Proceedings of the 15th International Heat Transfer Conference, IHTC-15
August 10-15, 2014, Kyoto, Japan

IHTC15-9246

# CFD STUDY OF NON-GUIDED LAMINAR MIXED CONVECTION OF A HIGH PRANDTL NUMBER FLUID IN A TRANSFORMER WINDING-LIKE GEOMETRY

**Tor Laneryd[1*], Jurjen Kranenborg[2], Yuhe Jiao[2], Andreas Gustafsson[2]**

[1]ABB Corporate Research, Västerås, Sweden
[2]ABB Transformers, Ludvika, Sweden

## ABSTRACT

Computational fluid dynamics is used to study the unsteady laminar mixed convection of a high Prandtl number fluid around a heat-generating finite cylindrical shell oriented with its axis in the vertical direction, with axial cooling ducts both inside and outside the shell, and with radial cooling ducts interspersed at regular intervals throughout the shell. This geometry is relevant to applications in large oil-filled power transformers. The incompressible Navier-Stokes system of equations with the Boussinesq formulation of buoyancy is used to obtain temperature and flow field for typical design parameters and operating conditions. The flow field is characterized by a zigzag pattern similar to the one obtained by blocking the inner and outer cooling ducts at selected locations with fluid guides, but becomes unsteady in some cases, with the exact number of reversals of the flow direction depending strongly on the mass flow rate of the system. The average magnitude and the location of the hotspot temperature are significantly different from the fluid-guided case and the non-guided case generally shows less efficient cooling. As a consequence it requires much more effort to predict the position and level of the hotspot temperature (the key limiting design parameter) for non-guided transformer windings as compared to their guided counterparts.

**KEY WORDS:** Numerical simulation and super-computing, Convection, Thermal Management, Power transformer, Hot spot temperature.

## 1. INTRODUCTION: TRANSFORMER COOLING

**1.1 Power transformers** A power transformer is an important component in an electrical power transmission and distribution network, where it serves to step up and step down the voltage levels in order to minimize the mostly current-induced losses in the transmission lines. The transformer is based on the principles of induction, in essence consisting of a steel core with a number of winding coils wound around it, so that energy is transferred from one winding to the other whilst the voltage level is changed. Due to the high voltage levels that are commonly used for large-area networks, the transformer is filled with an insulation fluid like mineral oil, characterised by its high Prandtl number.

The insulation of the current-carrying windings and other metallic parts is further enhanced through oil-impregnated paper wound around the winding conductors and cellulose-based pressboard in between the windings and between other components. These cellulose-based insulation materials have proven to allow for effective and economical insulation solutions for the multiple decade lifespan expected of a transformer, but it assumes that the unavoidable ageing processes due to paper and oil chemistry can be kept under control. Since these ageing process rapidly accelerate with temperature, the thermal design of a power transformer in general and the windings in particular is very important. In this context the winding hotspot temperature (i.e. the hottest temperature that occurs in the insulation paper wrapped around the winding conductors), average

*Corresponding Author: tor.laneryd@se.abb.com

winding temperature and tank oil temperatures are important design parameters and strictly limited by maximum values as defined through international standards [1]. Figure 1 gives a geometrical impression of the main components of a transformer, showing in detail the cross-section of the windings whose thermal behaviour is the subject of study in this article.

**1.2 Cooling modes** Although the technical operation principle of a power transformer has not changed during the last era, a power transformer has evolved into an extremely optimized product where losses are typically less than 0.3% of the transmitted power level. However, given the large power level of today's transformers (exceeding 1000 MW) the amount of heat that is generated in the transformer is significant and therefore the transformer needs to be cooled effectively. There are several ways to cool a transformer (each of which may be optimal for a given customer specification), which can be classified as follows:

- Oil Natural (ON): Oil flows through both the winding and the external heat exchanger through natural convection. No pumps are applied.
- Oil Forced (OF): Oil flows through the winding by natural convection and through the external heat exchanger through forced convection, where the cooled oil is pumped into the tank.
- Oil Directed (OD): Oil flows through the winding by forced convection through a piping system connecting the coolers and the pump to the active part

For the discussion in this article which focuses on the thermal behaviour of the windings only, it is sufficient to state that the above cooling modes can be associated with lower, medium and higher oil flow rates through the windings. The resulting oil flow rate ranges were checked to correspond with realistic values that can be expected in power transformer designs.

**1.3 Winding thermal design** In practice both non-guided and guided windings are used in winding thermal design. Figure 2 presents a cross section of the winding structure with the two winding variants present and which also forms the basis for the geometry used in the CFD simulations. The geometry consists of a number of discs (in which the copper conductors are present) which in case of a high-voltage winding may be covered with paper insulation material. The discs are separated by horizontal ducts (for electrical insulation as well as to allow a cooling fluid to flow) and have two main vertical ducts at their sides as well. When the width of the winding disc is relatively small the non-guided winding may be used to advantage for manufacturing since fluid guides do not need to be placed (and sometimes their supporting insulation collar structure can be absent as well) which leads to an economical solution regarding materials cost and manufacturing effort. However, when the winding disc width becomes much larger than its height it is advantageous to try to maximize the convective heat transport along the dominating surface areas of the discs bounding the horizontal ducts. For that reason fluid guides are often placed in an alternating pattern in the main vertical ducts to the sides of the discs, such that the winding gets divided in a number of passes where the oil flow direction has an enforced direction, with the flow direction alternating from one pass to the other, as shown in the picture of Figure 2.

