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Numerical Vibroacoustic Analysis of a Large Transformer Tank Including Structural and Noise Optimization

Max Gillet, Bruno Bosnjak, Hermann Landes, Neeraj Kewat

Summary **—This paper presents a coupled numerical vibroacoustic analysis of a 380 MVA power transformer intended for railway transport. The study employs finite element modelling of the transformer active part, oil, tank and surrounding air. Harmonic analysis at predetermined frequencies is performed to capture the complex coupled dynamic interactions between the transformer's structural components and acoustic fluid, namely insulating oil and air. The harmonic analysis uses a fully coupled approach between the structural and acoustic domain where the partial differential equations representing the vibroacoustic behaviour of the transformer are presented in a single matrix without the need to iteratively solve separate matrices for each physical domain. This approach yields a field solution that enables simple evaluation of the pressure distribution inside oil, tank vibrations and sound pressure radiated outside the tank. Some optimization solutions are introduced to prevent local resonance modes and to treat influence of air cavities.**

Keywords **— Vibroacoustic, FEM, Multiphysics, Modelling, Transformer**

I. Introduction

The continuous expansion of urban areas has led to an increasing number of high voltage substations being installed in the vicinity of residential areas. Therefore, audible noise emitted by power transformers has evolved from secondary design consideration into a governing design constraint. Transmission system operators and utilities often impose stringent noise limits to ensure regulatory compliance and public acceptance, thereby strongly influencing transformer design and validation processes [1]. Conventional noise mitigation measures, such as external acoustic barriers or enclosures, remain effective but typically mean significant drawbacks in terms of mass, cost, footprint, and long-term maintenance [1]. This context has reinforced the industrial need for predictive numerical tools - notably Finite Element Method (FEM) and Boundary Element Method (BEM) approaches to assess and optimize vibroacoustic behaviour at early design stages, with the objective of reducing reliance on corrective solutions [2, 3].

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Simulation of noise generation and propagation in power transformers results from complex multiphysics process, beginning with electromagnetic excitation and ending with acoustic radiation to the surrounding environment. Numerical modelling of this process should consider several coupled physical domains: electromagnetic sources, including core magnetostriction and Lorentz forces acting on the windings, the resulting mechanical vibration of the active part and the fluid–structure interaction with the insulating oil [2], [4]. In most advanced simulation practice, electromagnetic forces are typically transferred as volumetric or nodal loads onto detailed structural models, which are subsequently used to determine the vibration of the transformer tank [2]. A critical aspect of this modelling chain is the accurate representation of excitation mechanisms, since vibrations originating from different parts of the active assembly—such as high-voltage and low-voltage windings—are not strictly additive, and their phase relationships can significantly affect tank surface velocities and the resulting sound radiation [2].

Within the overall vibroacoustic behaviour, transformer tank represents primary radiating structure. Its mechanical design plays an important role in determining the acoustic signature of the unit, as the tank may either attenuate or amplify the internal excitation through structural resonances [5], [6]. The radiation efficiency of transformer tanks is strongly frequency dependent and influenced by parameters such as wall thickness, global geometry, and layout of various mechanical structures on the tank. Previous investigations have shown that appropriate tuning of tank stiffness, particularly through the design and spacing of stiffeners, can mitigate sound radiation by avoiding coincidence between structural eigenmodes and dominant excitation frequencies. In addition, the presence of insulating oil introduces mass-loading and damping effects that tend to shift tank eigenfrequencies toward lower values and attenuate higher-frequency vibration components [4], [7].

Several studies have investigated structural design strategies aimed at controlling tank resonance and reducing noise emission. Alternative tank concepts such as curved tank walls have been reported to improve structural stiffness without increasing radiated sound power [8]. Other works supported by FEM-based vibroacoustic analysis and validated through measurements have demonstrated that targeted modification of tank wall stiffness and stiffener layouts can shift critical eigenfrequencies away from dominant excitation bands and achieve measurable reductions in sound pressure levels [9], [10]. While these approaches provide valuable guidance for designing conventional transformer tanks, they are generally developed for relatively simple and flat structures and do not fully address configurations in which the tank itself assumes a primary structural role.

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For large power transformers, additional mechanical and logistical constraints influence vibroacoustic design. In the present paper, authors investigate vibroacoustic behaviour of a 380 MVA power transformer equipped with "Schnabelcar" tank, as presented in Fig. 1, that can be directly installed between train bogies as a self-standing wagon. This design, mainly requested in Germany, includes massive reinforcements, heavy beams, transport-related niches, external cavities filled with air and train-profiled shape which play a non-negligible role on transformer vibroacoustic behaviour compared to standard transformer tanks. Assessments of the influence of tank panel modal behaviour, tank cover trapezoidal shape and external cavities formed by reinforcements on external radiated sound pressure field are presented and potential optimization solutions will be proposed.

