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ANALYSIS OF THE COOLING DESIGN IN ELECTRICAL TRANSFORMER

Toste Azevedo[a]*; Joel Mendes[b] and Eugenio Santis[b]
*Author for correspondence
[a] Instituto Superior Técnico
Mechanical Engineering
Pav. Mecanica I
Av. Rovisco Pais
1049-001 Lisboa
Portugal
E-mail: toste@ist.utl.pt

[b] Siemens
Dep. Engenharia
Rua da Fábrica
2715-981 Almargem do Bispo
Portugal
E-mail: joel.mendes@siemens.com
E-mail: eugenio.santis@siemens.com

ABSTRACT

This work presents the application of a CFD model Fluent to simulate the cooling system of a transformer with natural oil circulation. The cooling fluid passes through channels between coils and is cooled in fins that are in contact with ambient air. Two types of cooling systems are compared namely fins installed in the transformer tank or external fins in panels. Due to the complexity of the system, simplified situations were defined representing the flow in channels and the natural circulation of oil in a cavity. A grid independency study was performed for these situations in order to define the minimum resolution to be used in the model for the transformer. The simulation of the complete transformer was initially performed with a coarser grid to anticipate the conditions in the channels and cavity. The grids used in the simulations presented in the present paper were defined taking into account the results from the parametric studies and the possibility to analyse several parameters: Location of the fins, presence of supports, thickness of channels and thermal conductivity of coils. Geometrical approximations were introduced respecting the cross section areas, while heat transfer in the fins were modelled as a volumetric heat sink based on the overall heat transfer coefficient between the cooling fluid and ambient air. The numerical results were in line with typical expected temperatures although higher than some values observed specially for the oil within the fins. The modification of parameters and the fin cooling system had a small impact on maximum temperature levels. The conductivity of the coils that is very different between foil and wire windings however has a major impact on temperature distribution and increases the maximum value about 10ºC.

INTRODUCTION

The configuration of transformers is essentially determined by the needs of the electrical field. There are two main types of construction, in discs for high voltage (>60 kV) and capacity (>40 MVA) or in layers for smaller transformers or a combination of both construction principles. The cooling of the transformers in general is made through the vertical channels with air for small transformers (dry system) or using a cooling fluid that is dielectric and in general has a large viscosity that changes with temperature.

With the objective of getting a better insight on the problem and to develop procedures to estimate the operation temperature of transformers, the present work presents numerical simulations of a representative transformer that can be assembled with fins installed at the transformer cage or in fin panels. This study should also examine the influence of modifications in the channel thickness and other construction parameters. The work was promoted by Siemens Portuguese factory and was developed at Instituto Superior Técnico.

The analysis of cooling systems in transformers can be performed with two main techniques, namely the use of flow network models or using Computational Fluid Dynamics (CFD) models. The present work presents the results obtained with the application of the later technique. It should be kept in mind however that the simplicity of the flow network models is favoured by the companies projecting and building transformers so further developments should consider them.

Zhang et al [1] used network models to calculate the oil circulation by natural convection in a disk type transformer and combined it with calculations for the temperature distribution within the windings. The results lead to the conclusion that the flow is mainly in the vertical direction and that the coupling between the two thermal-hydraulic and thermal model of the windings is important.

The analysis of heat transfer within transformers using CFD based models was already reported by Smolka and Nowak [2] who used a Fluent model for the case of dry cooling systems that is cooling by air. The turbulent flow was modelled with the k-ε model and was characterised by an upward flow through the channels between the windings and the nucleus. Around the coils the air descends in the transformer tank. The model for flow and heat transfer was coupled to the electromagnetic model carried out with the program Electromagnetic solver Ansys/Emag, to consider the interactions between temperature and the material properties, both thermal and electric and hence calculate the specific losses of coils and nucleus more accurately.

Mufuta and Bulck [3] performed simulations for oil circulation in a representative section of a disk transformer with mixed forced/natural convection. The oil flow is in the laminar regime with Reynolds numbers between 15 and 120 and maximum velocities of 0.03 m/s for a 25 MVA transformer. The use of grid refinement close to the walls is very important to the calculation of friction and heat transfer. The dimensioning criteria, for class A materials, was the limitation for the maximum temperature to 65ºC above ambient or 115ºC.

