THEORETICAL AND EXPERIMENTAL ANALYSIS OF HIGH-FREQUENCY VIBRATION ISOLATION FOR ELECTRIC MOTOR

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Modern variable speed electric motors often produce unacceptable high levels of vibration due to the harmonics of switching frequency in power source. Two-stage mounting system can isolate high-frequency vibration very effectively by using two sets of isolators and an intermediate mass. But the weight and dimension of the intermediate mass may be unaffordable in many applications. In this paper the effect of mounting system with simplified intermediate masses on electric motor high-frequency vibration isolation is studied theoretically. Qualitative analysis results show that mounting system with simplified intermediate mass may obtain better isolation effect than conventional mounting system at high frequencies. A two-stage mounting system with simplified intermediate mass and hard elastic upper set of isolators was manufactured and tested on an electric motor. Experimental results show that over 40 dB isolation effect can be obtained at the switching and harmonic frequencies.

1. Introduction

Modern electric motors often use inverters to control rotational speed. The inverters usually implement AC-to-DC and DC-to-AC conversions to alter the frequency of the power supply. In the DC-to-AC conversion, the inverter switches the DC voltage at high frequencies, to output the desired waveform to control the motor speed. So the output waveform inevitably contains harmonics related to the switching frequency. These harmonics act on the motor and produce unacceptable levels of noise. Lewis presented the structure-borne noise level of an AIM naval propulsion motor supplied by a PWM converter, which showed that vibration at 2 kHz carrier frequency was dominant in the spectrum. In Widdle’s experiments revealed that the fluctuating component of motor output torque was dominated by switching harmonics of multiples of roughly 5 kHz.

In most cases, the high-frequency vibration of electric motor is unwanted and should be attenuated, either by optimization of motor design, or by vibration insulation methods. In naval applications, isolating the vibration of electric motors from ship hull by mounting systems is a very extensive and effective measure. Ideally the vibration transmitted to the hull should be attenuated to such low levels that it would neither affect the operation of other onboard devices such as sonar system, nor induce excessive radiated underwater noise. Machinery resiliently supported by rubber or mental isolators constitutes a most common mounting system. But at high frequencies, standing-wave resonances occur in isolators and the isolation effect would be diminished. Experiments conducted on engine mounting system showed that about 30 dB attenuation of vibration acceleration...
can be obtained at most at frequencies above 1 kHz.\textsuperscript{4} With regard to onboard electric motor with high-frequency vibration exceeding 120 dB, much more attenuation is needed. To this end two-stage mounting systems should be applied to obtain over 40 dB isolation effect.

Two-stage mounting system is composed of upper and lower sets of isolators, between which an intermediate mass is inserted. Most intermediate masses adopt frame designs incorporating extended structural members, which lead to weight and space penalties that may be unaffordable in many applications. A comprise is using simplified intermediate mass which is smaller and lighter than conventional one, and may have better high-frequency isolation effect due to less resonances, as pointed out by Ungar.\textsuperscript{5}

This paper presents theoretical and experimental analysis of high-frequency isolation effect of a simplified two-stage mounting system for electric motor. The isolation effect of conventional and simplified two-stage mounting system are compared qualitatively by theoretical model. A prototype of simplified mounting system for an electric motor of 9-ton weight was manufactured and tested. Compared with conventional two-stage mounting system, the prototype adopts a different design which is characteristic of large-stiffness upper isolators, low-frequency air springs as lower isolators and two simplified intermediate masses on each side of the motor. Experimental results showed that over 45dB isolation effects can be obtained at the switching and harmonic frequencies.

2. Theoretical model

\begin{figure}[h]
  \centering
  \includegraphics[width=0.4\textwidth]{conventional.png}
  \caption{Schematics of a conventional two-stage mounting system.}
  \label{fig:conventional}
\end{figure}

\begin{figure}[h]
  \centering
  \includegraphics[width=0.4\textwidth]{simplified.png}
  \caption{Schematics of a simplified two-stage mounting system.}
  \label{fig:simplified}
\end{figure}

A conventional and a simplified two-stage mounting system are shown schematically in Figs. 1 and 2. It can be seen that the weight and dimensions of the simplified system are much smaller that of the conventional system. In the high frequency range, the isolation effect of the mounting system is determined by the acoustic wave characteristics in the structure of each component which can be represented by mechanical mobility. We denote the mobility of the machinery as $M_s$, the upper isolator as $M_{11}$, the intermediate mass as $M_m$, the lower isolator as $M_{12}$, the base as $M_R$.

The isolation effect of the mounting system is evaluated by force transmissibility $E$, which is the ratio of forces transmitted to the base with and without mounting system. Suppose that the upper and lower isolators are massless, then the isolation effect can be express as

\begin{equation}
E = \left[1 + \frac{M_{11} + M_{12}}{M_s + M_R} + \frac{(M_{11} + M_s)(M_{12} + M_R)}{M_m(M_s + M_R)}\right].
\end{equation}
By using Eq. (1), the transmissibility of conventional and simplified of two-stage mounting system can be compared qualitatively. Considering that the mobilities of other components in Eq. (1) are invariant, then the isolation effect is only determined by $M_m$. Apparently a smaller mobility means a better isolation effect. To simplify analysis, the intermediate masses of conventional and simplified mounting system are considered as uniform thin plate and beam respectively.

