Open Access

Contemporary Research Analysis Journal

e-ISSN: 3050-5909
p-ISSN: 3050-5895

Volume 02 Issue 11 November 2025
CrossRef DOI: 10.55677/CRAJ/03-2025-Vol02I11

Page no: 694-709

The Cooling Problem in a Very High-Power Electrical Transformer: Contribution to Heat Flow and Transfer Predictions in Innovative Technologies

Emanuel Eduardo Pires Vaz
Plant engineer, professor of applied mathematics by FEUP, IEEE, NYAS.

ABSTRACT: This article provides an overview of the current state of knowledge regarding heat transfer technology in high and very high power transformers, focusing on transformer cooling, heat dissipation without insulation deterioration, and the calculation of temperatures for the hottest points of the winding. It also discusses the rotation of the heat flux field, which induces additional forces that alter the heat transfer pattern of the wall channel, causing secondary radial fluxes with high temperature gradients. Erosion of ceramic and metallic materials on the lamination surfaces is outlined. Reference is made to the cooling of the surfaces of the cooling fluid tank and cooling tubes, as well as the hottest points and temperature transients. Attention is given to heat dissipation in the cooling fluid medium. Finally, calculations of the transient response are made without considering the delay due to the transport of the insulating fluid, and the influence of this in a very high power transformer is noted.

KEYWORDS: Bronze; Synthetic seawater; Passivation disruption; Buffering; Deoxygenation

1 INTRODUCTION

Energy losses in a transformer manifest as heat in the core and coils and some secondary radial fluxes caused by high temperature gradients. This heat must be dissipated without allowing the windings to reach a temperature that causes excessive deterioration of the insulation.

For the user, temperatures in a transformer are important for determining the amount of overload and the time of application of that overload, as well as the amount of "service life" of the transformer that will be destroyed by operation at various temperatures. The designer must be able to predict the temperature at all points in a transformer. Temperature determination and evaluation must be taken into account at all stages of the transformer's service life. The heat paths in a transformer are so complex that precise calculations of all temperatures within the transformer are not practical. Since temperature is a primary factor in determining the service life of the transformer insulation, we must pay the utmost attention to determining its value and finding ways to design it economically, within certain limits.

To calculate the temperature of the hottest point in a winding, the process is as follows:

- Calculation of the temperature drop in the copper through the insulation.
- Calculation of the temperature drop between the coil surface and the coolant.
- Calculation of the coolant temperature rise.

High-power transformers are necessarily immersed in a liquid or gas, and the heat from the core and coils must flow into this fluid. We will analyze the problems of heat flow transfer in fluids involved in the design and maintenance of transformers. We present several theoretical topics and technical comments from experience on the cooling of high and very high power electrical transformers.

2. EXPERIMENTAL DETAILS

In order to review the heat transfer technology for internal passages it is important to first look on the actual design of blades for transformer cooling. Figure 1 shows a typical example of such blades.

The external heat transfer of the blade provides the boundary conditions for the internal cooling problem. If one considers the external heat transfer at midspan of a blade which is subjected to pure convective heat transfer without film cooling, a lot of experimental data are available. This is even true if we consider additionally roughness effects on the blade surface. If the blade is cooled additionally by applying film cooling, the situation is much more complicated. The heat flux into the surface is given by

$$q_F = h_F(T_{aw} - T_{Surf}) \quad (1)$$

The heat flux q_F should always be smaller than q_0, which is the heat flux without film cooling:

$$q_0 = h_0 (T_{r\infty} - T_{rw}) \quad (2)$$

Otherwise, film cooling would have a negative impact on the overall cooling of the blade. Injecting cold air into the boundary layer will basically have two different effects: On the one side, the coolant injected into the main stream will reduce the driving temperature difference for the heat transfer. This effect can be expressed, in a non dimensional form, by the adiabatic film cooling effectiveness:

$$\eta_{ad} = (T_{aw} - T_{r\infty})(T_c - T_{r\infty}) \quad (3)$$

On the other side, the cooling jets entering and disturbing the boundary layer will increase the heat transfer coefficient, so that h_F is usually larger than h_0. Because of the growing importance of film cooling for blade design, the subject has been studied extensively over the past forty years and a larger part of the results is available in the open literature. Several review articles concentrate on flat plate configurations with film injection through slots, cylindrical or shaped holes. Because of the larger number of parameters influencing the film cooling, such as hole geometry, blowing and momentum flux ratio, density ratio, temperature ratio, angles of the cooling holes, curvature effects, upstream boundary layer effects, main stream turbulence effects, effect of main stream pressure gradient and the effect of surface roughness and system rotation, numerical methods and correlations have been developed to predict the adiabatic film cooling effectiveness and the increase in heat transfer coefficients. Several numerical methods can be found in literature, ref 7,13,19.

See in figure 1 some theoretical topics, comments and practical technical valuable data on transformer cooling are the aim of the present study.

After the external boundary conditions are known, the internal heat transfer can be discussed. The common methods for enhancing the internal heat transfer coefficients have to be chosen by the designer and are based on the cooling requirements, internal Reynolds number, and space limitations and manufacturing aspects; see ref 8,9,10,12, 15,18.

As it can be seen from fig 1, the radial aligned coolant channels are connected by 180º turns. Current trends are towards increasing number of channels, so that the flow and heat transfer characteristics around bends and turns are of high interest.

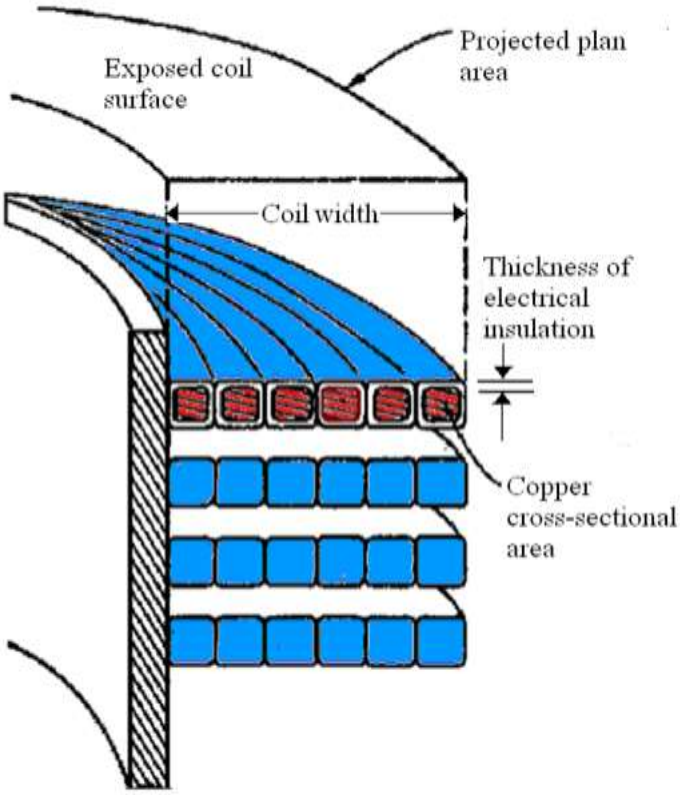

Fig. 1. Paths of heat flow in sandwiched coils

Figure 2 show us flow visualization for a 180º bend. The flow characteristics in the bend region are quite complex and secondary flow and separation zones lead to complex three-dimensional flow structures, and are highly geometry dependent.