Following this introduction, the next section presents the numerical model used and discusses the approximations that were introduced for the representative cross section of the transformer. The preliminary numerical simulations section presents numerical results, firstly of simple test cases used to evaluate the influence of the grid resolution and then for the transformer. The results for the transformer include a comparison with test data, the influence of the fin cooling system and the influence of the channels thickness. Finally conclusions are summarized in the last section.

NOMENCLATURE

A	[1/m^2]	Permeability
c_P	[J/kg.K]	Specific heat
D_h	[m]	Hydraulic diameter
g	[m/s^2]	Gravity acceleration
k	[W/m.K]	Conductivity
x	[m]	Axial coordinate
r	[m]	Radial coordinate
Q	[W/m^3]	Volumetric heat source
t	[m]	Channel thickness
U	[W/m^2.K]	Global heat transfer coefficient from oil to air in fins
u_x	[m/s]	Axial velocity
u_r	[m/s]	Radial velocity
V	[m/s]	Velocity
β	[1/m]	Thermal expansion coefficient
ρ	[kg/m^3]	Density
μ	[kg.m/s]	Viscosity

NUMERICAL MODEL

This section presents the numerical model used in the Fluent program. As the real transformer geometry is three dimensional and complex, approximations were introduced to consider an equivalent axissymmetric configuration explained in geometrical approximation section. The model adaptation required to represent heat transfer and fluid flow of the cooling fluid within the fins are presented.

Numerical model

The Fluent numerical model was used in the two-dimensional axissymmetric version for steady state conditions with constant properties except for viscosity due to the large variations of this property with temperature. The mass, momentum on x and r directions and energy balances are written in the most general form used as:

$$\frac{\partial(\rho u_x)}{\partial x}+\frac{1}{r}\frac{\partial(r\rho u_r)}{\partial r}=0 \tag{1}$$

$$\rho u_x\frac{\partial u_x}{\partial x}+\rho u_r\frac{\partial u_x}{\partial r}=-\frac{\partial p}{\partial x}+\left[\frac{\partial}{\partial x}\left(\mu\frac{\partial u_x}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu\frac{\partial u_x}{\partial r}\right)\right]-\frac{\mu}{A}u_x+\rho g_x \tag{2}$$

$$\rho u_x\frac{\partial u_r}{\partial x}+\rho u_r\frac{\partial u_r}{\partial r}=-\frac{\partial p}{\partial r}+\left[\frac{\partial}{\partial x}\left(\mu\frac{\partial u_r}{\partial x}\right)+\frac{\partial}{\partial r}\left(\frac{\mu}{r}\frac{\partial}{\partial r}(ru_r)\right)\right]-\frac{\mu}{A}u_r \tag{3}$$

$$\rho\left(u_x\frac{\partial(c_pT)}{\partial x}+u_r\frac{\partial(c_pT)}{\partial r}\right)=\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial x}\left(rk\frac{\partial T}{\partial r}\right)+Q \tag{4}$$

These equations apply to the cooling fluid zone and they include in the vertical direction momentum balance a term accounting for the gravity mass force. This term coupled with the hydrostatic pressure gradient term was expressed as a function of the expansion coefficient and temperature gradients by the Boussinesq approximation.

The momentum balances also include a viscous resistance term defined with the permeability A that is considered to represent the influence of the fin walls where they are installed as it will be described below. The heat transfer through the fins in the equivalent model is considered in the energy balance as a volumetric source. The energy balance equation is also applied in the solid zones, specifically with heat sources in the nucleus and coils.

For other situations that were simulated in two-dimensional cartesian coordinates the equations can be revealed by replacing the radius by unity. For these cases the objective was the comparison with correlations and the viscosity was also considered as a constant to allow for a better comparison.

Geometrical model approximation

The transformer being considered is a 2.5 MVA three phase transformer that has three coils installed in line surrounding the three legs of the nucleus that is closed in loops. The coils geometry and the nucleus leg are approximately cylindrical so an axissymmetric model was considered. The channels are built between the coils using small wood bars 5 mm thick placed every 50 mm to guarantee the coil separation and in the model their presence was neglected so a continuously open channel was considered. Figure 1 presents a sketch of the geometry considered as well as the dimensions.

