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Non-stationary thermal model of indoor transformer stations

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Abstract The paper presents a thermal model of an indoor transformer station. The model includes generation of heat and its transfer from the transformer active parts to the surrounding air, natural air ventilation through ventilation holes, and convection and conduction transfer of heat through room parts. All parameters of the model are discussed and the procedure, based on measurements, for the determination of natural air ventilation parameters is described. The model is tested using measurements on a typical indoor 10 kV/0.4 kV transformer station. The results of the proposed thermal model and those of a simplified approach from Standards are compared.

Key words Power transformer, Loading, Natural ventilation, Thermal model, Indoor transformer station

1 Introduction

The basic criterion for transformer loading is the temperature of the hottest spot of the solid insulation (hot-spot). It must not exceed the prescribed value in order to avoid irreversible insulation faults. Also, long-term insulation ageing, depending on the hot-spot temperature diagram, should be less than the planned value. The hot-spot temperature depends on the load loss (i.e. on the current) diagrams and on the temperature of the external cooling medium. A hot-spot temperature calculation procedure is given in the International Standards [1]. A series of papers, e.g. [2, 3], have been published in order to establish a more accurate calculation procedure than the one in [1]. In [4–6], the algorithm for calculating the hot-spot temperature of a directly loaded transformer, using data obtained in a short-circuit heating test, is given. These papers propose improvements in the modelling of thermal processes inside the transformer tank, i.e. in the calculation of the temperature rise of the transformer parts with respect to the temperature of the air surrounding the transformer.

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When a transformer operates in an enclosed space, the temperature of the surrounding air is higher than the outside ambient air temperature. Since the temperatures inside the transformer tank directly depend on the surrounding air temperature, its value must be accurately calculated.

Standards [1] give some approximate recommendations for taking into account the rise of the surrounding air temperature with respect to the outside ambient air temperature. However, exact calculation methods have to be based on an energy balance equation, where the following two components are dominant: (i) heat transfer from the transformer cooling surfaces to the surrounding air and (ii) heat transfer by natural air flow through input and output ventilation holes. The second component, representing the dominant room cooling process, cannot be easily described analytically. The precise analytical expression for the power of heat transfer by natural ventilation cannot be established based on theoretical considerations only. Reference [7] gives the results of steady-state measurements for the front-wall entrance ventilation hole location. The following effects concerning the entrance ventilation hole are analysed: size of ventilation holes; shape of ventilation holes (height and width ratio) at the fixed surface value; the jalousie type; and the vertical position of the entrance ventilation hole. Also, the influence of the location of the exit ventilation holes is investigated. Although in that research a wide range of the real cases is analysed, there is no formula that can be applied to the arbitrary configuration of the room and ventilation holes. A general procedure for obtaining the precise formula is proposed in this paper. The procedure requires measurements of the local temperature values in the room and configuration of the ventilation holes considered. In [8], the effect of wind and rain on the cooling process is investigated.

The complete model, developed in this paper, includes not only the two components quoted in the previous paragraph, but also components of the heat transfer through the room parts. The model can be used for temperature calculation on the arbitrary change of current and outside air temperature. Its use enables: (a) precise design of the ventilation holes for predefined loading conditions, and (b) precise calculation of the temperature for defined ventilation holes: (b1) calculation of a transformer maximal load leading to normal insulation ageing or (b2) calculation of a transformer life sacrifice and estimation of the possibility of loading beyond the rated load.

2
Thermal model

The complete thermal model can be represented by the thermal circuit shown in Fig. 1.

The thermal model, i.e. the equivalent thermal circuit diagram, is based on the energy balances for the following elements (bodies): I windings; II oil, core and tank; III cooling air; IV door; V walls; and VI ceiling. This thermal circuit represents an extension of the widely used thermal circuit with two nodes [4], which is sufficient when the temperature of the air surrounding the transformer (ϑ_{ai}) is known. Rises in temperature of the quoted elements with respect to the outside ambient air temperature (ϑ_{ao}) are denoted by θ_{Cu}, θ_{Oil}, θ_{ai}, θ_{d}, θ_{w} and θ_{c}, respectively; the respective temperature values are indicated by ϑ with the same indexes. The temperature inside each of the bodies is not constant, so the temperatures assigned to the nodes represent the values in the characteristic body spot [4, 5]. Generally, the energy balance for the separate bodies includes the following components: (a) heat generation inside the body; (b) heat transfer from the other bodies; (c) heat transfer to the other bodies; and (d) heat accumulation inside the body. The heat generation sources are the power losses inside the transformer – in the winding (copper) P_1 and in the iron core and auxiliary constructive parts P_2. The heat transfer between bodies is as follows:

1. From winding to oil – conduction through the solid winding insulation and convection from outer solid insulation surface to the oil. This is represented by the heat conductance Λ_1, which is temperature dependent. A generally accepted form of this dependence [1, 4] is

$$\Lambda_1 = a_1(\theta_{Cu} - \theta_{Oil})^{a_2} \ , \tag{1}$$

where a_1 and a_2 are parameters.