II. Modelling Approach

The harmonic analysis simulations should provide detailed information and predictions of the transformer vibrations and the acoustic emission characteristics of the complete assembly. This will be especially relevant for fundamental (100 Hz) and first two harmonics of the network frequencies at which the dominant transformer noise emissions are typically observed. Fully coupled vibroacoustic calculation approach was used, providing full vibroacoustic response in a single computational step without requiring additional iterative calculations between different physical domains in order to reach a satisfactory level of accuracy.

The basis of the simulations lies on the resolution of the linear partial differential equations describing the structural vibrations of the transformer assembly and the pressure waves in the oil as well as in the ambient air. The coupling between these physical domains has also to be integrated in the models. Assuming linear behaviour as well as isotropic solid material properties, structural vibrations (1) and acoustic wave (2) equations can be written as follows:

$$\frac{E}{2(1+\upsilon)}\left(\nabla\cdot\nabla\vec{u}+\frac{1}{1-2\upsilon}\nabla(\nabla\cdot\vec{u})\right)+\vec{f}_V=\rho\frac{\partial^2\vec{u}}{\partial t^2} \quad (1)$$

$$\nabla^2\psi=\frac{1}{c^2}\frac{\partial^2\psi}{\partial t^2} \quad (2)$$

The material properties *E, n, r,* and *c* being respectively elasticity module, Poisson's ratio, density, and the speed of sound in the structural and acoustic media, whereas $\vec{u}$ and φ are the structural vibrations and the acoustic velocity potential. Using the velocity potential instead of the acoustic pressure results in a symmetric finite element formulation while acoustic pressure and the velocity potential are related by means of $p=\rho\varphi$.

The coupling between the structural and acoustic parts is given by the fact that an acoustic medium only acts via a pressure force on the structural part and that at the acoustic-solid interface the normal velocities of the acoustic and structural parts must be the same. These coupling conditions are described by equation (3):

$$v_n=\vec{n}\cdot\frac{\partial u}{\partial t}=-\frac{\partial\psi}{\partial n}=-\vec{n}\cdot\psi \quad (3)$$

Assuming a time-harmonic behaviour, standard finite element formulations are then applied to the above coupled system of partial differential equations, leading to the algebraic system of equation (4) supplied with boundary conditions describing physical setup:

$$\left[\begin{pmatrix}K_{UU} & 0\\ 0 & K_{\psi\psi}\end{pmatrix}+j\omega\begin{pmatrix}C_{UU} & C_{u\psi}\\ C_{u\psi}^T & C_{\psi\psi}\end{pmatrix}-\omega^2\begin{pmatrix}M_{UU} & 0\\ 0 & M_{\psi\psi}\end{pmatrix}\right]\begin{pmatrix}\vec{u}\\ \vec{\psi}\end{pmatrix}=\begin{pmatrix}\vec{F}\\ 0\end{pmatrix} \quad (4)$$

It should be noted that in the formulation chosen by authors, the acoustic velocity potential in (4) describes wave effects in two different media: one in the oil and the other in the ambient air. In a free-field radiation problem, which is the case considered in this paper, the impact of ambient air on the vibrations of the tank walls is assessed to be very small and might then be neglected. Therefore, an alternative formulation might have been chosen, in which a boundary element formulation (BEM) would be chosen for the calculation of the acoustic radiation, thus ignoring the effect of the radiation on tank vibrations. This approach is not valid for the coupling between the oil and structural vibrations, as the presence of oil has a significant impact on vibrations amplitudes (hydrostatic pressure). Moreover, the use of BEM formulation would have led to unsymmetric acoustic problem, increasing unnecessarily complexity of equations to solve. Consequently, finite element resolution method using symmetric approach was used for this study.

Simulation model must include the whole chain of energy transfer taking place inside the transformer. Starting with vibrations of the windings, two different paths must be considered: structural vibrations transferred to the tank by means of windings and active part support as well as pressure waves generated and propagated inside the oil. These waves excite the tank walls to vibrate which will then finally radiate acoustic waves into the ambient air. This double fluid-structure interaction (active part to oil then oil to tank) requires specialized simulation tools combined with a modelling strategy that keeps the models practical to calculate.

