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Convection and radiation cooling of overhead power lines - laboratory verification using thermography

Abstract

The thermal behavior of overhead power lines depends upon physical parameters, such as surface emissivity and line dimensions, as well as weather conditions. In this paper, the results of the convection and radiation cooling of a conductor that simulate a power line are presented. Laboratory experiments were conducted and the results were compared with the data obtained using empirical formulae from the literature. Both the laminar and the turbulent airflow were investigated.

Keywords: Overhead power line, convection, laminar and turbulent airflow, radiation, IR thermography.

1. Introduction

Extensive research has been conducted in the area of the thermal loading of overhead power lines [1-7]. The loading of the line results in the increasing of the line's temperature. This temperature is a function mainly of loading and cooling of the line but also is a function of the conductor material properties, the line's diameter, surface state and ambient meteorological conditions [7].

The ampacity of a given overhead power line depends mainly on convection cooling contrary to underground cables where heat is dissipated mainly by conduction [8-9]. Weather conditions and line dimensions are the most important factors of line's cooling [10]. Overhead power lines may be subjected to wind, resulting in forced convective cooling. For a given wind speed, the parallel blowing wind results in a lower convective heat loss than wind blowing perpendicularly to the overhead power line. Cooling is not only dependent on wind speed and direction, but also on the turbulence of a flow.

As air velocity increases, the effect of the natural convective heat transfer on the total heat transfer diminishes while the forced convective heat transfer increases [11]. With low wind speed, natural and forced convection may occur at the same time. Morgan [12] has shown that for conductors with an outside surface temperature rise above ambient not exceeding 20 K, as is often the case in outdoor power lines installations, the natural convective cooling can be ignored when the wind velocity exceeds 1 m/s.

Many researchers performed studies on natural [12-13] and forced [14-18] heat convection from the horizontal cylinders. Except for analytical and experimental techniques, numerical methods [19] were extensively used. Although the subject has been studied broadly for many years, discrepancies in all of the data still exist due to various conditions the results were obtained.

2. Theoretical considerations

In this section, theoretical and empirical approaches are developed concerning the determination of the heat transfer coefficients of overhead power lines.

Heat transfer phenomena are complex for overhead power lines. Regardless of the particular nature of the convection heat transfer process and using Newton's law of cooling, the appropriate heat rate equation is of the form [7]:

$$q' = \pi D_e h_{t,exp} \Delta T, \ \frac{\mathrm{W}}{\mathrm{m}} \tag{1}$$

q' denotes heat losses per meter and is proportional to conductor's surface temperature rise above ambient ΔT. D_e is conductor external diameter. From eq. 1, total heat transfer coefficient $h_{t,exp}$ can be determined from the experimental data by:

$$h_{t,exp} = \frac{q'}{\Delta T \pi D_e}, \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{2}$$

Evaluation of the total heat transfer coefficient can become quite complex. Previous versions of IEC 60287 [1] standard considered only natural convection; however, a more recent version [20] provides estimation method for both natural and forced convection. According to IEC 60287, the total heat transfer coefficient could be calculated by [1]:

$$h_{t,IEC} = h \cdot (\Delta T)^{\frac{1}{4}}, \ \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{3}$$

where h is the heat transfer coefficient embodying convection, radiation, conduction, and mutual heating. It is given by the following analytical expression [1]:

$$h = \frac{Z}{(D_e)^g} + E, \ \frac{\mathrm{W}}{\mathrm{m^2K^{5/4}}} \tag{4}$$

Constants Z, E, and g are given in Table 1. Although the IEC 60287 standard method is concerned for its validity, improvements have been presented by Sedaghat et al. [21] for conductors under general operating conditions.

