Performance Optimization and Loading Rate-dependency of Friction Dampers with Non-metallic Friction Materials

Jingwei Gao¹, Yue Yuan¹*, Tianyi Qiu¹, Chun-Lin Wang¹*, Zhe Qu²

1. Key Laboratory of Concrete and Prestressed Concrete Structures of the Ministry of Education, Southeast University, Nanjing 210096, China
2. Key Laboratory of Earthquake Engineering and Engineering Vibration, Institute of Engineering Mechanics, China Earthquake Administration, Harbin 150080, China

Abstract: Non-Asbestos Organic (NAO) is widely used in automobile braking systems due to its low cost and stable performance. Meanwhile, NAO is also suitable for seismic energy dissipation of building structures, but further research was required. Configuration and loading rate are significant factors affecting the hysteretic performance of the friction damper. Thus, cyclic loading tests were conducted to evaluate the friction behavior of friction dampers with non-metallic friction pads (Non-Asbestos Organic (NAO) and Polytetrafluoroethylene (PTFE)), and the friction pairs were NAO-Steel and PTFE-Steel, respectively. A total of 13 specimens were designed to comparatively analyze the effects of the following factors on the mechanical performance of the friction damper: the configuration of the friction damper, loading rate, and reloading process after a reassembly. Firstly, some unnecessary friction needed to be avoided to keep the stable performance of the damper, such as the contact friction between the slot of the inner steel plate and friction pads or bolts. Secondly, the normal force of both NAO and PTFE friction dampers reduced gradually during the loading process. The former showed a higher cumulative loss rate, which increased with the initial nominal pressure. Again, the dynamic friction coefficients of NAO and PTFE friction dampers tended to be constant under high-speed loading, which was remarkably higher than that under low-speed loading. The dynamic friction coefficients of NAO and PTFE friction dampers were suggested to be 0.229 and 0.121 for structural analysis, respectively. Finally, compared with the first test of the friction damper, the reloading test results after a reassembly showed excellent reusability of both NAO and PTFE friction dampers.

Keywords: Friction damper; Non-metallic friction material; Loading rate; Dynamic friction coefficient; Static friction coefficient; Reassembly

*Correspondence to: Chun-Lin Wang(chunlin@seu.edu.cn), Yue Yuan(seuyuanyue@163.com); School of Civil Engineering, Southeast University, Nanjing, 210096, China.
1 Introduction

In recent decades, various energy dissipation devices have been widely applied to the damage reduction of building structures due to earthquakes and the rapid repair of the damaged regions. Many researchers have suggested installing friction dampers in structures to dissipate energy, and previous studies have shown that the friction dampers had stable mechanical behavior, large initial stiffness, and additional damping provided for the structure [1-3]. The friction force can also be the main source of the lateral resistance of the structure [4, 5]. Meanwhile, there will be no material yield and fatigue problems of the friction damper, which was easy to replace [6].

The Pall-typed frictional damper [7, 8] was the earliest friction energy dissipation device proposed. Recently, friction dampers have been incorporated into more structures to provide energy dissipation, such as the connection of the precast wall-slab-wall structure [9], the corner joint of the self-centering shear wall [10], and the energy dissipation bearings of bridges [11, 12]. The configuration of the friction damper affects its mechanical performance. For the friction damper in reference [13], the distance between the steel plates used to clamp the friction pad was slightly less than the thickness of the friction pad, resulting in the deformation of the steel plates and the reduction of the normal pressure on the friction pad by about 20%. In references [14, 15], the friction pads of some specimens were fixed between the steel plates by glue bonding or bolt clamping, and the damper could not work properly because of the bond failure or fracture of friction pads. Therefore, the configuration of the friction damper, which directly affects the performance, is a research point worthy of attention.

In terms of friction materials, friction pads have been widely made of metals, such as brass[16-18], bronze alloy [19], and aluminum alloy [20]. However, metal friction pads are expensive, and metal friction causes unbearable noise. Some metal friction pairs also need special processing to avoid galvanic corrosion. As a non-metallic material, Non-Asbestos Organic (NAO) has been used to manufacture brake pads in the automotive industry [21, 22]. Relevant studies showed that the friction behavior between NAO and metal was stable, and the strength degradation was slight [23]. Compared to metal brake pads, NAO brake pads had lower wear rates and braking noise [24], and there was no corrosion problem [15]. In addition, the price of NAO materials has gradually decreased, which is suitable for the production of friction dampers in civil engineering. Reference [14] adopted NAO friction dampers to improve the energy dissipation capacity of self-centering moment connections; in reference [25], NAO friction dampers were used to enhance the seismic performance of the steel-timber hybrid shear wall; reference [26] selected the NAO-Stainless steel friction pair for base isolation systems, and its friction performance was stable under constant pressure. Still, there is relatively little
research, and the application of NAO needs further verification.

The choice of friction material is also affected by the mechanical parameters (e.g., friction coefficient) of the friction damper. The friction coefficient, which is considered to be independent of sliding rate, is generally designed to follow the classical Coulomb friction law. However, reference [27] showed that some friction dampers using different friction materials had loading rate-dependency under low-speed loading (≤10mm/s). Reference [28] also observed through tests that the friction coefficient was related to the loading rate, but there was no quantitative research. Reference [15] completed the mechanical tests of five friction pairs composed of non-metallic composite materials and steel. The results showed that the friction coefficient of some friction pairs increases with loading rate due to the wearing process and visco-plastic phenomena. It should be noted that the friction dampers installed in the building structure generally bear a loading rate greater than 50mm/s [29]. Due to limitations of the test device, the maximum loading rate in reference [15] was 27mm/s, which is difficult to meet the demand of structural friction damper on loading rate.