Unlike conventional power transformer tanks, in Schnabel car tank standing waves can also occur in cavities of horizontal stiffening elements. These hollow stiffeners are visible along the tank long sides, as indicated by red arrows on Fig. 1. The Schnabel car tanks also have hollow vertical stiffeners indicated by blue arrows on Fig. 1. Standing waves occurring in these cavities can amplify the radiated noise and additional mechanical measures are performed to prevent noise amplification due to this phenomenon.

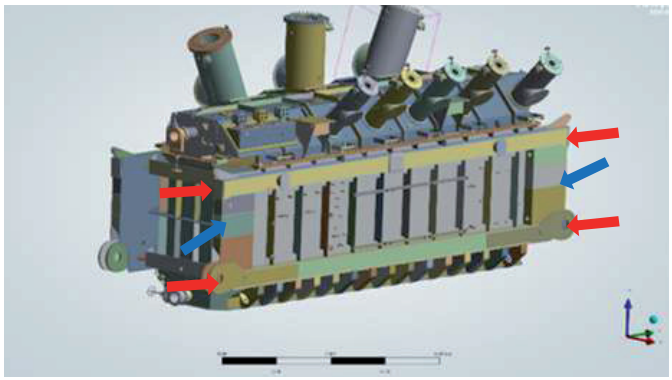

Fig. 1. Computer-aided design (CAD) model of the transformer

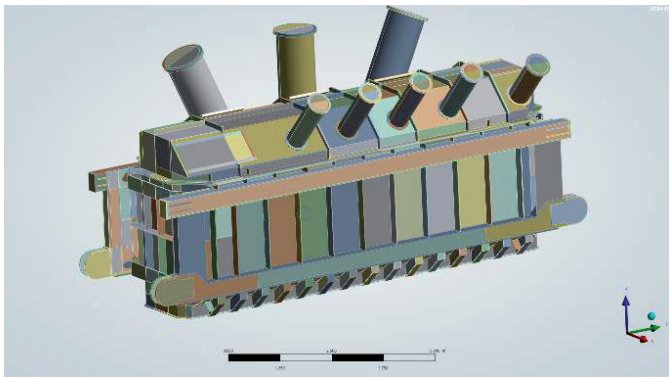

Fig. 2. Simplified transformer model

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Due to the high complexity of the complete setup, as can be observed on Fig. 1, direct use of the available CAD model as a basis for the finite element model is not feasible. Instead, a simplified model has been established as presented on Fig. 2. Although its complexity is slightly reduced, all the relevant parts of the transformer contributing to noise generation are included. It can be noted that this reduced model has required further minor modifications and adaptations to ensure continuity of the volume geometry. This model has been used as the basis for a finite element model, integrating transformer tank walls, insulation oil, enclosed air, and surrounding air in a single simulation model.

Fig. 3 shows the modelled transformer embedded in a semi-spherical hull representing ambient air. The developed FE model is made up of second order tetrahedral elements and consists of 5.2 million nodes and 3.4 million elements. The requested calculation time for one frequency run lasts approximately 30 minutes using a high-capacity computer (256 GB of RAM and 16 cores). To reduce computational complexity, the use of hierarchical elements based on Legendre polynomial approximations has also been introduced in the model.

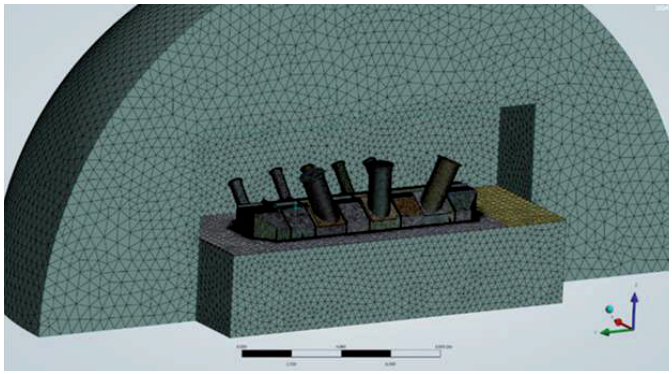

Fig. 3. Finite element mesh of transformer and surrounding air

The simulation of electromagnetic forces, including electromagnetic forces taking place in core and causing excitation of winding vibrations would add a significant degree of complexity to the model as it would require at least one additional separate computation step. For such large transformer, load sound power level -generated by windings- are generally dominant vs. no-load sound power levelgenerated by transformer magnetic core, due to design with low induction level and use of noise-improved grade-oriented steel grades. Furthermore, consideration of electromagnetic excitation in the models would not significantly affect results and conclusions of the study as authors have chosen to adopt a comparative approach rather than trying to obtain absolute values simulating actual sound level radiated by this large power transformer.

