Design of a Self-Repairing Breakaway Mechanism for High Speed Industrial Doors

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ABSTRACT

The design of a “self-repairing” breakaway mechanism for Horman industrial door models 1600 and 2600 is presented. The mechanism replaces a breakaway tab design currently in use, which requires complete replacement when used. Additionally, the new design reduces overall repair time to less than 30 seconds. Minimum system requirements, method of design and final prototype test results are considered.

1. INTRODUCTION

i. BACKGROUND

Hormann-Flexon manufactures high-speed garage doors for industrial settings, such as shown in Figure 1. The doors consist of sections of PVC fabric joined with aluminum ribs, which fit inside supporting tracks on either side. Common outdoor installations, such as that displayed in Figure 1, require that the doors withstand wind loads up to 65 mph. To accommodate frequent forklift traffic, the doors are designed to retract at a rate of 5 feet per second.

Figure 1: Typical installation of industrial doors. Outdoor installations require the doors to withstand up to 65 mph winds.

Forklift collisions with the doors often resulted in damage to the bottom panel, so a device to allow this panel to “break away” under these conditions was designed. The breakaway mechanism consisted of plastic tabs inserted into either end of the bottom-most rib of the door, as illustrated in Figure 2.

Figure 2: Tab breakaway mechanism. The tabs were designed to break when the door was impacted. However, the high frequency of collisions made this design inconvenient and inefficient.

Under sudden impact, these tabs would break inside the tracks and allow the bottom panel to swing free, thereby minimizing collision damage to the door. Replacement of the tabs required that the broken pieces inside the rib be removed before new tabs could be installed. This process proved to be inconvenient due to the overall amount of time required, in conjunction with the high frequency of incidents. Furthermore, extra replacement tabs had to be purchased, which contributed to higher maintenance costs for the doors.

A new breakaway design was desired that would both streamline the repair process (i.e., repair time must be reduced to less than 30 seconds), and minimize overall maintenance costs (be “self-repairing”). Additional requirements for the design stated that the mechanism must be
entirely mechanical (no electrical or mechatronic components), must be able to be repaired by an untrained operator, and must remain “unbroken” under forces resulting from the expected wind loads on the door.

2. Objective

The objective of this project was to design a new breakaway mechanism for Hormann-Flexon door models 1600 and 2600 as per the functional requirements set forth by Hormann, and to develop and test a proof of concept prototype. Design documentation, including 3-D CAD models, drawings, supporting calculations, and test results, were to be delivered to Hormann upon completion of the project.

3. Design Process

i. Force Characterization

The fundamental design requirement was that the device must remain “unbroken” under the forces arising from 65 mph wind loads on the door. Pertinent design choices including device material and initial device concepts were based largely on this constraint.

The forces which the device would be required to withstand were calculated by first determining the pressure corresponding to the 65 mph wind speed. A factor of safety was incorporated into the design by using a wind speed of 70 mph in the calculations. Bernoulli’s equation was used to determine the wind speed equivalent pressure:

\[ P_1 + \frac{1}{2} \rho V^2 = P_0 \]  

\( P_1 \) is the ambient gauge pressure (equal to zero), \( \rho \) is the air density, and \( V \) is the wind speed. The resulting pressure \( (P_0) \) was then treated as a distributed load on the bottom panel of the door. An equivalent force \( (F) \) was then determined from the product of the wind pressure \( (P_0) \) and the area of the door \( (A) \):

\[ F = P_0 A \]  

Since the panel was assumed to be restrained equally at each of the four corners, the reaction force \( (R) \) at each point was assumed to be one-fourth of the total load:

\[ R = \frac{1}{4} F \]  

For a 70 mph wind, reaction forces of 280 lbf were calculated. To include a factor of safety, this figure was increased by 10%, giving a design force of 308 lbf. Based on this result, initial design concepts were generated.

ii. Design Selection

Three preliminary designs were considered before a final design was selected and refined. The group consulted with Hormann engineers to evaluate each proposed design and eliminate unfeasible concepts.

The first design proposed consisted of a magnet inserted into the rib, and a metal I-shaped piece, which would fit inside the track opening (Figure 3). The magnet would lock onto the I-shape and hold the panel in place in the track.

![Magnet design. A magnet would be inserted into the base rib and would lock onto the I-piece in the track. The magnets required to satisfy the loading constraints would be unreasonably large and expensive for this to be considered a viable solution.](image)

After presenting this design to Hormann, several issues were exposed. The size of the magnet required to withstand the calculated forces would be far too large to fit in the limited amount of space, and could also inhibit operation of the door. Additionally, the cost of the magnets was unrealistically high.

The second design presented consisted of a metal rod with a spherical end, which would be
attached to the rib, and a rubber gasket, which would sit inside the track (Figure 4). The spherical end of the rod would be inserted into the gasket, where friction between the components would prevent the separation of the pieces.

The small size of the design was ideal for the application, but the force required to set the device would be equal to that required to separate it. This was physically unacceptable, as the device would have to withstand over 300 lbf of separation force.

