Descrease in noise and vibration of single-phase asynchronous motors in rolling stock

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Abstract. The article describes the magnetic noise during the operation of a single-phase asynchronous motor due to the vibration of the stator package of the motor under the action of vibration-absorbing forces due to the interaction of harmonics of a magnetic field of various origins in the air gap of the machine. As a normalized parameter of vibration created by an electric machine, the effective value of the vibration velocity is adopted. Electric motors used in electric rolling stock operate in more difficult conditions in comparison with general industrial ones. In the electric rolling stock, various auxiliary electric machines are used. Despite the relatively low level of magnetic vibration of high frequency, due to its determining effect on the magnetic noise of single-phase motors, this component is of independent interest for research.

1. Introduction
Single-phase machines have several specific features in terms of the formation of magnetic components of vibration and noise. This is due to the presence, along with a directly rotating, also rotating system of the first and higher harmonics of the magnetic field. The presence of the main backward rotating harmonic of the field causes the pulsation of torque with a double frequency of the network, which leads to significant tangential vibration of the stator of a single-phase motor [1, 2, 4]. Interactions of the higher harmonics of direct and reverse sequence systems can lead to an increase in the radial component of magnetic vibration and magnetic noise of single-phase motors compared to three-phase ones.

It is known that the magnetic component of the noise of asynchronous machines is caused, first of all, by high-frequency oscillations (1000-8000 Hz) of the stator yoke under the influence of radial electromagnetic forces caused by the interaction of higher harmonics of the field of various origin [3, 5, 7]. It is natural to assume that in single-phase machines all kinds of interactions between harmonics of not only the direct but also the reverse field, as well as between the harmonics of both fields will lead to the appearance of a large number of radial electromagnetic forces that excite magnetic noise and vibration, i.e. to increase noise and vibration of single-phase machines in comparison with symmetric three-phase ones [2, 6, 7, 9].

2. Methods
Existing methods for calculating magnetic noise and vibration of asynchronous machines have unsatisfactory convergence with experience. There is no direct experimental evidence of the theoretical premises underlying these techniques, the correct calculation of individual quantities, in
particular, inductions, interacting harmonics, forces, deformations [9].

The electrical industry is the only industry where the issues of regulation and methods for controlling noise and vibration are resolved at the level of state standards. In this regard, the requirements for vibroacoustic characteristics of electric machines are included in all types of regulatory and technical documentation (TU, TK, GOST for specific types of engines, etc.). These requirements, in some cases, are very stringent, which entailed the need to develop special design methods, improve technology, create and improve research methods and calculate vibration and noise.

The aim of this work is to study the sources of high-frequency components of magnetic noise and vibration of asymmetric electromagnetically single-phase machines, to identify the degree of increase in magnetic vibration and noise of single-phase machines in comparison with symmetric three-phase ones.

Magnetic noise is the result of the vibration of the stator pack of the engine under the action of vibration-absorbing forces due to the interaction of harmonics of a magnetic field of various origins in the air gap of the machine. As a standardized quantity for noise estimation, the so-called sound level (dBA) is used, which is a frequency-adjusted (correction A) total sound pressure level measured with an elastic machine installation without the additional mass. Correction A is established as international and consists of decreasing the measured noise levels at frequencies below 1000 Hz, i.e. in reducing the sensitivity of the measuring device at low frequencies, which is consistent with the peculiarities of human noise perception. Therefore, when studying magnetic vibrations from the point of view of their influence on the magnetic noise of the engine, it is advisable to consider only high-frequency components, since low-frequency components do not affect the practically important acoustic characteristic of the machine - the sound level.

3. Results and Discussions

As a normalized parameter of vibration created by an electric machine, the effective value of the vibration velocity is adopted. In contrast to the sound level, the overall level of vibration velocity is equally determined by both high-frequency and low-frequency components of vibration. Thus, despite the relatively low level of magnetic vibration of high frequency, because of its decisive influence on the magnetic noise of single-phase motors, this component is of independent interest for research. Electric motors used in electric rolling stock operate in more difficult conditions in comparison with general industrial ones. Increased vibration also affects the work, so the requirements for electric motors in an electromotive composition are very high. In the electric rolling stock, various auxiliary electric machines are used [5, 9, 12, 15]. These machines can use single-phase induction motors. These engines are also affected by various internal and external forces. To justify the calculated expressions of the dynamic vibration damper, the dimensions of the latter are determined by the value of the required stiffness of its elastic element $k_b$, determined by the expression;

$$k_a = \frac{k_e \left( \frac{V_T}{V_T} - 1 \right)}{\eta_g (1 - \eta_e)} \quad (1)$$

Included in this expression, the engine installation stiffness $K_g$ and the dynamic coefficient $\eta_g$ are related to the natural oscillation frequency of the engine $f_g$, which characterizes the engine installation conditions during operation [16, 17, 21].