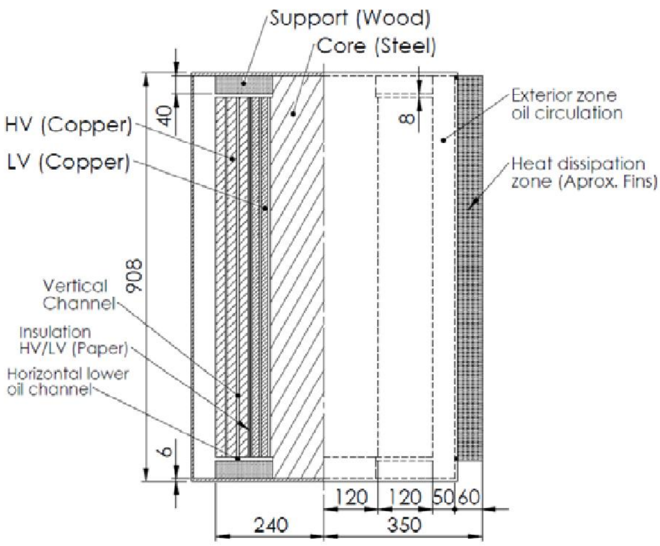

Figure 1 (a) Materials and indication of zones in the model. Fins are represented in right side (b) Dimensions.

There are two main types of cooling systems for this type of transformers in this capacity range. In the first one considered fins are placed directly in the tank and have an internal thickness of 8 mm so the cooling oil may flow between the channels formed and dissipate the heat to ambient air. The other system has sets of fin panels with similar thickness (8mm) and tubular connections at the top and bottom to distribute and collect the cooling oil.

For the capacity considered the total heat transfer area for natural convection of the air is similar and therefore the cross sectional area of the fin zone is similar. The equivalent outer cylindrical region has a thickness of 60 mm as represented in figure 1a) on the right side. The main difference between the fins on the tank or in panels is that in the later case there is a wall dividing the two zones with two openings at the top and bottom with thickness providing similar cross section as the ones from the connecting tubes.

Table 1 presents the main material properties for all the solid materials considered. Oil properties are presented in table 2. The viscosity was represented as a function of temperature by: $\mu = 2386{,}2e^{-0{,}032\cdot T}$ kg m/s.

Table 1 Properties of the materials considered.

Property	Copper	Steel	Wood	Paper	Resin	Units
ρ[Kg/m^3]	8933	8030	700	930	1190	Kg/m^3
c_p [J/kg.K]	385	502.5	2310	1340	-	J/kg.K
k [W/m.K]	401	60	0.173	0.18	0.16	W/m.K

Table 2 Oil properties.

Property	ρ [Kg/m^3]	c_p [J/kg.K]	k [W/m.K]	β [1/K]
Value	889	1909	0.145	7.50x10^{-4}

Numerical simulation of the fins

The geometrical equivalent area for the fins is defined in the previous section and figure 1, keeping the same cross section area. The real heat transfer area is lost in the approximation and therefore it was introduced through a volumetric heat source. The same applies to the effect of friction of the oil flow in the fin walls. This was represented in the model considering a porous media with porosity of one but with a permeability that was defined based on the comparison of the pressure loss term indicated in equation 2 and 3 with the pressure loss due to friction in a channel expressed below:

$$\frac{\Delta p}{L} = \frac{C}{\mathrm{Re}_{Dh}} \cdot \frac{1}{2} \cdot \rho \cdot \frac{v^2}{D_h} = \frac{C \cdot \mu}{\rho \cdot v \cdot D_h} \cdot \frac{1}{2} \cdot \rho \cdot \frac{v^2}{D_h} = \frac{C}{2} \cdot \frac{\mu \cdot v}{D_h^{\,2}} \tag{5}$$

where C is 96 and the hydraulic diameter D_h is equal to twice the channel thickness. Comparing this expression with the viscous term from the momentum equations allows for the conclusion that the permeability A should be:

$$A = \frac{2 \cdot (2t)^2}{96} = \frac{t^2}{12} \left[m^2\right] \tag{6}$$

This permeability was introduced in the porous media pressure loss law considering the inertial coefficient zero since the flow is laminar. For the radial direction however due to the approximations the geometrical model has 60 mm while in the real system the fins height are 360 mm for the fins on the tank and 250 mm for the fins on panels. Therefore the permeability factor was reduced by the length factors to simulate higher radial pressure drop.