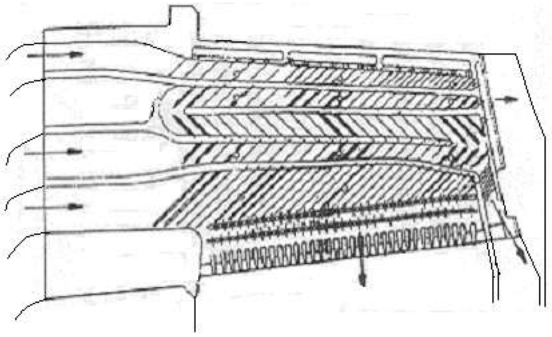

Fig. 2. Typical cooling characteristics of a modern transformer fin.

3. RESULTS AND DISCUSSION ON ANOTHER APPROACH

Rotation induces additional forces on the flow field and alters the flow pattern and heat transfer distribution in the internal cooling channels. Coriolis forces introduce cross stream secondary flows, which cause different heat transfer pattern on the individual channel walls. Buoyancy forces are important at high rotational speeds and coolant temperature gradients and cause radial secondary flows. On the "high pressure side" heat transfer is generally increased above the stationary value and on the "low pressure side" heat transfer is generally decreased. See figure 3 and ref. 15.

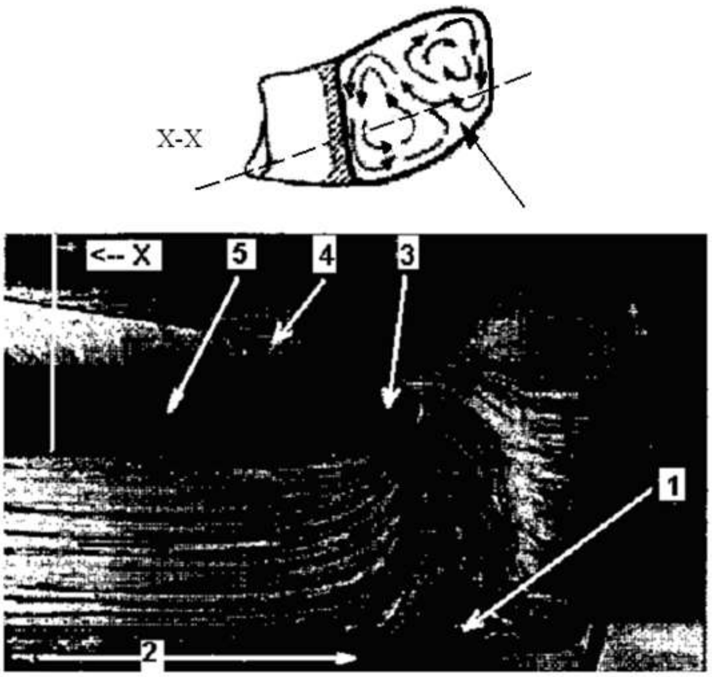

Fig. 3. Flow visualization for a flow in a bend: 1 Regions of low velocity.2 Flow direction.3 Area of flow separation. 4 Separation line. 5 Flow attachement point

Figure 4 shows us the concept of a new combined cooling system. It consists of a heat exchanger, coolant transportation lines. Cooling air is pre-cooled and/or heat pipes are introduced to the lines, in either case, the heat exchanger is used as a heat releasing device. This system is based on the idea that coolant mass flow rate can be reduced when the coolant is pre-cooled by introducing as heat exchanger.

For transformer maintenance, it is inevitable that particles floating in the atmosphere enter the engine due to the difficulty of inlet filtering. The particle motion is determined by the interaction and particle and boundary impaction. The erosion rate is strongly depended on the particle velocities, size, density and impact angles. This erosion produces oxidized aluminium particles.

If the mass ratio of erosion particles is small, the particles are known to be negligible effects on the oil flow.

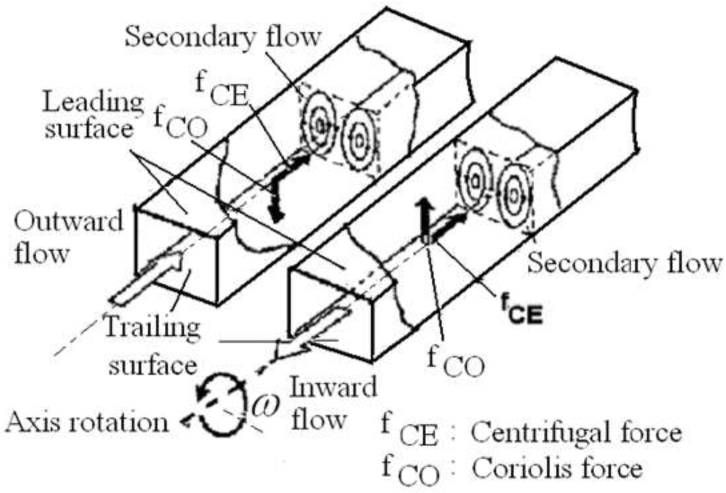

Fig. 4. Effect of Coriolis and Centrifugal forces on the flow

The particle flow field is analyzed to trace their trajectories by Lagrangian method. The governing equation of particle motion in oil flow is given by

$$\frac{d\vec{V}_P}{d\theta} = C_D \frac{R_{eP}}{24\, Stk}\left(\vec{V}_P - \vec{V}\right) \qquad (4)$$

Where

$$C_D = \frac{24}{R_e} \quad \leftarrow R_e < 1$$

$$C_D = \frac{24}{R_e}\left(1 + 0.0916\, R_e\right) \quad \leftarrow 0.1 \le R_e < 5.0$$

$$C_D = \frac{24}{R_e}\left(1 + 0.158\, {R_e}^{2/3}\right) \quad \leftarrow \quad 5.0 \le R_e$$

$$Stk = \frac{\rho_P\, d_P^2\, c_s}{18\, \mu\, L}$$

To calculate the particle impact possibility on the blade, the particle impact efficiency is defined as

$$\eta = \frac{number\ of\ \ particles\ colliding\ on\ the\ blade}{total\ number\ of\ \ particles\ \ floating\ in\ the\ \ flow} \qquad (5).$$

In equation 5, the denominator presents the total number of particles entering the calculation domain and the numerator presents the number of particles colliding on the pressure side surface of blade. Figure 5 presents the particle impact efficiency for several flow inlet angles. As the particle size, i.e., Stokes number, increases, the particle impact efficiency has a form of step function and there is a critical value of a particle size determining whether the particle impact occurs. However, actually the particle impact efficiency cannot follow the step function as shown in fig. 5.