2. From oil to surrounding air – convection heat transfer from the transformer tank to the surrounding air predominantly determines this heat transfer component. It is described by the temperature-dependent heat conductance Λ_2:

$$\Lambda_2 = b_1(\theta_{Oil} - \theta_{ai})^{b_2} \ , \tag{2}$$

where b_1 and b_2 are parameters.

3. Air inside the room is cooled by natural ventilation and convection heat transfer from the air to room parts (door, walls and ceiling). The first cooling component, being dominant, is represented by the heat conductance Λ_{vent}. The initial form of this thermal conductance is derived from the Hoppner formula [9], defining the size of the ventilation holes:

$$S = \sqrt{\frac{13.2P_1^2R}{\theta_{aex}{}^3H}} \ , \tag{3}$$

where S is the surface of input, i.e. output ventilation holes (m^2) (it is assumed [9] that they are equal), P_l is the power transfer by natural ventilation (kW), R is the hydraulic resistance to the air circulation, H is the distance from middle high point of the transformer to the lower edge of the ventilation-out hole, and θ_{aex} is the temperature rise of the air on the outlet ventilation holes with respect to the air on the inlet ventilation holes ($\theta_{aex} = \vartheta_{aex} - \vartheta_{ao}$). This formula, solved by P_l, becomes

$$P_l = c\theta_{aex}^{1.5} \ , \tag{4}$$

where c is the constant:

$$c = \sqrt{\frac{H}{13.2R}}S \tag{5}$$

This means that the thermal conductance describing this heat transfer component is

$$\Lambda_{vent} = c\theta_{aex}^{0.5} \ . \tag{6}$$

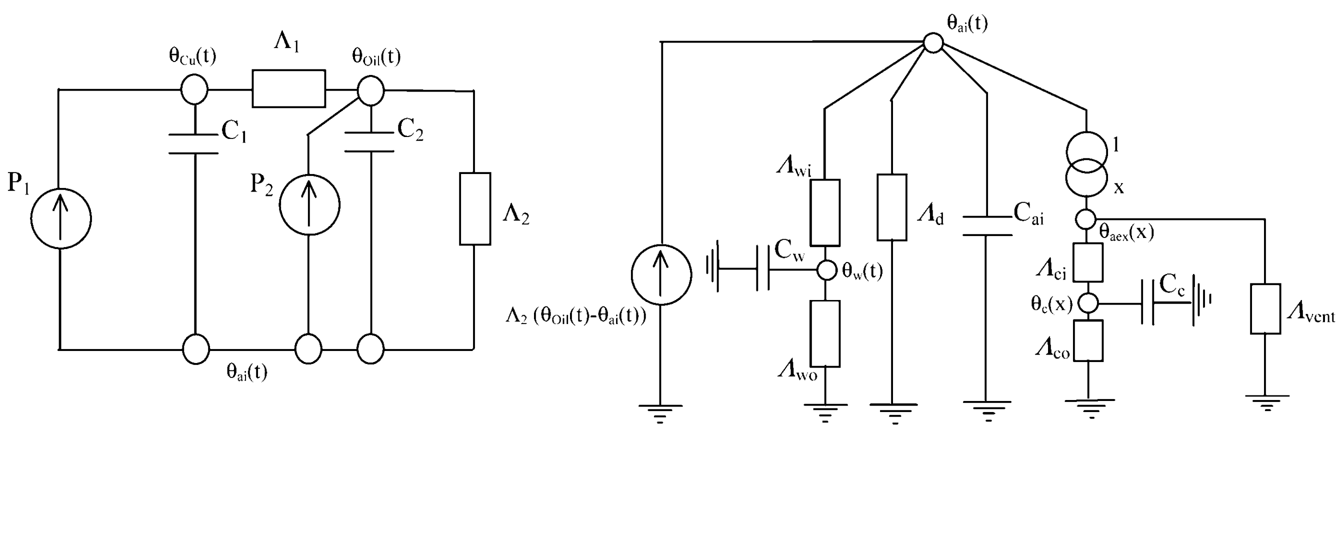
The value of θ_{aex} can be calculated as the product of coefficient x and the rise in cooling air temperature θ_{ai}. Initially, the coefficient x is taken to be equal to 2, since the temperature of the cooling air (mean value of air temperatures below and above the transformer) is assumed to be equal to

$$\vartheta_{ai} = \frac{\vartheta_{ao} + \vartheta_{aex}}{2} = \frac{\vartheta_{ao} + \vartheta_{ao} + \theta_{aex}}{2} = \vartheta_{ao} + \frac{\theta_{aex}}{2} \ . \tag{7}$$

4. The heat transfer through the door (of surface S_d) is represented by a single thermal conductance Λ_d. It can be calculated using the formulae

$$\Lambda_d = 1/[1/(\alpha_1 S_d) + R_{\lambda d} + 1/(\alpha_o S_d)] \ , \tag{8}$$