For heat transfer similar reasoning was used to deduce the volumetric heat source considered in the fin zones. As the volume of the oil within the fin channels is correct the heat source is defined providing the same heat transfer area that exists. The volumetric heat release is then defined as:

$$Q = \frac{-Area \cdot U \cdot \left(T_{Oil} - T_{Air}\right)}{Volume} = \frac{-2 \cdot U}{t} \cdot \left(T_{Oil} - T_{Air}\right) \tag{7}$$

Thus the heat source can be defined by two terms one constant considered explicitly and the other proportional to the variable under consideration T_{Oil} with a negative multiplying factor so it is considered implicitly to improve convergence. The implementation of the heat source is done through a UDF (User Defined Function) of Fluent.

The global heat transfer coefficient represents the result of all thermal resistances including convection of the oil, conduction through the wall and natural convection of the ambient air and other factors. From all the resistances the air convection outside is the higher and therefore defines the value of U. Using correlations for heat transfer from fin arrays led to estimates of the external coefficient of the order of 3 to 4 W/m^{2}K but this value is not consistent with known average heat fluxes and surface temperatures. Taking these values into account the global heat transfer coefficient is estimated as 8 W/m^{2}K.

RESULTS AND DISCUSSION

Preliminary numerical simulations

Before carrying out the simulation of the transformer model with the approximations presented above, a preliminary approximation of the geometry was considered without the fins area but with a high heat transfer coefficient at the tank surface as the ratio between the two areas is about 50. As initially a heat transfer coefficient for the air was estimated as 4 W/m^{2}K the simulations led to very high temperatures inside the transformer and small velocities of the oil as a constant viscosity was considered. In any case the initial simulation led to a first idea about the flow field and temperature distribution. Furthermore simulations were performed for two simplified situations corresponding to a cavity with uniform temperatures in the side walls and a channel with imposed heat flux. These simulations were used to evaluate the effect of the grid resolution in the numerical solution.

Table 3 presents the main characteristics of the numerical grids used to simulate the cavity and the calculated Nusselt number. Grids 1, 2 and 3 are uniform while grids 4 and 5 are refined closer to the wall. From the results it can be observed that the non-uniform grids have a much better performance than the uniform grids as expected and the Nusselt number that is calculated for the finer grids is about 10.

Table 3 Number of control volumes in both vertical and horizontal directions and Nusselt number calculated.

Cavity Mesh	**1**	**2**	**3**	**4**	**5**
Total number of cells	225	3600	57600	3600	19200
Divisions on X	5	20	80	20	60
Divisions on Y	45	180	720	180	320
Nusselt number calculated	4.90	11.63	10.12	10.07	10.12

The calculated value (10.07) with grid 4, that was considered acceptable, is about 5% lower than the value calculated from a correlation for this situation [4]. This implies that from the point of view of the natural convection side the temperature inside the transformer is over predicted. The number of non-uniform control volumes recommended for the transformer grid in the area of the fins was therefore higher than 20 in horizontal direction and 180 in the vertical one.

For the flow in channels a typical 5 mm/s obtained from the preliminary simulations was used to define a test case with flow in a 10 mm hydraulic diameter channel. In this case several grids were used for the 0.9m long channel with small number (<200) of control volumes in the axial direction, with 200 and with 400. Changing the radial resolution from 4 up to 80 control volumes 18 grids were defined and the results of the Nusselt number at the middle of the channel is represented as a function of the number of control volumes in figure 2. There are differences in the 200 and 400 axial volumes if they expand (E) only in the flow direction or if they also contract (C) towards the exit. The later condition applies to the grid used for the transformer since the vertical channels are then connected to transversal channels where the vertical control volumes are small.

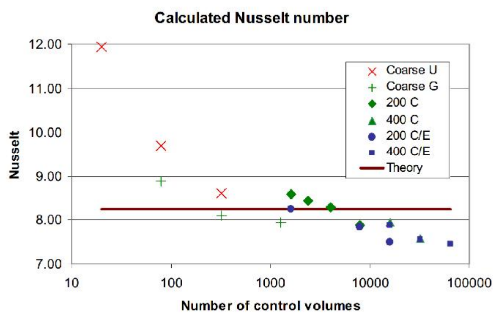

Figure 2 Calculated Nusselt number for the different grids compared with the theoretical value (8.235 [4]).