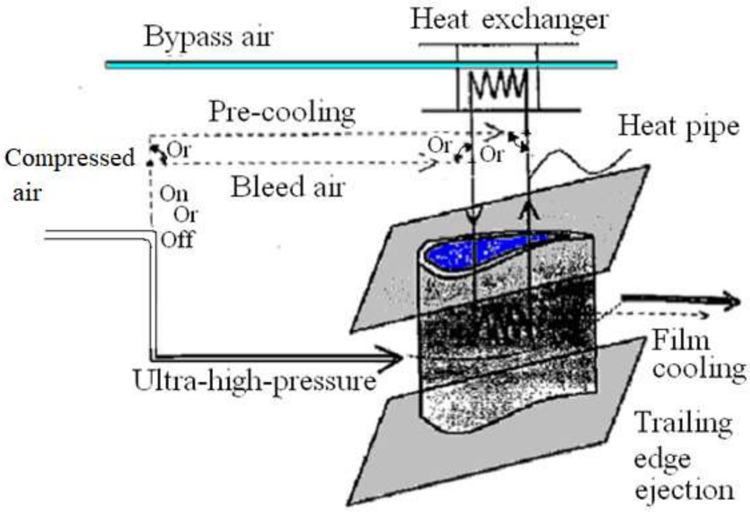

Fig. 5. A new combined cooling system

Figure 6 shows the erosion rates of two different materials, such as soft metal and ceramic, for the blade surface. The results show that erosion rates depend strongly on the surface material, particle impact angles, and particle sizes. In the figure, the erosion rate is given as a relative value for comparison. Erosion rates are very high along the pressure-facing blade surface for both materials. Erosion rates are very high at the leading edge due to the impact angle and the large variation in momentum of the impacting particles. The ceramic material exhibits erosion rates approximately 4 times higher at the leading edge because brittle materials generally exhibit a high erosion rate with large impact angles. Coating the blade's leading edge with ceramic material is not recommended.

Published numerical calculations on convective heat transfer on the outer surfaces of blades are performed considering either a uniform wall heat flux or a uniform wall temperature: these are the boundary conditions. Significant errors can occur in the calculated loads due to the lack of consideration of internal blade cooling. Figure 7 shows a sketch of the heat transfer from the hot oil side to the inside of the blade. An energy balance for the model in Figure 8 gives:

$$q = \frac{T_{2\,cell} - T_{1b}}{\frac{D}{2k_{air}} + \frac{1}{h_1} + \frac{t_{wall}}{k_{wall}}} \qquad (6)$$

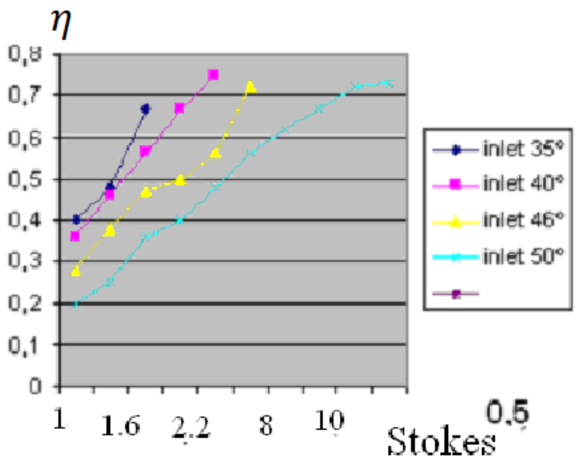

Fig. 6. Particle impact efficiency

The equations governing this study are solved by an explicit three-dimensional time-marching integration method for compressible flow using the Navier-Stokes equations. The cell-centered finite-volume method is used for discretization of the equations. For more calculation details, see reference 2 in the literature.

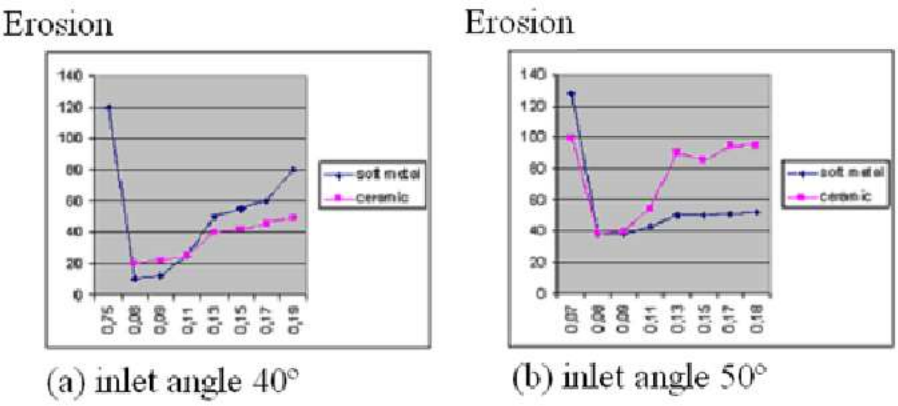

Fig.7. Erosion rates on the pressure side surface for two different materials at Stk=18.6

- The continuity equation describes conservation of mass:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left[\rho u_j\right] = 0 \qquad (7)$$

- The momentum equation:

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}\left[\rho u_i u_j + p\delta_{ij} - \tau_{ji}\right] = 0 \qquad (8)$$

- The energy equation:

$$\frac{\partial}{\partial t}(\rho e_0) + \frac{\partial}{\partial x_j}\left[\rho u_j e_0 + u_j p + q_j - u_i \tau_{ij}\right] = 0 \qquad (9)$$

- The total internal energy, e_0, is defined by:

$$e_0 = c_v T + \frac{u_i u_i}{2} \qquad (10)$$

- The stress tensor, τ_{ij}, for compressible flow reads

$$\tau_{ij} = \mu\left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right] + \xi \frac{\partial u_k}{\partial x_k}\delta_{ij} \qquad (11)$$

Where δ_{ij} is the Kronecker`s delta and ξ is the second viscosity, which is calculated as $\xi = -\frac{2}{3}\mu$.

In the energy equation the heat flux vector, q_j is given by Fourier's law:

$$q_i = -k\frac{\partial T}{\partial x_j} \qquad (12)$$

The temperature distribution in the blade metal is the result of several effects: internal heat transfer by convection, external convection, and conduction through the metal itself. Consequently, although the results of external flow heat transfer alone provide important information, they do not allow for the correct evaluation of the metal temperature.

