CALCULATION OF MECHANICAL STRESSES IN HERMETICALLY SEALED TRANSFORMERS

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ABSTRACT

In this paper, firstly a corrected model is suggested for oil temperature rise in hermetically sealed transformers and based on this model, oil pressure increase and resulted mechanical stresses in ribs of radiators are calculated. Using this approach and the dynamic temperature rise simulation method represented in IEEE C57.91-1995, fatigue problem due to dynamic stresses are analyzed and its effect in design and choose of appropriate tank and rib material is treated. This suggested method is applied to a 500kVA sealed type transformer manufactured in Iran and accuracy of calculations is investigated.

INTRODUCTION

One of the most important problems in operation of conservator type distribution transformers is gradual deterioration of transformer insulating oil affected by moisture and environmental pollutants existing in air. Conventional dryer materials such as silica gel are not suitable in contaminated areas and climates with high humidity and they are saturated and lose their quality in a short time. It is obvious that the deterioration of oil properties can jeopardize transformer and decrease its life time.

Hermetically sealed transformers, as another type of oil-immersed transformers for use in contaminated areas, have been suggested in 1970s. This type of transformers, don’t have conservator and the transformer oil cannot expand easily. In hermetically sealed transformers, the oil pressure is higher than the atmosphere pressure and despite of conservator type transformers, the tank of these transformers act as a pressurized vessel. Due to lack of any contact between oil and environment, aging of oil will be deferred and the transformer encounter lower risk in comparison with conservator type transformers. Because of these advantages, in some standards, use of hermetically sealed transformers up to 1600 kVA is compulsory in oil, gas and petrochemical plants especially in onshore or offshore installations [1].

As mentioned above, tank of hermetically sealed transformers must tolerate higher pressure and due to elimination of conservator, oil temperature variations leads to expansion and contraction of transformer tank and so this affects transformer tank design, behavior and aging.

Transformer tanks are manufactured in corrugated form for efficient heat transfer and oil cooling. In hermetically sealed transformers, the ribs of corrugated tank have the essential rule in cooling and tolerating pressure variations. In this paper, firstly a corrected model is represented for heat transfer problem in oil immersed transformers and pressure fluctuation mechanism is modeled and based on this model mechanical stresses and fatigue problem in transformer tank are investigated.

OIL TEMPERATURE INCREASE MECHANISM

Generally, Oil temperature increase models in transformers are categorized in two approaches:

1. Lumped Modeling that oil temperature is calculated in definite points of transformers (usually in bottom, top and middle points).

2. Distributed Modeling that based on solving Laplace differential equation in all points. This model requires complicated numerical methods such as FEM.

Although second method is more accurate than first one, but the lumped model is accurate enough in majority of practical applications and has been accepted as a standard method [2,3]. So in this paper, lumped modeling is used for analysis Oil temperature increase.

It is clear that transformer MV and LV windings and core losses are the main sources of heat and this generated heat increases the oil temperature. This oil temperature increase leads to increase in oil pressure in hermetically sealed transformers. As shown in figure 1, this generated heat is transferred by oil and through transformer tank ribs and other metal parts dissipated in environment. In this condition, the oil temperature variation can be represented by below differential equations [4, 5]:

\[ P_{le} + P_{cu} = m_o c_o \frac{d\theta_{TO}}{dt} + h_o (\theta_{TO} - \theta_{CT}) \]

\[ h_o (\theta_{TO} - \theta_{CT}) = \frac{A_L}{T} (\theta_{CT} - \theta_{CO}) \]

\[ \frac{A_L}{T} (\theta_{CT} - \theta_{CO}) = h_A (\theta_{CO} - \theta_A) \]

where:

- \( P_{le} \) and \( P_{cu} \) : Core and Winding losses (Watts)
- \( m_o \) : Oil weight (kg)
- \( c_o \) : Oil Specific Heat Capacity (J/kgK)
- \( h_o \) : Oil and Air Convectional Heat Transfer Coefficient respectively (W/m²K)
- \( \lambda \) : Transformer Tank Conductive Heat Transfer Coefficient (W/mK) and Tank Thickness (meter)
- \( \theta_{A}, \theta_{CO}, \theta_{CT}, \theta_{TO} \) : Ambient temperature, Tank exterior and
HEAT TRANSFER MODELING IN STEADY STATE CONDITIONS

In steady state conditions, temperature variation versus time will be zero. It is important that heat transfer coefficient in equation (1) is a function of oil temperature and this dependency is expressed by an experimental coefficient i.e. Nusselt Number [5]:

\[ h_o = \frac{N_u \times k_o}{L} \]  