Therefore, radial and axial displacements have been imposed on windings external surfaces (described as plain cylinders to simulate winding vibrations). Same displacement amplitudes have been used for all explored frequencies, this assumption being somewhat unrealistic as the displacements for the fundamental frequency are typically larger than for higher harmonics. However, as the studied system is linear, higher frequencies displacements can easily be scaled. A phase shift between axial and radial displacements for each winding has been introduced in the model to keep sources in constant volume conditions, and a phase shift of 120° between the three windings displacements has been applied to represent real behaviour of three-phase transformers. Thanks to this approach, a complete linear system arises for simulations, decreasing computation time for a significant extent.

Besides these excitations, the only additional boundary conditions applied to the model are caused by the assumptions of free-field radiation of the tank positioned on a hard reflective floor. This one is modelled thanks to zero displacement condition applied at the bottom faces of the tank, whereas either absorbing boundary conditions or perfectly matched layers can be used to obtain the free-field radiation conditions at the outer boundary of the enclosing air.

Various types of results can be obtained using FEM software, from raw data corresponding to physical quantities to more advanced output obtained after post-processing operations. In this paper, authors have chosen to notably present sound pressure fields (real part; representing acoustic active field) for various frequencies. Thanks to these primary results, it is possible to derive other types of results, such as noise levels prescribed by IEC 60076-10 standard [11], i.e. at one meter distance from radiating surface. These results permit to estimate sound pressure and power level of the apparatus as they are measured during factory acceptance tests and at sites.

III. Main Results

Simulations have been run for the model described in previous section and the following materials were used: the tank made of steel, the windings made of homogenized orthotropic material itself resulting from mixing copper, pressboard and paper, the winding supportive structure made of laminated wood (clamping plate) and steel, and the oil characteristics were set to obtain sound celerity in the range of 1350 $m.s^{-1}$ as described in Table I. For the harmonic analysis, many frequency steps were computed, majority of them located on and around first harmonics generally radiated by power transformers, i.e. 100, 200 and 300 Hz for 50 Hz network, as used in Europe. It is well-known that noise generated by windings, also called load-noise is predominantly composed of the first harmonic. The goal of this study is to obtain information on global Schnabel car tank vibroacoustic behaviour, which is why simulating higher harmonics through windings has been used, despite it is not physically correct when considering the global transformer multiphysics behaviour.

Some typical results of these simulations are presented in Fig. 4–7:

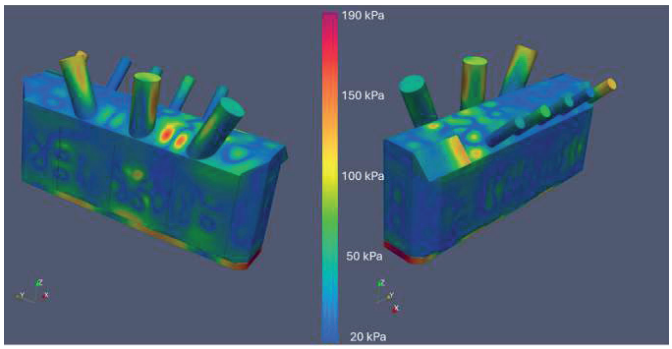

Fig. 4. Pressure distribution in oil, at interface with inner side of tank for 100 Hz frequency excitation for HV (left) and LV (right) sides

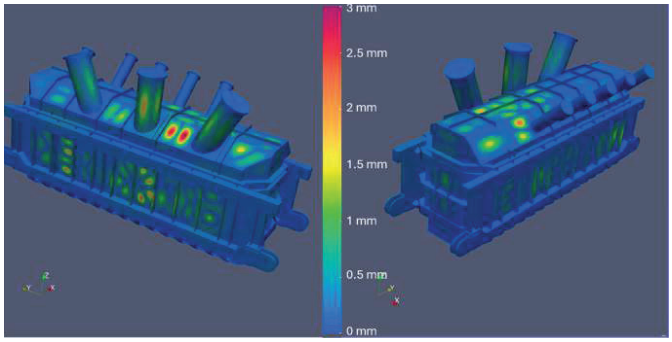

Fig. 5. Tank displacement calculated at 100 Hz for HV (left) and LV (right) sides

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The local pressure maxima in the oil can be clearly identified in Fig. 4, whereas the vibration hot spots on the tank walls are visible on Fig. 5. It is interesting to note that pressure peaks calculated in the oil at the interface with the tank do not necessarily correspond to vibration maxima on tank walls, although some correlation exists in many cases. As an example, oil pressure hotspot located at transformer cover on high voltage (HV) side (Fig. 4, left) implies high displacements of the tank in front of it (Fig. 5, left), whereas high pressure level calculated inside LV turret at the upper right of Fig. 4 right (orange coloured) doesn't generate high level of vibrations of the turret envelop (Fig. 5, right).