The driving-point mobility of an infinite plate in flexure is \[ M_{mp} = \frac{1}{8\sqrt{Dm_p}}, \] where $D = Eh^3/12(1 - \nu^2)$ is the bending stiffness per unit length and $m_p = \rho h$ is the mass per unit area of the plate.

The driving-point mobility of an infinite beam in flexure is \[ M_{mb} = \frac{1}{2(\frac{E}{I})^{1/4} \omega^{1/2} m_b^{3/4}(1 + j)}, \] where $E$ is the elastic modulus and $I = h^3/12$ is the second moment of area per unit width of the beam.

By comparing Eqs. (2) and (3), one can immediately know that for conventional dimensions, the driving-point mobility of a beam is much smaller than a plate at high frequencies, because the frequency $\omega$ appears in the denominator of Eq. (3).

The selection of stiffness or natural frequencies of upper and lower isolators is also an important problem in mounting system design. Eq. (1) shows the effect of upper and lower isolators’ mobilities on isolation effect. Take massless rubber isolator for example, its mobility is \[ M_{fr} = \frac{j\omega h k}{ES}, \] where $h$, $E$, $S$ are thickness, modulus and section area of rubber material in the isolator, $k$ is a constant related to isolator geometry. Eq. (4) shows that the mobility of rubber isolator is inversely proportional to the modulus, which should be as low as possible to increase isolation effect. But the decrease of modulus is limited by the requirements of static deflection and stability of mounting system in many applications. To improve stiffness of mounting system, it is desirable to increase the stiffness of upper isolators to ensure that $M_{I2} \gg M_R$ from the perspective of isolating base from vibration. Another concern in selection of isolators is wave effect which is hard to predict in practice. Assume that the isolator is an uniform rod, its resonance frequencies is \[ \omega_i = i\pi\omega_0\sqrt{\frac{m}{\rho Sh}}, \] where $\omega_0$ is fundamental frequency, $m$ is the mass of machinery, $\rho$ is the density of elastic material. Eq. (5) shows that the smaller the material density, the larger the interval of resonance frequencies. So elastic material of small density such pressured air used by air spring may help to reduce wave effect.

3. Experiment

3.1 Experimental apparatus

The simplified two-stage mounting system prototype and the electric motor are shown in Fig. 3. The electric motor is a 600 kW induction motor of 9-ton weight. It uses solid-state inverter to
alter the frequency of the power supply, with the switching frequency being 2.5 kHz. The mounting system prototype is installed between the motor and the base structure.

Twenty rubber isolators are placed under the motor and provide a maximum load capacity of 10 tons. The natural frequency of upper isolator of conventional mounting system usually ranges from 3 Hz to 10 Hz. But this is not necessary for electric motor for two reasons. One is that low frequency upper isolator decreases the system stability, and thus may be disadvantageous in naval applications. Ship machinery mounted by such isolators tends to be damaged in heavy sea and must use highly flexible connections with other onboard devices. The other reason is that the main vibration of electric motor is at high frequencies and can be attenuated considerably by large stiffness isolators. So the mounting system prototype uses rubber isolator with natural frequency of about 30 Hz as upper isolator.

The simplified intermediate mass is placed under the rubber isolators on each side of the motor. Conventional designs are usually an integrated steel frame instead of two separate beam-like structures in Fig. 3. But the weight of integrated frame often exceeds 30% of machinery weight and the dimension of it would claim more installation space, which are both adverse in naval applications. In comparison, two simplified intermediate masses of the prototype weigh about 350 kg which amount to only 3.9% of the motor weight, and apparently are much more compact and can be easily installed.

Ten air springs are installed between the intermediate masses and the base, with an angle of 30 degrees relative to the horizontal plane. One reason of choosing air spring is that it has a natural frequency of about 5 Hz, which is low enough to isolate vibration induced by major exciting frequencies of the motor. The other reason is that wave effect at high frequency is usually not obvious in the air spring because it uses pressured air as elastic medium.

The base is designed to represent finite impedance structure as in most applications. The isolation effect of mounting system is often diminished by the flexibility of source and base.

To measure isolation effect of the prototype, three groups of accelerometers are arranged on the mounting system prototype. Fig. 4 shows the arrangement of accelerometers at the motor feet, intermediate mass and base structure respectively.

### 3.2 Experimental results

The motor operated at the rotational speed between 100 r/min and 400 r/min and the vibration of the mounting system was recorded. The vibration measured at 200 r/min is shown in Fig. 5 from which it can be seen that peaks occur 2.5 kHz, 5 kHz and 10 kHz, i.e. the switching and harmonic frequencies.
As has been predicted, the isolation effect of rubber isolators is neglectable at low frequencies between 0-500 Hz because of its high natural frequency. But at high frequencies above 1 kHz, the isolation effect is quite obvious. Tab. 1 gives the high-frequency isolation effect of rubber isolators at different motor speeds. The 1/3 octave bands listed in Tab. 1 are selected because the motor’s high-frequency vibration energy is dominant in these bands. The isolation effect of each frequency band is relatively stable under different motor vibration levels at different speeds. Average isolation effect of over 15 dB can be obtained except at 5 kHz band, in which the isolation effect is obviously less than that of other bands for 6-9 dB. It can be reasonably inferred that wave effect may occur in the rubber isolators at about 5 kHz.