In order to further explore the mechanical performance of the NAO friction damper, eight specimens were processed in this paper, which selected one of the most common configurations [14, 15, 30], as shown in Fig. 1. Firstly, the key characteristics of the damper configuration were discussed through tests. Then, the variation of the friction coefficient was quantitatively analyzed within the range of 0.2mm/s~60mm/s loading rate. Finally, considering the post-earthquake maintenance of the damper, the friction coefficient changes were compared before and after the reassembly of the friction damper.

In addition, considering the diversity of structure forms, friction dampers with different friction coefficients are required. The friction coefficient of Polytetrafluoroethylene (PTFE) friction pads is relatively low (about 1/3~1/2 of NAO friction coefficient [25]), which may make PTFE have a different application in civil engineering. Therefore, five PTFE specimens were also subjected to evaluate the effect of the configuration, loading rate, and reassembly, respectively.

2 Test program

2.1 Specimen design

Fig. 1 shows the configuration of the friction damper, which is composed of a 12mm-thick inner steel plate, two 12mm-thick outer steel plates, 6mm-thick friction pads, and an 18mm-thick fixed plate. The friction pad was placed into a 3mm-thick recess on the left side of the outer steel plate to constitute a friction unit. The friction units and the inner steel plate feature bolt holes and a slot, respectively. Two high-strength bolts with a diameter of 12mm provide
normal force pass through the bolted holes of the friction units and the slot in the inner steel plate. The other four high-strength bolts clamp the right side of the outer steel plate on the fixed plate. A friction damper has two sets of friction pairs, each consisting of an inner steel plate and friction pads. The inner steel plate moves along the left and right direction and slips relative to the friction pad during loading, resulting in friction energy dissipation. It should be noted that two high-strength nuts are used for each high-strength bolt passing through the friction pads to minimize the loss of normal force of high-strength bolts.

![Fig.1. Configuration of the friction damper](image)

Each friction unit consists of friction pads and an outer steel plate. As shown in Fig. 2, four types of friction units were labeled as friction unit x-Y in the paper. The x and Y were the serial numbers for the friction pad and the outer steel plate, respectively, as shown in Fig. 3 and Fig. 4. The design method and improvement details of the friction units are as follows:

1) Severe local abrasion was observed in friction unit 1-A after loading, caused by a significant scraping between the slot edges in the inner steel plate and the friction pads near the bolt holes. The detailed discussions about friction unit 1-A were in Section 3.2. Thus, the Type 1 friction pad in friction unit 1-A was divided into two rectangular friction pads (Type 2 friction pad), and the clear distance $L_b$ between two friction pads was 16mm, as shown in Fig. 3 (b). The friction unit 2-B was proposed based on the Type 2 friction pad.
2) A friction damper, which was composed of two friction units 2-B and an inner steel plate with a 13mm-width slot (Fig. 5(a)), exhibited a remarkable scraping between the high-strength bolts and the slot sidewall during the loading history. More detailed analyses were presented in Section 4.1. Therefore, the clear distance $L_b$ between two friction pads increased from 16mm to 26mm (Type 3 friction pad, Fig. 3(c)), and the slot width $W$ in the inner steel plate increased from 13mm to 21 mm (Inner steel plate W-21, Fig. 5(b)) for the following tests.

Fig. 3 Types and geometric dimensions of friction pads

Fig. 4 Types and geometric dimensions of outer steel plates

Fig. 5 Types and geometric dimensions of inner steel plates

3) The length of the Type 3 friction pad was reduced from 184mm to 150mm (Type 4 friction pad, Fig. 3(d)) to study the influence of friction interface pressure on the hysteretic performance of the friction damper. The friction unit 4-C consisted of Type 4 friction pads and
a Type C outer steel plate (Fig. 4(c)). The influence of friction interface pressure was analyzed in Section 4.3.

The effects of the friction materials and the designed tightening torques of high-strength bolts were further studied based on the above configurations of friction damper. Two non-metallic friction materials were selected: Non-Asbestos Organic (NAO) and Polytetrafluoroethylene (PTFE). The NAO friction pads were glued with structural epoxy in the recesses of the outer steel plates, and the PTFE friction pads and outer steel plate were bonded with a professional glue for PTFE produced by VALIGOO in China. The high-strength bolts passing through the friction pads featured an identical tightening torque, and three torques \( T \) were selected: 105N·m, 135N·m, and 165N·m.

A total of 13 specimens were designed and tested in this paper, and their main parameters were given as listed in Table 1. The \( MxWyTz \) naming rules were adopted for the specimens. \( M \) was the first letter of the friction material, which could be N (NAO) and P (PTFE); \( x \) represented the configurational type of the friction pad, as shown in Fig. 3; \( W_y \) represented the slot width of the inner steel plate was \( y \) mm; \( T_z \) meant that the design torque value of the high-strength bolt was \( z \) N·m. For example, N4W21T105 was the specimen featured Type 4 friction pads, the corresponding friction material of NAO, the slot with a width of 21mm, and a designed tightening torque \( T \) of 105N·m.

