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Determination of natural convection heat transfer coefficient over the fin side of a coil system

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***Abstract*—This paper presents a thermal study to define the appropriate correlations allowing the determination of the convective heat transfer coefficient over large parallel rectangular fins for a permanent magnet synchronous generator coil. For this purpose, an experimental setup is developed for both horizontal and vertical orientations and different input currents. The experimental results are compared with the analytical method, based on correlations proposed in the literature, which are generally limited for a small range of heat sinks. The results show that the analytical calculation based on Jone's correlation for the horizontal case and Tari's correlation for the vertical case, have good agreement with the experimental data. These correlations are experimentally validated for the calculation of the natural convection coefficients of large rectangular fins arrangement too.**

Keywords—Empirical correlation; Heat transfer; Natural convection; Plate fin array.

I. Introduction

The extended surface, which is called a fin is the preferred cooling method in the natural convection mode to enhance the heat transfer rate between the surface and the cooling fluid in electrical machines and other electrical devices. The fin and heat sink technologies are a subset of the passive cooling methods and have several advantages over the active ones, such as energy saving, affordability, reliability, and ease of manufacturing [1].

Rectangular cross section shaped plate fins on a flat base are the most common types of fins used in different electrical devices. In the natural convection mode, the characteristics of fins, e.g., length, height, and spacing between the fins play an important role on the maximum heat transfer rate. Therefore, there is a number of studies dedicated to find out the optimal parameters of the fin's geometry. Based on these findings, other studies have tried to define empirical correlations to determine the convection coefficient [2]. The parallel rectangular cross section fins on a flat base are generally used in vertical and horizontal configurations. There are many studies focused on finding the amount of natural heat transfer either experimentally or analytically. E.g., Jones and Smith [3], Van Del Pol and Tierney [4], Elenbaas [5], Rao [6], Baskaya [7] and Tari [2], [8].

On the other hand, according to other studies, e.g., Rao [6], Ahmadi [1] and Boglietti [9], between 20 to 40 % of the total heat transfer is extracted by the radiation phenomenon. Thus, the radiation heat extraction in parallel with the natural convection has a great effect on the total heat dissipation.

As mentioned above, there is a number of empirical correlations to calculate the natural heat transfer from parallel arrangement of rectangular cross section plate fins on a flat base in both horizontal and vertical configurations. Each of these correlations have been developed based on different ranges of Rayleigh number (Ra), Prandtl number (Pr) and fin's characteristics. Most of these correlations have been developed and used for small fins' size and spacing. In large electrical machines and devices, the fins' size and spacing are increasing. Therefore, it is important to find the appropriate correlation among the existing ones, which can describe the heat dissipation in these cases.

In this paper, we determine the appropriate correlations for large electrical machines with rectangular flat fin arrangements in horizontal and vertical orientations by means of analytical and experimental methods, based on the correlations proposed in the literature, which are generally limited to a small range of heat sinks. We also consider the impact of the fin on the natural heat transfer and the temperature of the plate fin array. For this purpose, one segment of the stator coil of a permanent magnet synchronous generator consisting of rectangular fin arrangements is investigated in both horizontal and vertical configurations.

II. Empirical Correlations

There are many empirical correlations for the rectangular fin arrangement with flat base. In this paper, the empirical correlations based on the scientific research of Jones and Smith [3], Van Del Pol and Tierney [4] and Tari [2], [8] are discussed. According to Fig. 1, the configurations investigated are divided into two main categories: rectangular isothermal fins on a horizontal surface and rectangular isothermal fins on a vertical surface [10].

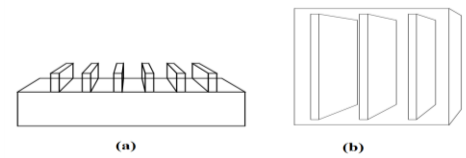

Fig. 1. Investigated configurations: a) rectangular isothermal fins on a horizontal surface. b) rectangular isothermal fins on a vertical surface.

