Modeling and Design of Anti-Impact Vibration System for Clamp-On Optical Current Transformers

YE Yuan-bo¹, ZHENG Hao¹, WANG Wei¹, WANG Gui-zhong², LI Hong-bo² and ZHANG Guo-qing³

¹ State Grid AnHui Electric Power Company, Hefei 230601, China;
² Harbin Institute of Technology at Zhangjiakou, Zhangjiakou 075421, China;
³ School of Electrical Engineering and Automation, Harbin Institute of Technology, Harbin 150001, China.
E-mail: guoqingz@126.com; yeyb7079@ah.sgcc.com.cn; wgz2003@sina.com; lihongbo19790127@126.com

Abstract. Optical current transformers, which possess excellent transient characteristics, are able to accurately reflect the full waveform of primary current, and have a wide range of potential applications. The clamp is the sensing unit of the optical current transformer that is mounted directly to the gas-insulated switchgear (GIS) casing and has advantages such as simple structure and ease of maintenance. However, vibration of the casing due to the operation of the GIS circuit breaker has a significant effect on the output of the optical current transformer, and may cause the malfunction of protective relays. This paper proposes the use of damping rubber as the main means of reducing vibration in transformers. A vibration damping system for the clamp-on optical current transformer was designed using the optimal parameters obtained from the modeling and analysis of an anti-impact vibration system. A test system was set up to perform experiments on the designed vibration damping system.

1. Introduction

Optical current transformers(OCTs) are based on Faraday’s magneto-optical effect. They have garnered wide industrial attention due to their numerous advantages; some of which include they do not experience saturation, have wide frequency bandwidth, wide dynamic range and natural insulation, and are not affected by electromagnetic interference. A clamp-on OCT mounted to a gas-insulated switchgear (GIS) casing has a simple structure, is easy to maintain, and can be replaced under live-line conditions, thus having a wide range of potential applications[1-3].

An optical current transformer is composed of numerous elements, and the optical components in its sensing unit are highly sensitive to vibration. Impact vibration may cause deformation of components in the sensing unit and lead to relative internal displacements, resulting in changes in output. In more serious scenarios, it may even cause the malfunction of protective relays, thus threatening the grid security [4-10]. This is shown in figure 1.

This paper proposes a method that can enhance the ability of clamp-on optical current transformers to resist impact vibration. Using a BE-type rubber pad as a vibration isolation material placed between the busbar casing and the measuring ring of the optical current transformer allows the system to absorb vibrational energy and reduce the effect of impacts on the optical measurement elements. The vibration isolation capability of the system depends on the properties of the insulation material and the
design of the vibration damping system. In this paper, modeling and analysis of the anti-impact vibration system was performed, and the system was experimentally tested and validated.

2. Sources of vibration and its effect on the performance of optical current transformers

Electronic transformers are composed not only of primary elements such as primary sensors and primary converters, but also of secondary parts such as secondary conversion and merging units. Hence, vibration tests on electronic transformers should include vibration tests on both primary and secondary parts. According to Clause 8.13.1 of Standard No. GB/T 20840.8, vibration tests on secondary parts should be performed in accordance with Standard No. GB/T 2423.10 titled Environmental Testing for Electric and Electronic Products. During the operation of its primary parts, the electronic transformer will experience two types of vibrations. First, vibration resulting from jitters in conducting wires due to electrodynamic forces caused by short-time current when the system experiences a short circuit. The second type of vibration is caused by the operation of switches, such as circuit breakers and isolating switches. For the first type of vibration, Clause 8.1.3.2 of Standard No. GB/T 20840.8 stipulates that the short-time current test and composite error test should be performed simultaneously, whereas operating time tests and vibration fatigue tests are performed for the second type of vibration. Clause 8.13.4.3 of Standard No. 20840.8 also stipulates corresponding provisions with regard to the vibration fatigue test.

Vibration affects electronic instrument transformers in two ways. First, by direct effects, which lead to the malfunction of protective relays due to incorrect (or invalid) output generated by electronic transformers during and after vibration. Second, by cumulative effects, which lead to excess outputs from meters and protective relays due to fatigue in electronic transformers caused by the accumulation of long-term vibration.

![Waveform of impact signal](Figure 1. (a) Waveform of impact signal when the circuit breaker is switched on (b) Waveform of impact signal when the circuit breaker is switched off.