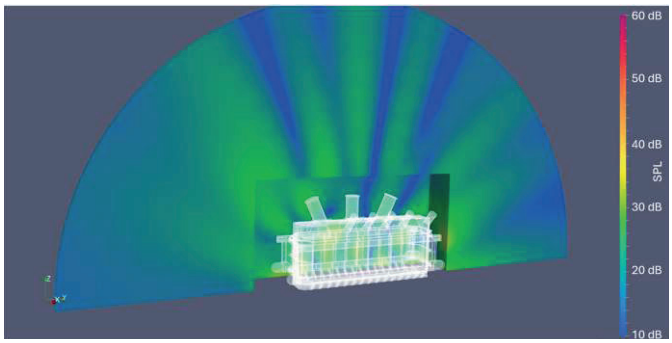

Fig. 6. Radiated sound pressure field at 100 Hz, centre section

In Fig. 6 and Fig. 7, the radiated sound pressure field is shown for a centre cross section view of the model at 100 and 200 Hz, respectively, same pressure range being used for both figures. The radiated sound pressure levels and sound field structures strongly depend upon the frequency. The sound pressure field for 200 Hz excitation shows higher levels and complexity in the near field, as well in the far field where many more alternances of high pressure/low pressure areas are observed.

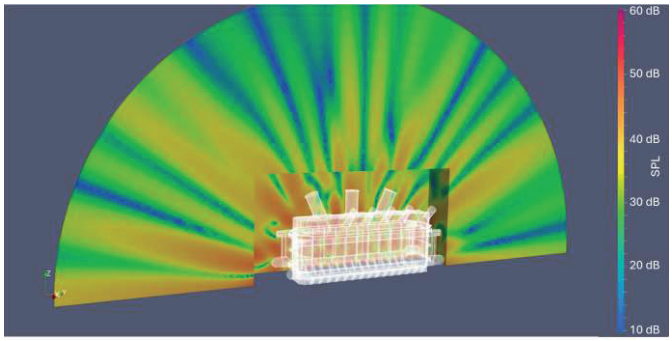

Fig. 7. Radiated sound field at 200 Hz, centre section

IV. Analysis

Pressures calculated in the oil and the surrounding air as well as displacements of the tank permit the investigation of vibroacoustic behaviour. Therewith, these simulations can be used to optimize transformer Schnabel car tank vibroacoustic and consequently reduce radiated noise. In addition to possible improvements, authors propose to use developed FE model to assess the influence of structural and fluid damping characteristics as well as oil characteristics on radiated sound levels. The following analyses are performed using a comparative approach thanks to FE parameters variations such as geometry modifications, material characteristics and model properties to derive learnings. The simulation of accurate absolute values of sound levels isn't intended in this article. It has to be noted that no measurement result that could support or confirm calculation results are available during the writing of this article.

As this was proposed in [9], the method consisting of stiffening areas on the tank wall where displacement peaks are detected has been investigated. One additional vertical stiffener has been introduced in the model, and the harmonic analysis has been rerun. Initial results and corresponding hot spot highlighted by white circle are presented on Fig. 8 left, while those including additional stiffener configuration are shown on Fig. 8. right:

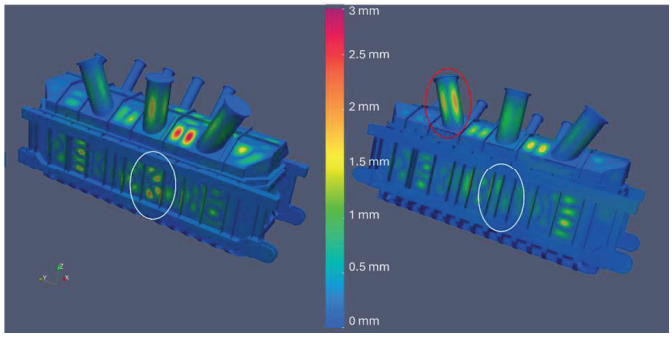

Fig. 8. Tank displacement hot spot computed with initial design (left) and impact of additional stiffener (right)

It is observed that tank displacements are locally reduced, on tank side and even more significantly on tank cover where local mode with two lobes almost disappear. However, it is also observed that this relatively small modification reinforced tank displacement amplitude for the left side turret, as indicated by red circle on Fig. 8 right. This component will then radiate more energy, which will unfortunately be impacting sound pressure field not only on one lateral side, but all around the transformers due to its top position. This example of modification highlights the fact that local stiffening can significantly decrease local resonance amplitudes, but the impact of such modifications must be globally considered risking moving amplitude "hot spots" to areas which might have a larger impact on radiated overall sound power levels.