(2)

Which \( k_o \) is oil conductive heat transfer coefficient (W/mK) and L is length of transformer tank rib. Nusselt number in equation 2 is expressed based on Prandtl and Grashof numbers as follows:

\[ N_u = C (Pr \times Gr)^n \]  

(3)

\[ Pr = \frac{c_o \times \mu_o}{k_o} : Prandtl Number \]

\[ Gr = \frac{L^3 \rho_o \cdot \beta_o}{\mu_o^2} : Grashof Number \]

Which:

- \( g \): Gravity Constant
- \( \mu_o \): Oil Dynamic Viscosity (kg/sec)
- \( \rho_o \): Oil density (kg/m³)
- \( \beta_o \): Oil thermal expansion coefficient (1/K)

The numbers of C and n are chosen based on fluid conditions corresponding table (1)[6]:

<table>
<thead>
<tr>
<th>Fluid condition</th>
<th>n</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>0.25</td>
<td>0.59</td>
</tr>
<tr>
<td>Turbulent</td>
<td>0.33</td>
<td>1</td>
</tr>
</tbody>
</table>

In equation 3, oil characteristics such as density, viscosity, thermal capacitance and thermal expansion coefficient are function of oil temperature and these parameters are represented in tables [5,6]. However it can be expressed by relatively precise equations. These equations may be extracted from oil properties tables by interpolation method. In this article, below equations are suggested for this purpose:

\[ c_o = a_1 + b_1 \theta_o \]

\[ \rho_o = \rho_o e^{-b_2 \theta_o} \]

\[ \mu_o = \mu_o \left( \frac{k}{\theta_o + 273} \right) \]

\[ k_o = a_2 - b_2 \theta_o \]

Multipiers and exponent in equation (4) are dependent of oil type and may be supplied by oil manufacturer. Practically oil thermal expansion coefficient is closely constant in wide range of transformer operation temperatures and we adopt it as a constant value. As shown in figure 1, generated heat by windings and core of transformer, is transferred through oil in convectional form to transformer tank and ultimately to air. Heat transfer mechanism is dominantly conductance in metals and conductive heat transfer coefficient is independent from temperature variations. This fact has been considered in equation 1.

In existing heat transfer models of transformer, the effect of tank is ignored [6,7,8] but in this article we consider tank conductance in suggested model. Heat transfer mechanism in air is essentially convection and the radiation is negligible. It should be mentioned that, behavior of heat transfer in air is turbulent. In transformer oil, in spite of air, heat transfer behavior is laminar caused by low viscosity variations versus temperature. Behavior of fluids is important in selecting C and n in equation (3) and calculating of Nusselt Number.

Definition of heat transfer coefficient of air is also similar to oil, but this coefficient is approximately constant for air in wide range of temperature variation [5]. Also other parameters in equation (4) unless air thermal expansion coefficient is nearly constant. Air thermal expansion coefficient is the reverse of average temperature of inner and outer sides of transformer tank [6]:

\[ \beta_A = \frac{2}{\theta_{CO} + \theta_A} \]  

(5)

Other equations for air are similar to oil:

\[ h_A = \frac{N_u \times k_A}{L} \]

\[ N_u = C (Pr \times Gr)^a \]  

C = 0.1  \ n = 0.33

\[ Pr = \frac{c_A \times \mu_A}{k_A} : Prandtl Number \]  

(6)

\[ Gr = \frac{L^3 \rho_A \cdot \beta_A}{\mu_A^2} : Grashof Number \]

Which:

- \( c_A \equiv const. \rho_A \equiv const. \mu_A \equiv const. k_A \equiv const \)

Now, for calculating of oil average point temperature in steady state condition, temperature derivation is assumed.
It is obvious that (7) is a non-linear equation and so we need to use an iterative method for calculating of oil temperature. By determination of this parameter, it is possible to determine the temperature of other places in transformer. It is necessary to choose an initial value for oil temperature and based on this initial selection.

**CALCULATION OF OIL EXPANSION IN HERMETICALLY SEALED TRANSFORMER**

Now, we can estimate approximately increasing of oil volume due to temperature rise. Regarding the relationship between oil density and its volume, oil expansion can be calculated as follow:

\[
\Delta V = \frac{m_o}{\rho_o((\theta_{TO1})^2)} - \frac{m_o}{\rho_o((\theta_{TO2})^2)}
\]

(8)

That \(\Delta V\) is volume variation due to temperature variations. Based on this volume increment, oil pressure increment can be obtained in different operational conditions dependent on load and ambient temperature variations. As a result, regarding these calculations, it is possible to investigate tank strength and durability against pressure tensions. For this purpose, we select one of tank ribs and treat the effect of pressure increase on exerted tensions. As shown in figure (2), we assume that the distribution of applied force due to oil expansion is uniform. This force causes parabolic deflection of rib and two corners of rib act as supports for "beam" as shown figure (2).