Fig. 1. Thermal circuit diagram



which includes the convection heat transfer from air inside the room to the inner door surface, thermal conduction through the door (calculated using the well-known expression for a plain wall: $R_{\lambda d} = 1/\lambda_d d_d/S_d$, where λ_d is the thermal conductivity of material, and d_d is the door thickness) and convection heat transfer from the outer door surface to the outside ambient air. The inner heat transfer coefficient α_i depends on the velocity of natural air streaming v. It can be calculated using the formulae [8]

$$\alpha_i = 3.958 + 4.304v \;\; \mathrm{W/m^2K} \;, \tag{9}$$

where the air velocity is equal to

$$v = \sqrt{\frac{gH\theta_{aex}}{273 + \vartheta_{aex}}} \;\; \mathrm{m/s} \;, \tag{10}$$

where $g = 9.81\ \mathrm{m/s^2}$. The outer heat transfer coefficient α_o depends on the wind velocity. The value corresponding to the low wind velocities (0.6 m/s), $\alpha_o = 6.54\ \mathrm{W/(m^2\ K)}$, is adopted.

5. The heat transfer through the walls (of surface S_w) is represented by a standard non-stationary model of a plain wall [*RC–T* scheme of thermal resistances (conductances) and thermal capacitance C_w]. The inner and outer thermal conductances are:

$$\Lambda_{wi} = 1/[1/(\alpha_i S_w) + R_{\lambda w}/2] \tag{11}$$

$$\Lambda_{wo} = 1/[R_{\lambda W}/2 + 1/(\alpha_O S_w)] \tag{12}$$

6. The heat transfer through the ceiling (of surface S_c) is represented in the same manner as that through the walls. The inner and outer thermal conductances are:

$$\Lambda_{ci} = 1/(1/(\alpha_i S_c) + R_{\lambda c}/2) \tag{13}$$

$$\Lambda_{wo} = 1/[R_{\lambda c}/2 + 1/(\alpha_o S_c)] \tag{14}$$

The thermal capacitance is denoted by C_s. Since the surrounding air temperature for the ceiling is approximately equal to the temperature of air on the outlet ventilation holes, the equivalent voltage transformer exists in the thermal circuit diagram (Fig. 1).

The heat accumulation inside the body of windings, the body of oil, core and tank and the body of air inside the room are modelled by thermal capacitances C_1, C_2 and C_{ai}.

The thermal model parameters and their determination on the basis of measurements results will be discussed in Sect. 4.

3 Description of measurements

Measurements were carried out on a typical distribution transformer station 10 kV/0.4 kV, with a 1,000 kVA transformer placed in the prefabricated concrete housing. The total surface area of the entrance ventilation jalousie was 0.95 m^2, while the out ventilation hole was on the roof edge (of the 1.2-m^2 surface). The temperature values were recorded in 15 spots (three spots on the entrance ventilation holes, three spots on the exit ventilation holes, the oil in the pocket, the top and bottom of the radiator, the inner surface of the wall, door and ceiling, and the ambient room temperature at the middle high point of the transformer in three spots); also, the values of the load current were recorded. Each measurement lasted for at least 24 h; the experiments were carried out for a completely free entrance ventilation jalousie and after that for 75%, 50% and 25% of its free surface. The size of the output ventilation hole was not changed and had a constant value of 1.2 m^2. For data recording, a computer-controlled acquisition system was used.

As an example, Fig. 2 shows the time change of the current, and Fig. 3 shows the time change of the temperatures, under normal operation conditions (with a fully opened entrance ventilation hole). The measurements started on 11 February 2000, at 16 h 15 min. The entering air temperature is the average value of the measured temperatures on the entrance ventilation holes, and the ambient air (cooling air) temperature is the average value of three values obtained by sensors at the middle high point of the transformer.