A further point of investigation specific to Schnabel car design is the presence of air cavities existing within tank design. These hollow structures directly implemented on tank walls might be subjected to several acoustics problem within one object. Standing waves can occur within the enclosed air since their rectangular shape with parallel surfaces naturally admits numerous cavity modes. Secondly, the external surface of these cavities, made of plain steel, can admit numerous plate modes able to amplify noise emissions. Finally, these cavities could also act as Helmholtz resonators, as some of them aren't fully closed in Schnabel car tank design for manufacturing reasons. Contrary to cavity and plates modes, Helmholtz resonances will tend to damp vibroacoustic energy and are often used as noise reduction solutions when discrete frequencies need to be attenuated.

The study of the coupling between these three phenomena offers an interesting topic for vibracoustic design investigations and possible structural optimization. In the present study, this problem has been addressed by creating an alternative model in which air has been removed from the cavities indicated in Fig. 1 by red and blue arrows. No significant differences were obtained in sound pressure levels as well as for structure displacements by the removal of the air in the model for all frequencies investigated. Some waves patterns are observed at 300 Hz inside the cavities as observed in the lower ones in Fig. 9, without significantly influencing radiated energy.

Frequencies for which efficient energy transfer and damping phenomena occur are probably located away from main frequencies evaluated in this study. Further modelling works are required to deeper investigate this coupled problem, for instance by running a modal analysis of the cavity structures, thus revealing potential critical internal resonance frequencies. Solutions to treat such am-

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plification problems thanks to FE vibroacoustic parametric studies could include the reshaping and resizing of cavities dimensions, the introduction of internal stiffeners and opening or closing of cavities, as well as the parametric study of acoustic absorption coefficient of cavities' internal surfaces.

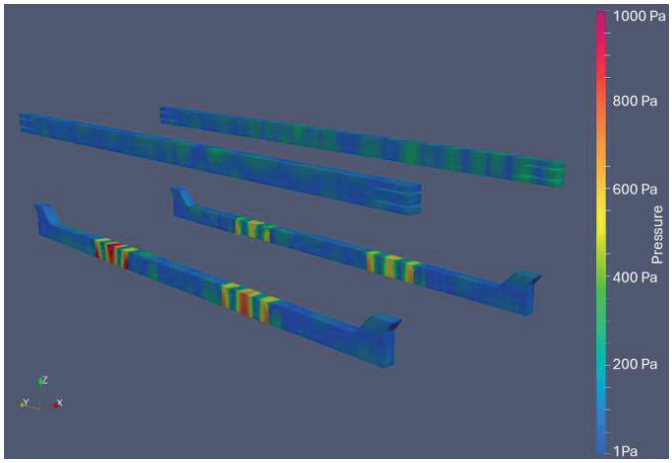

Fig. 9. Pressure distribution inside cavities at 300 Hz

On top of possible damping brought by Helmholtz resonators, global damping introduced in FE models deserves to be investigated for vibroacoustic problems. Tank response to excitation and therewith radiated noise strongly depend on properties associated with the modelled materials. Values of elasticity modulus, Poisson's ratio, and density of different structural components are well-known and mastered, but mechanical losses and consequent damping factors in the tank cannot fully be described in the FE models with accuracy. When predictive FE models are developed, damping properties are often tuned at the end of modelling process according to calibration measurements carried out for that purpose. The situation is even more uncertain for oil damping characteristics, which suffer from lack of consolidated input data. It is proposed by authors to compare the impact of global damping properties, by introducing in the FE model two different damping coefficients. The first value used is a loss factor of 1% applied to the whole model, considered as a standard value (used for all other simulation configurations presented in this paper). The second value of loss factor modelled is 2%, considered as high damping values. Fig. 10 presents sound pressure level inside the oil at tank interface for 100 Hz frequency for 1% (left) and 2% (right) global damping, while Fig. 11 presents corresponding displacements calculated on tank structure.