An implicit Navier-Stokes solver, whose accuracy in applying heat transfer has been previously evaluated, was used for calculating the external flow. The Navier-Stokes equations are written in a general curvilinear coordinate system, $\xi\eta$:

$$U_t + (F - F_v)_\xi + (G - G_v)_\eta = 0 \qquad (13)$$

For the vector of conservative variables and for the flux vector we have:

$$U = J^{-1}[\rho, \rho u, \rho v, e] \qquad (14)$$

$$F = J^{-1}[\rho u, \rho\bar{u}u, \xi_x p, \xi_y p, (e+p)\bar{u}] \qquad (15)$$

$$G = J^{-1}[\rho v, \rho\bar{v}u, \eta_x p, \rho\bar{v}v, \eta_y p, (e+p)\bar{v}] \qquad (16)$$

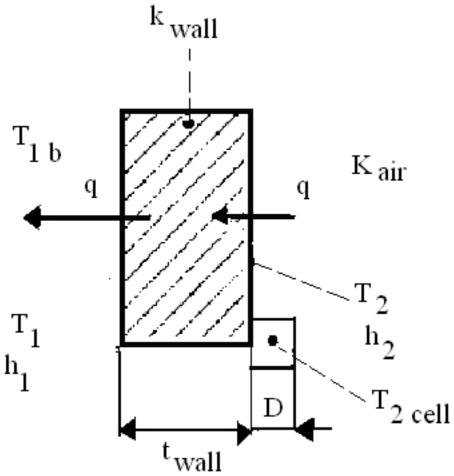

Fig. 8. Heat flux through a wall

The conduction equation within the solid was discretized into a finite volume formulation on an unstructured triangular grid. The integral formulation on each triangular cell is as follows:

$$\int_\Omega \frac{\partial T}{\partial t} = \int_{\partial\Omega} \frac{k}{\rho c}.\nabla T.n \qquad \int_\Omega \frac{\partial T}{\partial t} = \int_\Omega \frac{k}{\rho\, c}\nabla T\, n \qquad (17)$$

The implicit discretized Euler formulation is then:

$$\Delta T^n = \Delta t \sum_{1,3} A_i\, q_1 = \Delta t \sum_{1,3} A_i \frac{\rho\, c}{k}[\nabla(T^n + \Delta T^n)]n_i \qquad (18)$$

Where $\Delta T^n = T^{n+1} - T^n$ and A_i is the area of cell face **i**. Neumann boundary conditions are easily implemented due to the finite volume integral formulation. The coupling between the fluid and the solid is obtained through an exchange of boundary conditions. The fluid flow solutions, both internal and external, are obtained with an estimated temperature distribution along the wall; these two estimators are used as a boundary condition for the solid conduction problem, resulting in a new temperature distribution along the walls. You can see a portion of such a solution to the problem in reference 11.
An analytical solution to the Navier-Stokes equations in the fluid domain, Ω_f is used to solve the Dirichlet boundary conditions at the interface and the Poisson conduction equations in the solid domain, Γ with Neumann boundary conditions at the solid-fluid interface:

$$f_s(T) = 0 \quad \text{on} \quad \Omega_s \qquad \wedge \qquad k_s\frac{\partial T}{\partial n} - q_f = 0 \quad \text{on} \quad \Gamma \qquad (19), (20)$$

To solve this analytical problem applied to transformers I recommend ref. 7 where Dirichlet and Neumann problems have been treated and a particular case can be read in ref. 4.

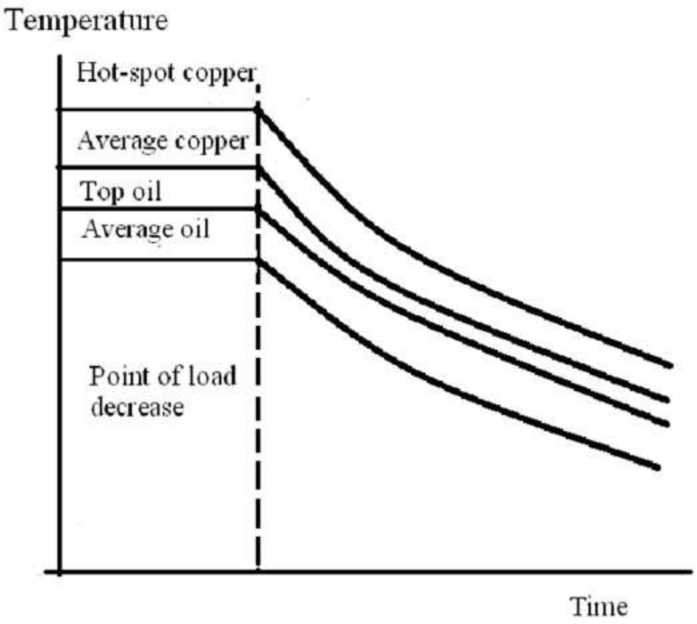

Fig. 9. Copper and oil temperatures after a decrease in load.

4. HEAT FLOW FROM OIL TO AIR WITH FORCED AIR OR FORCED - OIL COOLER.

There is a significant increase in heat transfer from the tank and radiators if air is blown over the surface with cooling fans. This application is generally done empirically due to the different natures of the tank surfaces, cooling tubes, radiators, and the fans themselves, and there is no basic formula to guide the application of the fans. There are many differences in the various forced oil-air cooler arrangements. Generally, two distinct arrangements are used:

• Use of tubular or finned radiators, which allow a large oil flow when the pumps are off, so that the transformer can operate at 100% load without fans or pumps, and at 167% load with fans and pumps.

• Use of a finned tube cooler, in which the oil and air flow is so restricted that very little heat is dissipated with pumps and fans out of operation. Typical cooling systems dissipate only about 5% of their rated capacity under these conditions, meaning that large transformers overheat just as quickly as if there were no cooling system at all.

5. TEMPERATURE TRANSIENTS AFTER LOAD CHANGES

The most important factor that determines the life of a transformer's insulation is the maximum winding temperature, that is, the hottest spot. The theory presented so far has considered all temperatures as constant, that is, under steady-state conditions. However, for transient conditions, temperature calculations must take more factors into consideration, since ultimate steady-state conditions may never occur, but maximum temperatures need to be determined.

Heating of conductor. The losses in the copper increases as the square of the current. For the first instant all the increment of losses goes toward heating the copper according to the following relation:

$$Temp.\,increases, ^\circ C = \frac{(watts\ increment)(time\ in\ sec\ )}{(lb\ of\ copper)(Effe\ cap)} \qquad (21)$$

Here we follow Montsinger and Clem, who determined the temperature rises and effective heat capacity of insulated copper.

$$\text{Effective thermal capacity} = 180 + 120\frac{A_i}{A_c} \qquad (22)$$

Where

A_c= cross sectional area of conductor

A_i= cross-sectional area of conductor insulation

Substituting in eq.21 we have:

$$Temp.\,increases, ^\circ C = \frac{(watts\ increment)(time\ in\ sec\ )}{(lb\ of\ copper)\left(180+120\frac{A_i}{A_c}\right)} \qquad (23)$$

6. DISSIPATION OF HEAT INTO THE COOLING MEDIUM

The temperature would continue to increase according to eq. 23 where it is not for heat dissipated into the cooling medium.

$$\Delta = \frac{1}{K}\left(\frac{watts\ dissipated}{area\ of\ winding\ surface}\right)^n \qquad (24)$$

Where:

n =0.8 for free convection and n = 1.0 for forced convection.

K = constant of proportionality for type of cooling.