The numerical results show, as expected, that the calculated values converge to a common value for all the grid types. The calculated value for the finer grids however is smaller than the value calculated from theory. This is related to the fact that for the constant heat flux case at the channel walls there is a constant average temperature increase, while in the simulations there is no heat transfer through the inlet and outlet. Using the coarser grids with gradual increase of the control volume dimensions reasonable results are obtained with 1 to 2 thousand elements and therefore this was the grid resolution considered for half the channels in the transformer. Using 200 elements in the vertical direction more than 16 control volumes in the radial direction were used for the channels.

Transformer simulations

Based on the experience from the preliminary tests and the objective of performing parametric tests, a general grid was prepared dividing initially the volume in 27(radial) by 11(axial) zones. Figure 3 presents a sketch of the zones and the numerical grids within these. In the vertical direction 200 nodes were used while for the horizontal direction the cavity, fins and channels had respectively 28, 30 and 16 control volumes. The grid represents the 5 mm channel and when this is increased to 7 and 10 mm, the number of volumes is increased by 10 and 24.

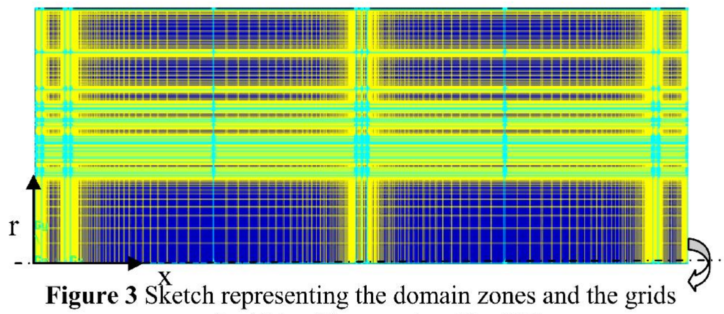

Figure 3 Sketch representing the domain zones and the grids used within. (Figure rotated by 90º)

Figure 4 and 5 presents the numerical results of the simulations performed for the case of fins in the tank and fins in panels with the main difference being that in the first case the fin zone (right side of the figure) is opened to the cavity, while in the case of fins in panels there is a dividing wall with openings in the top where the oil enters the fins and exits in the lower opening to enter below the coils.

The case with dividing wall has more resistance to flow, including the restrictions of the connecting tubes and therefore leads to lower values of the oil velocity through the fins. The lower velocities are also observed within the channels when compared with the case of fins in the tank. When the fins are directly connected to the tank there is a clear difference of the velocity within the fins and in the cavity where the values are close to zero. The same applies when there is a dividing wall as expected. In the cavity zone there is only a minor zone close to the coils where there are upwards velocities and therefore this zone should be minimized if possible to save on the amount of cooling oil.

The maximum temperature values are observed at the nucleus and than the low voltage coils. The temperature in the coils has large variations in the vertical direction but negligible variations in the radial direction. In fact based on the theoretical temperature distribution in plates with internal heat generation leads to the conclusion that the temperature differences in the horizontal direction is of the order of 0.01 K for the copper conductivity. These small differences lead to numerical difficulties in calculating the heat flux from differences of very similar values and it was observed that using single precision, convergence and a correct global overall heat balance was difficult to achieve.

For the case of transformers with external fin panels there are some measured results available from Siemens and therefore a direct comparison of the measured and calculated values is reported in table 4. It can be observed that the calculated values are in general over predicted. The table indicates the absolute differences between calculated and measured values and the relative differences for the temperature difference between the local values and the ambient air temperature considered as 300 K in both experimental and simulated cases.

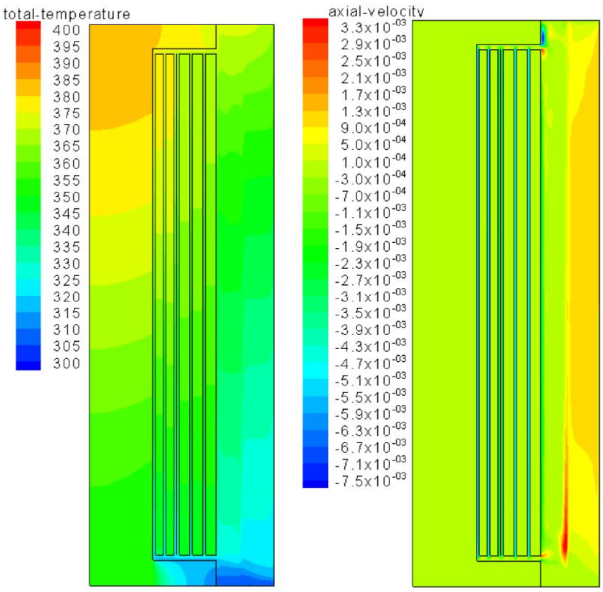

Figure 4 Calculated temperature and axial velocity with fins in the tank (without dividing wall).