3. Mathematical modeling of clamp-on optical current transformers

3.1 Physical Structure of the Vibration Damping System

A clamp-on optical current transformer is composed of a measuring ring and a busbar casing. The measuring ring comprises precision optical measuring elements in a ring-shaped structure. Impact vibration can result in relative displacements among the optical measuring elements. Changes in coupling efficiency leads to changes in light intensity, which in turn generates surge currents which may trigger the malfunction of protective relays. The busbar casing has a cylindrical structure and is composed of numerous devices including busbars, switches, circuit breakers, and insulators. Vibration primarily results from jitters in busbars when switches in the casing are operated or short-circuited. This type of vibration is known as impact vibration. In actual applications, the clamp-on measuring ring wraps around the busbar casing, as shown in figure 2.
Figure 2. Physical structure of the clamp-on optical current transformer.

When a switch in the casing is either being turned on or off, the busbar strongly jitters, causing impact vibration in the casing. Impact vibration results in the relative displacement of optical elements in the clamp-on measuring ring, which in turn causes sudden changes in the measured value of current, leading to the formation of surge currents. To reduce the effect of impact vibration on the measurement precision of optical elements and the resulting damage to equipment, the equipment should be equipped with impact protection or vibration isolation. In actual applications, placing a vibration damping rubber pad between the busbar casing and the clamp-on measuring ring can effectively buffer the instantaneous effects of impact vibration. After adding the vibration damping rubber pad, it was found that the vibration response (equivalent to current) of the clamp-on optical current transformer was significantly reduced, as shown in figure 3.

Figure 3. (a) Surge current before adding vibration isolation  (b) Single surge current after adding vibration damping rubber pads.
3.2 Modeling and Analysis of Anti-Cyclic Vibration of the Vibration Damping System

The study of the vibration characteristics of a rigid body aims to investigate the changes in speed, acceleration and displacement of the vibration isolators over time. Considering that the vibration isolation material is rubber and that the vibration isolation mode is a single-degree-of-freedom vibration isolation system, the simplified mechanical model of the anti-vibration system of the optical current transformer is shown in figure 4.

Figure 4. Simplified mechanical model of the anti-impact vibration system.

In this model, k is the elasticity coefficient in units of N/m; c is the damping ratio in units of N/(m/s^2); x is the displacement of the vibration isolator, which is the clamp-on measuring ring; u is the displacement of the source of vibration, which is the busbar casing; and m is the mass of the vibration isolator.

In vibration mechanics, we define the natural frequency as \( \omega = \sqrt{\frac{k}{m}} \), the critical damping coefficient as \( c_c = 2 \sqrt{mk} \) and the damping ratio as \( \xi = \frac{c}{c_c} \).

In the vibration isolation system, transmission rates are often used to measure the effect of vibration. Among these, the vibration transmission rate and impact transmission rate are usually considered. The lower the transmission rates, the better the vibration isolation[11].

Vibration transmission rate is defined as the ratio of the amplitude of the response to the amplitude of the input excitation:

\[
T = \frac{R_{out}}{R_{in}}
\] (2.1)

Impact transmission rate is defined as the ratio of the maximum amplitude of the transient response to the peak of impact excitation:

\[
T_{pulse} = \frac{R_{out-max}}{R_{in-max}}
\] (2.2)

By performing force analysis based on a simplified mechanical model, where the resultant force is

\[
m \ddot{x} = m \ddot{x} = k[(-x) - (-u)] - f_f
\] (2.3)

and the linear damping force is

\[
f_f = c(v_x - v_u) = c(\dot{x} - \dot{u})
\] (2.4)

the following equations of motion of a passive vibration isolation system can be obtained:

1. Absolute displacement equation:

\[
m \dddot{x} + c \dddot{x} + k \dddot{x} = c \dddot{u} + ku
\] (2.5)
From which we obtain the frequency characteristics:

\[ G_i(\omega) = \frac{k + j\omega c}{k + j\omega c - j\omega^2} \]  

(2.6)

and the amplitude characteristics:

\[ R_i(\omega) = \left[ \frac{\omega^2 (1 + 4\xi^2 \omega^2)}{(\omega_s^2 - \omega^2) + 4\xi^2 \omega^2} \right]^{\frac{1}{2}} \]  

(2.7)

By substituting the frequency ratio, \( g = \frac{\omega}{\omega_s} \), into the equation above, we obtain the absolute vibration isolation transmission rate:

\[ T_i(g) = \left[ \frac{1 + 4\xi^2 g^2}{(1 - g^2)^2 + 4\xi^2 g^2} \right]^{\frac{1}{2}} \]  

(2.8)

Each corresponding value of \( \xi \) has a characteristic curve of absolute transmission rate, \( T_i(g) \) against frequency ratio \( g \). By setting different representative values of \( \xi \), the curves corresponding to these values can be plotted on the same axes, as shown in figure 5.

