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Dynamic thermal model of kiosk oil immersed transformers based on the thermal buoyancy driven air flow

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ABSTRACT

This paper presents a dynamic thermal model of an indoor transformer station – transformer, high voltage and low voltage compartments are placed in a kiosk with inlet and outlet ventilation openings. The well-known dynamic thermal model of an oil immersed power transformer is extended with thermal models of walls and ceiling of the kiosk and natural air ventilation through the ventilation holes. The influence of wind velocity and the direction to convective heat transfer on each of the outer surfaces is taken into account. The solar calculator is developed and applied to determine the sun irradiation on each of the kiosk surfaces, taking into account shadows on some of the walls. The natural ventilation is modeled using equilibrium of pressure produced due to thermal buoyancy and pressure drop on the air path. The model is validated by comparing calculation results with the results of measurements on the transformer kiosk, with different surfaces of ventilation inlet and outlet openings. The model can be used in design phase to optimize ventilation openings (jalousies). Another application of the model is the estimation of the maximum possible load in forecasted ambient conditions, possible being applied in scope of smart grid concept.

1. Introduction

The hottest spot of the solid insulation (hot-spot) is the major practical benchmark of transformer loading. It influences long-term insulation ageing and it is the measure of the risk of irreversible insulation faults if the hot-spot temperature exceeds certain value. The hot-spot temperature depends on the load (the current) and on the temperature of an outer cooling medium. In the majority of the cases with oil as the inner cooling medium and air as the outer cooling medium, there is no difference between ambient temperature and the temperature of air cooling outer surfaces of the transformer. Consequently, the thermal models in literature focus on the dynamic heat transfer from the active parts (the windings and the core) to the oil and from the oil, over cooling surfaces (the radiators or the compact coolers) to ambient air. There are standards [1,2] which deal with this topic. We also published the paper [3], proposing an alternative to the approach in [1] for solving overshoot hot-spot to top pocket oil temperature difference.

In cases for indoor transformers, the temperature of air cooling the transformer is higher than outside air temperature. That is why the thermal model should be extended with the heat transfer in the room where the transformer is positioned.

This item is addressed in literature and standards (for example in [1,4]), where the classification of the type of the enclosure and rough recommendations for correction of top oil temperature due to the heating of the enclosure are given. In [1] the correction for increase in ambient temperature due to the enclosure is given for different types of the enclosure depending on the transformer rated power and the number of transformers in the enclosure.

This paper offers the dynamic thermal model of the indoor oil immersed power transformer and the enclosure. In our previous publication [5], we presented such kind of the thermal model. In [5] a matter of concern was the prefabricated concrete enclosure, typically used in Power Distribution Company, Belgrade, Serbia, while in this paper it is the kiosk substation with the transformer inside. From the perspective of the thermal model, these two constructions are very similar. The model presented in this paper is more physically based than the model from [5]: heat transfer due to air mass transfer (natural ventilation) is based on theoretical (physical) equilibrium of produced pressure due to thermal buoyancy and pressure drop on the air path, while in [5] it is based

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on the equation for thermal conductance with the parameters determined empirically.

2. Air flow and the air temperature

Fig. 1 presents qualitatively the distribution of air flow and its temperature inside the kiosk. For the qualitative explanation, one inlet opening and two outlet openings (types B and D) are analyzed.

Due to a small component of air flow (Q_{w2}), flowing downwards near the tank and mixing with the outside air (of temperature ϑ_{ao}) entering through the inlet opening (Q_{ao}), the temperature of the mixed air (ϑ_{aom}) is somewhat higher than ϑ_{ao}. The air heats up in the radiators and goes mostly upwards to the ceiling (ϑ_{ai}), where it slightly cools down. Further cooling down of the air happens on the vertical walls ($\vartheta_{aD} < \vartheta_{ai}$, $\vartheta_{aB} < \vartheta_{ai}$); it is not necessarily that $\vartheta_{aB} < \vartheta_{aD}$, since there may be a direct air flow from the ceiling to the opening B. In this model, the approximation $\vartheta_{aB} = \vartheta_{aD} = \vartheta_{ai}$ was implemented.

In the model no difference of the temperatures ϑ_{aD} and ϑ_{aB} is considered. Also, Q_{w2} is not calculated – instead, it is assumed that 90% of heat transferred from the radiators is transferred outside (the air of flow Q_{aD} and temperature ϑ_{aD}, i.e. the air of flow Q_{aB} and temperature ϑ_{aB}), while 10% goes back and mixes with the air entering the kiosk through the inlet openings (the air of flow Q_{ao} and temperature ϑ_{ao}, mixes with the air of flow Q_{w2} and temperature ϑ_{wj}); flow of the mixed air is Q_r and temperature ϑ_{aom}. The initial value 90% was adopted in an arbitrary way and later tuned to achieve a good agreement of the calculated and the measured values of ϑ_{aom}.

Small part of air can also flow directly from the ceiling downwards to the free space in the kiosk. In terms of total energy and flow balance, this component is small, but the temperature of this air flow component (ϑ_{room}) represents the value which is measured by the temperature sensor in the inner space of the kiosk.