A. Rectangular isothermal fins on a horizontal surface

Figure 1a shows rectangular fins on a horizontal surface. In 1970, Jones and Smith derived an empirical correlation to determine the natural heat transfer from horizontal fins [3]. They assumed the fins as U-shape horizontal channels and based on this assumption they developed their correlation by using the dimensionless Nusselt number

$$Nu = 0.00067 \cdot Gr \cdot Pr \cdot \left\{1 - e^{(-7640/\mathrm{Gr}\cdot\mathrm{Pr})^{0.44}}\right\}^{1.7}, \qquad (1)$$

where Gr is Grashof number and Pr is Prandtl number.

In this case, they have defined the fin space (S) as the characteristic length. According to (1), the Nusselt number is determined without considering the fins' size.

While in [8], Tari and Mehrtash defined an empirical correlation for the natural heat transfer from upward horizontal plate-fin heat sinks according to the fin characteristics. For this purpose, they defined a modified Grashof number as:

$$Gr' = Gr \cdot \left(\frac{H}{L}\right)^{0.5} \cdot \left(\frac{S}{H}\right)^{0.38}, \qquad (2)$$

where H (m) and L (m) are respectively the fin height and length. The Nusselt number is then expressed as:

$$Nu = 0.0915 \cdot (Gr' \cdot Pr)^{0.436}. \qquad (3)$$

B. Rectangular isothermal fins on a vertical surface

The rectangular isothermal fins on vertical base plate is the common heat sink configuration. A number of research have been published about calculating the natural heat transfer from this configuration. One of the earliest research about this configuration is back to Van De Pol research in 1973. In [4], he also described the vertical fin configuration as a U-shape vertical channel. In this case, the Nusselt number was defined as:

$$Nu = \frac{r}{H} \cdot \frac{Gr \cdot Pr}{Z} \cdot \left[1 - e^{-Z \cdot \left(\frac{0.5}{(r/H) \cdot Gr \cdot Pr}\right)^{0.75}}\right], \qquad (4)$$

where Z is defined as:

$$Z = 24 \cdot \frac{1 - 0.483 \cdot e^{-0.17/\alpha}}{\left[(1+\alpha/2) \cdot \left(1 + \left(1 - e^{-0.83 \cdot \alpha}\right) \cdot \left(9.14 \cdot \sqrt{\alpha} \cdot e^{-465 \cdot S} - 0.61\right)\right)\right]^3}, \qquad (5)$$

α is the channel aspect ratio and r (m) the characteristic length.

In [2], Tari and Mehrtash introduced a new empirical correlation for the calculation of the Nusselt number from a vertical heat sink. They defined a new modified Grashof number as:

$$Gr' = Gr \cdot \left(\frac{H}{L}\right)^{0.5} \cdot \left(\frac{S}{H}\right), \qquad (6)$$

and based on the modified Grashof number, they defined the Nusselt number as:

$$Nu = 0.0929 \cdot (Gr' \cdot Pr)^{0.5} \text{ for } Gr' \cdot Pr < 250, \qquad (7a)$$

$$Nu = 0.2413 \cdot (Gr' \cdot Pr)^{1/3} \text{ for } 250 < Gr' \cdot Pr < 10^6. \qquad (7b)$$

Finally, the natural convection coefficient h_c is calculated from the Nusselt number as [11]:

$$h_c = \frac{Nu \cdot k}{L_c}, \qquad (8)$$

Where k (W/m K) is the fluid thermal conductivity and L_c (m) is the characteristic length of the cooling surface.

III. Experimental Setup and Procedur

The objective of the experimental work in this paper is to assess the natural heat transfer coefficient from the fin section of the stator coil of a permanent magnet generator in the horizontal and vertical orientations. Yet another objective is to compare the analytical data to experimental data for finding the appropriate empirical correlation in both cases and verify the accuracy of these correlations.