**Figure (2): Force exerted to a single rib**

In these conditions, deflection of ribs can be illustrated by below differential equation [9,10]:

\[
\frac{d^4 y}{dx^4} = \frac{M(x)}{EI}
\]

(9)

Which

- \(y\): rib deflection function
- \(M(x)\): flexible Moment

I: Moment of Inertia
E: Elasticity (Young) Module

Loading of ribs in this condition is similar to loading of a two end fixed beam and with assuming fixing of two ends of beam as boundary condition, we can solve (9) for calculating \(y\). So:

\[
M(x) = \frac{F W^2}{12} + \frac{F x^2}{2} - \frac{F W x}{24} + \frac{F x^4}{12}
\]

(10)

\[
y(x) = \frac{1}{EI} \left( \frac{F W^2 x^2}{24} + \frac{F x^4}{24} - \frac{F W x^3}{12} \right)
\]

Which "F" is the force exerted ribs, " Width" of ribs and "x" is distance variable from inner side of rib.

For calculation of pressure increase, volume increase of any rib must be calculated. Regarding figure (2) volume increase can be expressed as follows:

\[
\Delta V = L \times \Delta A
\]

(11)

Which \(\Delta A\) is the under curve area in figure (2). This can be determined by integrating of \(y(x)\) [9]:

\[
\Delta A = \int_0^L y dx = \frac{F W^5}{720 EI}
\]

(12)

\[
I = \frac{LT^3}{12}
\]

Using Equation (12) and with considering the number of ribs and pressure exerted to both sides of ribs, we can achieve the total volume increase of transformer:

\[
\Delta V = N \Delta A = \frac{2NF W^5}{60 E T^3}
\]

(13)

Which \(N\) is the number of ribs and \(T\) is thickness of tank. Based on relationship between force and pressure, we can express oil pressure increase versus oil volume:

\[
F = L \Delta P
\]

\[
\Delta V = \frac{N A P L W^5}{30 E T^3}
\]

Then the pressure increase is:

\[
\Delta P = \frac{30 E T^3 \Delta V}{N L W^5}
\]

(15)

So, by knowing oil temperature increase and resulted oil volume we can calculate oil pressure deviations in transformer.

As seen in (15), increasing the number of ribs can decrease rise of oil pressure. Also larger dimension of ribs, especially width of ribs, causes lower pressure increase and mechanical stresses.

The maximum stress is occurred in center of rib. This stress is calculated as below [9]:

\[
\sigma_{max} = \frac{M(x) = \frac{W}{2}T}{2I}
\]

(16)

\[
M(x) = \frac{W}{2} = \frac{F W^2}{12} + \frac{F W^2}{8} - \frac{F W W}{4} = -\frac{F W^2}{24}
\]

\[
I = \frac{LT^3}{12}
\]
So:
\[\sigma_{\text{max}} = -\frac{W^2 \Delta P}{4T^2} = \frac{-15EATV}{2NLW^3}\]  \hspace{1cm} (17)

This stress must be smaller than yield strength of selected material for transformer tank ribs.

As shown in (17) increase in number of ribs and dimension of ribs reduces mechanical stress, however the increase in thickness of tank causes higher mechanical stress.

**DYNAMIC STRESSES AND FATIGUE**

Practically, transformer oil temperature is continuously fluctuating in all times due to periodical load and ambient temperature variations. So the hermetically sealed transformer tank ribs encounter oil pressure variations and these pressure stresses can expedite tank fatigue in comparison with conservator type.

**Dynamic Analysis of Heat Transfer in Transformer**

Dynamic Modeling of heat transfer mechanism is performed through solution of differential equation (1). Several Models have been represented for transient analysis of oil temperature rise. These models essentially calculate winding hot spot temperature and in the next step, obtain transformer oil temperature in top and bottom points. One of the most successful and well-known methods is the model represented in IEEE C57.91 Appendix G. In this method, oil and winding temperatures in different points such as top, bottom of transformers are measured and defined as base condition. Using parameters in base condition, the oil temperature can be estimated in other loading and ambient temperature conditions. It should be noticed that in this standard in despite of this paper, heat transfer in tank and adjacent air and also the geometry of tank are ignored. In this paper, we use the suggested flowchart in IEEE C57.91 for dynamic analysis.