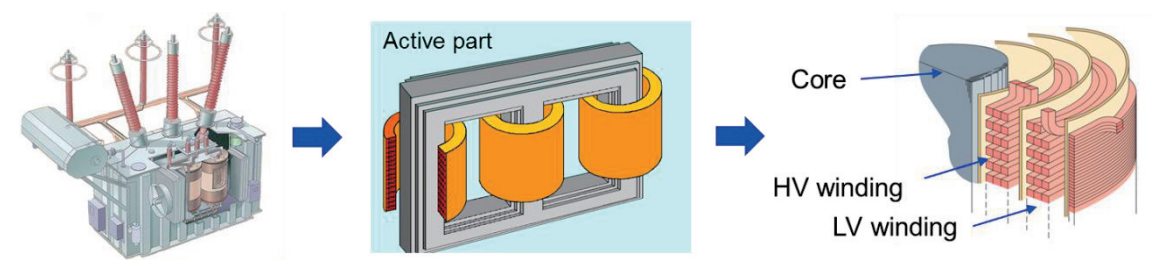


**Fig. 1** HV and LV (High / Low Voltage) windings as part of a power transformer

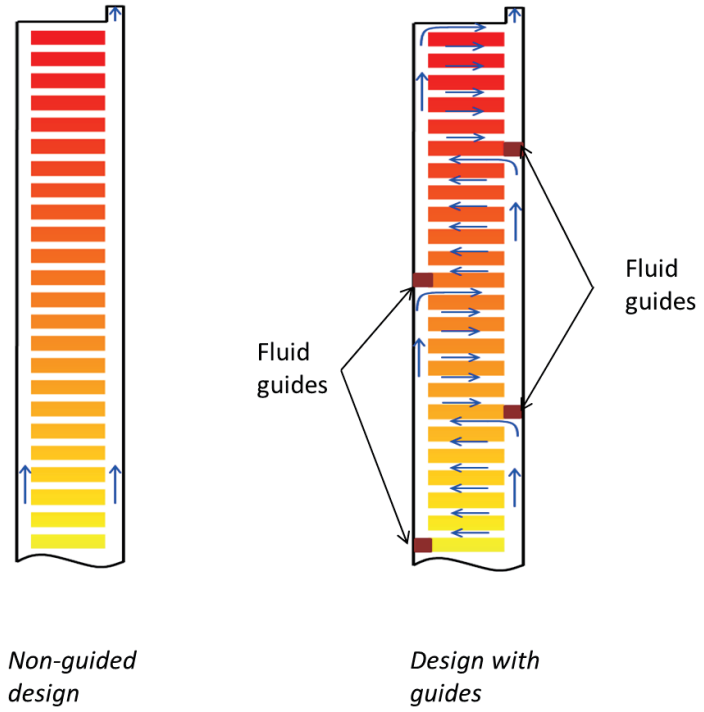


**Fig. 2** Two-dimensional geometries of disc windings (vertical cross-cuts of the windings shown in Figure 1) as used in the model study for non-guided (left) and guided (right) windings.

**1.4 Available literature on CFD for transformer windings** There is a limited number of studies published on CFD applications for transformer windings, and they mainly focus on windings with fluid guides. The first CFD-based study of the thermal performance of transformer windings was published by Kranenborg et al. [2] in which it was shown that due to the high Prandtl number of the insulation oil hot streaks occur that may affect the position of the winding hotspot temperature. Furthermore it was shown that internal buoyancy (i.e., temperature-induced oil density differences inside the windings) plays a major role in determining the actual oil flow pattern and the position and strength of the hotspot temperature for all relevant cooling modes. A detailed study by Torriano et al. [3] performed detailed numerical investigations on how the numerical models, mass flow rate and inlet temperature profile affect the temperature distribution of a 2D oil-guided transformer winding, also confirming the effect of hot streaks on the hotspot temperature. Later this study was extended [4] with simulation for a 3D disc winding revealing differences with 2D winding models which can be corrected for in the 2D model by proper compensation of mass flow rate and wetted area. Finally, Skillen et al. [5] performed CFD calculations to predict the winding temperature distribution using a 2D oil-guided disc winding model where it was found that hot streaks can be present in the oil in both horizontal and vertical ducts, and they found that these streaks affected the temperature distribution in the passes between fluid guides above. Chen et al. [6] compared three winding types for a given transformer specification and concluded that the configuration with fluid guides provided for the best cooling. Finally, a study by Jiao [7] focused specifically on non-guided windings and its thermal behavior as a function of design parameters, revealing a large range of features and a behavior that was more diverse than that for oil guided windings. In particular the fluid mass flow rate was shown to have a significant impact on both average winding and hotspot temperatures.

## 2. WINDING CFD MODEL

**2.1 Geometry and characteristic parameters** A geometry is considered which consists of a finite cylindrical shell, positioned with its axis in the vertical direction. The shell is generating heat with a constant loss

density, and it has axial cooling ducts both inside and outside the shell where a cooling fluid is driven upwards through buoyancy. The shell is confined between adiabatic vertical cylinders that determine the width of the axial cooling ducts, and located in a tank filled with the cooling fluid. The tank temperature may be different than the temperature of the fluid entering the axial cooling ducts. Radial cooling ducts are interspersed at regular intervals throughout the extent of the shell, where the cooling fluid is also able to flow. The solid region between two horizontal cooling ducts is denoted as a disc. Solid obstacles located in the axial cooling ducts, denoted as guides, which obstruct the flow and guides it into the radial cooling ducts, may or may not be present in the geometry, although the main purpose of the study is to investigate the absence of fluid guides. This geometry is an ideal representation of a power transformer winding being cooled by insulation oil, and the geometrical and physical parameters (given in Table 1) are chosen accordingly.