<table>
<thead>
<tr>
<th>No.</th>
<th>Specimen</th>
<th>Friction pads</th>
<th>Friction unit</th>
<th>Inner steel plate</th>
<th>Friction interfaces</th>
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<td></td>
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<td></td>
<td></td>
<td></td>
<td>( T/(\text{N·m}) )</td>
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<tr>
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<td>3-A</td>
<td></td>
<td>105</td>
</tr>
<tr>
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<td></td>
<td>3-A</td>
<td></td>
<td>135</td>
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<td>165</td>
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<td></td>
<td>4-C</td>
<td></td>
<td>105</td>
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<td></td>
<td>135</td>
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<td>P1W21T135</td>
<td></td>
<td>1-A</td>
<td></td>
<td>135</td>
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Note: \( T \) is the designed tightening torque of the high-strength bolt passing through friction pad; \( N_0 \) denotes the nominal normal force of two high-strength bolts on the friction interface; \( p \) indicates the initial nominal pressure on the friction interface.
The tightening torque of each specimen was recorded by the calibrated torque wrench. Each bolt exhibited a nominal normal force of approximately 43kN, 54kN, or 65kN when the design tightening torque $T$ was reached. The initial nominal pressure $p$ of the counterpart friction interface is listed in Table 1, and the $p$ met the recommended value of 5MPa~15 MPa in the code [27].

2.2 Test setup and loading protocol

The cyclic loading tests were conducted based on an MTS electrohydraulic testing machine with a loading capacity of ±500kN and a displacement capacity of ±75 mm. As shown in Fig. 6(a), the friction damper was placed vertically, and its inner steel plate and fixed plate were fixed at the top clamp and the bottom clamp of the testing machine, respectively. It was stipulated that the downward movement of the actuator was positive, and the upward movement was negative. As shown in Fig. 6(b), the normal force of the high-strength bolt was recorded by loading cells with the measurement range from 0 to 100kN.

![MTS Base + Bolt A 1(2) Bolt B 3(4) Actuator](image)

(a) Test setup (b) Arrangement of transducers

Fig. 6. Test setup and arrangement of transducers

Two pairs of displacement transducers were mounted on the top and bottom clamps, respectively. The former collected the axial displacement of the loading end of the damper, and the latter measured the possible mechanical slip of the fixed end. The difference between them was the slip displacement of the friction interface. In order to improve the measurement accuracy, two displacement transducers were arranged at the loading and fixed ends, respectively. Take the average value, and the calculation formula is as follows:

$$S = \frac{1}{2} [(s_1 + s_2) - (s_3 + s_4)]$$

(1)

Where $S$ denotes the slip displacement of the friction interface (i.e., the relative displacement of both ends of the damper); $s_1$ and $s_2$ denote the displacements measured by two
upper displacement transducers; $s_3$ and $s_4$ denote the displacements measured by the two lower displacement transducers, as shown in Fig. 6(a).

As illustrated in Fig. 7, the loading protocol controlled by displacement included four stages, composed of sine waves with varying amplitudes and frequencies. Stage 1 consists of three loading cycles, each of which featured a displacement amplitude of 5mm and a constant frequency of 0.01Hz. The purpose was to check the test setup and data acquisition devices, and the static friction coefficients of friction dampers were obtained. Stage 2, with a constant frequency of 0.05Hz, exhibited two sub-stages with a variable displacement amplitude and a constant displacement amplitude. In the first sub-stage of Stage 2, the displacement amplitudes were 5mm, 10mm, and 15mm in series, and each displacement amplitude was applied for three cycles [31]. The second sub-stage of Stage 2 was composed of ten cycles with a displacement of 10mm to analyze the stability of the damper [30]. The average loading rate of Stage 2 was less than 3mm/s, which belonged to the range of quasi-static loading [13]. Meanwhile, the change of main parameters, such as the normal force of friction interface, were compared between variable amplitude and constant amplitude loading. Stage 3 included variable and constant amplitude loading under a loading frequency of 1Hz, which was similar to the fundamental frequency of high-rise buildings [32], and the counterpart amplitude and cycle times were identical to those of Stage 2. The hysteretic behavior was contrasted with Stage 2 to evaluate the difference in hysteretic performance under high-speed loading. Stage 4 was the repetition of Stage 3, which was used to compare the variation of friction coefficient under repeated high-speed loading. Fig. 7 also shows the loading rates of each stage, which defined the low speed ranged from 0.2mm/s to 3mm/s, and the high speed ranged from 20mm/s to 60mm/s. The time interval between each stage ranged from 2 min to 5min to reduce the influence of temperature on the test result.

Furthermore, to evaluate the reusability of the friction damper, some specimens were re-assembled and re-tested one week after the first test. The detailed discussions of the re-tested specimens were presented in Section 6.

![Fig. 7. Loading protocol](image-url)
3 Test results

3.1 Hysteretic behavior

Based on the original test data, the hysteretic curves of specimens with the friction pads of both NAO and PTFE (labeled NAO specimen and PTFE specimen, respectively) are shown in Fig. 8 and Fig. 9, respectively. The abscissa $S$ of those figures represented the relative displacement of both ends of the specimen, and the $S$ was measured by the displacement transducers; the ordinate $F$ was the axial force of the friction damper provided by the testing machine. The hysteretic curves of the NAO and PTFE specimen were plump in shape, indicating that the friction dampers had excellent energy dissipation performance. As shown in Table 2, the axial force $F$ increased significantly with the loading frequency.