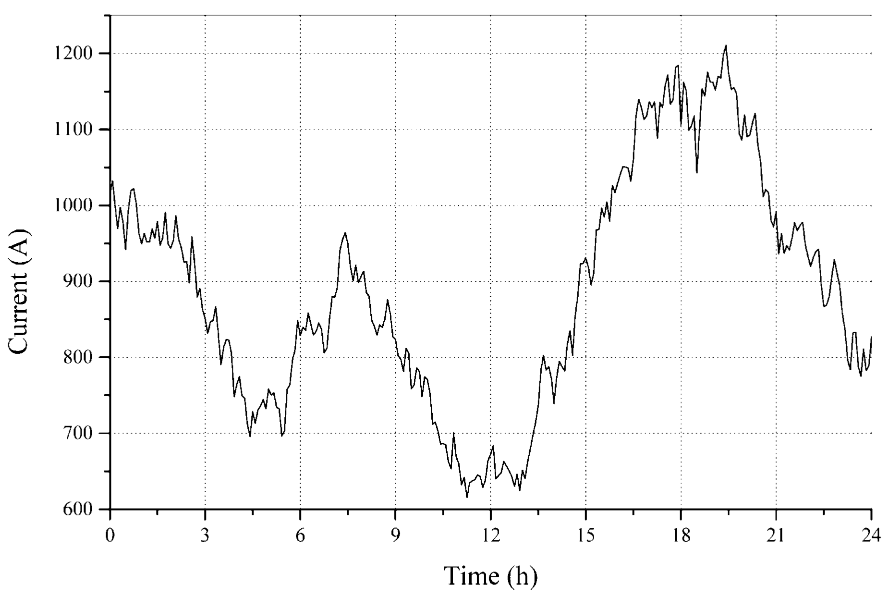


Fig. 2. Daily diagram of measured current load

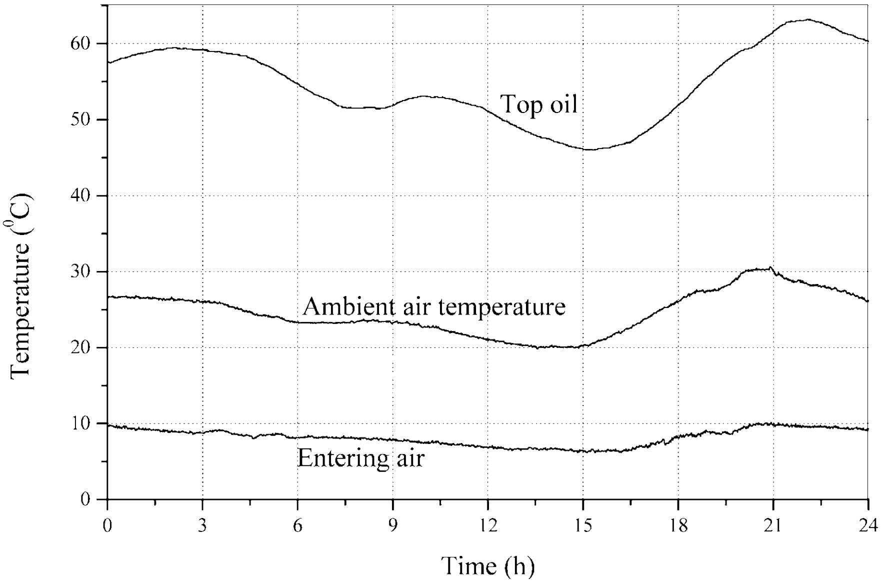


Fig. 3. Daily diagram of measured temperatures

4 Discussion on thermal model parameters

4.1 Parameters of the transformer

The parameters of the transformer may be obtained from short-circuit heating tests [4, 5, 10]. The experiment is usually carried out in spacious locations, where there are no big changes in the air temperature, and the air heated by the cooling surfaces of the transformer streams only upwards. The parameters of thermal conductivity Λ_2 (Fig. 1) are accordingly determined for conditions when the fluid with "undisturbed mass" (for natural air flow, with zero velocity) surrounds the transformer. The use of at least three sensors (placed at a level about halfway up the cooling surfaces) and prevention of rapid variations in the readings are recommended by the Standard [11].

However, when the transformer operates indoors, considerable circular motion of the air occurs, causing the heated air to mix partly with the cold air entering through the entrance ventilation hole instead of exiting through the exit ventilation holes. The "undisturbed mass" of the fluid in this case does not actually exist. Therefore, it is necessary to choose and assign the characteristic air temperature to the node representing the cooling air (for example, average value from three spots around the middle of the radiator, between the radiator and the wall). The thermal conductivity Λ_2 has a new set of parameters, which have to be determined according to the selected characteristic air temperature value. The procedure for their determination is based on the minimization of the integral-square error objective function. The top oil temperature is calculated from a difference equation, based on a differential equation written using the method of node potentials for a node corresponding to θ_{Oil}. The temperature of the top oil in the initial instant is equal to its measured value, whereas it is calculated in the remaining n−1 equidistant instants (measuring period is Δt) from the previously calculated values of the oil temperature and measured room temperature ϑ_{aim} and measured current loads I, using the formula

$$\vartheta_{\mathrm{Oil,i+1}} = \vartheta_{\mathrm{Oil,i}} + \frac{\Delta t}{C_2}\left(P_{\mathrm{Cur}}\left(\frac{I_{\mathrm{i}}}{I_{\mathrm{r}}}\right)^2 + P_{\mathrm{Fer}} - b_1\left(\vartheta_{\mathrm{Oil,i}} - \vartheta_{\mathrm{aim,i}}\right)^{b_2+1}\right) , \quad (15)$$

where P_{Cur} is the rated power loss in the winding (18,628 W) and P_{Fer} is the rated iron power loss (2,250 W). More accurate formulae for the power loss calculation can be used [12], and the transient thermal process in the winding can be taken into account, but the simplified approach adopted has sufficient precision for this application. The criteria function which should be minimized is

$$f = \sum_{i=2}^{n}\left(\vartheta_{\mathrm{Oil,i}} - \vartheta_{\mathrm{Oilm,i}}\right)^2 , \quad (16)$$

where $\vartheta_{\mathrm{Oilm}}$ are measured values of the top oil temperature. The Nelder-Mid simplex method [available as a MATLAB Version 5.3.1 (1999) with SIMULINK Version 3.0.1 (The MathWorks, Inc., MA, USA) system function] can be used to determine the optimal values of b_1, b_2 and C_2. The measured values of current, temperature of top oil and ambient temperature are used as the input data. Note that the thermal capacity C_2 cannot be calculated as the product of the specific heat and the mass because of the non-homogeneous temperature distribution.