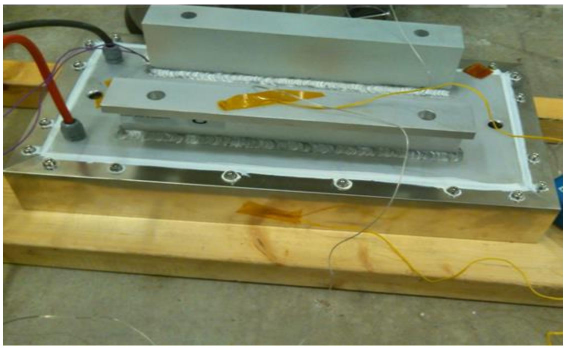

Fig. 2. The coil module used in the experimental work.

From previous research works [12] and [13], the stator coil used in this investigation consists of six different faces as shown in Fig. 2. To consider the natural convection from the fin's side, the heat flux flow should be confined only to the fin side. To achieve this, we created an insulation box according to the dimensions of the coil by means of foam insulation boards with thickness of 10 cm. Figure 3 shows the coil box and the fins' side of the coil in vertical and horizontal configurations. Since the foam insulation material has low thermal conductivity k =0.3 (W/m K), the thermal flux flow is restricted to the open surface. Therefore, the box is operating as a semi-closed calorimeter. Another important point about the box is the temperature operation point; as the foam insulation board can handle temperatures up to 90℃, during the experiment, the coil temperature should not exceed that temperature. Furthermore, in order to protect the test bench from external heating source and bulk fluid motion as well as increasing the accuracy of the results, the experimental setup is located in a closed room and the environment temperature of room is monitor by an extra thermocouple.

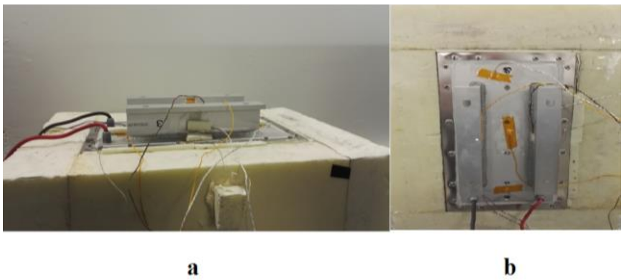

Fig. 3. The used configurations in the experiment: Coil module inside the coil box in a) horizontal position b) vertical position.

The fin side of the coil module is constructed from aluminum with a thermal conductivity of 171 (W/m.K) and an emissivity of 0.5. The test bench is designed to measure the natural convection coefficient from the fin's side of the coil module. For this purpose, we use the DC test method. The DC test is a common experimental method for determining the convection coefficient of heat transfer. In this test, the loss of the coil is confined to the joule loss of the coil winding and the power can be easily calculated from the measured electric quantities. In the calculations, we accounted for the variations in the winding electrical resistance, as the winding resistivity is temperature dependent [12], [14]. During the experiments, in addition to the power input to the coil module, the temperature of the fins side is measured at different locations. The Coil module is supplied through a digitally-controlled DC power supply. The voltage and the current are measured to determine the input power, which is equal to the heat power. Seven K-type thermocouples are installed by means of adhesive material in various locations on the fin side of the coil module. The ambient temperature is also measured by means of a K-type thermocouple. For the purpose of increasing the accuracy of the temperature measurement and minimizing the contact resistance between the thermocouple and the coil module surface, we used a commercial thermal paste. The average temperature of these seven thermocouples is assumed to be the mean temperature of the fin side of the coil module. This assumption is based on the observed measurements, where the mean difference between the temperatures of the seven thermocouples is 1% and is less than 0.5% in most cases. During the experiments, all the temperature data are collected by means of a Graphtec GL200 logger. The experiment has been carried out for each current input until the system reached its steady state condition. For each of the configurations, the experimental procedure is repeated for five different current inputs: 10, 12, 15, 17 and 20 Amps.

IV. The Analysis Method of Experimental Data

The total heat produced in the coil is equal to the total input power. Thus, the total heat in Watts (W) is defined as:

$$Q_T = V \cdot I \,, \tag{9}$$

where V (V) is the input voltage and I (A) the input current.