Δ = temperature difference copper to oil ºC

Equation 23 accordingly may be written:

$$\frac{Temp.increases, ℃}{sec} = rate\ of\ rise\ temp = \frac{watts\ generated - watts\ dissipated}{(lb\ of\ copper)\left(180+120\,\frac{A_i}{A_c}\right)} \qquad (25)$$

Electricity laws permits us to write:

$$Watts\ generated, at\ 75℃ = I^2 \times (resistance, at\ 75℃)(1+K_e) \qquad (26)$$

Where $K_e = ratio\ eddy \frac{loss}{I^2} \times R, at\ 75℃$

$$\text{Watts generated}\ \ (\text{at}75℃ + \text{T}_\text{c}) = \text{I}^2 \times (\text{resistance, at } 75℃)\left(\frac{75+\text{T}_\text{c}+234.5}{75+234.5} + \text{K}_\text{e}\,\frac{75+234.5}{75+\text{T}_\text{c}+234.5}\right) \qquad (27)$$

Combining equations 23 and 28 and using mathematical notation we have:

$$\frac{\text{dT}_\text{c}}{\text{dt}} = \frac{\text{watts generated} - (\text{KT}_\text{c})^{1/\text{n}}(\text{area of winding})}{(\text{lb of copper})\left(180+120\frac{\text{A}_\text{i}}{\text{A}_\text{c}}\right)} \qquad (28)$$

Where "watts generated" may be calculated by eq. 27.

This equation presents a most awkward integration of **n** that is other than 1.0. Transformer engineers simply assume for this problem that **n=1.0** and K to give an approximately correct answer. Assuming the watts generated as a constant and equal to the value corresponding to an average expected value of T_c, the variables can then be separated and the integration performed:

$$T_c = T_{f\,c} - \left(T_{f\,c} - T_{i\,c}\right)exp(-C) \qquad (29)$$

The calculation of the constants gives us:

$$C = \frac{t\ K \times area}{(lb\ copper)\left(180+120\frac{A_i}{A_c}\right)} \qquad (30)$$

Where

$T_{f\,c} = \frac{\text{watts}}{area \times K}$

which represents the final winding rise over that would be attained at infinite time; $T_{i\,c}$ = initial winding rise, and $K = \frac{\text{watts}}{area \times T_{f\,c}}$.

7. HEATING OF THE COOLING LIQUID

A very good assumption for liquid-filled transformers is that the temperature of the cooling medium remains constant while the temperature of the copper increases. The liquid, however, absorbs heat from the copper and dissipates it into the air or water according to similar laws, and the temperature of the liquid will change in a manner similar to that in which the copper changed according to equation 29. The main difference is that, in the final equilibrium condition, the parts of the liquid that are in the heating and cooling zones reach, as shown, an average temperature approximately that of the oil, while a considerable mass of liquid above the coils reaches the maximum temperature of the oil.

The thermal capacity of the complete transformer is much greater than the thermal capacity of the winding, and is usually expressed in watt-hours per °C, rather than watt-seconds. The final oil temperature equation is similar to equation 29:

$$T_0 = T_{0\,i} + \left(T_{0\,f} - T_{0\,i}\right)\left[1 - exp\left(\frac{-t(watt\ loss)}{T_{0\,f}(thermal\ capacity)}\right)\right] \qquad (31)$$

Where $T_{0\,f}, T_{0\,i}$ are final and initial values of top oil temperatures.
Thermal capacity is the effective thermal capacity or the sum of 1.33 watt-hour per ºC per gal of oil +0.04 watt-hour per ºC per lb of tank +0.06 watt-hour per ºC per lb core and coil.

From these problems which were studied using a variety of research –tools we have conclude the last practical formulas ready to apply in transformer maintenance . Figure 9 is given in the ASA Guides for Loading C57.92-880.

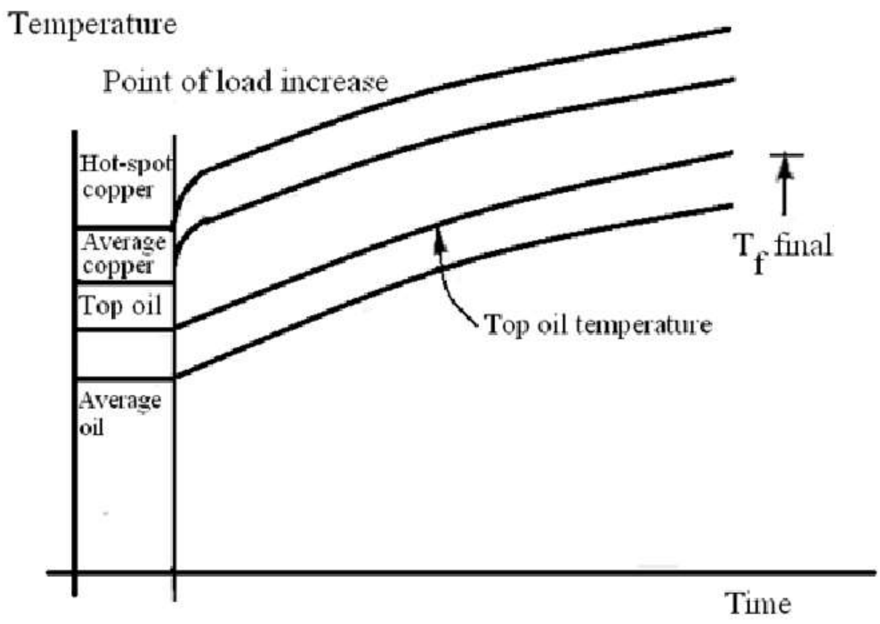

Fig.10. Copper and oil temperatures after an increase in load

8. CALCULATION of the temperature transient response without considering the delay due to the transport of the insulating fluid.
For many reasons, we cannot present the exact calculation, hence several approximations are presented. Here, we present a method that we use when studying nuclear power plants.
The power outputs are on the order of 125 MW. The calculation applies to a very high-power transformer with oil cooling and sporadically also nitrogen and air fans. All used by cooling layers; our model regarding the heat transfer equations perfectly adapts to what is studied in power plants, as I said.
Let's look at Figure 12 and derive the heat transfer equations:

$$\begin{cases} q = q_1 + q_2 \\ q_2 = q_3 + q_{out} - q_{in} \end{cases} \qquad (32)$$

$$\begin{cases} q = C_1 \frac{d\theta_U}{dt} + \frac{\theta_U - \theta_D}{R_1} \\ \frac{1}{R_1}(\theta_U - \theta_D) = C_2 \frac{d\theta_D}{dt} + C_2 \frac{\omega 1}{L}(\theta_D - \theta_{in}) \end{cases} \qquad (33)$$

Putting $\theta_U - \theta_{in} = V_1$, $\theta_D - \theta_{in} = V_2$ e $\frac{L}{C_2\omega_2} = R_2$,
the previous system can be written as:

$$\begin{cases} q = C_1\frac{d\,V_1}{d\,t}+\frac{1}{R_1}(V_1-V_2) \\ \frac{1}{R_1}(V_1-V_2) = C_2\frac{d\,V_2}{dt}+\frac{1}{R_2}V_2 \end{cases} \qquad (34)$$

where V_2 represents the temperature rise of the oil.