**Table 1.** Isolation effect of rubber isolators at high frequencies.

<table>
<thead>
<tr>
<th>Motor speed</th>
<th>1/3 octave band</th>
<th>2.5 kHz</th>
<th>5 kHz</th>
<th>8 kHz</th>
<th>10 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 r/min</td>
<td></td>
<td>18</td>
<td>9</td>
<td>15</td>
<td>18</td>
</tr>
<tr>
<td>200 r/min</td>
<td></td>
<td>17</td>
<td>9</td>
<td>15</td>
<td>19</td>
</tr>
<tr>
<td>300 r/min</td>
<td></td>
<td>17</td>
<td>10</td>
<td>15</td>
<td>19</td>
</tr>
<tr>
<td>400 r/min</td>
<td></td>
<td>15</td>
<td>9</td>
<td>16</td>
<td>19</td>
</tr>
</tbody>
</table>

The air springs exhibit excellent isolation effect at either low or high frequencies in the range of 10-10 kHz. The isolation effect at 20 Hz even reaches 41 dB, making the vibration of base structure as quiet as background. The high-frequency isolation effect of air springs is presented in Tab. 2. The isolation effect is 18-32 dB at different speeds, which is generally much higher than that of rubber isolators. No obvious wave effect is observed in the air springs in 10-10 kHz.

**Table 2.** Isolation effect of air springs at high frequencies.

<table>
<thead>
<tr>
<th>Motor speed</th>
<th>1/3 octave band</th>
<th>2.5 kHz</th>
<th>5 kHz</th>
<th>8 kHz</th>
<th>10 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 r/min</td>
<td></td>
<td>18</td>
<td>32</td>
<td>26</td>
<td>30</td>
</tr>
<tr>
<td>200 r/min</td>
<td></td>
<td>20</td>
<td>31</td>
<td>26</td>
<td>28</td>
</tr>
<tr>
<td>300 r/min</td>
<td></td>
<td>20</td>
<td>31</td>
<td>26</td>
<td>30</td>
</tr>
<tr>
<td>400 r/min</td>
<td></td>
<td>23</td>
<td>32</td>
<td>26</td>
<td>29</td>
</tr>
</tbody>
</table>
The total high-frequency isolation effect of the two-stage mounting system prototype is listed in Tab. 3, by adding the isolation effect of rubber isolators and air springs inTabs. 1 and 2. It can be seen that over 40 dB vibration attenuation was obtained at 5 kHz, 8 kHz and 10 kHz, i.e. the frequencies at which the motor’s most vibration energy are focused and three highest peaks occur in the spectrum. The isolation effect at 2.5 kHz, the switching frequencies, is about 37 dB which is less than that at other frequencies. But note that the motor’s vibration is also much less at 2.5 kHz, and the base vibration has been attenuated to about 70 dB, a level which is very quiet in most applications.

<table>
<thead>
<tr>
<th>Motor speed</th>
<th>2.5 kHz</th>
<th>5 kHz</th>
<th>8 kHz</th>
<th>10 kHz</th>
</tr>
</thead>
<tbody>
<tr>
<td>100r/min</td>
<td>36</td>
<td>41</td>
<td>41</td>
<td>48</td>
</tr>
<tr>
<td>200r/min</td>
<td>37</td>
<td>40</td>
<td>41</td>
<td>47</td>
</tr>
<tr>
<td>300r/min</td>
<td>37</td>
<td>41</td>
<td>41</td>
<td>49</td>
</tr>
<tr>
<td>400r/min</td>
<td>37</td>
<td>41</td>
<td>42</td>
<td>48</td>
</tr>
</tbody>
</table>

4. Conclusions

The high-frequency vibration of modern variable speed electric motor related to switching frequency often exceeds 120 dB and needs to be attenuated effectively. A simplified two-stage mounting system is proposed in this paper to isolate high-frequency vibration of electric motor. This system uses beam-like intermediate masses instead of frame structure of conventional system to reduce resonances, as well as weight and dimensions. The possible advantage in high-frequency isolation of beam-like intermediate mass is analyzed qualitatively by theoretical model. A simplified two-stage mounting system prototype was manufactured and tested on an induction motor. The upper isolators are rubber isolators with natural frequency of 29 Hz, and the lower isolators are air springs with natural frequency of 5 Hz. Experimental results show that the rubber isolators and air springs attenuate high-frequency vibration effectively, except that the possible wave effect occurs in rubber isolators at 5 kHz. Over 40 dB isolation effect in total can be obtained at 5 kHz, 8 kHz and 10 kHz, i.e. the switching and harmonic frequencies.

REFERENCES