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<td>$F_{\text{axi}} / \text{kN}$</td>
<td>36.5</td>
<td>53.4</td>
<td>50.2</td>
<td>21.4</td>
<td>27.9</td>
<td>30.3</td>
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<tr>
<td>$F_{\text{ists}} / \text{kN}$</td>
<td>42.3</td>
<td>58.0</td>
<td>59.0</td>
<td>28.9</td>
<td>36.3</td>
<td>40.7</td>
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<tr>
<td>$F^\prime / \text{kN}$</td>
<td>51.6</td>
<td>69.4</td>
<td>71.2</td>
<td>37.6</td>
<td>44.7</td>
<td>50.5</td>
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<table>
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<th>P3W21T105</th>
<th>P3W21T135</th>
<th>P3W21T165</th>
<th>P4W21T105</th>
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<tbody>
<tr>
<td>$F_{\text{axi}} / \text{kN}$</td>
<td>27.9</td>
<td>12.0</td>
<td>17.1</td>
<td>18.3</td>
<td>7.0</td>
<td>9.1</td>
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<tr>
<td>$F_{\text{ists}} / \text{kN}$</td>
<td>35.4</td>
<td>13.8</td>
<td>18.8</td>
<td>18.9</td>
<td>9.3</td>
<td>12.5</td>
</tr>
<tr>
<td>$F^\prime / \text{kN}$</td>
<td>44.0</td>
<td>19.9</td>
<td>25.5</td>
<td>25.1</td>
<td>17.0</td>
<td>22.7</td>
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</table>

Note: $F_{\text{axi}}$, $F_{\text{ists}}$, and $F^\prime$ are the average values of the positive axial force at zero displacements when the loading frequency is 0.01Hz, 0.05Hz, and 1Hz, respectively.
Fig. 8 $F$-$S$ hysteretic curves of NAO specimens

Fig. 9 $F$-$S$ hysteretic curves of PTFE specimens
3.2 Test phenomena

After loading, the deterioration of friction pads and inner steel plates were observed, respectively. The main phenomena were summarized as follows:

Both specimens N1W21T135 and P1W21T135 featured two friction units 1-A, and the friction materials of the former and the latter were NAO and PTFE, respectively. As shown in Fig. 10(a), obvious scrapes were observed on the friction pads, and the scrapes were more severe near the bolt holes. The damage schematic diagram of friction unit 1-A was illustrated in Fig. 10(b). The main cause of the severe scrapes was that the red zones of the friction pad were scraped repeatedly by the slot edges in the inner steel plate. Thus, friction unit 2-B featured two rectangular friction pads based on friction unit 1-A, as depicted in Fig. 2.

Fig. 11 shows the deterioration of the friction pads and the inner steel plates of specimens B4W21T105 and N4W21T105 after their respective loading histories. No obvious abrasions were observed on the PTFE friction pads and the inner steel plates, and some floccules adhered to their surfaces. Visible abrasions were found on the surfaces of the NAO friction pads and the corresponding inner steel plates with tiny NAO powders deposited. The deterioration of other specimens was similar after loading.
4 Evaluation on friction behavior of the friction damper

4.1 Slot width of the inner steel plate

Two batches of specimens with different widths $W$ of the inner steel plate were designed as listed in Table 1. The first batch was composed of three specimens with $W = 13\text{mm}$: N2W13T105, N2W13T135, and N2W13T165. The second batch with $W = 21\text{mm}$ included N3W21T105, N3W21T135, and N3W21T165. The clear distances $L_b$ of Type 2 and Type 3 friction pads were 16mm and 26mm, respectively (Fig. 3).

Fig. 12 Comparison of the $F$-$S$ hysteretic curves of specimens with different slot widths
Fig. 12 compares the hysteretic curves of the two groups of specimens in the second cycle of each displacement amplitude. The two batches of specimens showed stable hysteretic performance, while the specimens with $W=13$mm exhibited a larger axial force and a more dissipative hysteretic curve.

Fig. 13 further compares the axial force $F_{0+}$ of the specimens under different loading rates, where the $F_{0+}$ is the positive axial force at zero displacements in the second cycle of each hysteretic curve. The loading rate $v$ showed a correlation between the frequency $f$ and the displacement amplitude $A$. For the loading cycles with $f=0.05$Hz, the loading rates $v$ were 1mm/s, 2mm/s, and 3mm/s, respectively, when $A=5$mm, 10mm and 15mm. Moreover, the loading rate $v$ were 20mm/s, 40mm/s, and 60mm/s for the loading cycles with $f=1$Hz, respectively, when $A=5$mm, 10mm, and 15mm.

The $F_{0+}$ of specimens N2W13T105, N3W13T135, and N3W13T165 exhibited about 50.0%, 61.5%, and 45.7% higher than those of N1W13T105, N1W13T135, and N1W13T165 on average, respectively. Meanwhile, the $F_{0+}$ of specimens with $W=21$mm increased with the increase of designed tightening torque $T$ and loading rate $v$, but the $F_{0+}$ of specimens with $W=13$mm, such as N2W13T165 and N2W13T135, were similar when the loading rate $v$ was the same. For specimen N2W13T135, obvious abrasions were observed on the slot sidewall in
the inner steel plate and the counterpart bolts, as shown in Fig. 14. The above abrasions increased the axial force of specimens with \( W = 13 \text{mm} \), and affected the parameter analysis of the specimen performance. All specimens with \( W = 21 \text{mm} \) did not show a similar abrasion phenomenon as those with \( W = 13 \text{mm} \), so the subsequent parameter analyses were based on the specimens with \( W = 21 \text{mm} \).

4.2 Normal force on the friction interface

The variation of normal forces during the loading process was given in Fig. 15, where the abscissa represents the number of cycles in each stage, and the ordinate represents the normal force \( N \) on the friction interface.

As shown in Fig. 15(a), the normal forces of NAO specimens slowly decreased with the increased numbers of cycles when \( f = 0.05 \text{Hz} \), while the normal forces exhibited a significant degradation with the increased numbers of cycles when \( f = 1 \text{Hz} \). As shown in Fig. 15(b), the normal forces of PTFE specimens showed a negligible variation during the loading protocol. It was considered that the PTFE-Steel friction pair had a small friction coefficient, and the friction force exhibited a slight disturbance to the normal force. Furthermore, the surface of friction pads showed tiny abrasions (Fig. 11(a)), which had little influence on the clamping length of the bolt.