Tab. 1. Values for constants Z, E and g in free air

Parameter	Z	E	g
Single cable	0.21	3.94	0.60

Heat losses generated in a conductor are dissipated to the environment due to convection and radiation. Over the years, several simple formulae have been developed to account for the convection and radiation heat transfer outside the conductor surface. Neglecting conductive heat transfer, which is very small in free air, total heat transfer coefficient is given theoretically by [7]:

$$h_t = h_c + h_r, \ \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{5}$$

where h_c and h_r is convection and radiation heat transfer coefficients, respectively.

Radiation heat transfer coefficient h_r can be calculated by [7]:

$$h_r = \varepsilon\sigma_B(T_s + T_a)(T_s^2 + T_a^2) \approx 4\varepsilon\sigma_B T_{av}^3, \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{6}$$

where ε is emissivity coefficient of the surface and σ_B is Stefan-Boltzmann constant, equal to $5.67 \cdot 10^{-8}$ W/m^{2}K^4. T_s and T_a are conductor surface and ambient temperature (in K), respectively. T_{av} is the average value of T_s and T_a (in K).

In case of laminar airflow, many empirical formulae have been proposed. An approximate value of the forced convection coefficient can be obtained from Neher and McGrath [10]:

$$h_{c,Neher} = 2.87\sqrt{\frac{v}{D_e}}, \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{7}$$

where v is air velocity (m/s).

Forced convection coefficient h_c can be expressed in terms of Nusselt number as:

$$h_c = \frac{k \cdot Nu}{D_e}, \ \frac{\mathrm{W}}{\mathrm{m^2K}} \tag{8}$$

where k is the air heat conductivity (W/m·K).

Hilpert [14] proposed an empirical expression for Nusselt number in case of forced convection:

$$Nu = CRe_D^m Pr^{1/3} \tag{9}$$

where parameters C and m are given in Table 2. Re_D is the Reynolds number while Pr is the Prandtl number. Re_D is calculated by:

$$Re_D = \frac{vD_e}{V} \tag{10}$$

V is the kinematic viscosity of air (m^2/s).

Tab. 2. Constants of Hilpert expression

Re_D	C	m
0.4-4	0.989	0.330
4-40	0.911	0.385
40-4000	0.683	0.466
4000-40000	0.193	0.618
40000-400000	0.027	0.805

The formula suggested from Zukauskas [15] takes the form:

$$Nu = CRe_D^m Pr^n \left(\frac{Pr}{Pr_s}\right)^{1/4} \tag{11}$$

All properties are evaluated at T_a, except Pr_s , which is evaluated at T_s (temperature of the conductor's surface). Values of C and m are listed in Tab. 3. If $Pr \leq 10$, $n = 0.37$ while if $Pr \geq 10$, $n = 0.36$.

Tab. 3. Constants of Zukauskas expression

Re_D	C	m
1–40	0.75	0.4
40–10^3	0.51	0.5
10^3–2x10^5	0.26	0.6
2x10^5–10^6	0.076	0.7

Churchill et al. [17] proposed an equation for both laminar and turbulent convection:

$$Nu = 0.30 + \frac{0.62 Re_D^{1/2} Pr^{1/3}}{\left[1+(0.4/Pr)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{Re_D}{282000}\right)^{5/8}\right]^{4/5} \tag{12}$$

All properties of eqns. (8) and (12) are evaluated at the mean boundary layer temperature T_f, termed the film temperature:

$$T_f = \frac{T_s + T_a}{2}, \ \mathrm{K} \tag{13}$$

3. Experimental measurements

In order to examine the cooling of overhead power lines, two experiments were conducted in a laboratory low-speed wind tunnel. Fig. 1 presents the experimental apparatus. In the first experiment, the laminar flow was used, while during the second measurement, the turbulences were introduced for forced cooling of conductor. For the purpose of this work, a low speed wind tunnel was used. Proper wind tunnel design ensured that the air is brought into the test section with uniform velocity and very little turbulence at least for velocities up to 5.0 m/s. Inversion of the airflow direction in the wind tunnel results in a turbulent airflow.