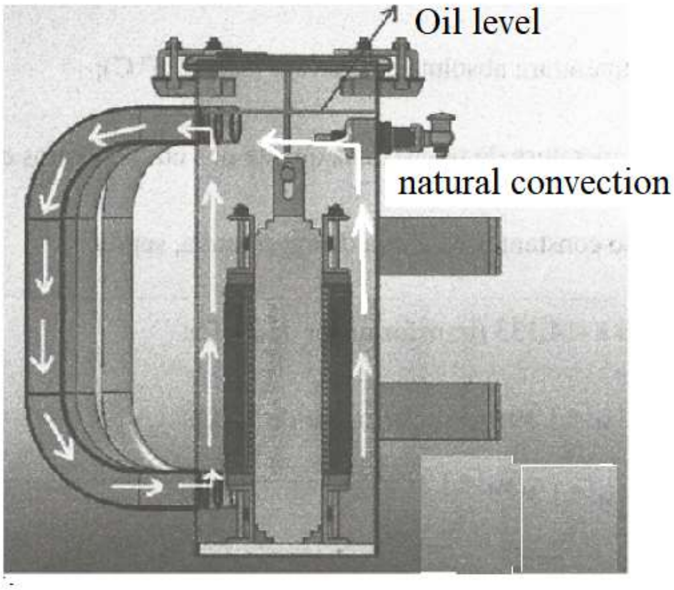

Fig. 11. Cooling in a power transformer

It is simpler to consider an analog electrical circuit, as in Figure 13.
The temperature variations of the water, due to a sudden drop in speed, can now be calculated. By decreasing the speed to a certain value ω_2, R_2 will correspondingly increase to a value that can be called ΔR.

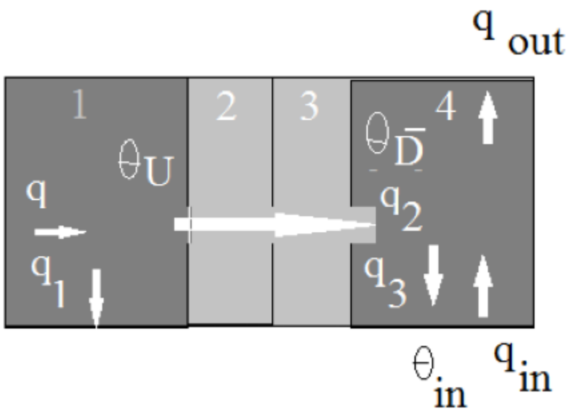

Fig. 12. Diagram for determining the heat transfer equations according to the notation in the text.

This can be simulated electrically by opening the switch in Figure 8. According to Thévenin's theorem, the variations in this system can be calculated by introducing a current q in S, which has the same amplitude but opposite direction to that which passed through S before being interrupted. Our goal now is to try to find the variation in V_2, which we will denote from u. According to Figure 14, the Laplace transform for u can be written as:

$$u = \frac{\frac{1}{pC_2}\left(R_1+\frac{1}{pC_1}\right)\Delta R\, q}{\left(R_1+\frac{1}{p\,C_1}+\frac{1}{p\,C_2}\right)\left[\Delta R+R_2+\frac{\frac{1}{p\,C_2}\left(R_1+\frac{1}{p\,C_1}\right)}{R_1+\frac{1}{p\,C_1}+\frac{1}{p\,C_2}}\right]}\cdot\frac{1}{p} \qquad (35)$$

This equation is due to Rydén.

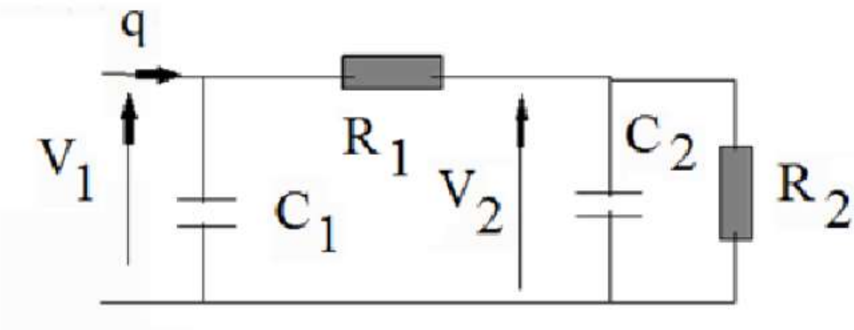

Fig. 13. Electrical circuit for the case of temperature variations expressed with respect to the last system of equations.

$$u = \left(1 - \frac{\omega_2}{\omega_1}\right)\frac{q}{C_2}\cdot\frac{p+\frac{1}{R_1C_1}}{p^2+\left(\frac{C_1+C_2}{R_1C_1C_2}+\frac{\omega_2}{\omega_1}\cdot\frac{w_1}{L}\right)p+\frac{\omega_2}{\omega_1}\cdot\frac{\omega_1}{R_1C_1L}}\cdot\frac{1}{p} \qquad (36)$$

The time function will have the form:

$$u(t) = A[1 - a\,exp(-\,\beta_1 t) - b\,exp(-\beta_2 t)]. \qquad (37)$$

To find the inversion, the constants in the last equation in **u** must be determined.

Determining the constants.

$$\boldsymbol{R_1 \approx R_U + R_{oil} + R_{revest.}} \qquad \textbf{(38)}$$

For the aluminum alloy tank, we have:

$$\mathrm{R_{revest} \approx 0{,}03 \times R_U} \qquad \textbf{(39)}$$

From which, disregarding this portion in relation to R_U, we have:

$$R_1 \approx R_U + R_{oil} \approx R_{oil}.$$

$$R_U = \frac{\theta_{U_{médio}} - \theta_1}{q} = \frac{273+31-(273+27)}{125\times10^6} = 0{,}024 \times 10^{-6}\ K/W \qquad (40)$$

For reference oils, the web search (Paratherm HE, 40°C) gives the conductivity values: $0{,}131\,\frac{W}{m\,°K}$.

For an oil height of approximately 2 meters, we have:

$$R_{oil} = 15{,}2672\ °K/W$$
$$\boldsymbol{R_1 \approx 15{,}2672\ °K/W.}$$

The transformer temperature is assumed to vary between 27°C and 35°C.