Accordingly, the total heat is extracted by means of the natural convection and radiation phenomena. Thus, the total heat can be described as:

$$Q_T = Q_c + Q_r \,, \tag{10}$$

where Q_c (W) is the amount of heat extracted by natural convection and Q_r (W) is the amount of heat extracted by radiation.

According to [15] and [16] the heat extraction coefficient h_e is calculated as:

$$h_e = \frac{Q_T}{(T_s - T_a)\cdot A} \quad , \tag{11}$$

Where T_s (℃) is the mean temperature of the fin side of the coil module, T_a (℃) is the ambient temperature and A (m^2) is the fin side surface area.

The total heat extraction coefficient is defined as the sum of the convection h_c and radiation h_r coefficients:

$$h_e = h_c + h_r \ . \tag{12}$$

The radiation coefficient is defined as [16]:

$$h_r = \varepsilon \cdot \sigma \cdot \left(T_s^2 + T_a^2\right) \cdot \left(T_s + T_a\right) , \tag{13}$$

where ε is the emissivity of the surface and $\sigma = 5.67\times10^{-8}$ (W/m^2 K^4) is the Stefan-Boltzmann constant.

A. Uncertainty Analysis of Experimental Results

In this section, we determined the total accuracy of experimental data according to the accuracy of the measurement instruments. During the experiment, the voltage and current are measured with the TTi QPX1200S. The accuracy of the voltage and current readings are 0.1% and 0.3% respectively. Furthermore, the standard accuracy of the K-type thermocouple is 0.75 %. According to [1], the power uncertainty is evaluated as:

$$\omega_{Q_T} = \left[\left(\frac{\partial Q_T}{\partial V}\cdot \omega_V\right)^2 + \left(\frac{\partial Q_T}{\partial I}\cdot \omega_I\right)^2\right]^{0.5} , \tag{14}$$

where ω_{QT}, ω_V and ω_I are the uncertainties in the total input power, voltage and current.

This leads to the uncertainty for the computed convection coefficient as:

$$\omega_h = \left[\left(\frac{\partial h}{\partial Q_T}\cdot \omega_{Q_T}\right)^2 + 2\cdot\left(\frac{\partial h}{\partial T}\cdot \omega_T\right)^2\right]^{0.5} , \tag{15}$$

where ω_T is the uncertainty in the temperature measurement.

V. Analytical Calculation Method

Figure 4 shows the actual shape of the fin side of the coil module and the modified shape used in the theoretical calculations. The actual fin is divided into two sections; the fin section and the flat plate surface. The module consists of two fins spaced by S=86 mm. The fin's height is H=62.7 mm, its thickness is D_{fin}=10mm, and its length is L_{fin}=265 mm. The width of the L-shaped plate on top of the fin is W_L=38 mm. The width of the coils module is W=222 mm and its length L_{coil}=419 mm. Therefore, the convection coefficient for each section is calculated separately and finally, the equivalent convection coefficient is calculated according to the area of each section. It should be noted that the actual setup consists of L-shaped fins. However, the upper part of the fin is for mechanical support purpose only. The effect of this part on the heat convection coefficient has been estimated through 2D finite element computations. It turns out that this part participated in the heat transfer, as it dissipated 13.8 % of the total heat in average, resulting in a temperature difference of 3.7 %. Therefore, this shape has been replaced in the experimental calculations by an equivalent increased length of the fins, so that the same amount of heat is dissipated through the additional length. However, this change in the length affected very little the calculation of the heat transfer coefficient (less than 1% difference, which is lower than the measurement uncertainty).

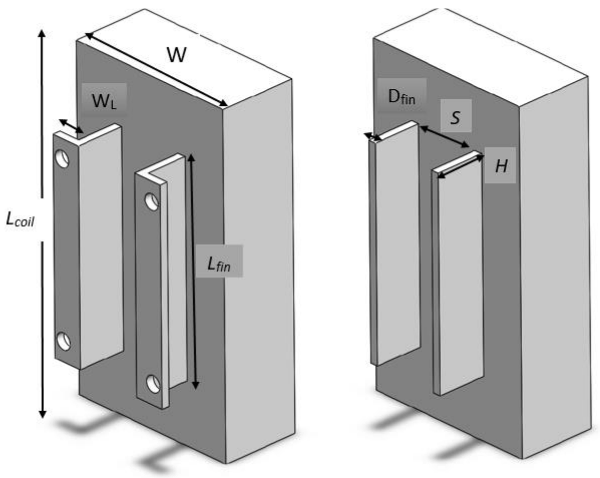

Fig. 4. Illustration of the investigated setup: (a) Actual module format (b) Simplified format. The indicated dimensions are given in the text.