**Table 1** Characteristic geometrical and physical parameters

| Quantity | Description | Value |
|---|---|---|
| $D$ | shell inner diameter | 1000 mm |
| $N_{\text{disc}}$ | number of discs in shell | 100 |
| $W_{\text{disc}}$ | disc width | 50 mm |
| $H_{\text{disc}}$ | disc height | 10 mm |
| $W_{\text{duct}}$ | vertical duct width | 5 mm |
| $H_{\text{duct}}$ | horizontal duct height | 5 mm |
| $\dot{S}$ | loss density in shell | 200 kW/m³ |

The cooling fluid and the solid material of the guides and the discs are assumed to have constant material properties (given in Table 2), in order to be able to compare the effect the characteristics of the system for different operating conditions.

**Table 2** Material parameters

| Quantity | Description | Fluid | Disc | Guide |
|---|---|---|---|---|
| $\rho$ | density | 900 kg/m³ | 5000 kg/m³ | 1000 kg/m³ |
| $\mu$ | viscosity | 0.004 kg/m·s | -- | -- |
| $c_p$ | heat capacity | 2000 kg/J·K | 400 kg/J·K | 1000 kg/J·K |
| $k$ | thermal conductivity | 0.1 W/m·K | $\begin{cases} k_r = 200\ \text{W/m·K} \\ k_z = 1\ \text{W/m·K} \end{cases}$ | 1 W/m·K |
| $\beta$ | expansion coefficient | 0.001 K$^{-1}$ | -- | -- |

**2.2 Governing Equations and boundary conditions** The appropriate governing equations are the Navier-Stokes equation for incompressible flow [8] with the Boussinesq formulation of buoyancy.

$$\frac{\partial}{\partial t}\rho + \nabla\cdot(\rho\boldsymbol{u}) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho\boldsymbol{u}) + \rho\boldsymbol{u}\cdot\nabla\boldsymbol{u} = -\nabla p + \nabla\cdot\tau + \rho g\beta(T - T_{\text{ref}}) \tag{2}$$

$$\frac{\partial}{\partial t}\left(\rho c_p T\right) + \nabla\cdot\left(\rho\boldsymbol{u}c_p T\right) = \nabla\cdot(\boldsymbol{k}\nabla T) \tag{3}$$

The constant density $\rho$ for the reference state is defined as the inlet temperature, i.e. temperature conditions for the fluid at the bottom of the tank.

For cases with natural convection through the shell (N and F cooling modes as defined in Table 4), a pressure gradient is imposed over the boundary so that

$$\Delta p = (1 - \beta\, \Delta T_{\text{tank}})\rho_{\text{ref}}\, gH \tag{4}$$

In equation (4), $\Delta T_{\text{tank}}$ signifies the temperature difference between the tank and the reference state which is defined as inlet boundary conditions.

For cases with forced convection through the shell (the D cooling mode as defined in Table 4), the velocity is set to a fixed value at the inlet boundary so that the required mass flow is obtained, i.e.

$$u = u_{\text{in}} \tag{5}$$

All external walls are considered adiabatic with the no-slip boundary condition for the fluid flow.

**2.3 Numerical method and Mesh** The calculation is performed using the commercial software Ansys Fluent 14.0 utilizing the unsteady two-dimensional axisymmetric double precision solver with a fixed time step of 0.05 s. The two-dimensional approach is a good approximation as long as the mass flow rate is increased so that the flow velocities are of the correct magnitude. This is in line with the conclusions from [4]. For the calculations with fluid guides the analysis is done with the steady solver. A mesh of around 200'000 cells is generated to describe the full winding geometry. Although this represents a sparser grid than the one presented in [2], steady-state comparisons with the mesh from [2] show the validity of the conclusions in the present study.

## 3. RESULTS

**3.1 Flow and temperature distribution as a function of mass flow rate** The thermal results of the heat generating shell without fluid guides reveal a surprisingly large number of features which appear to be strongly dependent on the cooling mode, i.e. the flow rate through the winding. Figure 3 shows a typical distribution of the temperature in the solid shell and the cooling fluid, revealing local temperature maxima occurring at regular intervals. Although the flow is from the bottom to the top of the shell, and heat is added to the fluid when it passes the discs, the disc temperature does not increase monotonically, indicating that an alternating flow field is present. The position of the hotspot temperature is not in the top disc position but appears in a disc that is on a lower level. For this particular case the converged model solution is time-independent.

The effect of varying the mass flow rates through the shell (representing N, F and D cooling modes respectively for increasing flow rates, where the discriminating parameters for these cases are given in Table 4) is depicted in Figure 4, where distributions of the (maximum) disc temperature rise above inlet temperature and velocity magnitude of the flow through the horizontal ducts are presented. Apart from the flow rate and the inlet boundary condition all other model parameters are identical. The picture reveals a spatially patterned (but not exactly periodic) distribution of the temperature which appears to be largely controlled by the flow magnitude in the horizontal ducts. This is to be expected since the disc interface with the horizontal ducts comprises the larger part of the total cooling surface of the discs.

