity and skin effect are much controlled which leads to the reduction of power consumption in the load.

## Nomenclature

$A_s$ – surface area, [m$^2$]
$C_p$ – specific heat at constant pressure, [kJkg$^{-1}$°C$^{-1}$)
Gr – Grashof number, [–]
$h$ – convective heat transfer coefficient [Wm$^{-2}$°C$^{-1}$]
$I$ – current, [A]
Nu – Nusselt number, [–]
Pr – Prandtl number, [–]
$R(t)$ – AC electric resistance as a function of temperature, [Ω]
$T$ – ambient air temperature, [°C]

*Greek symbols*

$\varepsilon$ – emissivity

$\rho$ – density, [gm$^{-3}$]

$\sigma$ – Stefan-Boltzmann constant, [Wm$^{-2}$K$^{-4}$]

$\tau$ – thermal time constant, [s]

## References

[1] Ho, S. L., *et al*., Calculations of Eddy Current, Fluid and Thermal Fields in an Air Insulated Bus Duct System, *IEEE Trans. on Magnetics., 39* (2006), 1, p. 101

[2] Ho, S. L., *et al*., A 3-D Study of Eddy Current Field and Temperature Rises in a Compact Bus Duct System, *IEEE Trans. on Magnetics., 42* (2006), 4, pp. 987-990

[3] Ho, S. L., *et al*., Calculations of Eddy Current, Fluid, and Thermal Fields in an Air Insulated Bus Duct System, *IEEE Trans. on Magnetics., 43* (2007), 4, pp. 1433-1436

[4] Labridis, D. P., Dokopolos, P. S., Electromagnetic Forces in Three-Phase Rigid Busbars with Rectangular Cross Section, *IEEE Trans. on Magnetics., 11* (1996), 2, pp. 793-800

[5] Thirumurugaveerakumar, S., *et al*., Experimental and Analytical Study on the Bus Duct System for the Prediction of Temperature Variations Due to the Fluctuation of Load, *Journal of Electrical Engineering and Technology., 9* (2014), 6, pp. 2036-2041

[6] Thirumurugaveerakumar, S., *et al*., Experimental and Analytical Study of Effect of Forced Convectional Cooling of Bus Duct System in the Prediction of Temperature Rise, *International Journal of Advances in Engineering and Research.,10* (2015), 21, pp. 42202-42208

[7] Robert Coneybeer, T., *et al*., Steady-State and Transient Ampacity of Bus Bar, *IEEE Trans. Power Delivery., 9* (1994), 4, pp. 1822-1829

[8] Klimenta, D. O., *et al*., An Analytical Algorithm to Determine Allowable Ampacities of Horizontally Installed Rectangular Bus Bars, *Thermal Science, 20* (2016), 2, pp. 717-730

[9] Kim, J. K., *et al*., Temperature Rise Prediction of EHV Gis Bus Bar by Coupled Magneto-Thermal Finite Element Method, *IEEE Trans. on Magnetics, 41* (2005), 5, pp. 1636-1639

[10] Bottauscio, O., *et al*., Numerical and Experimental Evaluation of Magnetic Field Generated by Power Bus Bar Systems, *IEEE Transactions on Proc. Gener, Transm, Distrib.*, *143* (1996), 5, pp. 455-460

Paper submitted: April 7, 2016
Paper revised: June 19, 2016
Paper accepted: June 22, 2016

© 2018 Society of Thermal Engineers of Serbia
Published by the Vinča Institute of Nuclear Sciences, Belgrade, Serbia.
This is an open access article distributed under the CC BY-NC-ND 4.0 terms and conditions