![Figure 5. Absolute transmission rate curves.](image)

Based on the figure above, it was found that only when \( g > \sqrt{2} \), and the absolute transmission rate, \( T_i(g) < 1 \), will the system possess vibration isolation. As the frequency ratio \( g \) increases, \( T_i(g) \) decreases, and the effect of vibration isolation of the system gradually improves.

### 3.3 Analysis of Anti-Impact Vibration in the Vibration Damping System

Impacts can be categorized into two types, base impacts and impacts caused by external forces. The mathematical descriptions of both types are essentially the same, and the variations in acceleration of anti-impact objects are also consistent [12]. In this paper, the vibration isolation material involved is BE type rubber material, which possesses linear damping properties. For an impact vibration isolation damping system, the operating equation of the system can be represented by:

\[ m\ddot{x} + c\dot{x} + kx = f(t) \]  

(2.9)

which can be simplified to:

\[ m\ddot{x} + c\dot{x} + kx = 0 \]  

(2.10)
The initial conditions are: \( x(0) = 0, \quad \dot{x}(0) = v_0 \)

There are two impact resistance indicators:

1. \( \delta_{\text{max}} \) is the maximum absolute displacement of the clamp-on measuring ring from the base of the busbar casing, \( x(t) \). It indicates the amplitude characteristics of the output response of the vibration damping system.

2. \( a_{\text{max}} \) is the maximum absolute acceleration of the clamp-on measuring ring. It indicates the relative magnitude of impact force.

In vibration mechanics, the constraint of the vibration damping system is:

\[
\delta_{\text{max}} a_{\text{max}} \geq 0.5v_0^2 \tag{2.11}
\]

When the values of \( \delta_{\text{max}} \) and \( a_{\text{max}} \) are small, the amplitude and impact force of the system response are also small. The vibration isolation at this point is the most effective. When both \( \delta_{\text{max}} \) and \( a_{\text{max}} \) reach the optimal values, their product is also at a minimum [13-14].

By solving these equations, the displacement response can be obtained as:

\[
x(t) = \frac{V_0}{\omega_d} e^{-\xi \omega_d t} \sin \omega_d t = \frac{V_0}{\omega_h \sqrt{1-\xi^2}} e^{-\xi \omega_h t} \sin \omega_h t \tag{2.12}
\]

where \( \omega_d = \omega_h \sqrt{1-\xi^2} \).

By finding the second-order derivative of \( x(t) \), the acceleration of the system response can be obtained:

\[
a = \ddot{x}(t) = \frac{V_0 \omega_d}{\sqrt{1-\xi^2}} e^{-\xi \omega_d t} \sin(\omega_d t - 2 \arctan \frac{\sqrt{1-\xi^2}}{\xi}) \tag{2.13}
\]

Hence, the following can be obtained:

\[
\delta_{\text{max}} a_{\text{max}} = \begin{cases} 
\frac{V_0^2 \sqrt{1-\xi^2}}{\omega_d^2} e^{-\xi \omega_d t} \sin(\omega_d t - 2 \arctan \frac{\sqrt{1-\xi^2}}{\xi}) & (0 \leq \xi \leq 0.5) \\
2 \xi V_0^2 e^{-\xi \omega_d t} & (0.5 < \xi < 1) 
\end{cases} \tag{2.14}
\]

To determine the effectiveness of the damping effect, the parameter for measuring the damping effect is defined as:

\[
\phi = \frac{\delta_{\text{max}} a_{\text{max}}}{v_0^2} \tag{2.15}
\]

Analysis shows that the smaller the value of \( \phi \), the better the vibration damping properties of the system. A characteristic curve of \( \phi \) against \( \xi \) was plotted to find the value of \( \xi \) where the value of \( \phi \) is at a minimum. In theory, the damping effect is the best at this point. The curve is shown in figure 6.
Figure 6. Characteristic curve of $\phi$ against $\xi$.

Based on this figure, it was found that when $0 < \xi < 0.4$, $\phi$ decreases as $\xi$ increases, indicating the damping effect is improving. When $\xi > 0.4$, $\phi$ increases with $\xi$, indicating that the damping effect is deteriorating. Hence, when $\xi = 0.4$, the minimum value of $\phi = 0.52$ is obtained. This value satisfies the constraint of the vibration damping system, $a_{\text{max}} \delta_{\text{max}} \geq 0.5v_0^2$, and $\phi > 0.5$. This demonstrates that the system achieves the best damping condition when $\xi = 0.4$.

3.4 Design of Rubber Vibration Dampers

The vibration damping system of the clamp-on optical current transformer was designed based on the analysis above. Considering the differences between rubber dampers and other vibration dampers, this vibration damping system has the following advantages:

(1) It can be rotated about the x-, y-, and z-axes and has the effect of a six-way spring. Therefore, it can meet the requirements for stiffness in all directions by appropriately designing its structure and adjusting the properties of the rubber.