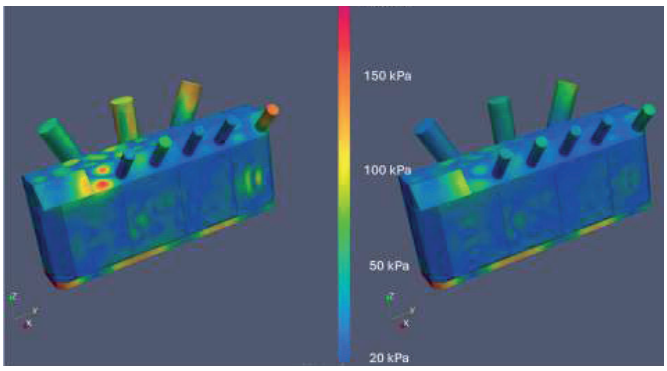

Fig. 10. Oil pressure calculated at 100 Hz for 1% and 2% damping

While for the tank displacement amplitudes, the observed vibration patterns remain relatively stable with a reduction of the peak amplitudes approximately by a factor two, the impact on the pressure distribution inside the oil is much more significant. On top of global linear pressure amplitude decrease, changes in pressure

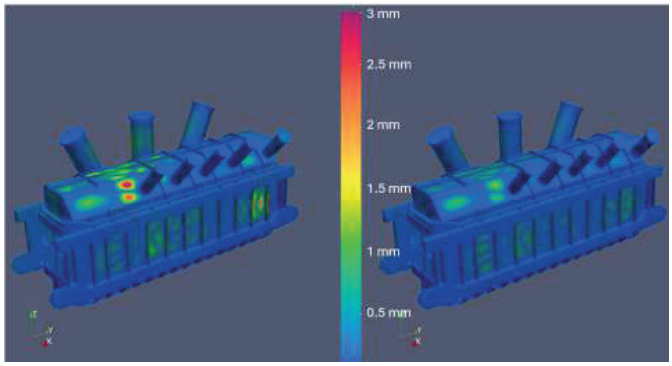

Fig. 11. Tank displacements at 100 Hz for 1% and 2% damping

distribution are observed in the results, notably in turrets located on the tank cover. This result highlights the importance of controlling damping properties when performing vibroacoustic FE studies.

Vibroacoustic FE studies of power transformers require the modelling of oil. Various characteristics have been used in FE vibroacoustic former simulations works, for instance in [10, 12], but no sensitivity study for this parameter was discussed. Two main types of oil are available for power transformers: mineral oils that are mostly used worldwide, and ester oils (synthetic and natural) that are increasingly frequently used because of their environmentally friendly characteristics. Investigation was performed to determine impact of this parameter by introducing different oil characteristics in the developed model, by considering set of input data representing mineral, natural ester and synthetic ester oil types. Table I summarizes main oil characteristics used in the model for this parametric evaluation, and corresponding computed overall sound pressure levels (L_{pa}) as they are measured following IEC standard [11]:

TABLE I.
SIMULATION INPUTS AND RESULTS FOR VARIOUS OIL CHARACTERISTICS

Oil characteristics / Sound pressure levels	Units	Mineral oil	Natural ester oil	Synthetic ester oil
Density	$kg.m^{-3}$	865	900	970
Bulk modulus	GPa	1.57	1.66	1.90
Speed of sound	$m.s^{-1}$	1347	1358	1400
Calculated L_p at 100 Hz	dB	0	0.4	-1.5
Calculated L_p at 200 Hz	dB	0	-0.4	-4.6
Calculated L_p at 300 Hz	dB	0	0	-0.9

Sound pressure levels presented in Table I have been normalized, the reference value for each frequency being determined for the mineral oil. Positive values mean noise increase compared to mineral oil, negative values a noise decrease. Results show that synthetic ester oil, which is more than 10% denser than standard mineral oil, generally tends to reduce noise emissions compared to mineral oil for the three main frequencies of interest, in range of 0 to 4.5 dB reduction depending on considered frequency. These results highlight that oil characteristics have a significant impact on radiated sound pressure levels, and that such characteristics are changing pressure distribution which implies a frequency dependency. Further calculations would be required to determine whether the difference in magnitude is applicable to all types of transformers. However, it is interesting to observe that eco-friendly ester oil product is also noise-friendly, giving an additional advantage to synthetic ester oil vs. mineral oil. These results show that oil properties used in FE model are as the damping quite sensitive and should be used with a lot of care in case FE models are intended to be used for noise prediction.