The winding heat phenomena are treated in a simplified manner, since it was not convenient to obtain the data necessary for its exact modelling. The calculation procedure and the required thermal parameters are taken from the standard [1]. In addition, a winding thermal time constant of 5 min is adopted. In practice, this influences only the copper temperature, as the influence on the oil and the air inside the room is negligible.

4.2
Ratio of air temperatures on output ventilation holes and in the room

The rise of air temperature on the outlet ventilation holes θ_{aex} is obtained by multiplying the value of the rise of cooling air temperature θ_{ai} by a coefficient x. The value of x is adopted to achieve the minimal deviation between values $\theta_{aex}(t)$ and $x\theta_{ai}(t)$. Temperatures $\theta_{ai}(t)$ and $\theta_{aex}(t)$ represent average values of the measured temperatures. Each of the temperatures is measured at three points.

4.3
Convection heat transfer coefficients

Basically, Eqs. (9) and (10) are used to determinate the heat transfer coefficients by the natural convection to the inner surfaces of the room compartments, and from the external surfaces, the value of 6.54 W/(m^2 K) is used. Some corrections of these values are done using the best-fit criteria of the calculated and measured inner surface temperatures of the room compartments. These corrections are done also bearing in mind the position of the transformer station (for example, the position of one wall causes a decreased air flow over its outer surface). It should be noted that these parameters do not affect the temperature of the cooling air to a great extent (see Fig. 7), and for that reason do not significantly affect the temperature inside the transformer. Therefore, the exact determination of these parameters has no great significance.

4.4
Parameters of heat transfer by natural ventilation

As mentioned in the Introduction, it is necessary to carry out the measurements in an enclosed space. Based on the results of measurements, the precise functional dependence of the heat conductivity Λ_{vent} can be established. The functional dependence obtained can then be used for the calculation of the temperature at arbitrary current load and ambient temperature changes, for any transformer station of the same type.

Based on the measurements described in Sect. 3, the functional dependence of the heat conductivity Λ_{vent} is obtained. The influence of the entrance ventilation jalousie size is described analytically. First, Eq. (6) was used to calculate the thermal conductivity Λ_{vent}, and the value of coefficient c was adjusted for each measurement in order to get the values of θ_{ai} obtained by the simulation as close as possible to the measured ones. After that, the validity of Eq. (5) was checked, i.e. the correlation between coefficient c and the surface of the entrance ventilation hole S was analysed. It should be noted that two parameters from Eq. (5), namely H and R, have approximately constant values, i.e. depending slightly on ventilation hole S.

5
Results obtained from experiments

Based on the completed experiments, the values of all the parameters described are adjusted. The analytical expression for coefficient c is derived. The proposed thermal model is validated by comparison with experimental results, and is also compared with the model from the standard [1].

5.1
Values of the model parameters

The parameters of thermal conductivity Λ_2, obtained from measurements with the entrance ventilation jalousie opened 50%, are: $b_1 = 22.737$, $b_2 = 0.76277$ and $C_2 = 935.12$ W h/K.

The thermal capacity C_{ai} is small and could therefore be neglected without significant loss of calculation precision. The value C_{ai} is adopted in an approximate manner as the product of specific heat and mass of air inside the room ($C_{ai} = 3.82$ W h/K).

The values of coefficient x for each state of entrance ventilation jalousie are given in Table 1.

Coefficients α from the air inside the room to the inner surfaces of the room compartments were based on Eqs. (9) and (10). Coefficient α was doubled as the correction for the walls. Coefficients α from the external surfaces to the outside air are estimated in the following way: wall, 5 W/(m^2 K); door, 6.54 W/(m^2 K); ceiling, 10 W/(m^2 K). The groups of values (surface, thickness, thermal conductivity, specific heat) for the walls, ceiling and door are respectively: [11.4 m^2, 5 cm, 1.1 W/(m K), 1.4 kJ/(kg K)], [4.88 m^2, 5 mm, 0.116 W/(m K), 0.837 kJ/(kg K)], [1.15 m^2, 3 mm, 231 W/(m K), –].

The values of coefficient c, determined for each of the four states of the entrance jalousie, are shown in Fig. 4 (bold curve). The dependence of c from surface S clearly shows the deviation from linearity (as presumed in the Hopner formula); it approximately fits the exponential function:

$$c = k_1\left(1 - e^{-k_2 S_{rel}}\right) \quad , \tag{17}$$

where S_{rel} is the value of surface expressed relative to the complete surface of 0.95 m^2. The minimal sum of square deviations of this function from the four spots defined based on measurements is attained from the value of coefficients $k_1 = 64.899$ and $k_2 = 5.37$.

This result leads to the conclusion that the surface of the entrance ventilation hole can be smaller than the surface of the exit ventilation hole (which in this case is 1.2 m^2); the same can be concluded observing function $cx^{1.5}$, which is of interest if we calculate the power transfer by natural ventilation via the temperature rise θ_{ai}. In other words, the increase of entrance ventilation hole has a smaller effect as its size approaches the size of the exit ventilation hole. The validity of this conclusion can be assessed by repeating the described measurements and calculation procedure on other transformer stations.