The thermal capacities of copper and oil are:

$$\mathrm{C_1 = c_U\rho_U A_D L = 0{,}0055 \times 8{,}925 \times 1{,}5 \times 0{,}6 \times 2 = 0{,}0884\ Ws/K} \qquad (41)$$

$$C_2 = c_D\rho_D A_{D_2} L = \begin{cases} 2000 \times 850 \times 45 \times 10^{-4} \times 2 = 15300\,\frac{Ws}{K} \leftarrow stage\ 1 \\ 2000 \times 850 \times 60 \times 10^{-4} \times 2 = 20400\,\frac{Ws}{K} \leftarrow stage\ 2 \end{cases} \qquad (42)$$

Where ρ is the density, c the specific heat, and A_D the cross-sectional area of the oil. The Laplace transform of the temperature transients of the oil exiting the tank and then cooled to ambient temperature in the two stages can be written as:

$stage\ 1\ (65\ MW)$,

$$u[p] = 4238{,}37\left(1-\frac{\omega_2}{\omega_1}\right)\frac{p+0{,}7410}{p^2+\left(0{,}7410+\frac{\omega_1}{2}\cdot\frac{\omega_2}{\omega_1}\right)p+\frac{\omega_1}{2{,}6992}\cdot\frac{\omega_2}{\omega_1}}\cdot\frac{1}{p} \qquad (43)$$

$125\ MW\ Stage$,

$$u[p] = 6127{,}45\left(1-\frac{\omega_2}{\omega_1}\right)\frac{p+0{,}7410}{p^2+\left(0{,}7410+\frac{\omega_1}{2}\cdot\frac{\omega_2}{\omega_1}\right)p+\frac{\omega_1}{2{,}6992}\cdot\frac{\omega_2}{\omega_1}}\cdot\frac{1}{p} \qquad (44)$$

These equations do not take into account the delay in coolant transport; we will highlight the influence of oil transport delay on the results later. The problem can be solved in reference 19 at some point. If we use the Arrhenius formula given in this text for the service life of a continuously operating transformer, the conditions involved in the cooling design are severe, but worthwhile; the average lifespan of a transformer in terms of cooling is approximately 73 years. We should always follow the manufacturer's maintenance program. Here we use the following notation:

θ_U, average temperature of the copper core; θ_D, average temperature of the oil; θ_{in}, temperature of the incoming oil; q, power developed; q_1, power required to create transients in the transformer core temperature; q_2, power output through the tank and cladding (aluminum); q_3, power supplied to the cooler; $q_{out} - q_{in}$, cooler power; R_1, thermal resistance of the transformer material; C_1, thermal capacity of the core and winding; C_2, thermal capacity of the coolant; w_1, normal flow rate of coolant ($\left(3{,}33\ \frac{m}{s} for\ 65\ MW\right)$ and ($5{,}5\frac{m}{s} for\ 125\ MW$); L, height of the coolant column (2 meters).

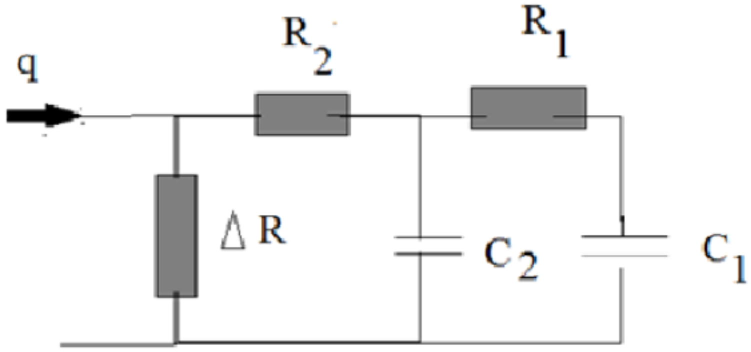

Fig. 14. Analog electrical circuit for calculating temperature variations expressed in the equation for **u**

9. THE INFLUENCE ON COOLANT TRANSPORT DELAY CALCULATIONS IN A VERY HIGH POWER TRANSFORMER

The Rydén transient time function for an analogous problem, even at the powers used in a D_2O reactor, can be expressed by:

$$u(t) = \left(1-\frac{\omega_2}{\omega_1}\right)\frac{q}{C_2}\ f(t) \qquad (45)$$

This study, although far from complete, aims to give an overview of the current state of the art in the field of heat transfer technology in a high power transformer. Over the last forty years, the inlet temperature for industrial transformers has been steadily increasing to increase thermal efficiency and overload requirements. It is subjected to a strong thermomechanical load that affects the durability of the transformer. The design of a modern industrial transformer requires specialized knowledge in thermodynamics, heat transfer, stress analysis, materials science and manufacturing technology, and the designer tries to balance the often conflicting requirements of heat transfer versus stress, ease of manufacture and costs. The work described in this article is an experimental investigation of heat transfer from the main flux to the surface of the cover, which may be applicable to ceramic and metal transformer heat sinks. Heat transfer measurements were performed for the experimental conditions of a uniform flux or a uniform wall temperature.

Power losses in a transformer appear as heat in the core and coils. This heat must be dissipated without allowing the windings to reach a temperature that causes excessive deterioration of the insulation.

For the user temperatures of a transformer, these are important for determining the amount of overload and overload time that can be applied, as well as how much of the transformer's "service life" has been or will be destroyed by operation at various temperatures. The designer must be able to predict the temperature at all points in a transformer. Temperature determination and evaluation must be taken into account at all stages of the transformer's life. The heat path in a transformer leads us to calculations so complex and requiring precision of all temperatures within the transformer, which is not practical. Since temperature is a primary factor in determining the life of transformer insulation, it is essential to make every effort to determine its value and find economical ways to design within those limits.

To calculate the temperature of the hottest point in a winding, proceed as follows:

• Calculation of the temperature drop through the insulation in the copper.

• Calculation of the temperature drop between the coil surface and the cooling medium.

• Calculation of the temperature rise of the cooling medium.

Large transformers are necessarily immersed in a liquid or gas, and the heat from the core and coils must flow to this medium. We will analyze the fluid transfer problems involved in the design and maintenance of transformers. Experiments were carried out in factory laboratories to confirm theoretical treatments on fluid heat transfer. See Figure 10 for some theoretical topics, comments, and valuable practical technical data on transformer cooling, which are the objective of this study.

In the continuation of our study, we will rely on modern complex cooling systems that include sophisticated fin systems and cooling fluid film..

Figure 11 shows the heat flow paths in a highly cut fin. The study of this type of fin was carried out in the work of doctoral students Maria da Conceição de S. James and S. Teresa C. D. and Maria Drost, with Professor A. A. Pires de Carvalho from the University of Porto as a visiting professor, for aggregation at our University of Jerusalem a Nova. Following these studies, the esteemed professor presented a value for q, the fulcrum of these studies, which we will mention below (Eq. 46).

The published numerical calculations of convective heat transfer on the outer surfaces of the fins are performed for a uniform wall heat flux or a uniform wall temperature boundary condition. Significant errors can occur in the calculated loads if the internal cooling of the fin is not considered. An energy balance for the model in Figure 12 gives:

$$q = \frac{T_{2\,cell} - T_{1b}}{\frac{D}{2k_{air}} + \frac{1}{h_1} + \frac{t_{wall}}{k_{wall}}} \qquad (46)$$

The process equations are solved by an explicit 3D time march method for compressible flow bases in the Navier-Stokes equations. For the discretization of the equations, the cell-centered finite volume method is used. For more details, consult the literature.