**Fatigue Analysis of Transformer Tank**

Alternative loading of transformer and resulted expansion and contraction can expedite transformer tank wear out. So it is necessary to consider fatigue effects in material selection.

There are various approaches for fatigue analysis of materials with alternative loading. In this paper, Tensile Strength Correlation Method [11] is used for this purpose. As described in [11,12], we need to select the factor of \( \Phi \) as fatigue ratio. This parameter is defined as follow:

\[\Phi = \frac{S_e}{\sigma_T}\]  \hspace{1cm} (18)

In (18), \( S_e \) is endurance limit in Mega Pascal and \( \sigma_T \) is ultimate tensile strength of tank material. Regarding experimental endurance limit vs. tensile strength curves and characteristics of steel, fatigue ratio is a value between 0.4-0.6. So we can determine endurance limit based on this factor in (18). Also we can use Mohr Method [12] in materials with ultimate tensile lower than 1400MPa that include all types of steels used in transformer tank manufacturing. In this method \( S_e \) defined as follow:

\[S_e = 0.506\sigma_T \times f(1, 0.138)\]  \hspace{1cm} (19)

In (19), \( f(1, 0.138) \) is the stochastic variable with logarithmic distribution and its mean and deviation are 1 and 0.138 respectively.

In equation (19) the value of \( f \) is nearly 1, so we have:

\[S_e \approx 0.5\sigma_T\]  \hspace{1cm} (20)

Now, it is necessary to determine the number of expansion and contraction cycles in transformer life time. Assuming normal operation condition for transformer and load variation similar to standards [7] and [8], the transformer will be encountered the daily expansion and contraction due to load variation. So the daily load variation provides two cycles of stress in transformer tank. This will be repeated in utilization period of transformer and must be considered for all of life time e.g. 30 years.

The main factor in fatigue assessment of materials is fatigue strength. This strength must be less than ultimate tensile strength. If the number of stress cycle is more than 1000 times in life time of transformer, the fatigue strength can be calculated as below [12]:

\[S_f = aN^b\] \text{fatigue Strength}

\[a = \frac{\sigma_T^2}{S_e}\]
\[b = -\frac{1}{3} \ln\left(\frac{F\sigma_T}{S_e}\right)\]  \hspace{1cm} (21)

\[F = \frac{(\sigma_T + 345)(2000)^b}{\sigma_T}\]

It is obvious that the (21) is a nonlinear equation because of non-linear relationship between \( F \) and \( b \). So we have to use an iterative method to determine fatigue strength. As mentioned above, this value must be smaller than ultimate tensile strength:

\[S_f \leq \sigma_T\]  \hspace{1cm} (22)

**CASE STUDY**

For practical verification of suggested model and evaluation of transformer tank design procedure, a 500kVA, 200/0.4kVA hermetically sealed transformer manufactured by Iran Transfo Co. is selected [13]. The transformer has 82 ribs and the dimension of ribs is 70cm×21cm. Other dimensions are shown in figure (3). The material of tank is steel sheet type St 12.03 and its thickness is 1.2mm. Thermal conductance factor of this material is 52 W/mK.

The weight of different parts and losses are given in table (2).

<table>
<thead>
<tr>
<th>Weight of Oil(kg)</th>
<th>Weight of Core and coil(kg)</th>
<th>No-load Losses(W)</th>
<th>Short Circuit losses in 75°C(W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>435</td>
<td>1167</td>
<td>1000</td>
<td>7800</td>
</tr>
</tbody>
</table>
Because of diversity of oil properties, we adopt the equations mentioned in IEEE C57.91 Appendix G:

\[ c_o = 1960 + 4.005 \theta_o \quad \rho_o = 884 \cdot e^{-0.00084 \theta_o} \quad \beta_o = 0.00084 \]

\[ \mu_o = 0.13573 \times 10^{-5} e^{3.297.3 \theta_o + 273} \quad k_o = 0.124 - 1.525 \times 10^{-4} \theta_o \]

Similarly, thermal properties of air can be extracted from heat transfer tables in reference [5]:

\[ c_A = 1007, \quad \rho_A = 1.1038 \]

\[ \mu_A = 17.76 \times 10^{-6}, \quad k_A = 0.0277 \]

**Steady State Condition Analysis**

1. **Nominal Load Condition and 25°C Ambient Temperature**

As a base condition, firstly we analyze oil temperature rise and oil expansion in nominal load condition and ambient temperature is assumed 25°C. We suppose the oil temperature is equal ambient temperature in initial condition and after transformer starting, oil temperature will increase gradually to achieve steady state condition.