**Table 4** Case description

| Cooling mode | Boundary condition | Mass flow rate | Reynolds number |
|---|---|---|---|
| F (OF) | $\Delta T_{\text{tank}}$ = 0K | 0.991 kg/s | 142.7 |
| N (ON) | $\Delta T_{\text{tank}}$ = 10K | 0.546 kg/s | 78.6 |
| D (OD) | $\dot{m}$ = 2 kg/s | 2.000 kg/s | 288.1 |

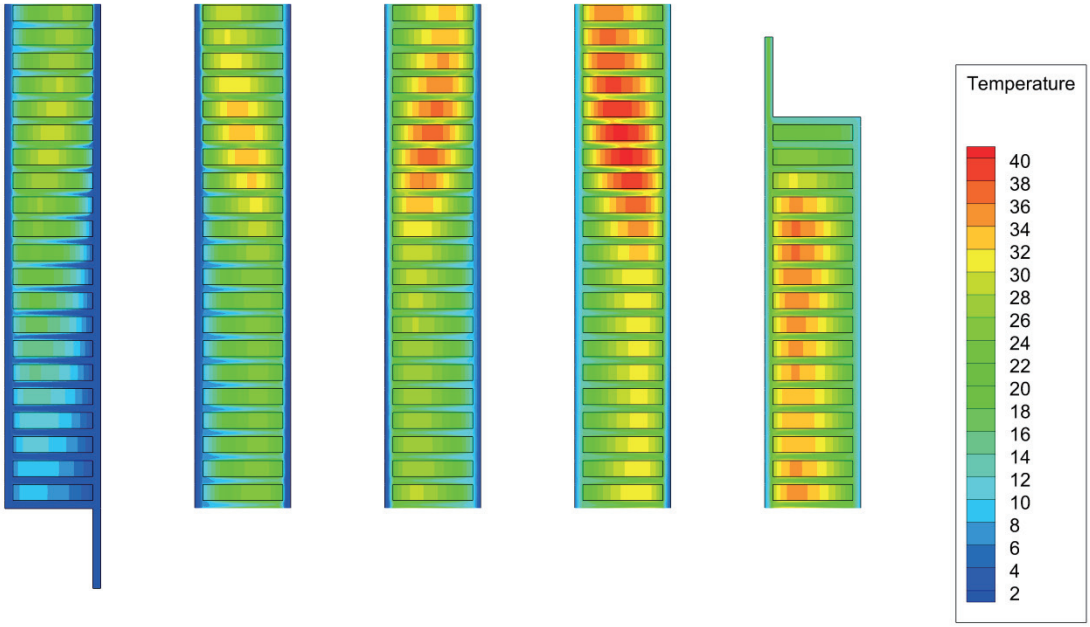


**Fig. 3** Temperature contours of the full winding model in the F cooling mode.

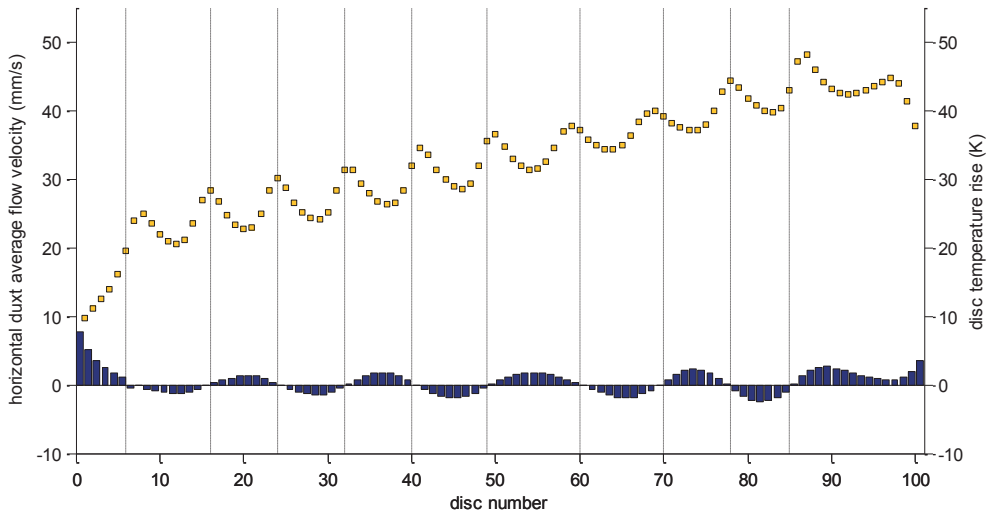


**Fig. 4a** Velocity and temperature profile for the shell in N cooling mode (time snapshot)

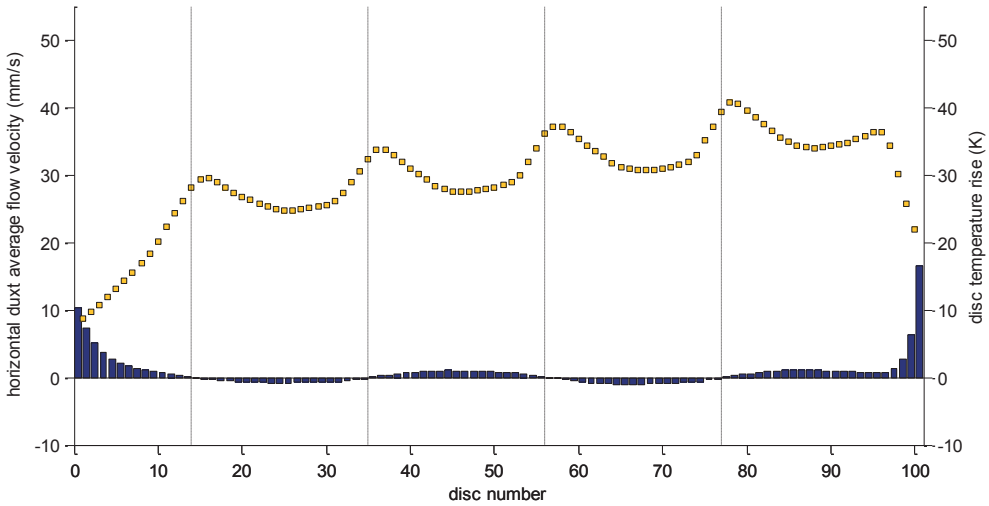


**Fig. 4b** Velocity and temperature profile for the shell in F cooling mode

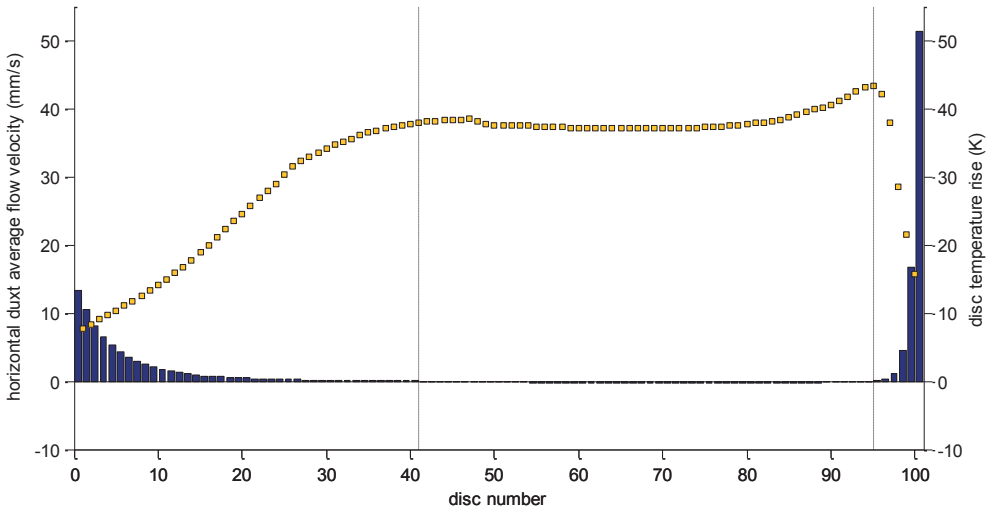


**Fig. 4c** Velocity and temperature profile for the shell in D cooling mode

The number of flow patterns (whose boundaries can be associated with flow reversals of the fluid flow in the horizontal ducts and are marked by vertical dashed lines in the figure) appears to be strongly dependent on the mass flow rate, with the number of flow patterns decreasing with increasing mass flow rate. Additionally, with an increase in total mass flow rate the horizontal flow velocities in the horizontal ducts do not tend to increase in all but the ducts close to the shell bottom and top areas (and the difference in duct velocities between the center of the shell and the areas close to the inflow/outflow areas increases with mass flow rate as well). Consequently, the cooling of the discs does not significantly improve (as is reflected by the average shell temperature) and – most importantly – the hotspot temperature stays roughly on the same level. This implies that there appears little "direct control" of the cooling efficiency of non-guided windings (whose characteristic geometry dimensions are mainly set by electrical and mechanical requirements). Furthermore, a small change of a low velocity duct due to tolerances in design or material dimensions/properties may then have a large impact on the resulting temperature.