(2) It has a moderate damping performance, and possesses two different properties, i.e., attenuation and energy absorption. Its damping effect is good, and can easily pass through the resonance zone. It can also be applied to both low and high frequencies.

(3) It performs various functions including vibration damping, buffering, and acoustic insulation.

(4) Its shape is easily varied. It is also light, and is easy to install and disassemble. Its price is reasonable due to its good performance and easy installation.

The vibration damping system uses rubber as its main vibration damping material. The rubber pad was placed between the measuring ring of the clamp-on optical current transformer and the GIS casing as shown in figure 7.

Figure 7. Rubber vibration damper.
From equation (2.8), based on Snowdon’s derivation [15], \( T_1(g) \) is as follows:

\[
T_1(g) = \left[ \frac{1 + 4\xi^2 g^2}{(1 - g^2 K_n/K)^2 + 4\xi^2 g^2} \right]^{1/2} \tag{2.16}
\]

where \( K_n \) is the dynamic stiffness at the resonance frequency \( f_n \) and \( K \) is the dynamic stiffness of the vibration damping material.

The most basic parameters reflected in the design of the rubber damper are its stiffness and damping ratio. Because the damping ratio of rubber is almost constant (though it varies in a specific small range), only data on its stiffness required processing.

To obtain excellent vibration isolation, the vibration isolation system must first have an appropriate intrinsic frequency. After determining the quality of the system, its intrinsic frequency is mainly dependent on the dynamic stiffness, whereas the stiffness of the rubber damper is primarily determined by the elastic modulus and geometric dimensions of the rubber pad. Vibration dampers with complex shapes can be seen as the result of combining several vibration dampers in series or in parallel. The dynamic stiffness of a single cylinder on the rubber pad was observed when it was compressed.

The relationship between the dynamic elastic modulus and the dynamic stiffness of a cylindrical rubber pad is as follows:

\[
G = (Kh/A)(3 + 4.935S^2)^{-1} \tag{2.17}
\]

\[
E = (Kh/A)(1+1.645S^2)^{-1} \tag{2.18}
\]

\[
K = 3G(A/h)(1+1.645S^2) \tag{2.19}
\]

In these equations, \( h \) and \( d \) are the thickness and diameter of the rubber cylinder, respectively:

\[
A = \pi d^2 / 4 \quad \text{is the cross-section area of the rubber cylinder;}
\]

\[
S = d / 4h \quad \text{is the shape factor}
\]

Constrained area of rubber (area compressed by load)

Free surface area (area of unstressed sides)

A damping rubber pad with length 150mm, width 55mm, and base thickness 6mm was chosen. The base of the pad comprised cylinders with two different sets of dimensions, a \( 8\times2 \)mm cylinder and a \( 5\times3 \)mm cylinder, with a central spacing of \( 10\times10 \)mm between them. This is shown in figure 8. The thickness of any two cylindrical points on both sides of the damping pad did not exceed 12mm, whereas the thickness of the damping pad during compression was 10mm, which is the engineered gap between the clamp-on optical current transformer and the tooling.

![Figure 8. Dimensions and structure of the damping rubber pad.](image-url)
The elastic modulus of the rubber is mainly adjusted using fillers, whereas the damping factor of rubber is primarily determined by the type of rubber. The damping properties of the damper are mainly determined by the elastic modulus and damping ratio of rubber. Therefore, the dynamic stiffness of the rubber material is key to selecting the appropriate material. The dynamic properties of rubber usually refer to the dynamic elastic modulus and the loss factor measured under sinusoidal vibration. It directly decides the dynamic stiffness and damping factor of the rubber damper.

4. Testing the vibration damping system of the clamp-on optical current transformer

4.1 Setting up the Vibration Test System
To simulate on-site vibration and test the performance of the designed vibration damping system, a vibration test system for the clamp-on optical current transformer was set up. After assembling the sensor head of the clamp-on transformer, which consisted of the designed rubber damper and the GIS casing, the assembled set was placed on top of a laboratory vibration table, as shown in figure 9.

![Figure 9. Clamp-on optical current transformer and vibration test table.](image)

The RC-2000 STI vibration controller was used to add impact excitation. The control signal used in this experiment was the typical impact signal MIL-STD-810F.