A steel conductor (length l =0.40 m and diameter D_e=1.2 mm) was located inside the wind tunnel, perpendicularly to the airflow; due to the sufficiently large aspect ratio l/D_e, axial conduction losses are negligible [18]. Then, the conductor was fed with a constant DC electrical current (I=6 A) until a thermal steady state. The air velocity in the wind tunnel was increased by a step of 0.5 m/s in the range from 0 to 5 m/s. The air and ambient temperature was close to 300 K. The fan speed was controlled by varying frequency supplied to the motor. Since the conductor is located indoors, the effect of solar radiation is neglected.

During the experiments, conductor's temperature, voltage drop and electrical current were automatically recorded every 1 second. Surface temperature of the conductor was measured directly by the infrared camera Cedip Titanium 560M with relative accuracy 1% of the chosen temperature range. To avoid flow disturbances, the infrared camera was located outside the experimental cell. Thus, surface temperatures had to be measured through a window fixed up in one of the cell walls of the wind tunnel.

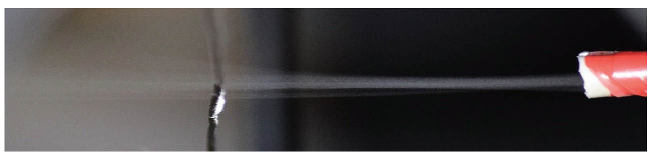
In thermal equilibrium, having measured the voltage drop across the conductor and the dissipated electric current, the dissipated power step q', which is the conductor's Joule losses, can be calculated by:

$$q' = \frac{V \cdot I}{l}, \ \frac{\mathrm{W}}{\mathrm{m}} \tag{14}$$

V is the voltage drop (V), I is the electric current (A), and l is the conductor's length (m).

In Fig. 2 the laminar (a) and turbulent (b) flows are visualized while air is moving from right-to-left and left-to-right directions. In both experiments the inlet temperature of air was the same – the room temperature in the laboratory.

Fig. 1. Laboratory arrangement



(a)

(b)

Fig. 2 a) Laminar, and b) turbulent flow in the tunnel while the air moves in different directions

In Fig. 3 the dissipated Joule losses versus air velocity are presented for both experiments. The fact that q' turns out to be smaller for turbulent flow seems to be unrealistic at first sight. It must be noted here that a constant current was fed through the conductor. In turbulent flow, the heat transfer coefficient is higher (Fig. 5), hence the conductor temperature is reduced, too. Due to the positive temperature coefficient of the conductor electric resistance, less power will be dissipated in case of turbulent flow.

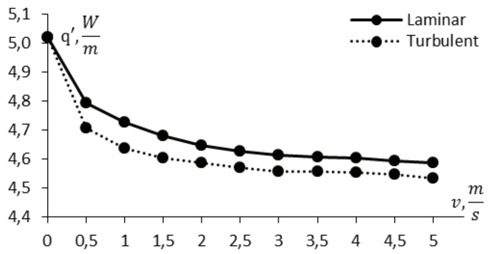


Fig. 3. Dissipated Joule losses versus air velocity during laminar and turbulent airflow

4. Results

When a constant heat flux is generated and a steady-state condition is reached, the experimental heat transfer coefficient h_t can be directly determined using eqn. (1). Using the experimentally estimated dissipated power and the corresponding temperature rise, the experimental total heat transfer coefficients were calculated by using eqn. (2). According to IEC 60287 standard, for natural convection only, the total heat transfer coefficient was calculated from eqn. (3) using the heat transfer coefficient obtained from eqn. 4. The relative difference between the experimental results and the IEC 60287 standard is approximately 20% (Table 4).

Subsequently, for air velocities (0.5 – 5.0) m/s, the total empirical heat transfer coefficients were calculated by eqn. (5). Convection heat transfer coefficients were calculated both by Neher and McGrath formulae, and using eqn. (8). The Nusselt numbers are calculated three times by using formulae (9), (11) and (12). Then, each convection coefficient summed up with the corresponding radiation one – eqn. (6). The resulted coefficients are the total heat transfer coefficients.