**3.2 Steady vs. non-steady behavior** In this study both steady and time-dependent behavior was observed, the sole discriminating factor being the mass flow rate. For a low mass flow rate (N cooling mode), the solution proved to be time-dependent where the net mass flow rate appeared to swing within roughly 20% of its maximum value as is shown by Figure 5. Its effect on the flow and temperature distribution pattern is that the turning points of the

flow shift slightly and that the flow in the horizontal ducts changes in magnitude. Figure 5 also shows that the effect on hotspot temperature is significantly smaller (less than 1K) but the latter distribution also shows that the temporal pattern is far from purely cyclic and is characterized by several timescales. The shorter time scale correspond to the changes in mass flow rate, whereas the longer time scale is more chaotic and no cyclic behavior was observed. The reason for the varying mass flow rate is that the average fluid temperature in the winding is also changing in time with a corresponding variation in buoyancy that is taken into account through the boundary condition in Equation (4).

Figure 6 shows the effect of the cyclic flow behavior in duct number 68 where the average velocity is close to zero. In contrast to the N cooling mode with relatively low mass flow rate, the F and D cooling modes corresponding to higher mass flow rates reveal a time-independent solution.

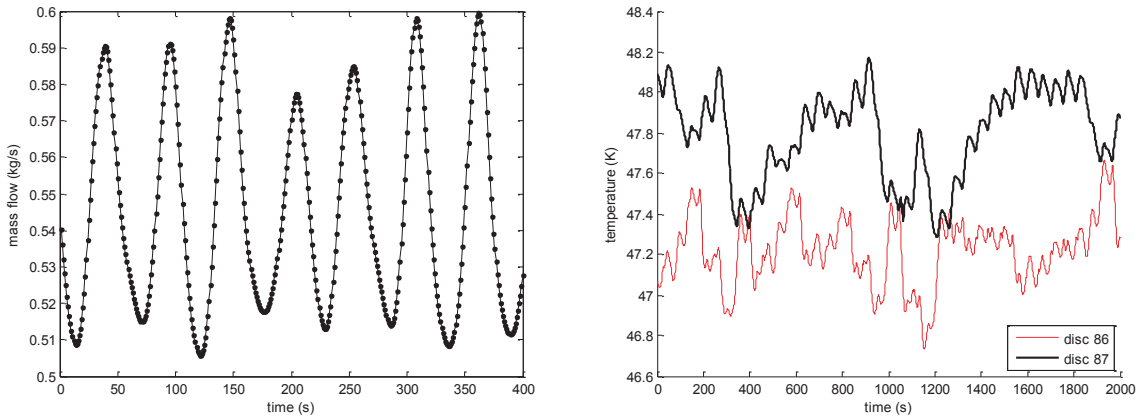


**Fig. 5** Time-dependent variation (for the N cooling mode case) of the net winding oil mass flow rate (left) and its effect on the position and variation of the hotspot temperature rise above inlet temperature (right) as monitored during a longer time interval.