Based on the procedure shown in figure (2), and using above mentioned data for transformer dimensions and oil properties, the steady state average point temperature is 76.5°C while the actual temperature measured by manufacturer is approximately 75°C. With knowing initial and final temperature of oil, we can calculate oil pressure increase by equation (15):

\[ \Delta P = \frac{30 \times 210 \times 10^3 (1.2 \times 10^{-5}) \times 0.01987}{82 \times 0.7 \times 0.21} = 11770 \text{ pascal} = 0.1177 \text{ bar(g)} \]

The oil pressure of tank is the summation of initial pressure and added pressure (ΔP). The maximum permissible oil pressure of tank is mentioned 0.5 bar(g) in manufacturer documents. So if the initial pressure is adjusted on 0.2 bar(g), the ultimate pressure will be 0.3177 bar(g) and this value is less than 0.5 bar(g). It should be mentioned that the nominal oil pressure is 0.35 bar(g) corresponding transformer datasheet.

2. **15% Overload Condition and 45°C Ambient Temperature**

At this condition, we assume that the transformer operates in 115% of nominal load and the ambient temperature is 45°C. Winding losses will be increased with the square of load and core losses are constant. Using the procedure represented in figure (2), the oil ultimate average temperature will be achieved 112.68°C and is higher than oil permissible temperature approximately 105°C.

With knowing initial and final temperature of oil, we can calculate oil pressure increase by equation (15):

\[ \Delta P = \frac{30 \times 210 \times 10^3 (1.2 \times 10^{-5}) \times 0.03522}{82 \times 0.7 \times 0.21} = 20870 \text{ pascal} = 0.2087 \text{ bar(g)} \]

If the initial oil pressure is adopted 0.2 bar(g), the oil pressure will be 0.4087 bar(g) and this value is higher than normal conditions.

At this condition, maximum exerted stress to rib can be calculated using (18):

\[ \sigma_{max} = \frac{0.21 \times 20870}{4 \times 0.0012} = 145 \text{ MPa} \]

The yield stress of steel type ST12.03 is 170 MPa and \( \sigma_{max} \) is smaller than this value. So transformer tank can tolerate probable stresses easily.

**Dynamic Analysis of Temperature and pressure Variations**

As mentioned before, the method described in IEEE C57.91 is used for dynamic modelling. It should be notified that for this method, we need a temperature measuring test results in nominal condition. This data have been supplied by manufacturer and will be available if requested. As an example, we use load and ambient temperature variation curves represented in IEEE C57.91 Annex G. The results of temperature variations of top, bottom and average points are shown in figure (4).

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This time constant is between 70 to 150 minutes dependent to oil type and transformer dimensions. It should be noted that core and coil time constant are about 10 to 20 minutes [8].

<table>
<thead>
<tr>
<th>Time(MIN)</th>
<th>Pressure(Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.4</td>
</tr>
<tr>
<td>500</td>
<td>1.0</td>
</tr>
<tr>
<td>1000</td>
<td>1.6</td>
</tr>
<tr>
<td>1500</td>
<td>0.4</td>
</tr>
</tbody>
</table>

Figure (5): Daily Oil Pressure Variation

Fatigue Analysis

For fatigue analysis, we assume that the transformer life time is 30 years and loading and applied stress curve are according figure (5). So the number of stress cycles in (21) is equal:

\[ N = 365 \times 2 \times 30 = 22000 \]

The ultimate tensile strength \( \sigma_T \) for steel type ST12.03 is 303 MPa. So solving nonlinear equation (21), the fatigue stress can be obtained:

\[ S_f = 221.69 \text{MPa} \]

This value is smaller than 303MPa. So the transformer tank can tolerate periodic stresses without deformation and considerable in mechanical properties.

As a result of these calculations, occurrence of harmful overpressure is not a contingent and probable event in transformer utilization period and hermetically sealed transformer can cope with these stresses.

CONCLUSION

Despite of conservator type transformers, hermetically sealed Transformers encounter higher oil pressures and this problem have to be considered in transformer design and operation. As described in this paper, the exerted stresses to ribs caused by oil temperature rise, is smaller than yield strength of tank material even in the worst ambient temperature and loading condition, and the safety margin is sufficient in all condition. Also the oil pressure increase and oil temperature rise will be reduced by increasing the number of ribs and enlarging their dimensions.

Regarding the periodic nature of transformer loading, the effect of fatigue in material selection must be considered. As shown in paper, occurred fatigue stresses are smaller than permissible ultimate tensile strength and as a result, all mechanical stresses can be tolerated easily by transformer.

REFERENCES