Table 1. Optimal coefficients x depending on opening state of the entrance jalousie

	Totally opened jalousie	Jalousie opened 75%	Jalousie opened 50%	Jalousie opened 25%
Coefficient x	1.58	1.4	1.32	1.28

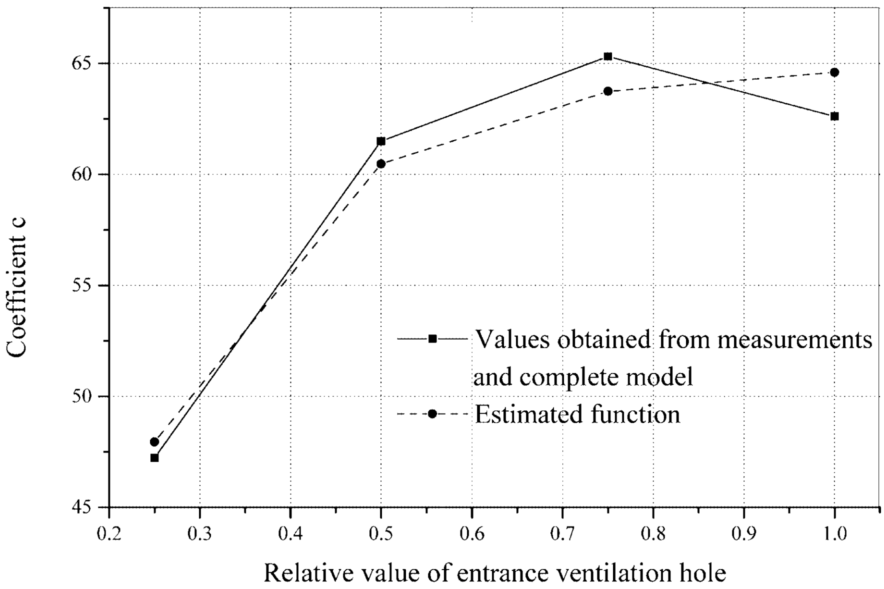


Fig. 4. Functional dependence for coefficient of natural ventilation

5.2 Model accuracy

A review of the model accuracy is given in Table 2. It contains the limit values of deviations from the measured temperatures for all measurements undertaken. In Figs. 5 and 6, the calculated and measured temperatures are shown, as well as their differences, for top oil (Fig. 5) and cooling air (Fig. 6), all for a fully opened entrance ventilation jalousie.

The accuracy of the temperature calculations achieved can be considered satisfactory, especially for the temperatures relating to transformer loading – top oil and cooling air. The actual maximal deviations between the result obtained with the model and the measurement results are lower than this in Table 2, because the values given in Table 2 include the measurement noise as well. The deviations of the temperatures of the inner surface room parts are higher, primarily because the effects of sun and wind are not considered. The effect of these factors can be included in the model in a relatively simple way, if the values of sun radiation intensity and direction and wind velocity are known. Detailed and precise thermal modelling of room parts was not done because the heat exchange through these is considerably smaller than the heat exchange by natural ventilation. As an example, the time change of these two cooling components for the situation with the totally opened entrance jalousie is shown in Fig. 7.

Besides the strong effect on the value of outer heat transfer coefficient (α_o), the wind also affects the heat transfer by natural ventilation. The transformer losses were calculated in a simplified way, which also introduces some error in the calculation of temperatures. In addition, the model of heat transfer from the transformer cooling surface (radiators) to the surrounding air is not ideal.

Table 2. Range of discrepancies (minimal negative and maximal positive values, expressed in K) between calculated and measured temperatures

	Totally opened jalousie	Jalousie opened 75%	Jalousie opened 50%	Jalousie opened 25%
Cooling air	−1.13	−1.83	−2.10	−1.28
	1.02	3.71	2.71	2.69
Top oil	−2.3	−1.01	−2.16	−1.15
	0.47	2.24	1.59	1.9
Wall	−0.98	−0.98	−1.61	−0.58
	0.09	4.15	3.36	7.43
Door	−1.64	1.17	−0.61	0.58
	−0.11	5.01	2.84	5.08
Ceiling	−4.42	−6.35	−3.9	−6.05
	0.8	4.37	2.51	4.34

5.3 Comparison of the results obtained by the proposed model and by the procedure in the Standard

Using the procedure from the International Standard [1], the temperature rise of the top oil with respect to the outside ambient air temperature (θ_{Oil}) is calculated as the product of value θ_{Oil}^* that would be achieved if the transformer were in free space and correction factor (k). This correction factor can be calculated by the formula

$$k = 1 + \frac{\theta_{ai,r}}{\theta_{Oil,r} - \theta_{ai,r}} \quad . \tag{18}$$

where $\theta_{Oil,r}$ is the temperature rise of the top oil with respect to the outside ambient air temperature, and $\theta_{ai,r}$ is the temperature rise of the cooling air with respect to the outside ambient air temperature, both values in the steady state under rated load. The value $\theta_{Oil,r} - \theta_{ai,r}$ is taken from the short-circuit heating test, done at the factory. For the value $\theta_{ai,r}$, certain rough characteristic values are given in [1], but it is recommended that the value should be determined experimentally for the individual case of transformer station configuration. It will be shown that the temperature rise $\theta_{ai,r}$ in the case of the transformer station analysed has a high value compared with the characteristic values given in [1], because the 1,000-kVA transformer is placed in a room which is designed for the 630-kVA transformer.