Fig. 15 Normal forces of friction interface under different loading frequencies
According to the loading protocol in Section 2.2, there was a short pause of the loading process after each loading stage and its sub-stages. The conversion of each sub-stages led to a decrease of the normal force from the 9th and 10th cycles of Stage 3 (and Stage 4), as illustrated in Fig. 15. Moreover, a reduction of the normal force was also observed at the end of each loading stage. For example, the normal force of the NAO specimen at the end of Stage 3 (Fig. 15(a)-(Ⅱ)) was greater than the corresponding normal force at the beginning of Stage 4 (Fig. 15(a)-(Ⅲ)). The normal force decrease of the friction interface caused by loading pause was also considered a normal force loss during the loading process.

Fig. 16 shows the cumulative loss rate of normal force after each loading stage, where the ordinate represents the decrease of the normal force divided by the initial normal force. The losses of Stage1 and Stage2 were added together because there were only three cycles in Stage1.

As given in Fig. 16(a), the cumulative loss rate of normal force on the NAO-Steel friction interface increased with the growth of designed tightening torque $T$, and the maximum cumulative loss rate of normal force was 23.6% of specimen N3W21T165. The $T$ of specimen N3W21T105 was the smallest compared with the other two specimens, so there was a tiny difference in the cumulative loss rate of each stage. The $T$ of specimens N3W21T135 and N3W21T165 were relatively large, and the cumulative loss rates under high-speed loading (Stage 3 and Stage 4) were higher than that under low-speed loading (Stage 2).

As given in Fig. 16(b), the three PTFE specimens showed lower cumulative loss rates of normal force than the NAO specimens. The maximum cumulative loss rate was only 4.7% (specimen N4W21T165), and the average cumulative loss rate was 3.2%. For PTFE friction dampers, the cumulative loss rate of normal force was relatively low, and the accidental factors in the assembly and loading process might significantly interfere with the lower loss rate, such as the flatness of the friction interface. Therefore, the cumulative loss rate of PTFE friction dampers did not show a similar variation to that of NAO friction dampers. In summary, the normal force and the loading rate had a negligible effect on the loss of normal force on the
PTFE-Steel friction interface.

4.3 Initial nominal pressure on the friction interface

The Type 4 friction pad was developed based on the Type 3 friction pad, and the length of the former was 150mm, while that of the latter was 184mm, as shown in Fig. 3. The initial nominal pressure $p$ in Table 1 was controlled by the area of the friction interface and the nominal normal forces $N_0$. To study the influence of initial nominal pressure $p$ on the friction performance of the damper, the counterpart specimens were divided into the following three groups:

**Group 1:** The specimens with the same nominal normal force $N_0$ and different areas of friction pad, including specimens N3W21T105 and P3W21T105 with Type 3 friction pads ($p=8.37$MPa, friction pad with larger area) and N4W21T105 and P4W21T105 with Type 4 friction pads ($p=10.28$MPa, friction pad with smaller area), respectively.

**Group 2:** The specimens with similar initial nominal pressures $p$ and different areas of friction pad, including specimens N3W21T135 and P3W21T135 with Type 3 friction pads ($p=10.51$MPa, friction pad with larger area) and N4W21T105 and P4W21T105 with Type 4 friction pads ($p=10.28$MPa, friction pad with smaller area), respectively.

**Group 3:** The specimens with the same area of friction pad and different nominal normal forces $N_0$, including specimens N3W21T105 and P3W21T105 ($p=8.37$MPa), N3W21T135 and P3W21T135 ($p=10.51$MPa), and N3W21T165 and P3W21T165 ($p=12.65$MPa), respectively.

During the loading history, the data acquisition of the axial force $F$ and the normal force $N$ were carried out simultaneously, and the friction coefficient $\mu$ of the friction damper was calculated as follows:

$$\mu = \frac{F}{2N}$$

Fig. 17 shows the variation of the friction coefficient $\mu$ with the initial nominal pressure $p$. To avoid interference from the peak point of the friction coefficient in the first loading circle, the positive friction coefficients $\mu_{0+}$ at zero displacements in the second cycle were selected for subsequent research.

The test results of NAO and PTFE specimens are shown in Fig. 17(a) and Fig. 17(b), respectively. The conclusions were as follows:

For Group 1, the $\mu_{0+}$ of the specimen with Type 4 friction pads was significantly lower than that of the counterpart specimens with Type 3 friction pads under the same frequency $f$ and amplitude $A$. The average decrease of $\mu_{0+}$ was about 15.8% (NAO friction pads) and 19.8% (PTFE friction pads), which showed that the $\mu_{0+}$ of the NAO and PTFE specimens decreased
with the decline of friction area when the nominal normal forces \( N_0 \) was the same.

For Group 2, the specimen with Type 4 friction pads showed a tinier \( \mu_0^+ \) than that of the specimen with Type 3 friction pads under the same \( f \) and \( A \). The \( \mu_0^+ \) of NAO and PTFE specimens reduced by 23.3% and 25.7%, respectively. The \( \mu_0^+ \) of NAO and PTFE specimens also decreased with the reduction of friction area when the \( p \) was similar.

For Group 3, the \( \mu_0^+ \) of all specimens were similar under the same \( f \) and \( A \), which showed that the friction coefficient was less affected by the initial nominal pressure \( p \) when the areas of friction pads were the same.