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Another specificity of Schnabel car tank is the profiled shape of tank cover, which could affect noise radiation compared to standard flat cover tanks as inclined parts of the cover could reinforce sound levels measured as per [11]. In order to quantify noise emitted by top side of the 380 MVA transformer, it is proposed in this study to add to the standardized 1/3 and 2/3 tank height prescribed contours at one meter distance from main radiating surface two other contours: one located at the level of the flat part of tank cover (3/3), and another line (Top) located one meter above the central part of tank cover and closed on 3/3. These contours are presented on Fig. 12 while Fig. 13 shows transversal cross-section view of sound pressure field calculated in the near and far fields:

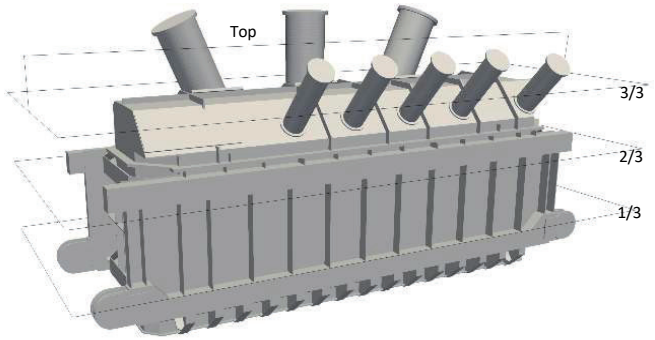

Fig. 12. Four contours used to determine tank pressure levels

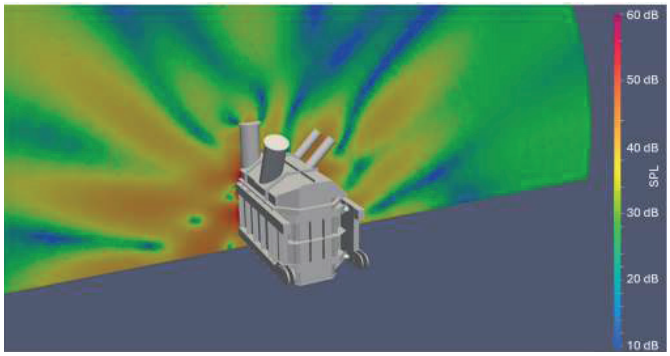

Fig. 13. Transversal cross-section view of sound pressure field at 200 Hz

Sound pressure levels determined for these four contours are presented in Table II. Presented results have been normalized, the chosen sound pressure level reference value being set to 0 dB for 100 Hz frequency step and 1/3 tank height contour calculation.

TABLE II.
CALCULATED AVERAGE SOUND PRESSURE LEVELS FOR DIFFERENT HEIGHT OF TANK

Sound pressure levels at 1 m	Units	Contour at 1/3 H_{Tank}	Contour at 2/3 H_{Tank}	Contour at 3/3 H_{Tank}	Top contour
L_p at 100 Hz	dB	0	2.3	5.7	7.2
L_p at 200 Hz	dB	2.8	8.9	3.4	11.8
L_p at 300 Hz	dB	9.1	8.5	8.7	5.1

Large discrepancies are obtained in calculated sound pressure results for the different contours as well as for frequencies computed. No clear trend is consequently observed; however, it appears obvious that the integration of upper contours in determination of average sound pressure and sound power levels significantly affects determined values. For 100 Hz component, noise emissions from the cover exceed ones determined as per IEC standard [11] for 3 to 7 dB. For 200 Hz component, results are more scattered, the maximum average sound pressure level being obtained for the top contour. This is confirmed by sound pressure distribution visualization on Fig 14., on which areas of high sound pressure are observed notably around LV and HV turrets. For 300 Hz, sound pressure levels of different contours are more homogeneous and the consideration of tank cover sound emissions doesn't affect results as currently determined due to IEC measurement procedure which focuses on lateral sides tank emissions. This analysis can't be generalized to all power transformers and does require further studies, in particular, the comparison between different tank cover shapes. However, authors recommend considering possible changes in measurement procedure to better account for noise generated by tank cover and elements positioned on it, such as turrets and conservators. This would help to improve the accuracy of noise environmental studies, in particular for far field predictions.

V. CONCLUSION

A large scale fully coupled vibroacoustic FE model of a 380 MVA Schnabel car transformer tank has been developed and applied. The results indicate that radiated sound levels are mainly driven by global tank structural resonances, which can be effectively mitigated and optimized when addressed through overall mechanical design rather than local modifications. The contribution of air-filled cavities integrated in the Schnabel car structure was found to be limited within investigated frequency range for studied design. In contrast, oil characteristics and global damping assumptions were identified as sensitive modelling parameters and should be selected with caution when FE vibroacoustic models are used for absolute sound levels prediction. The profiled tank cover was shown to have a significant influence on sound radiation. Future work could include an extended study of cavities influence on radiated sound levels as well as global consideration of tank cover sound emissions.

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