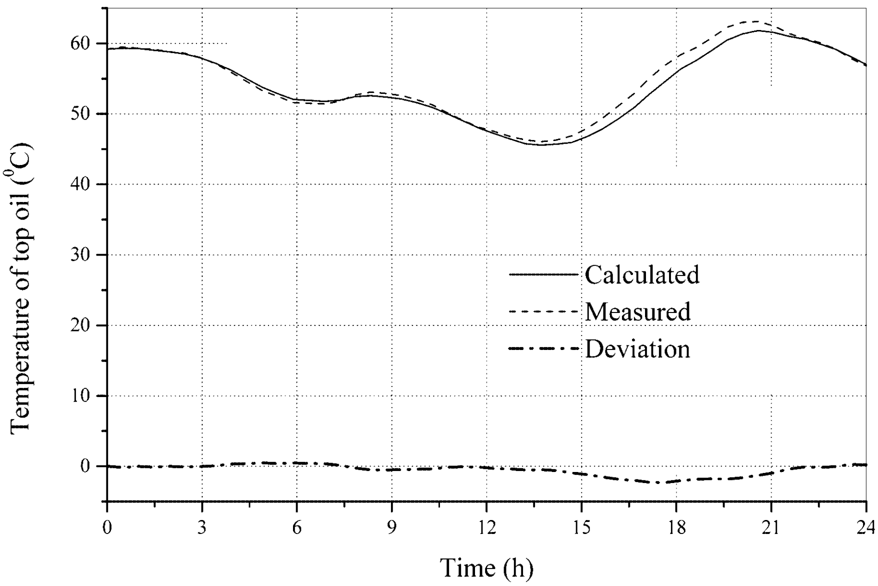


Fig. 5. Top oil temperature, totally opened jalousie

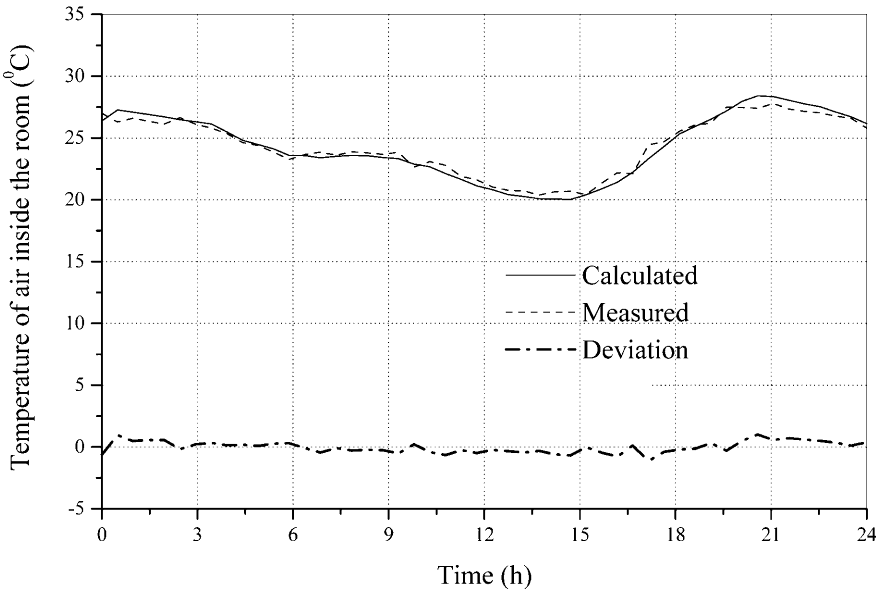


Fig. 6. Cooling air temperature, totally opened jalousie

The following procedure is applied. Based on the model proposed in this paper, the simulation of the heating process existing in the indoor transformer stations, starting from the cold state [θ_{Oil} $(t = 0) = 0$, θ_{ai} $(t = 0) = 0$], at constant rated load and external ambient temperature of 20 °C, is carried out. In this way, the values of $\theta_{\text{Oil,r}} = 76.4$ K and $\theta_{\text{ai,r}} = 28.4$ K are obtained as a result of simulations up to the temperature steady state, and on their basis the coefficient k: $k = 1 + 28.4/(76.4 - 28.4) = 1.592$. A simulation was done for the case of the totally opened entrance jalousie. Then, using the procedure from [1], the calculation of the top oil temperature is done for the daily load diagram shown in Fig. 2 and the daily outside ambient air temperature diagram shown in Fig. 3. Compared with the original algorithm [1], the influence of the non-linearity of heat transfer from the oil to the air is treated more precisely [4, 5]. In this way, the top oil temperature calculation error is reduced to that caused by the inaccurate calculation of the room temperature; the result is shown in Fig. 8.