![Fig. 17 Relationship between initial nominal pressure \( p \) and friction coefficient \( \mu_0^+ \)](image)

In summary, the reduction of the area of friction interface decreased the friction coefficient \( \mu \) of NAO and PTFE specimens. References [33, 34] also show that the classical Coulomb friction law applied to materials (such as metals) with yield strength, but other non-metallic materials do not necessarily follow this law. The variation of the \( \mu_0^+ \) in Fig. 17 (a) and Fig. 17 (b) showed that the friction behavior of the NAO-Steel friction pair and the PTFE-Steel friction pair did not follow the classical Coulomb friction law. Moreover, the loading rate-dependency of the friction coefficient was observed in Fig. 17 and discussed in Section 5.

5 Main parameters of friction damper

Fig. 18 (a) shows the axial force-displacement (\( F-S \)) hysteretic model of the friction damper[27]. \( F_a \) and \( S_a \) are activation force and activation displacement when the damper starts, respectively; \( S_d \) denotes the design displacement of the friction damper; \( F_0 \) is the axial force (sliding friction force) at the zero displacement, \( F_0^+ \) and \( F_0^- \) represent positive and negative axial forces, respectively; \( K_1 \) and \( K_2 \) are the initial stiffness and the second stiffness, which are approximately \(+\infty\) and 0, respectively. There will be a pause in the motion of the damper during the loading direction change, and the damper begins to move in reverse when the axial force is large enough. The above axial force is defined as the restart force \( F_t \), and \( F_{t2} \) and \( F_{t4} \) are the restart forces in the second and fourth quadrants of the hysteretic curve, respectively.
Furthermore, the $F$-$S$ hysteretic model can also be expressed as a friction coefficient-displacement ($\mu$-$S$) hysteretic model based on Equation (1). Where $\mu_a$, $\mu_0$, and $\mu_t$ are static friction coefficient, dynamic friction coefficient, and restart friction coefficient, respectively, obtained from $F_a$, $F_0$, and $F_t$ through Equation (1). Specifically, $\mu_{0+}$, $\mu_{0-}$, $\mu_{t2}$, and $\mu_{t4}$ are positive dynamic friction coefficient, negative dynamic friction coefficient, restart friction coefficient in the second quadrant and restart friction coefficient in the fourth quadrant, corresponding to $F_{0+}$, $F_{0-}$, $F_{t2}$, and $F_{t4}$, respectively.

### 5.1 Static friction coefficient

Two groups of specimens were analyzed to determine the numerical relationship between the static friction coefficient $\mu_a$ and the dynamic friction coefficient $\mu_0$ under different loading rates (Fig. 7): N3W21T105, N3W21T135, and N3W21T165 with NAO friction pads; P3W21T105, P3W21T135, and P3W21T165 with PTFE friction pads.

Fig. 19 shows their $\mu$-$S$ hysteretic curves of the above specimens in the first cycle of Stage 1. The friction coefficient of the NAO and PTFE specimens increased sharply near the zero displacement, i.e., the static friction coefficient $\mu_a$ was significantly greater than the corresponding dynamic friction coefficient $\mu_{0+}$. Table 3 summarizes the static friction coefficient $\mu_a$ and dynamic friction coefficient $\mu_{0+}$ and $\mu_0$ in the first cycle of Stage 1. It can be
seen that the $\mu_a$ of the damper with the same friction material was relatively close, and the average value $\mu_a$ of the $\mu_a$ could be taken as the design value of the static friction coefficient. The similarity was also represented between the positive dynamic friction coefficient $\mu_{0+}$ and negative dynamic friction coefficient $\mu_{0-}$, thus the dynamic friction coefficients had excellent symmetry along the x-axis of the hysteretic curve.

Table 3 Friction coefficients in the first cycle of Stage 1

<table>
<thead>
<tr>
<th>No.</th>
<th>Specimen</th>
<th>material</th>
<th>$\mu_a$</th>
<th>$\mu_a$</th>
<th>$\mu_{0+}$</th>
<th>$\mu_{0-}$</th>
<th>$\mu_2$</th>
<th>$\mu_4$</th>
</tr>
</thead>
<tbody>
<tr>
<td>S4</td>
<td>N3W21T105</td>
<td></td>
<td>0.177</td>
<td>0.124</td>
<td>0.123</td>
<td>0.136</td>
<td>0.133</td>
<td></td>
</tr>
<tr>
<td>S5</td>
<td>N3W21T135</td>
<td>NAO</td>
<td>0.180</td>
<td>0.178</td>
<td>0.133</td>
<td>0.133</td>
<td>0.135</td>
<td>0.135</td>
</tr>
<tr>
<td>S6</td>
<td>N3W21T165</td>
<td></td>
<td>0.176</td>
<td>0.126</td>
<td>0.127</td>
<td>0.122</td>
<td>0.123</td>
<td></td>
</tr>
<tr>
<td>S9</td>
<td>P3W21T105</td>
<td></td>
<td>0.092</td>
<td>0.069</td>
<td>0.072</td>
<td>0.081</td>
<td>0.086</td>
<td></td>
</tr>
<tr>
<td>S10</td>
<td>P3W21T135</td>
<td>PTFE</td>
<td>0.106</td>
<td>0.101</td>
<td>0.079</td>
<td>0.081</td>
<td>0.098</td>
<td>0.097</td>
</tr>
<tr>
<td>S11</td>
<td>P3W21T165</td>
<td></td>
<td>0.104</td>
<td>0.080</td>
<td>0.081</td>
<td>0.092</td>
<td>0.093</td>
<td></td>
</tr>
</tbody>
</table>

5.2 Restart friction coefficient

According to Fig. 8 and Fig. 9, there was a phenomenon of “sharp corner” in the second and fourth quadrants of the hysteretic curve, and the friction damper exhibited the state change of motion-static-reverse motion, where the restart force $F_t$ needed to be overcome from the static state to the reverse motion state [27]. References [35, 36] showed that the “sharp corner” of hysteretic curves was mainly caused by the momentary sticking of interfaces and the acceleration impulses, and different loading rates had a great influence on the size of the “sharp corner”. This paper further analyzes as follows. Table 3 shows the restart friction coefficients $\mu_2$ and $\mu_4$ in the first cycle of Stage 1, which correspond to the restart forces $F_2$ and $F_4$ in the second and fourth quadrants, respectively. Referring to Table 3, the restart friction coefficient $\mu_2$ and $\mu_4$ were similar and less than the static friction coefficient $\mu_a$ in the first loading cycle when the specimen was activated.