According to the theory of convection from a flat plate, the Nusselt number empirical correlations for the horizontal upward and vertical configurations are defined respectively as [17] and [18]:

$$Nu = 0.54(Gr \cdot Pr)^{0.25}, \quad (16)$$

$$Nu = \left\{ 0.825 + \frac{0.387(Gr \cdot Pr)^{0.25}}{\left[1 + (0.492/Pr)^{9/6}\right]^{8/27}} \right\}^2. \quad (17)$$

Therefore, the equivalent convection coefficient is defined as:

$$h_c = \frac{h_1 \cdot A_1 + h_2 \cdot A_2}{A_T}, \quad (18)$$

where h_1 (W/m^{2}C) is the convection coefficient from fin section, h_2 (W/m^{2}C) is the convection coefficient from the flat section, A_1 (m^2) is the surface area of the fin sections, A_2 (m^2) is the surface area of the flat plate. A_T (m^2) is the total surface area.

VI. Results and Discussion

The validation of the empirical correlations is made by comparison with experimental data for the five input currents for both horizontal and vertical configurations. We present the analytical and experimental results according to the coils module's orientation.

In the experimental part, the fin side of the coil module is studied in both horizontal and vertical cases. To determine the convection coefficient from the experimental data; first, the total heat Q_T is calculated by Eq. (9), then according to the ambient and surface temperatures of the case study, the total heat extraction coefficient h_e and radiation coefficient h_r are determined respectively by Eq. (11) and Eq. (13). Finally, the natural convection coefficient h_c is calculated by Eq. (12).

For the analytical part, according to the coil module orientation, the appropriate empirical correlations are used to calculate the natural convection coefficient h_c. The fin side of the coil module consists of two different sections: the flat part and the fin section. Therefore, the amount of the natural convection is calculated separately for each section. To calculate the natural convection coefficient h_1 of the model, according to the model's configuration, the Nusselt number Nu for the fin section was calculated by Eq. (1) and Eq. (3) in the horizontal case or Eq. (4), Eq. (7a) and Eq. (7b) in the vertical case. While for the calculation of the convection coefficient h_2 of the flat part, the Nusselt number is calculated by Eq. (16) or Eq. (17). Then, h_1 and h_2 are calculated by Eq. (8). Finally, the total natural convection coefficient h_c is calculated by Eq. (18).

For the ease of comparing the experimental and analytical results, the data for each configuration are plotted against the temperature difference between the ambient and the cooling surface. This plot is expected to help to choose the appropriate correlation depending on the configuration and the temperature range. It should be noted that the maximum uncertainty in the computed convection coefficients is 6.2%. The uncertainty values are displayed as error bars in Fig. 5 and 6.

Tables I shows the experimental data for the horizontal configuration.

TABLE I. Experimental Results for Horizontal Configuration

I (A)	T_s (°C)	T_a (°C)	Q_T (W)	h_e W/(m^{2}K)	h_r W/(m^{2}K)	h_c W/(m^{2}K)
10	27.7	18.2	15.23	10.09	3.53	6.56
12	32.4	18.9	22.38	10.37	3.63	6.74
15	38.6	18.2	35.97	11.07	3.73	7.34
17	43.9	18.3	47.34	11.58	3.83	7.75
20	53.4	19	68.16	12.44	4.03	8.41

Tables II and III show the analytical data for the horizontal case for which the natural convection coefficients h_1 for the fin section has been calculated based on Jones [3] and Tari [8] empirical correlations respectively.