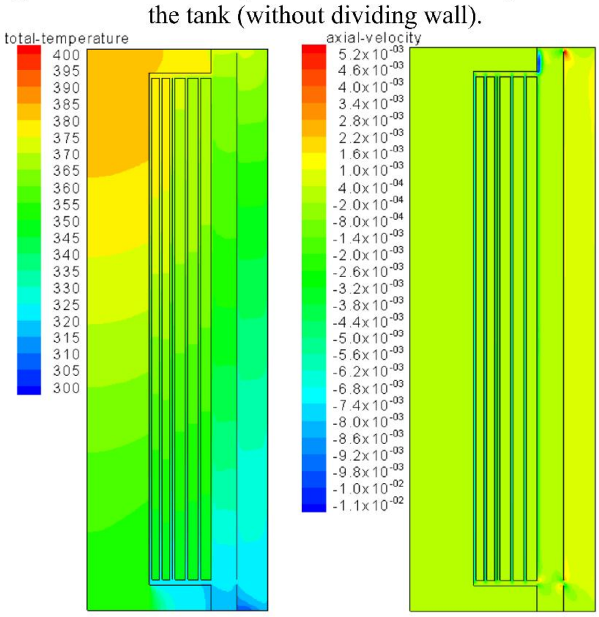

Figure 5 Calculated temperature and axial velocity with fins in panels (simulated with a dividing wall).

Table 4 – Comparison between calculated and measured temperatures in different locations in the transformer.

Location	Measured	Simulated	Difference	Variation %
T_{LT} Average low voltage	346.4	364.4	18.0	39%
T_{HT} Average high voltage	343.9	359.6	15.7	36%
$T_{HP\ LT}$ Hot point in low voltage	365.7	378.6	12.9	20%
$T_{Oil\ circ}$ Upper part of oil circulation	341.7	372.0	30.3	72%
$T_{RAD\ sup}$ Oil at the inlet to fin panels	335.5	370.9	35.4	100%
$T_{RAD\ low}$ Oil at the exit of the fin panels	315.4	322.5	7.1	46%
$T_{Oil\ ave}$ Average oil in fin panels	328.0	340.8	12.8	46%

The relative differences change from 20 to 100% but there is a good correlation between the values except for the 4th and 5th position. The larger difference in the 5th position is on the inlet temperature for the fin panels where a small deviation from the main flow may affect the measured value. At the 4th position close to the top, the transformer may lose some heat that was not considered in the simulations.

The lower oil temperature measured suggests that the oil flow through the fin panels is higher also observed by the smaller temperature variation when compared with the calculated value. This may also be a consequence of a higher external heat transfer coefficient. This can also be observed from the average oil temperature in the fin panels.

Several parametric tests were carried out to evaluate the influence of some construction parameters on the working conditions of the transformer. To evaluate the results the minimum and maximum temperature and the maximum velocity are reported for all the cases tested in table 5. The first test was to remove the support of the coils that are wooden and in reality do not cover the whole area of the coils. When these are removed the maximum temperature is lowered slightly although the minimum temperature increases. This is due to the better circulation of the oil that in general has slightly higher velocities in the channels. The maximum velocities that occurred at the entrance and exit between the coils and the supports however decrease due to the lower flow restriction.

The other test that was performed was to increase the thickness of the channels to 7 and 10 mm. When these modifications are done the maximum temperature decreases and the maximum velocity increases due to the better oil circulation. The differences between the results however are relatively small and the balance between the moderate improvement in heat transfer and the additional cost of the system should be further weighted.

Table 5 Calculated maximum and minimum values of temperature and maximum velocity in the domain.