In Fig. 4, the experimental and the empirical (from the literature) total heat transfer coefficients for laminar airflow versus air velocity are presented. As it can be seen, the experimental heat transfer coefficients differ no more than 5% from the calculated empirical ones. This demonstrates the validity of the existent literature. The RMS errors (%) between the considered empirical methods and the experimental results are summarized in Table 5.

Fig. 5 presents a comparison between the experimental total heat transfer coefficients due to laminar and turbulent airflow. As expected, the turbulent airflow gives rise to higher coefficient than the laminar one. In laminar flow, the air is flowing in the thin and relatively uniform boundary layer. During the turbulent flow there are laminar, transient and turbulent boundary layers with not well defined borders between them. In some cases they overlap and mixed. Since the energy transfer rate depends on the temperature difference, the layer of air gets warmer by the end of the rod and is removing heat at a lower rate than in the front part of the rod. The turbulent flow has wider boundary layer. This means that more fresh cold air will contact the surface resulting in the higher heat transfer rate due to a larger area of heat exchange and higher local air velocity because the extra vortexes in the flow. Table 6 lists the relative difference between the laminar and turbulent total heat transfer coefficients with respect to the laminar ones.

Tab. 4. Experimental and IEC 60287 heat transfer coefficients for natural convection

Method	Experiment	IEC 60287
h_t, W/m^{2}K	45.76	36.73

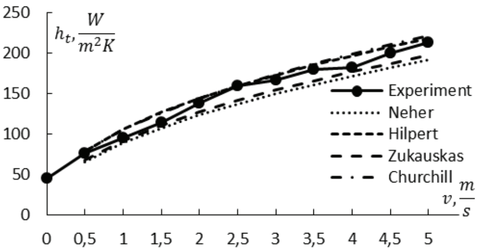


Fig. 4. Comparison between experimental and according to empirical formulae calculated total transfer coefficients for laminar airflow

Tab. 5. RMS errors of considered methods

Method	Neher	Hilpert	**Zukauskas**	Churchill
RMS error (%)	1.06	6.28	7.33	6.51

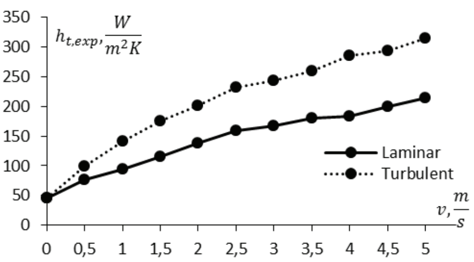


Fig. 5. Comparison between laminar and turbulent airflow experimental total heat transfer coefficients

Tab. 6. Relative differences of experimental total heat transfer coefficients

Air velocity (m/s)	0.5	1	1.5	2	2.5	3	3.5	4	4.5	5
Difference (%)	30.0	48.5	51.6	45.9	45.4	45.6	44.7	56.4	46.6	47.5

5. Conclusions

In this contribution, the cooling of overhead power lines were examined. In the laboratory, two experiments were conducted. For different air velocities, the conductor was heated until the thermal steady state condition was reached. The generated Joule losses were dissipated due to convection and radiation. During the experiments, the voltage drop, electrical current and the conductor temperatures were measured.

Knowing the dissipated heat and the conductor surface temperature rise above ambient, the corresponding heat transfer coefficients were calculated. Then, the experimentally estimated heat transfer coefficients were compared with the empirically ones published in the literature. It has been shown that the empirical calculation methods from the literature provide the valid results under certain conditions. One has to be very careful anytime the data from literature is used in practice, especially for natural convection.

Finally, it has been verified experimentally that turbulent airflow results in higher heat transfer coefficient (about 50%) than laminar one. When the airflow turbulence intensity increases, the heat transfer coefficient values $h_{t,exp}$ increase significantly.