To compare $\mu_t$ and $\mu_0$ in each loading stage, the change rate $\rho$ of $\mu_t$ relative to $\mu_0$ in the same hysteretic loop was defined as follows:

$$\rho = \left( \frac{\mu_t - \mu_0}{\mu_0} \right) \times 100\%$$

(3)

Where $\mu_0$ includes positive and negative dynamic friction coefficients $\mu_{0+}$ and $\mu_{0-}$, $\mu_t$ includes restart friction coefficients $\mu_2$ and $\mu_4$ in the second and fourth quadrant, respectively.

Fig. 20 shows the variation of the $\rho$ during loading, where the white area indicates variable amplitude loading; the gray area indicates constant amplitude loading. In Fig. 20(a), the maximum $\rho$ of NAO specimens in Stage 1 ($f=0.01\text{Hz}$) was 9.6% and decreased sharply; the loading rate of Stage 2 ($f=0.05\text{Hz}$) was in the low-speed range, and the $\rho$ was mostly negative,
i.e., $\mu_0$ was greater than $\mu_t$; the $\rho$ in Stage 3 and Stage 4 ($f=1\text{Hz}$, high-speed loading) was close to 0. In Fig. 20(b), the $\rho$ of PTFE specimens showed a significant decrease in Stage 1, and the maximum value of the $\rho$ was 23.9%; the $\rho$ in Stage 2 was about 10~15%; the $\rho$ in Stage 3 and Stage 4 was about 0~10%, and obvious degradation was observed during constant amplitude loading.

Through further comparison between Fig. 20(a) and Fig. 20(b), the change rate $\rho$ of NAO specimens was much smaller than that of PTFE specimens, and the $\rho$ of both NAO and PTFE specimens decreased to less than 10% with the increase of loading rate. In summary, the restart friction coefficient of the NAO and PTFE friction damper has little effect on the hysteretic
performance, which can be ignored in the design process.

Table 4 summarizes the change rate of the restart friction coefficient of friction dampers in references [13, 25, 37-41]. In references [25, 40], dynamic tests of the NAO-Steel and PTFE-Steel friction pairs were carried out, respectively, and the maximum change rates $\rho_{\text{max}}$ of the restart friction coefficient were similar to the corresponding results in this paper. Overall, the change rate of the restart friction coefficient was also affected by the configuration of dampers, different friction materials, loading protocol, loading rate, etc.

Table 4 Statistics of restart friction coefficient

<table>
<thead>
<tr>
<th>Friction pair</th>
<th>Loading protocol</th>
<th>Maximum loading rate (mm/s)</th>
<th>$\rho_{\text{max}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NAO-Steel [25]</td>
<td>dynamic loading</td>
<td>80</td>
<td>10.2</td>
</tr>
<tr>
<td>PTFE-Steel [40]</td>
<td>dynamic loading</td>
<td>10</td>
<td>14.3</td>
</tr>
<tr>
<td>Brake pad-Steel [13]</td>
<td>dynamic loading</td>
<td>3.3</td>
<td>0</td>
</tr>
<tr>
<td>Brass-Steel [41]*</td>
<td>dynamic loading</td>
<td>24</td>
<td>59.2</td>
</tr>
<tr>
<td>Steel-Steel [37]</td>
<td>quasi static</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>Composite material -Steel [38]</td>
<td>dynamic loading</td>
<td>60</td>
<td>0</td>
</tr>
<tr>
<td>Composite material -Steel [39]*</td>
<td>dynamic loading</td>
<td>120</td>
<td>30.5</td>
</tr>
</tbody>
</table>

Note: * denotes as rotational motion devices, and the rest as translational motion devices; $\rho_{\text{max}}$ denotes the maximum of the change rate $\rho$ in Formula (2).

5.3 Loading rate-dependency of the dynamic friction coefficient

Fig. 21 shows the second-cycle hysteretic curves of the two specimen groups under different loading amplitude in Stage 1 and Stage 2. As shown in Fig. 7, the loading rates of the above stages were in the low-speed range, and the values of loading rates are shown in Fig. 21(a)-(I). As depicted in Fig. 20(a), the dynamic friction coefficient of NAO specimens increased with the increase of loading rate. As depicted in Fig. 21(b), the loading rate showed a negligible effect on the dynamic friction coefficients of PTFE specimens under low-speed loading.
Fig. 21 $\mu$-$S$ hysteretic curves of friction dampers under low-speed loading

Fig. 22 $\mu$-$S$ hysteretic curves of friction dampers under high-speed loading

Fig. 22 shows the second-cycle hysteretic curves of the two specimen groups under different loading amplitude in Stage 3. As illustrated in Fig. 22(a), the dynamic friction coefficient of NAO friction damper varied little under high-speed loading. As illustrated in Fig. 22(b), the dynamic friction coefficient of PTFE friction dampers increased slightly with the increase of loading amplitude.