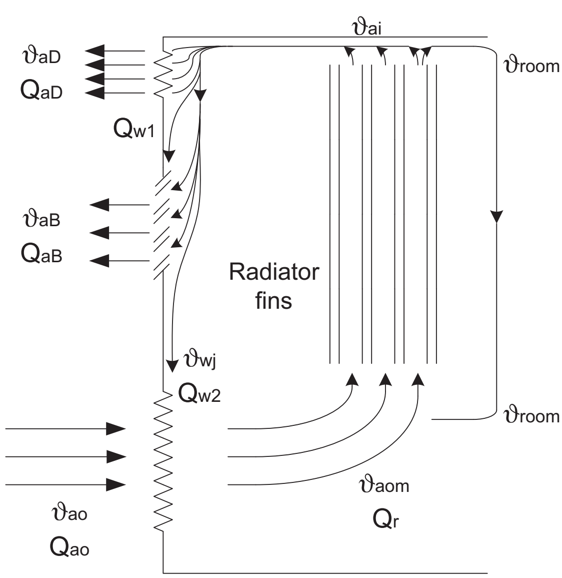

Fig. 1. The distribution of air flow in realistic case of more outlet openings. ϑ_{ao} – Ambient temperature outside the kiosk; ϑ_{aom} – Air temperature inside the kiosk near inlet opening; ϑ_{room} – Air temperature at the position of temperature sensor in the inner space of the kiosk; ϑ_{ai} – Air temperature exiting the radiators; ϑ_{aD} – Temperature of the air exiting the kiosk through opening of type D; ϑ_{aB} – Temperature of the air exiting the kiosk through opening of type B; ϑ_{wj} – Temperature of air near the wall at the position of inlet opening; Q_{ao} – Air flow rate through the inlet opening (type C); Q_r – Air flow rate through the transformer radiator; Q_{aD} – Air flow rate through the outlet opening of type D; Q_{w1} – Air flow rate near the wall between the outlet openings of type D and B; Q_{aB} – Air flow rate through the outlet opening of type B; Q_{w2} – Air flow rate near the wall just above the inlet opening (type C).

3. Buoyancy model

The natural air circulation is due to the thermal driving force (buoyancy), which appears as the consequence of air density change, decreasing with increase of air temperature.

If air flows in a closed flow loop, buoyancy is defined as the integral of density variations along the loop:

$$\Delta p_B = \oint (\rho - \rho_0) g_x dx \tag{1}$$

Using the Boussinesq approximation for the temperature dependence of density, previous equation can be written as follows:

$$\Delta p_B = \oint \rho_0 \beta (\vartheta_0 - \vartheta) g_x dx \tag{2}$$

ρ_0 – Air density at reference temperature (kg/m^3); 1.292 kg/m^3 at 0 °C
β – Volume expansion coefficient of air (1/°C); 0.003628196–9.866374 $\cdot 10^{-6} \cdot \vartheta_a$ at ϑ_a
g_x – Gravity (9.81 m/s^2)

Total flow rate in a flow circuit is obtained at the point where thermal buoyancy equalizes with the pressure drop on elements in the closed air flow loop:

$$\Delta p_D = \Delta p_B \tag{3}$$

Both buoyancy and pressure drop depend on air flow. In steady state the air flow (Q_r) attains a final value due to equilibrium of buoyancy and total pressure drop. Details of model implementation are explained at the end of Section 7.

Simplified diagram of air density change is presented in Fig. 2. It is assumed that air enters the kiosk through the inlet openings and temperature of air at the bottom of the fins is the same as that of the outside air (ambient) temperature. After this air splits equally between the transformer fins, where it is heated up and lifted up to the ceiling. It should be mentioned that velocity boundary layer is very small (much smaller than the distance between the fins), i.e. the air moves vertically, in the small space near the radiator surface – this is valid for both the solution with fins, being perpendicular to the transformer tank, and for the radiator with plates, positioned parallel with the transformer tank. It is supposed that

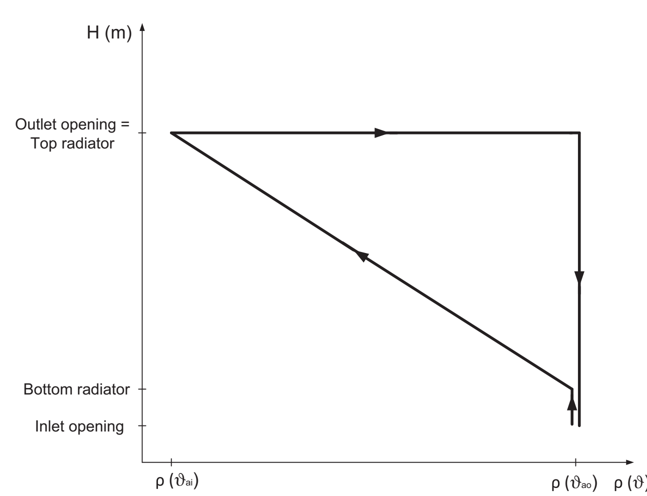

Fig. 2. The simplest model of change of air density along the loop.