Considering the refrigerant film, the delay in its transport necessitates a new heat balance problem. The heat flux in this case is:

$$q_F = h_F\left(T_{aw} - T_{Surf}\right) \qquad (47)$$

The heat flux must always be less than the heat flux without cooling of the film.

$$q_0 = h_0(T_{r\infty} - T_{rw}) \qquad (48)$$

Otherwise, cooling the film would have a negative impact on the overall cooling of the fin. Injecting cold air into the boundary layer will basically have two different effects:

- On the one hand, the coolant injected into the main flow will reduce the temperature difference in the direction of heat transfer. This effect can be expressed, in a non-dimensional way, by the adiabatic effectiveness of the film cooling:

$$\eta_{ad} = (T_{aw} - T_{r\infty})(T_c - T_{r\infty}) \qquad (49)$$

- On the other hand, cooling jets that enter and disturb the boundary layer will increase the heat transfer coefficient, so it is generally higher. Due to the increasing importance of film cooling for fin design, the subject has been extensively studied over the past forty years, and a greater portion of the results is available in the open literature. Several review articles focus on flat plate configurations with film injection through slots, cylindrical holes, or other forms. Due to the large number of parameters influencing film cooling, such as hole geometry, blow and momentum flow ratio, density ratio, temperature ratio, cooling hole angles, curvature effects, rising boundary layer effects, main flow turbulence effects, main flow pressure gradient effect, and surface roughness and system rotation effect, numerical methods and correlations have been developed to predict the effectiveness of adiabatic film cooling and the increase in heat transfer coefficients. Several numerical methods can be found in the literature.

Once the external boundary conditions are known, internal heat transfer can be discussed. Common methods for improving internal heat transfer coefficients should be chosen by the designer and are based on cooling requirements, internal Reynolds number, and space and manufacturing limitations; see literature.

The quotient: $\frac{q_F}{q_o} = \frac{h_F(T_{aW} - T_{surf})}{h_0\,(T_{r\infty} - T_{rW})}$ (50)

represents the influencing factor of coolant transport delay in a modern, ultra-high-power transformer. Rotation induces additional forces in the flux field and alters the flow pattern and heat transfer distribution in the internal cooling channels.
The following notation was used in this paragraph:

q_F heat flux; h_F heat transfer coefficient in the presence of film cooling; $T_{a\,W}$ adiabatic wall temperature; T_{surf} component surface temperature; q_0 heat flux without film cooling; $T_{r\,W}$ wall recovery temperature; $\eta_{a\,d}$ adiabatic film cooling effectiveness; T cell temperature; K thermal conductivity; C solid heat capacity; K_f fluid thermal conductivity; K_s solid thermal conductivity; q_j heat flux vector.

10. FINAL CONSIDERATIONS

This article summarized some recent research with applications in the flow of heat transfer in an oil-to-oil cooling system with compressed air of an electrical transformer. It is hoped that the references used here will adequately direct the interested reader to more details on the various works already published. The heat transfer characteristics on the surface of the casing are currently being investigated by us and many other research projects in doctoral programs at our university, with support from companies in the field. The casing may be subject to severe thermal stresses due to the large variation in heat transfer coefficients.

Although an exact analysis of temperatures and forces is difficult due to the small magnitudes of the temperature and force differences, only a partial analysis is possible, as some facts are evident.
The forces are due to differences in oil density, which in turn are due to differences in oil temperature.
The hot oil column starts at the bottom of the coil and extends to the top, the oil being heated to its maximum as it rises through the coil, but remaining at its maximum temperature until the top.
The colder oil column starts at the top of the cooling surface and extends to the bottom. These considerations mean that the hot oil column will have the least possible weight if the heating area can be located at the very bottom of the tank, so that the hot oil column can extend from the bottom to the top. Similarly, the cold oil column would have the greatest possible effect if the cooling area could be located at the very top of the oil column.

ACKNOWLEDGEMENTS

I thank my sons João Pedro and Hugo Vaz for their kindness in participating in some tests and collecting some data that we used in this article, and my wife, who reviewed the translation and always provides ongoing support in my studies. I also thank the dean of my university organization, Pantaleão, for the commitment shown in my participation in this matter.

REFERENCES

1. Clem, J.E.: Equivalent Circuits of Regulating Transformers, Trans. AIEE vol. 58, p. 871, 1939.
2. Dahlander, P. 1998. An engineering tool for film cooling simulations. Thesis for the degree licentiate of engineering 98/10. Department of Thermo and Fluid Dynamics, Chalmers University of Technology, Göteborg, Sweden.
3. Emanuel E. Pires Vaz; João P. B. Vaz; Hugo F.B.Vaz. Moldes Cunhos e Cortantes, Liv. Lopes da Silva Editora.
4. Emanuel Eduardo Pires Vaz. Tratamento matemático da elasticidade duma barra homogénea. Gazeta de Matemática. Lisboa nº 133-136, 1975-76.
5. Emanuel Eduardo Pires Vaz, João Pedro Barbosa Vaz, Hugo Filipe Barbosa Vaz. O problema do arrefecimento no projeto de um transformador elétrico de muito alta potência. QSP,7/8 2017 Portugal.
6. Emanuel Eduardo Pires Vaz, João Pedro Barbosa Vaz, Hugo Filipe Barbosa Vaz, Maria Arminda R.B. P. Vaz. O problema do arrefecimento no projeto de um transformador elétrico de muito alta potência – Parte 2. QSP, Nov 2021 Portugal.
7. Goldstein, R.J. 1971. Film Cooling. In Advances in Heat Transfer. Irvine, T.F. and. Hartnett Eds. Academic Press. New York 7: 321-379.
8. Harasgama, S.P. 1995.Aerothermal Aspects of Gas Turbine Flows. VLI-LS 1995-05. Heat Transfer and Cooling in Gas Turbines.
9. Hennecke, D.K. 1984. Heat Transfer Problems in Aero- Engines. Heat and Mass Transfer in Rotating Machinery, Hemisphere. Washington. 353-379.

10. Hall, K., Johnson, B., Weigand, B. & J.Richardson. 1995. An Advanced Blade Design for V84.3 Gas Turbines, VDI Berichte, 1185: 257-275.
11. Kato, M. and Launder, B.E. 1993. The modelling of turbulent flow around stationary and vibrating square cylinders. In Ninth Symposium on Turbulent Shear Flow. Vol. 1: 10-4-1 10-4-6. Kyoto.
12. Kays, W.M. & M.E. Crawford. 1993. Convective Heat and Mass Transfer. Mc Graw –Hill, Inc., New York.
13. Leontiev, A.I. 1999. Heat and Mass Transfer Problems for Film Cooling. J. Heat Transfer. 121: 509-527.
14. Maria Augusta Ferreira Neves. Matemática 12. Porto Editora.
15. Mochizuki, S.; J. Takamura,S.Yamawaki, W.J.Wang.1992. Heat transfer in serpentine flow passages with rotation.92-GT-190.
16. Notes on Física Complementar of J.M.M. R. Araújo 1970, UP.
17. Rogério S.S.Nunes. Computação de equações diferenciais.1967, CUP.
18. Schulenberg.T., Kopper, F. & J. Richardson. 1995. An Advanced Blade Design for V 84.3 Gas Turbines, VDI Berichte, 1185: 257-275.
19. VKI Lecture Series. 1982. Film Cooling and Turbine Blade Heat Transfer.VKI-LS 82-02.
20. Yeh, F.C. & F.S. Stepka.1984.Review and Status of Heat- Transfer Technology for Internal Passages of Air-Cooled Turbine Blades. NASA Technical Paper 2232: 1-33.