The positive dynamic friction coefficient $\mu_{0+}$ under low-speed and high-speed loading were collected from the hysteretic curves in Fig. 21 and Fig. 22, respectively. Furthermore, the above $\mu_{0+}$ were exhibited in Fig. 23. The variation of friction coefficient was as follows: when the loading rate $\nu$ was 0.2~3mm/s, the $\mu_{0+}$ of NAO specimens increased remarkably with the $\nu$, and the $\mu_{0+}$ of PTFE specimens were relatively concentrated; when the $\nu$ was 20~60mm/s, the $\mu_{0+}$ of both NAO and PTFE specimens changed slightly. Noteworthy stating that the $\mu_{0+}$ of NAO and PTFE specimens under high-speed loading was significantly greater than that under low-speed loading.
Fig 23 (a) also showed some data about friction coefficient in reference [15], and the corresponding loading rate was 3mm/s~27mm/s. The friction coefficients of 20 mm/s~60mm/s in this paper were close to those in reference [15], indicating that the friction coefficient of the NAO friction damper could be regarded as a constant under high-speed loading. According to the test data in this paper, the average value of friction coefficient was 0.229 when the loading rate was 20 mm/s~60mm/s. As shown in Fig 23 (b), the friction coefficient of the PTFE friction damper could also be simplified as a constant within the rate range of 20 mm/s~60mm/s, with an average value of 0.121.

In conclusion, for both NAO and PTFE friction dampers, the friction coefficient under high-speed loading ($v=20 \text{ mm/s}~60\text{mm/s}$) is significantly higher than that under low-speed loading ($v=0.2 \text{ mm/s}~3\text{mm/s}$). The friction coefficient under high-speed loading is suggested as the main parameter in the analysis of the structural friction damper, which can improve the accuracy of the structural seismic analysis.

6 Reusability evaluation of the friction damper

The reusability of friction dampers without significant damage was necessary to be investigated after an earthquake. One week after the test, the four specimens, N3W21T105, P3W21T105, N3W21T135, and P3W21T135, were re-assembled and re-tested (the 2nd loading) according to the loading protocol in Fig. 7. The friction pads was not replaced during the above process.

Fig. 24 shows the variation of the $\mu_0$ in each loading stage during the 1st and 2nd loading, where the gray and white area indicates low-speed and high-speed loading, respectively; the meaning of $\mu_0$ is illustrated in Fig.18 (b). The dynamic friction coefficient of the NAO specimens was increased during the 2nd loading, and the $\mu_0$ of specimens N3W21T105 (Fig.24 (a)) and N3W21T135 (Fig.24 (b)) increased by about 18.1% and 11.7% on average, respectively. However, the $\mu_0$ of the specimen with PTFE friction pads changed negligibly in the two loading
processes.

(a) \( T = 105 \text{ N·m} \)

(b) \( T = 135 \text{ N·m} \)

Fig. 24 Dynamic friction coefficient of the 1\textsuperscript{st} loading and the 2\textsuperscript{nd} loading

Fig. 25 Static friction coefficient of the 1\textsuperscript{st} loading and the 2\textsuperscript{nd} loading
Fig. 25 compares the static friction coefficients $\mu_a$ (as illustrated in Fig. 18 (b)) of the above four specimens in the two loading processes. During the 2nd loading, the $\mu_a$ varied within the range of ±10%, which proved that the re-assembled and re-tested had little effect on the $\mu_a$.

In summary, the reassembled friction damper after the 1st loading still has a similar performance to the unused friction damper, i.e., reloading has little effect on the static friction coefficient and dynamic friction coefficient of the friction damper.

7 Conclusions

Cyclic loading tests of NAO and PTFE friction dampers were carried out with loading rates ranging from 0.2mm/s to 60mm/s. Four friction units were compared to obtain optimized configurations of the friction damper. The influences of the loading rate were summarized for the main parameters of the hysteretic model. Some specimens were re-tested to evaluate their reusability. The main conclusions are as follows:

(1) NAO and PTFE friction dampers exhibited stable and plump hysteretic curves. In order to keep the stable mechanical performance of friction dampers, two configuration problems should be avoided: the friction pad damage caused by local scrapes of the slot edges in the inner steel plate, and the contact abrasion between the slot sidewall and high-strength bolts.

(2) The normal force on the friction interface degraded gradually during loading. The cumulative loss rate of normal force on NAO-Steel friction interfaces increased with the designed tightening torques. However, the PTFE-Steel friction interface showed a significantly small cumulative loss rate of normal force than the NAO-Steel friction interface.

(3) The dynamic friction coefficient of NAO and PTFE friction dampers under high-speed loading (20mm/s~60mm/s) converged to constant values, which were significantly greater than that under low-speed loading (0.2mm/s~3mm/s). Based on the test results, the dynamic friction coefficients for structural analysis under earthquake are suggested to be 0.229 and 0.121, respectively.

(4) In the conversion of the motion direction, NAO and PTFE friction dampers showed the transition from a static state to a dynamic state. The phenomenon of “sharp corner” appeared in the second and fourth quadrants of the hysteretic curve. The restart friction coefficients of NAO and PTFE friction dampers were sometimes slightly greater than the dynamic friction coefficient, and the change rate was generally less than 10% with the increase of loading rate.

(5) Re-assembled and re-tested processes exhibited a negligible effect on the static and dynamic friction coefficients of the PTFE friction damper. The static friction coefficient of the NAO friction damper also changed little, but the dynamic friction coefficient increased marginally.
Therefore, the re-assembled NAO and PTFE friction dampers showed excellent reusability without replacing the friction pads.

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