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6. References

[1] IEC 60287-2-1:2015 RLV: Electric cables – Calculation of the current rating – Part 2-1: Thermal resistance – Calculation of thermal resistance. 2015.

[2] IEEE 738-2012: Standard for calculating the current-temperature relationship of bare overhead conductors. 2013.

[3] Morgan V.: External thermal resistance of aerial bundled cables. IEE Proceedings C, vol. 140, no. 2, pp. 65-72, 1993.

[4] Slaninka P.: External thermal resistance of air-installed power cables. Proceedings of IEE, vol. 116, no. 9, pp. 1547-1552, 1969.

[5] Whitehead S., Hutchings E.E.: Current rating of cables for transmission and distribution. Journal of the Institution of Electrical Engineers, vol. 83, no. 152, pp. 517-557, 1938.

[6] Theodosoglou, I., Chatzipanagiotou, P., Chatziathanasiou, V., Boier, A., Rata, M.: Measurement and calculation of thermal characteristics of an overhead power line. Acta Electrotech Proc. 5th Int. Conf. Mod. Power. Syst., pp 474-478, 2013.

[7] Anders G.: Rating of electric power cables in unfavorable thermal environment. Wiley-IEEE Press, 2005.

[8] Chatziathanasiou, V., Chatzipanagiotou, P., Papagiannopoulos I., De Mey G., Wiecek B.: Dynamic thermal analysis of underground medium power cables using thermal impedance, time constant distribution and structure function. Appl. Therm. Eng., vol. 60, no. 1-2, pp. 256–260, 2013.

[9] Chatzipanagiotou, P., Chatziathanasiou, V., De Mey G., Wiecek B.: Influence of soil humidity on the thermal impedance, time constant and structure function of underground cables: A laboratory experiment. Appl. Therm. Eng., vol. 113, pp. 1444–1451, 2017.

[10] Neher J. H., McGrath M. H.: The calculation of the temperature rise and load capability of cable systems. AIEE, vol. 76, no. 3, pp. 752-764, 1957.

[11] Bergman T., Lavine A., Incropera F., DeWitt D.: Introduction to Heat Transfer. John Wiley & Sons, Inc., 2011.

[12] Morgan V.: The overall convective heat transfer from smooth circular cylinders. Advances in Heat Transfer, vol. 11, pp. 199-264, 1975.

[13] Boetcher S.: Natural convection from circular cylinders. Springer International Publishing, 2014.

[14] Hilpert R.: Wärmeabgabe von geheizten Drähten und Rohren im Luftstrom. Forsch Ing-Wes, vol. 4, no.5, pp. 215–224, 1933.

[15] Žukauskas A.: Heat Transfer from Tubes in Crossflow. Advances in Heat Transfer, vol. 8, pp. 93-160, 1972.

[16] Churchill S., Chu H.: Correlating equations for laminar and turbulent free convection from a horizontal cylinder. International Journal of Heat and Mass Transfer, vol. 18, pp. 1049-1053, 1975.

[17] Churchill S., Bernstein M.: A Correlating Equation for Forced Convection from Gases and Liquids to a Circular Cylinder in Crossflow. J. Heat Transfer, vol. 99, no.2, pp. 300-306, 1977.

[18] Morgan V.: The Overall Convective Heat Transfer from Smooth Circular Cylinders. Advances in Heat Transfer, Vol. 11, pp. 199-264, 1975.

[19] Brito Filho J.P.: Heat transfer in bare and insulated electrical wires with linear temperature-dependent resistivity. Applied Thermal Engineering, vol. 112, pp. 881–887, 2017.

[20] IEC 60287-2-3:2017: Electric cables - Calculation of the current rating - Part 2-3: Thermal resistance - Cables installed in ventilated tunnels, 2017.

[21] Sedaghat A., de León F.: Thermal Analysis of Power Cables in Free Air: Evaluation and Improvement of the IEC Standard Ampacity Calculations. IEEE Transactions on Power Delivery, vol. 29, no. 5, pp. 2306–2314, 2014.

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