To identify the relationship between the size of the vibration damper and the engine installation conditions, we transform the expression (1), bearing in mind two main cases that may occur in practice:

- sets the required multiplicity of tangential vibration reduction using a vibration damper ($\frac{V_T}{V'_T} = \text{const}$);
- set the maximum permissible value of the tangential vibration of the motor with a vibration damper ($V_T = \text{const}$).

Considering the first case, we substitute the values of the rigidness of the installation and the dynamic coefficient of the engine in the expression (1). In this case, for the frequency of the network $f_c = 50 \text{ Hz}$, we get

$$k_a = \frac{4\pi^2 \theta_g \left( \frac{V_T}{V_T'} - 1 \right)}{1 - \eta_a} \left( f_g^2 - 10^4 \right) \quad (2)$$

In this expression, the value

$$z = (f_g^2 - 10^4) \quad (3)$$

Describes the installation conditions of the engine, and the rest is a constant value for this engine [18, 19]. For ease of analysis, see Fig.1 the dependence $Z = F(f_g)$ is presented. From this dependence, it follows that in the pre-resonant region of the motor installation ($f_g < 100 \text{ Hz}$-elastic installation), with an increase in $f_g$, the $Z$ value decreases, which leads to a decrease, $\theta_g$ and $m_g$, therefore, i.e., the size of the vibration damper. But increasing $f_g$ increases the value of the tangential vibration of the engine $V_T$, and also $V_T'$, since it is set that ($\frac{V_T}{V_T'} = \text{const}$).

Following the expression $\theta_g$, a decrease with simultaneous $V_T'$ increase leads to a sharp increase in $g_m$, which means a significant increase in the mechanical stress in the elastic element of the vibration damper. The latter circumstance limits the reduction of the size of the vibration damper when approaching the resonance.

In the resonant region (the area of rigid motor installation, $f_g > 100 \text{ Hz}$), the $Z$ value increases with increasing $f_g$ (Fig. 1), which leads to an increase $k_a$ in the required size of the vibration damper, to maintain $\theta_g$ and $m_g$ a constant multiplicity of vibration reduction. Since the increase and decrease, $f_g$, $V_T$, and $V_T'$ increase, the stress in the elastic element of the $\theta_g$ vibration damper is significantly reduced and its size is determined only by the required efficiency of vibration cancellation [17, 19, 20].

It should be noted that with $f_g$ entrainment in the resonant region, the value of the tangential vibration of the engine itself decreases. Therefore, there is reason to believe that the requirement of invariability of the multiplicity of vibration reduction with the help of a vibration damper while increasing the rigidity of the engine installation is relatively rare in practice, namely, in the case of particularly strict restrictions on the permissible level of vibration. In the latter case, as well as when the engine is operating in conditions close to resonant, when the size of the vibration damper may increase significantly, due to the strength condition, it may be recommended (to reduce the required size of the vibration damper) to use a vibration damper in combination with vibration isolation.

Consider another case-condition $V_T = \text{const}$. In this case, we will substitute the value $V_T$ in expression (2). After simple transformations, we get:

$$k_a = \frac{4\pi^2 \theta_g}{1 - \eta_a} \left[ \frac{1}{V_T'} - \left( f_g^2 - 10^4 \right) \right] \quad (4)$$

In this expression, as in (2), the value $Z$, being a function of $f_g$, characterizes the conditions (stiffness) of the engine installation.
In the pre-resonant zone, the $Z$ value decreases as $f_g$ increases (Fig.1), which in this case leads, following (3), to an increase in the size of the vibration damper. It is obvious that the tension in the elastic element, following (1), this reduces the size of damper required are only defined valid value for engine vibration with a damper $V'_{T}$.

In the resonant zone, the $Z$ value increases with increasing engine installation stiffness, which leads to a decrease in the $k_b$ and the size of the vibration damper. In this case, the tension in the elastic element increases, which limits the reduction of the size of the vibration damper.

Thus, the size of the vibration damper depends on the installation conditions of the engine during operation and can be determined using the formulas given in the previous sections, taking into account specific requirements.

The design of the vibration dampener should provide convenient adjustment and adjustment of its own frequency [18, 21]. The vibration damper is adjusted by moving the load along the elastic element. The geometric shape of the load of the vibration damper and its elastic element are determined by specific conditions. For example, an elastic element in the form of a flat plate is preferable in the case of attaching a vibration damper to the engine edge using a bolted connection. In this case, the most appropriate is the cylindrical shape of the cargo. A vibration damper of this design was used in the tests in this work. It is necessary to emphasize the requirement of high reliability of the vibration damper attachment to the engine, since this determines the stability and, to a certain extent, the efficiency of operation damper’s [4, 20, 21].

When installed on the body of a single-phase motor, the vibration damper must be located:
- In a plane perpendicular to the axis of rotation of the engine, the axis of symmetry of the vibration damper must pass through the axis that coincides with the axis of rotation of the engine;
- the load symmetry axis of the vibration damper must be $90^0$ with an axis parallel to the engine rotation axis.