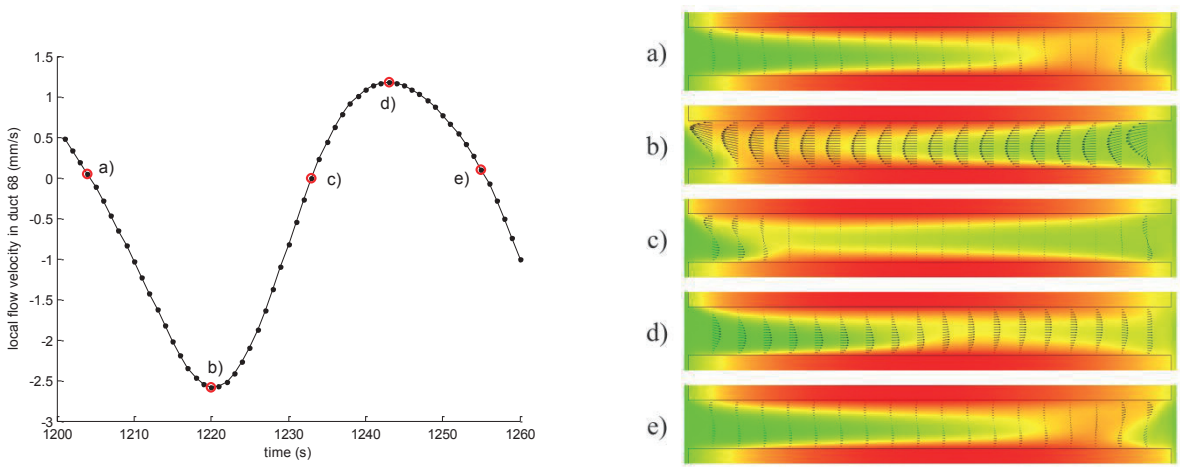


**Fig. 6** Local analysis of the time-varying flow and temperature behavior between disc 68 and 69 showing the flow magnitude variation in the duct in time (left) and the change of local velocity and temperature profile (right) at five snapshots indicated in the left picture.

**3.3 Thermal performance of non-guided vs. guided windings** In order to investigate the cooling efficiency of non-guided vs. guided geometries, the average and hotspot temperature results for the non-guided cases are compared with equivalent setups where a number of guides are added. Since guided flow is not the primary subject of the underlying study, only a set of predefined uniform guide distributions (i.e. fixed number of discs per pass) is regarded.

Figure 7 shows the average shell temperature and hotspot temperature as a function of the number of guides in comparison to the non-guided case. Since the primary effect of the guides is to divide the shell in sections and enhance the flow in the horizontal duct where the main disc cooling area is present, the effect of the enhanced cooling of the shell through the average shell temperature can be clearly seen. When looking in detail at the temperature profiles in the windings the increasing number of guides appears to lead to a more uniform temperature distribution in the winding in the sections, at least for the F (Figure 8) and D cooling modes, having a lowering of the hotspot temperature as a consequence.

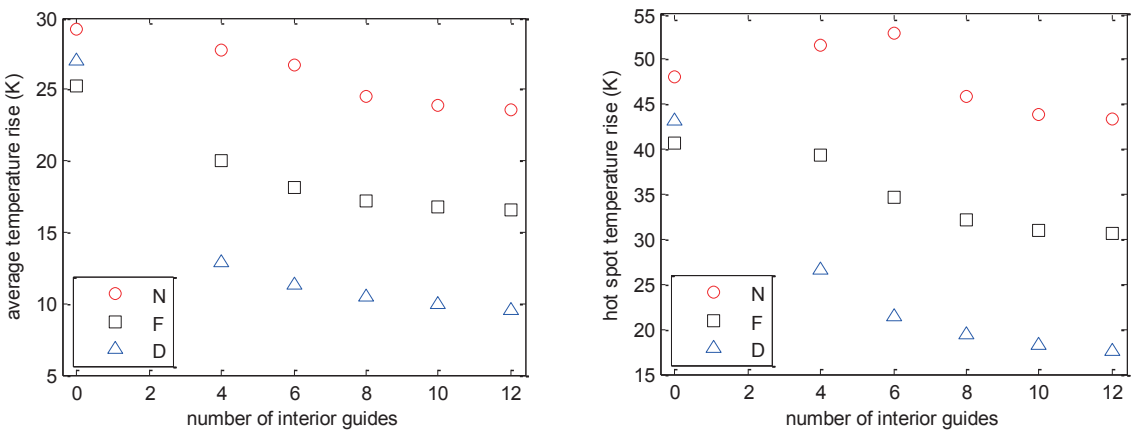


**Fig. 7** Change of calculated temperatures (left: winding average, right: winding hotspot) as a function of the number of oil guides applied. The zero oil guides case corresponds with the non-guided winding case.