TABLE II. ANALYTICAL RESULTS FOR HORIZONTAL ORIENTATION BASED JONES EMPIRICAL CORRELATION

I (A)	T_s (°C)	T_a (°C)	h_e W/(m²K)	h_r W/(m²K)	h_c W/(m²K)
10	27.7	18.2	8.87	3.53	5.34
12	32.4	18.9	9.47	3.63	5.84
15	38.6	18.2	10.16	3.73	6.43
17	43.9	18.3	10.66	3.83	6.83
20	53.4	19	11.36	4.03	7.33

TABLE III. ANALYTICAL RESULTS FOR HORIZONTAL ORIENTATION BASED TARI EMPIRICAL CORRELATION

I (A)	T_s (°C)	T_a (°C)	h_e W/(m²K)	h_r W/(m²K)	h_c W/(m²K)
10	27.7	18.2	10.07	3.53	6.54
12	32.4	18.9	11.02	3.63	7.39
15	38.6	18.2	12.15	3.73	8.42
17	43.9	18.3	12.96	3.83	9.13
20	53.4	19	14.05	4.03	10.03

Figure 5 shows the variation of the total heat extraction coefficient h_e with the temperature difference for five different input currents in the horizontal configuration. The average relative difference of the analytical data based on Tari [8] in comparison with the experimental one is 8.1% and the maximum relative difference based on this correlation is 12.9%. The average relative difference for analytical data based on Jones [3] empirical correlation is 9.1% and the maximum relative difference for these data is 12.1%. However, according to Table I and II, it is clearly shown that by increasing the temperature, the relative difference of the analytical data based on Jones correlation compared with the experimental data decreases. For the temperature difference of 34.4 (℃) the relative difference is 9.5% with Jones correlation while in the same situation it is 11.5% with Tari correlation. Furthermore, Fig. 5 shows that by increasing the temperature the analytical data curve based on Tari's correlation diverges from the experimental curve. It is interesting to note that the Jones empirical correlation has been developed without any dependence on the physical fin array's properties e.g., fin's height (H) and fin's length (L_{fin}).

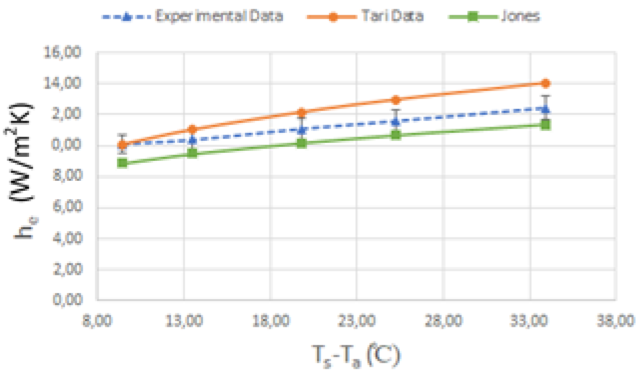

Fig. 5. Variation of the total heat extraction coefficient with the temperature difference for the horizontal orientation.

Tables IV shows the experimental data for the vertical configuration.

TABLE IV. EXPERIMENTAL RESULTS FOR VERTICAL CONFIGURATION

I (A)	T_s (°C)	T_a (°C)	Q_T (W)	h_e W/(m²K)	h_r W/(m²K)	h_c W/(m²K)
10	28.6	19	15.31	9.97	3.56	6.41
12	33.1	19.3	22.44	10.20	3.65	6.55
15	40.7	19.2	36.28	10.60	3.79	6.81
17	45.9	18.4	47.65	10.88	3.87	7.00
20	55.4	18.6	68.66	11.70	4.07	7.63

Tables V and VI show the analytical data for the vertical case for which the natural convection coefficients h_1 for fin section has been calculated based on Van De Pol [4] and Tari [2] respectively.

