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Influence of the Power Transformer Tank Design on the Sound Level

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Abstract

Low noise levels are today requested for power transformers in order to comply with customer specifications and environmental legislations. Manufacturers have to improve the acoustic performances of their products and also reduce the costs in the same time. Therefore it is necessary to predict sound levels with a sufficient accuracy for each transformer design.
Sound radiation of transformer tanks is influenced by a wide variety of parameters including placement, type, arrangement and dimensions of stiffeners, tank thickness, tank shape and type, insulating fluid or gas (mineral oil, ester or air). The vibration sources in transformers are located within an oil-immersed rib-stiffened transformer tank vessels made of thin and flexible steel plates. The vibrations generate sound pressure waves which are iteratively reflected from structural boundaries and discontinuities. They form standing patterns of vibrations, structural modes of the vessel and standing waves inside the fluid-filled acoustic cavity. If the corresponding natural frequencies of the structures are in the vicinity of the forcing frequencies, unexpectedly high sound levels are regularly observed, [1]. In order to avoid these unexpectedly high sound levels caused by the natural frequencies of the tank, optimization of the tank and tank stiffeners arrangement, dimensions and design has to be carried out.

Keywords: Power transformer tank; Fluid-structure interaction; Vibrations; Eigen-frequencies; Harmonic response; Sound pressure level

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1. Introduction

A finite element modelling will be used to study the vibro-acoustic behaviour of a transformer operation conditions, [2]. Results from the FEM analysis will be correlated with performed measurements in the test bay in accordance with [3]. Eigen-frequency analysis is performed in range from 80-310 Hz. Main goal is avoiding/shifting eigen-modes of tank plates in vicinity of 100, 200 and 300 Hz, where core excitation has local peak values. Also harmonic response analysis is performed in order to determinate in which frequency range sound radiation is the highest.

Nomenclature

$[C_s]$	damping matrix for solid
$[C_F]$	damping matrix for fluid
FEA	finite element analysis
FFT	Fast Fourier transform
f_f	external excitation vector in the acoustic fluid
f_s	external excitation vector in the structure
FSI	fluid-structure interaction
$[K_s]$	stiffness matrix for solid
$[K_F]$	stiffness matrix for fluid
$[M_s]$	mass matrix
p_e	pressure vector
PML	perfectly matched layer
$[R]$	coupled matrix (coupled conditions between the fluid and the structure)
u_e	displacement component vector
ρ_0	mass density

2. The General Acoustic Equations

In order to simulate fluid-structural interaction (FSI) problems, the structural dynamics equations are considered along with the Navier-Stokes equations of fluid momentum and the flow continuity equation. The discretized structural dynamics equation can be formulated using the structural elements. The fluid momentum (Navier-Stokes) equations and continuity equations are simplified to get the acoustic wave equation using the following assumptions:

- The fluid is compressible (density changes due to pressure variations)
- The fluid is irrotational
- There is no body force
- The pressure disturbance of the fluid is small
- There is no mean flow of the fluid (fluid is resting)
- The gas is ideal, adiabatic, and reversible

Concept of fully coupled finite element dynamic matrix equation:

$$\begin{bmatrix} [M_S] & 0 \\ \overline{\rho_0}[R]^T & [M_F] \end{bmatrix} \begin{Bmatrix} \{\ddot{u}_e\} \\ \{\ddot{p}_e\} \end{Bmatrix} + \begin{bmatrix} C_S & 0 \\ 0 & C_F \end{bmatrix} \begin{Bmatrix} \{\dot{u}_e\} \\ \{\dot{p}_e\} \end{Bmatrix} + \begin{bmatrix} K_S & -[R] \\ 0 & K_F \end{bmatrix} \begin{Bmatrix} \{u_e\} \\ \{p_e\} \end{Bmatrix} = \begin{Bmatrix} f_S \\ f_F \end{Bmatrix} \tag{1}$$