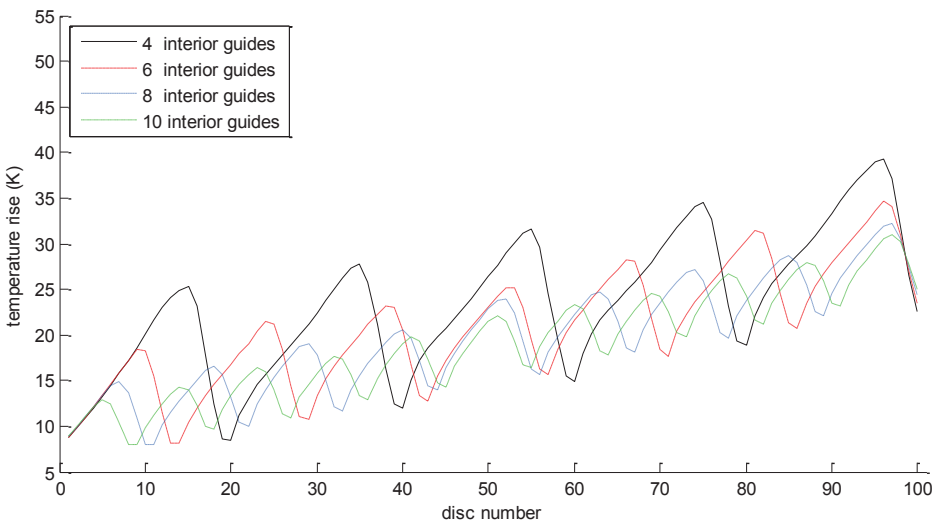


**Fig. 8** Distribution (left side: bottom, right side: top) of the maximum temperature in the discs for different numbers of guides, for the F cooling mode

The effect of additional guides diminishes the more guides are already present. Although the number of guides was not optimized, one may expect an optimal number of fluid guides exist since an increasing number of fluid guides ultimately creates a larger flow path length with viscous drag, with a negative effect on the mass flow rate.

Interestingly, Figure 7 shows that the hotspot temperature follows the same pattern as the average shell temperature for the F and D cooling modes, but the N cooling mode with a low number of guides show a potential for worse results as compared to the non-guided case, as is also clear from the detailed temperature distributions as shown in Figure 9. In particular the scenarios with 4 and 6 fluid guides show a larger number of local temperature maxima than the number of guides present. A detailed analysis of the velocity distribution as shown in Figure 10 reveals that local flow reversal (in passes between fluid guides) is the cause of the problem. In the regions where flow reversal occurs the velocities in the ducts are very low and consequently adjacent discs are insufficiently cooled, giving rise to local temperature maxima.

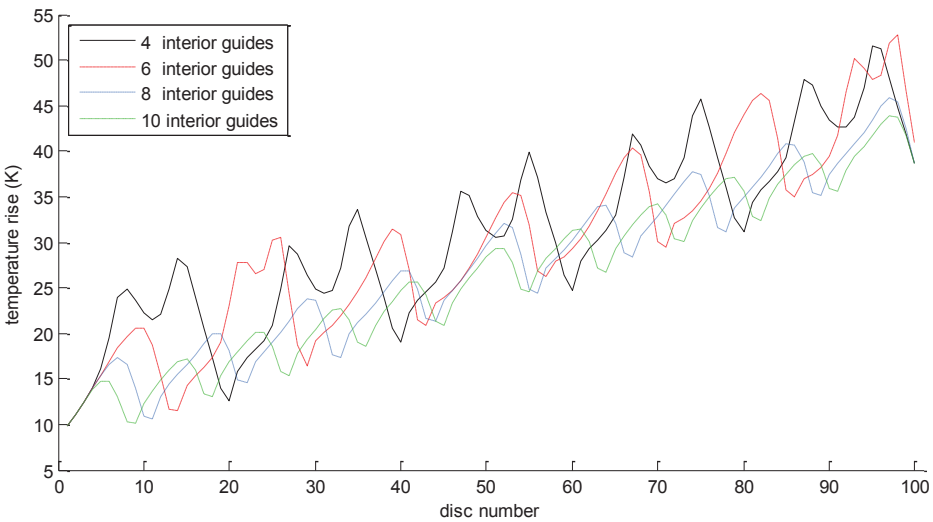


**Fig. 9** Distribution (left side: bottom, right side: top) of the maximum temperature in the winding discs for different numbers of fluid guides in the N cooling mode

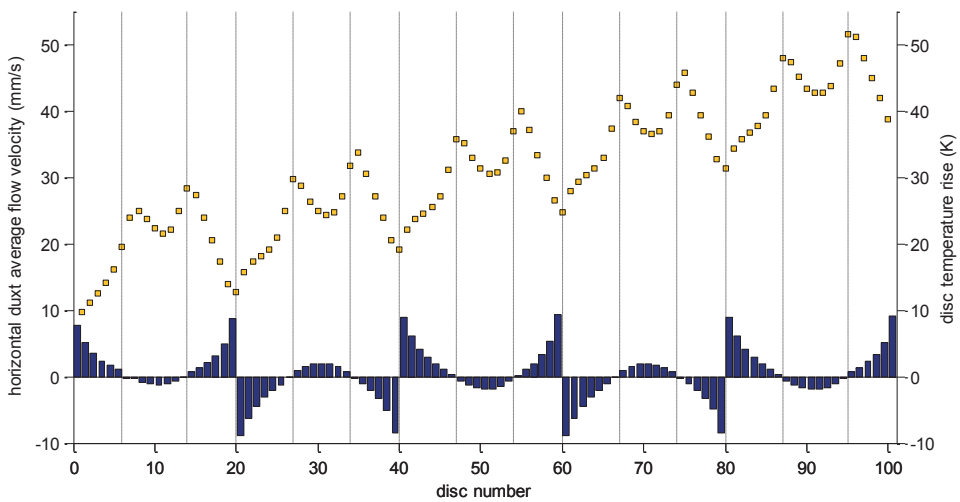


**Fig. 10** Example of distribution of disc temperature and horizontal duct velocities for the non-optimal case with 4 oil guidesin the N cooling mode, revealing flow reversal in the passes with associated local temperature maxima.