TABLE V. ANALYTICAL RESULTS FOR VERTICAL ORIENTATION BASED VAN DE POL EMPIRICAL CORRELATION

I (A)	T_s (°C)	T_a (°C)	h_e W/(m²K)	h_r W/(m²K)	h_c W/(m²K)
10	28.6	19	7.12	3.56	3.56
12	33.1	19.3	7.56	3.65	3.91
15	40.7	19.2	8.17	3.79	4.38
17	45.9	18.4	8.54	3.87	4.67
20	55.4	18.6	9.08	4.07	5.01

TABLE VI. ANALYTICAL RESULTS FOR VERTICAL ORIENTATION BASED TARI EMPIRICAL CORRELATION

I (A)	T_s (°C)	T_a (°C)	h_e W/(m²K)	h_r W/(m²K)	h_c W/(m²K)
10	28.6	19	8.02	3.56	4.46
12	33.1	19.3	8.63	3.65	4.98
15	40.7	19.2	9.47	3.79	5.68
17	45.9	18.4	10.00	3.87	6.13
20	55.4	18.6	10.71	4.07	6.64

Figure 6 shows the variation of the total heat extraction coefficient (h_e) with the temperature difference for five different input currents in the vertical orientation. Significant differences were found between the analytical data based on Van Del Pol [4] correlation and the experimental data. The mean relative difference of this empirical correlation is about 24% and the maximum one is 29%. The main reasons for this difference can be explained by the defined range for this correlation as well as by the fact that Van De Pol equations were derived for periodical channel whereas Tari's equation is valid for a single channel too. According to [4] and [19], this correlation is fitted for rectangular fins in the range of 0.33 $<H/S<$4 and 42$<L_{fin}/S<$10.6. Our case-study is not located within these ranges. However, as Fig. 6 shows, the difference between the analytical data based on Tari's correlations and experimental data is significant at lower temperatures, but as the temperature difference increases, the analytical data curve converges toward the experimental one. Thus, the appropriateness of this correlation is better at high temperature raises. The mean relative difference for this correlation is 12%. The maximum relative difference occurred at low-temperature rise and it amounts to 29%.

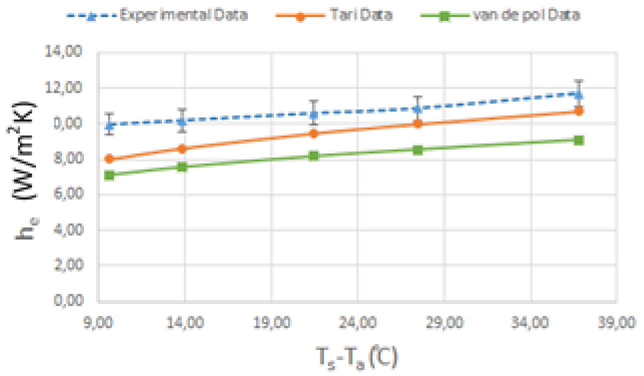

Fig. 6. Variation of the total heat extraction coefficient according to the temperature difference for the vertical orientation.

Another interesting finding is achieved by simple comparison between natural convection coefficients of the corresponding currents, which proves that the natural convection coefficient in the horizontal configuration is higher than in the vertical one, which means that the horizontal flat plate fins topology provides better natural cooling than the vertical one and thus can reduce the amount of the cooling power consumption as well as the surface temperature.

VII. Conclusion

The focus in this paper is on the determination of the natural convection heat transfer from parallel rectangular fins on a flat base plate of the coil of a large permanent magnet generator in both horizontal and vertical configurations, by means of experimental and analytical methods. For this purpose, a number of empirical correlations for both configurations was studied. Finally, according to a comparison of experimental data with empirical ones, the appropriate ones were selected. According to the selected correlations, the natural convection coefficients for both configurations were calculated and validated through an experimental test setup. The experimental data were collected for five different input currents corresponding to different temperature raises. All experiments were made at the steady state of the thermal system.

According to this study, it can be concluded that for the large rectangular fins in the horizontal configuration the results based on Jone's empirical correlation are in good agreement with the experimental one. In addition, the appropriateness of this correlation is enhancing with increasing temperature rise. For the vertical case, the analytical results based on Tari's empirical correlation have a good agreement with the experimental data.

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