The deviation in the calculated top oil temperatures based on the method given in [1] from the measured ones is in the range [0, 3.48] K, which means that the calculated temperatures are somewhat higher. The error range if the proposed new model is applied is [−2.3, 0.47] K, which means that the procedure in the standard has quite satisfactory accuracy. Nevertheless, it is important to notice that when we apply the procedure from [1], it is necessary to determine the value of the temperature rise $\theta_{\text{ai,r}}$ experimentally, especially in the cases when the cooling conditions are more difficult, meaning that the ventilation holes are not adequately positioned and sized.

6
Conclusions

The paper presents a complete thermal model of indoor transformer stations. The model comprises all the basic components of the thermal process, describing their non-stationary and non-linear characteristics. The model pa-

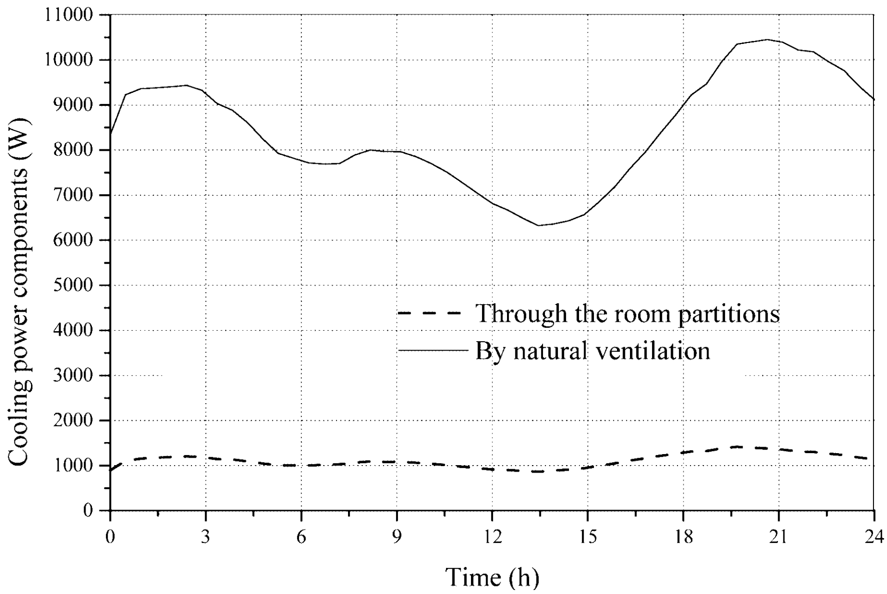


Fig. 7. Cooling power components, totally opened jalousie

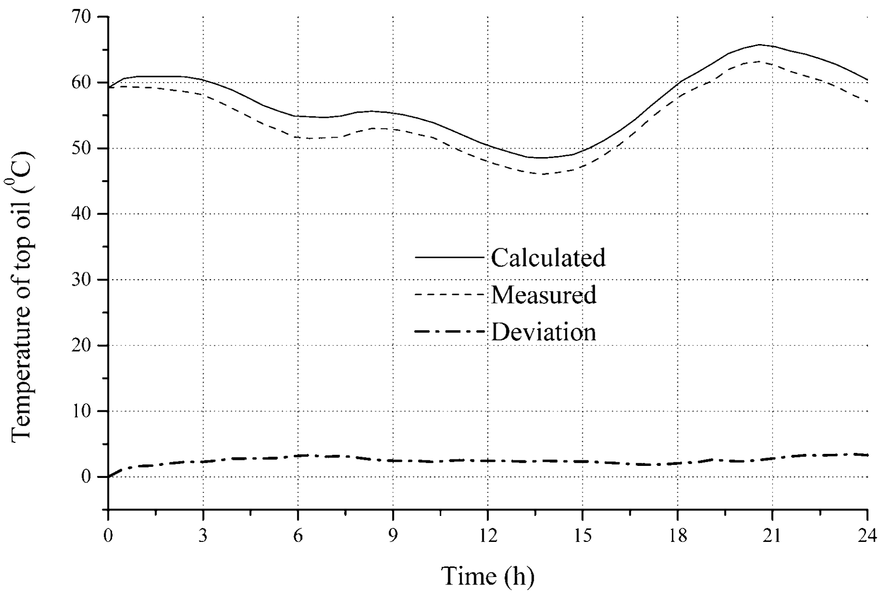


Fig. 8. Top oil temperature, totally opened jalousie, model from IEC Standard

rameters are discussed, and the experimental procedure for their determination is explained. The experimental base of the work is carried out on one typical 10 kV/0.4 kV distribution transformer station placed in prefabricated concrete housing. Based on these measurements, the analytical expression for the power of heat transfer by natural ventilation, which is the component that is the hardest to describe mathematically, is established. The formula can be applied to the transformer station considered, where the size of the input ventilation jalousie can be altered. Also, the measurements provided enabled the verification of the complete model. Repetition of measurements and calculations on other types of indoor transformer stations could lead to a satisfactorily accurate general formula for the natural air ventilation cooling.

Using the developed model, precise evaluation of additional temperature rises, insulation ageing and decrease in transformer loading capability due to the indoor (compared with the outdoor) operation can be analysed.

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