The latter condition is not necessary only in the case of spherical shape of the load and a circular cross-section of the elastic bond.

If the first condition is not met, in this case, the elastic force of the spring of the vibration damper $F_b$ will only compensate for the component of the vibration-inducing force $F_i$:

$$F_i = F_T \cos \beta_i$$  \hspace{1cm} (5)
Where: $F_T$ is the tangential ejection due to the pulsating moment; 
$\beta_1$ is the angle that determines the deviation from the first condition of the vibration damper. 

Laying out the unbalanced component of the vibration force

$$F_2 = F_T \sin \beta_1$$  (6)

we find an unbalanced tangential component of the vibration force:

$$\Delta F_{T1} = F_2 \sin \beta_1 = F_T \sin 2 \beta_1$$  (7)

and an additional radial component:

$$F_R = F_2 \cos \beta_1 = F_T \sin \beta_1 \cos \beta_1$$  (8)

Thus, failure to comply with the first condition for installing a vibration damper leads to a decrease in the efficiency of quenching tangential vibration, as well as to the appearance of an additional radial component of vibration with the frequency of the pulsating moment.

4. Conclusion

1. The dimensions of the vibration damper depend on the installation conditions of the engine during operation and can be determined by the formulas $k_s = \frac{4 \pi^2}{T_n^2} - \frac{f^2}{10^4}$ taking into account specific requirements.
2. When installed on a single-phase motor housing, the vibration damper should be located: - in the plane perpendicular to the axis of rotation of the engine, while the axis of symmetry of the vibration damper must pass through the axis that coincides with the axis of rotation of the engine; - the axis of symmetry of the load of the vibration damper should be $90^\circ$ (363 K) with the axis parallel to the axis of rotation of the engine.
3. Failure to comply with the first installation condition of the vibration damper leads to a decrease in the efficiency of damping tangential vibration, as well as to the appearance of an additional radial component of the vibration with the frequency of the pulsating moment.

References

[1] Adamenko N I 1975 Problems and methods of optimization of series of asynchronous machines of low power In the collection Problems of technical electrodynamics Vol 52 Kiev
[2] Alymkulov K. 1995 Single-phase asynchronous motors for small-scale electric drives Bishkek
[3] Marzas 1974 Investigation of noise and vibration and energy indicators of asynchronous motors powered by a single-phase network Kaunas
[4] Usmanhodzhaev N M 1980 Methods of speed control of single-phase capacitor asynchronous motors p 120 Energy
[5] Isamukhamedov Z Sh and Khadzhinova M U 1986 Multi-Rotor asynchronous motors for driving spindles of a cotton harvesting machine Tashkent
[6] Usmanhodzhaev N M, Sagitov P I and Belakovskii R I 1989 The Theory of multimotor asynchronous electric drive p 176 Tashkent
[7] Usmanhodzhaev N M 1995 Electrical machines and energy saving UZB. journal "Problems of Informatics and energy No 1 p 5
[8] Berdiev U T 2015 Peculiarities of operation of a frequency-controlled traction asynchronous motor Bulletin No 1 pp 78-80
[9] Aliyev I I 2004 Asynchronous motors in three-phase and single-phase modes p 128 Moscow, IP
"Radiosoft"

[10] Katsman M M 1984 Calculation and design of electric machines p 360 Moscow, Energoatomizdat

[11] Rajagopal M S, Seetharamu K N and Ashwathnarayana P A 1989 Transient thermal analysis of induction motors IEEE No 1 pp 62-70

[12] Retter G J The dynamic analysis of partially asymmetric machines Archiv für Elektrotechnik 60 pp 69-78

[13] Sabir A 1989 Theory of zero-sequence performance in induction machines without/with multiple armature reaction Electric Machines and Power Systems 17 No 5 pp 295-313

[14] Vas P 1978 Generalized transient analysis of induction motors Archiv für Elektrotechnik 60 pp 307-312

[15] Vas P 1981 Generalized zero-sequence and anti-zero-sequence performance of induction machines Archiv for Elektrotechnik 63 pp 357-361

[16] Vas P Modified symmetrical components theory and its application in the theory of asymmetric induction motors Period, polytechn. Elec. Eng 22 No 1 pp 3-12

[17] Vas P and Vas J 1977 Transient and steady-state operation of induction motors with stator asymmetries Archiv für Elektrotechnik 59 No 2 pp 121-127

[18] Vas P and Vas J Transient and steady-state operation of induction motors with stator asymmetries Archiv für Elektrotechnik 59 No 1pp 55-60

[19] Vaske P 1963 Über die Drehfelder und Drehmomente Symmetrischen Komponenten in Induktionsmaschinen Archiv für Elektrotechnik 48 No 2 pp 97-117

[20] Venkata Rao P 1964 Transient analysis of single-phase induction motors By ASIA publishing house p 146

[21] Semenchuk, Khechumyan V K 1990 Electrical. prome. Ser. 01. Electric machines: an Overview, inform. Vol 29 pp 1-80