Case	Tmax [K]	Tmin [K]	Vmax [mm/s]
Without wall	382.97	304.97	13.34
With wall *	384.12	307.64	12.90
without coil support	380.15	314.24	11.80
without coil support *	381.15	308.22	9.87
with t=7 mm	379.39	305.45	14.37
with t=7 mm *	374.77	311.25	18.10
with t=10 mm	377.06	306.01	18.30
with k_c=4 W/m.K *	391.35	305.38	12.15

The last simulation indicated in the table corresponds to lowering the conductivity of the material in the coils, once the copper wires are covered by resin and therefore the effective heat conductivity may decrease substantially. A decrease of two orders of magnitude was considered leading to an increase of about 10ºC on the maximum temperature although the maximum velocity is similar. The main temperature gradients remain in the axial direction although smaller gradients can also be observed in the horizontal direction. Figure 6 compares the heat flux profile from the low voltage coils to the oil for the two values of the conductivity. With the larger conductivity the temperature distribution in the coil is more uniform producing a large heat flux peak at the entrance of the channel (y~0.9m) and it decreases towards the upper part of the channel. When the conductivity is much lower there are more significant temperature gradients in the vertical direction and the temperature profile is almost parallel to the temperature profile of the oil in the channels. This corresponds to an almost uniform heat flux. The larger conductivity in the coils corresponds to the construction in foil conductor while the lower conductivity corresponds to the construction with wires.

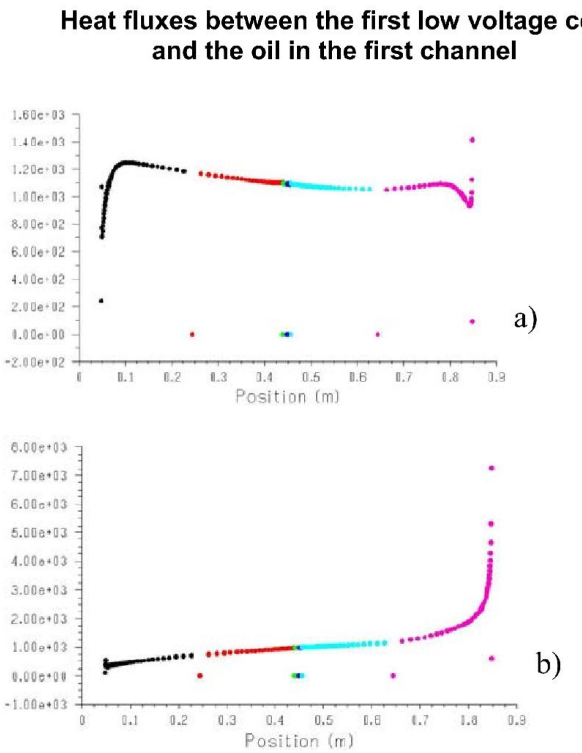

Figure 6 Calculated heat fluxes from the inner low voltage coil to the oil in the channel.
a) Low conductivity in the coil and **b)** High conductivity in the coil.

CONCLUSION

This paper presents to the best of our knowledge the first simulations of oil cooling electrical transformers based on *CFD* simulations. Due to the complexity of real transformers an approximate representation was conceived leading to an axissymmetric model with the fin zone represented by volumetric momentum and heat sinks.

The definition of grid refinement was based on simplified test cases representative of the phenomena within the transformer, namely a cavity and channels. For the cavity and for the flow in channels the heat transfer was under predicted by about 5%.

The simulated temperature distribution of the transformer showed that the higher temperature is at the nucleus as was expected and revealed that the low voltage coils had larger temperatures than the high voltage ones. The radial temperature gradients in the coils are negligible but the axial temperature gradients are very important to transfer heat from the upper part of the coils to the lower part where the oil temperature in the channels is lower.

The calculated temperatures are over predicted when compared with experimental test results from Siemens, although in line with typical expected values. The differences between the fins location at the tank or at external panels show a small impact on the heat dissipation. The use of fin panels had some flow restriction level in the connections with the tank.

The increase of the channel thickness led to a reduction of up to 9ºC on the maximum temperature and was higher for the case of fins in panels. The influence of the supports of the coils is smaller but it reduces the restrictions where the velocities are higher.

The influence of the thermal conductivity of the coils is very important since when it is lower, for wire compared to foil conductor coils, the heat flux is almost uniform to the oil while for higher conductivity it presents large variations with very high values close to the oil inlet. This information is very important for design where assumptions are made for the temperature or heat flux distribution.

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