A low number of guides also implies a large number of discs per pass, which in turn for the N cooling mode with its flow reversals in a pass implies that the flow distribution per pass begins to show non-guided behavior. Thus it is clear that, although guided windings generally perform better than non-guided windings (as shown in Figure 8), the determination of the number and position of the fluid guides should be done with care and requires proper modeling analysis.

## 4. CONCLUSIONS

In this article a method is presented for analyzing so-called "non-guided" power transformer disc-type windings where the orientation of the cooling oil flow surrounding the discs is not defined by fluid guides at certain positions in the winding. The simulated thermal performance (in terms of winding hotspot temperature and average winding temperature) of this particular winding geometry is compared with equivalent winding designs where a varying number of fluid guides are included. The results reveal significant differences in behavior between non-guided and guided windings that may have important consequences for thermal verification based on measurements during transformer acceptance testing.

From a qualitative perspective the analysis shows that non-guided windings show a larger number of flow types as compared to their guided counterparts. Non-guided windings may show non-steady behavior with significant variation in oil mass flow rate through the winding (approximately 20% as shown in the ON case, generated through internal buoyancy). As a consequence the value and location of the winding hotspot temperature varies as well, although the impact on temperature level is moderate for the case under study (approx. 1K for the hotspot temperature). This behavior contrasts with the fluid guided cases (assuming exactly the same geometrical dimensions and winding losses but with a varying number of fluid guides placed in the vertical ducts) which for the cases under study always show steady state behavior. Consequently, this study reveals that it is necessary to model pressure boundary conditions at both in and outflow boundaries instead of the more commonly used mass flow or velocity inlet conditions in order to allow for non-steady solutions. For two other cases with higher oil flow rates (OF, OD) the flow was shown to be time-independent.

For non-guided windings the flow and temperature distributions as a function of winding height tend to follow a repetitive zigzag pattern characterized by alternating areas with a weak, oriented flow between the discs, and local temperature maxima occur between the areas where the horizontal flow is extremely weak and changes direction with winding height. Close to the winding entry and exit the horizontal flow becomes more significant again. The number of alternating areas decreases strongly with increasing winding oil flow rate, with the oil flow in the vertical ducts becoming more dominant.

Contrary to what is expected and assumed in international standards [1], the hotspot temperature location is not necessarily at the top of the winding, but – depending on the oil flow rate – may be at the interface between the top-most and second flow area (ON, OF) or close to the top (OD). Since the winding flow rate depends not only on the winding design but also on other design parameters of the transformer as a whole, this will have the practical consequence that for non-guided windings it is a much more involved task requiring a strong experience base on physics and modeling tools to predict the actual position of the hotspot and determine an appropriate position of the hotspot temperature sensor as is used increasingly for verification during final testing.

In comparison to non-guided windings, fluid guided windings with the same geometry and heat loss conditions tend to be more efficiently cooled (regarding both hotspot temperature and average winding temperature) and the temperature distribution easier to predict because of the imposed zigzag oil flow pattern through the particular fluid guide placement. The cooling efficiency of fluid guided windings is dependent on the number of fluid guides and for the cases under study they become more efficiently cooled with an increasing number of fluid guides. In one case it is shown that with a very low number of fluid guides the thermal performance is actually worse than the non-guided winding case.

This study shows that CFD-based modeling approaches can play a very valuable role in understanding the thermal behavior of power transformer windings in general and non-guided discs windings in particular, It reveals that non-guided windings generally are less efficiently cooled as compared to their oil-guided counterparts and that the assessment of the hotspot temperature by direct measurement may be assumed to be more involved due to the larger range of possible flow modes in non-guided windings. The phenomena that are emergent from the results also imply that it is necessary to model the winding as a whole and that pressure boundary conditions for both winding inlet and outlet are required.

## REFERENCES


[1] IEC60076-2 (3rd ed), Power transformers – Part 2: Temperature rise for liquid-immersed transformers (2011). **Standard**

[2] Kranenborg, E.J., Olsson, C.O., Samuelsson, B.R., Lundin, L-Å., Missing, R.M., Numerical study on mixed convection and thermal streaking in power transformer windings, Eurotherm 2008, Eindhoven, The Netherlands (2008). **Journal Paper**

[3] F.Torriano, M. P., "Numerical study of parameters affecting the temperature distribution in a disc-type transformer winding," *Applied Thermal Engineering,* 30, pp. 2034-2044, (2010). **Journal Paper**

[4] F.Torriano, P. M. C., "Numerical investigation of 3D flow and thermal effects in a disc-type transformer winding," *Applied Thermal Engineering,* 40, pp. 121-131, (2012). **Journal Paper**

[5] Skillen, A. R. H. I. W. W., "Numerical prediction of local hot-spot phenomena in transformer windings," *Applied Thermal Engineering,* 36, pp. 96-105, (2012). **Journal Paper**

[6] Chen, S., Devaux, F., Saravolac, M., Le Friec, C., CFD Investigation on thermal performance of different types of windings for a 300MVA transformer, CIGRE 107 Canada Conference (2012), **Journal Paper**

[7] Yuhe Jiao, CFD Study On The Thermal Performance of Transformer Disc Windings Without Oil Guides, KTH, Stockholm (2012), **M.Sc. Dissertation**

[8] ANSYS FLUENT Theory Guide, Release 14.0, ANSYS, Inc., November 2011. Available at https://support.ansys.com/AnsysCustomerPortal/en_us/Knowledge+Resources/Online+Documentation/Previous+Releases/14.0 [accessed May 30, 2014] **E-books**