The pulse interval was set to 1.5 s. A fast activation rate was used, and the response level was set to 10%. A low-pass filter was not used, and the 10% increase/decrease step was used. The interrupt decline rate was 20dB/s. The control input channel was used. The range was set to 1.0V, and charge coupling was used, with sensitivity of 2.08pC/(g). The maximum number of points was 4096, and the drive voltage limit was 10V. The peak impact acceleration was 40g, and the impact duration was 10ms. The specific waveform is shown in Figure 9. The green line represents the curve of the standard waveform of the simulation output, whereas the two red lines represent its upper and lower limits. The white line represents the actual vibration curve fed back from the vibration table.

The setup of the entire vibration test system is shown in figure 10.
4.2 Vibration Test Results

We carried out tests to compare the effects of vibration damping. First, the clamp-on optical current transformer was mounted directly to the casing without any vibration damping measures, and a set of vibration responses was collected. Next, the vibration damper was placed between the transformer and the casing, with an identical vibration signal applied to the set, and a second set of vibration responses was collected. The effect of the vibration damping system was obtained by comparing the two sets of vibration responses.

Figure 11 shows the output waveform without vibration damping. It reached a maximum value of approximately 150A (the rated current was 2000A). This value was relatively high and may cause malfunction of the protective relays, thereby threatening grid security. Figures 12 and 13 show the corresponding output waveform of the optical current transformer vibration as the vibration damping measures were continuously improved. After implementing the optimal vibration damping measure, the corresponding output of the optical current transformer vibration was almost drowned out by noise and the peak value was less than 5A, as shown in Figure 14. The vibration transmission rate was less than 0.033, thus meeting standard requirements.
Figure 12. Output waveform without vibration damping on the optical current transformer.

Figure 13. Output waveform 1 after improving the vibration damping measure on the optical current transformer.

Figure 14. Output waveform 2 after improving the vibration damping measure on the optical current transformer.
5. Conclusion
In this paper, a comprehensive analysis was performed on the rubber vibration damping system of an optical current transformer, and a mathematical model of the vibration damping system was constructed. The effects of all the parameters in the model on the vibration damping effect were analyzed and the optimal parameters for the damping system were obtained. A damping rubber pad was designed in detail. By setting up the vibration test system, the test results indicate that the impact transmission rate of the vibration damping system was less than 0.033, thus meeting design requirements and solving the anti-vibration problem in clamp-on optical current transformers. This paper has laid the foundations for a wide range of applications for optical current transformers.

References
[1] Li Shen-wang. 2011, Research on the key technology of optical current transformer for gas insulated switchgear[D]. Harbin Institute of Technology.
[2] Liu Rui-fu and Shi Jin-ce. 1987, Optical fiber sensors and applications. China Machine Press, Beijing.
[3] B. B. Afanasyevskiy et al. 1989, Current transformers (China Machine Press, Beijing).
[4] Yu Wen-bin, Zhang Guo-qing et al. 2012, Analysis of resistance disturbance capability of optical current transformers[J]. Power System Protection and Control, 40 (12):8-12+18.
[5] Zhang De-sai, Luo Dao-jun, Peng Jian. 1999, Present Situation of research on Optical Current Transducer in Domestic & Foreign Countries[J]. Sichuan Electric Power Technology, (02):55-57+62.
[6] Shen Zhu and Luo Cheng-mu. 2001, Recent Progress in electronic current transformers. Automation of Electric Power Systems, 25 (22):59-63.
[7] Zhenping Wang. 2012, Effect of modulation error on all optical fiber current transformers, journal of sensor technology, 2, 172-176
[8] Li Fu-sheng. 2014, Research on all-digital of optical current sensing technology[D]. Harbin Institute of Technology.
[9] Li Yan-song, Zhang Guo-qing, Yu Wen-bin et al. 2003, Combined method to improve the accuracy of optical current transducer[J]. Automation of Electric Power Systems, (19):43-47.
[10] Xu Zhi-wei, Chen Tian-ning, Huang Xie-qing, Chen Hua-ling. 1999, Simulation of Vertical Impact Damper System[J]. Journal of Xi’an Jiaotong University, (07):68-72.
[11] Gu Yao-ying. 2009, Research of the design method of the vibration isolation system based on elastic base. Shanghai Jiaotong University.
[12] Ding Wen-jing. 2014, Theories of vibration damping(2ed). Tsinghua University Press.
[13] Wang Jian and Zhao Guo-sheng. 2016, MATLAB mathematical modeling and simulation. 
Tsinghua University Press.

[14] Li Ming-an and Qian Li. 2015, Modeling and simulation of dynamic systems using 
Matlab/Simulink, National Defense Industry Press.

[15] Wang Gui-yi. 1998, Designing principle and properties test of rubber damper[J]. SpecialPurpose 
Rubber Products, (06):42-47.