The third design considered consisted of a plastic bulb-shaped piece with a T-shaped end, which would slide inside the track, and a plastic clip attached to the rib (Figure 5). The device would function as a buckle, relying on friction and the elastic properties of the plastic clip to prevent the device’s separation.

The device did not permit the rotation of the rib about the track, which would occur in the event of a forklift collision, so a hinge had to be incorporated. Although the device as designed would not be functional, the underlying concept was judged to be feasible. In conjunction with the Hormann engineers, the group chose to focus on this design and incorporate the necessary modifications to provide a practical mechanism.

iii. DESIGN DEVELOPMENT

Due to the large forces anticipated in the mechanism, as well as the environment in which the device would be placed, the material chosen for the design was changed from a polycarbonate plastic to a stainless steel. The required hinge was integrated into the device by changing the plastic bulb-shaped piece to a steel sphere attached to the end of a rod. The rod was placed in a plastic T-shaped piece, which would sit inside the track (Figure 6). The clip was likewise modified to form a complementary clamp around the sphere. Additionally, the clip was designed to be constructed from two separate pieces, which would be bolted together at the pivot point. The required clamping force of the device was provided via a combination of extension springs placed on the clamping end of the device.

Figure 4: “Lollipop” design. The metal sphere on the end of the rod is inserted in the rubber gasket. Friction between the components holds the pieces together. However, the force required to separate the pieces is the same as that to set the device.

Figure 5: Clip design. The plastic clip clamps over the bulb-shaped piece, which is held in the track via a T-shaped end. The device relies on friction between the components and the shape of the bulb to remain together. The design does not allow for rotational movement of the rib, so a hinge must be incorporated.

Figure 6: Final design. The clip design was modified to incorporate a hinge and a robust clamping mechanism. A rod with a steel sphere at one end is held in the track by means of a plastic T. The clamping force in the clip is generated by use of extension springs attached to the clip.

iv. CLAMP SPRING DESIGN

The required clamping force of the mechanism was determined by first simplifying the mechanism to a two-dimensional static system, and then calculating the force necessary to prevent the clamp from opening under a 300 lbf load. Figure 7 shows a schematic of the clamp and the resulting free-body diagram.
Figure 7: Schematic and FBD of the clamp. The spring force $f$ had to be large enough to balance the moment caused by $F_A$ about pivot $P$.

The force of the spring had to be large enough to counter the moment of the separating force developed at point $A$. Treating the clamp as a rigid beam with a pivot about point $P$, the spring force $f$ in terms of force $F_A$ may be determined:

$$
\Sigma M_P = 0 \rightarrow F_A L - f l_1 = 0
$$

$$
f = F_A \frac{L}{l_1}
$$

(4)

Figure 8: Free-body diagram of the top half of the steel sphere. Due to symmetry, only one half of the forces on the device were considered in the calculations.

Figure 8 shows the free-body diagram of the sphere at point $A$. The forces $F_{x'}$, $F_{y'}$, and $F_f$ are calculated as:

$$F_{x'} = F \cos(\theta) + F_A \sin(\theta)$$

$$F_{y'} = -F \sin(\theta) + F_A \cos(\theta)$$

$$F_f = \mu_s F_{x'}$$

$$F_f = \mu_s (F \cos(\theta) + F_A \sin(\theta))$$

(5)

When the clamp begins to open, the sum of the forces in the $y'$ direction will equal zero:

$$\Sigma F_{y'} = 0 \rightarrow F_{y'} + F_f = 0$$

(6)

Substituting equations (5) into (6) and solving for the spring force $f$ yields:

$$f = \frac{L}{l_1} F \left( \frac{\sin(\theta) - \mu_s \cos(\theta)}{\cos(\theta) + \mu_s \sin(\theta)} \right)$$

(7)

Assuming an initial extension $d$ of the spring, the required spring rate may be calculated as:

$$k = \frac{f}{d}$$

(8)

The values listed in Table 1 were used to determine the necessary component values.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L$ (in)</td>
<td>2</td>
</tr>
<tr>
<td>$l_1$ (in)</td>
<td>1</td>
</tr>
<tr>
<td>$F$ (lbf)</td>
<td>308</td>
</tr>
<tr>
<td>$\theta$ (degrees)</td>
<td>30</td>
</tr>
<tr>
<td>$\mu_s$</td>
<td>0.3</td>
</tr>
<tr>
<td>$d$ (in)</td>
<td>0.5</td>
</tr>
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</table>

Using the values listed in Table 1, equation (7) yielded a necessary spring rate of 73 lbf/in. Since the load was to be distributed between two presumably equivalent springs, the required rate per spring was approximately 36 lbf/in.

After determining the appropriate spring sizes, the first prototype (Figure 9) was fabricated from 3/32” stainless steel sheet, and initial testing was conducted.

Figure 9: First prototype. Once the required spring forces were known, the appropriate springs were obtained and initial testing commenced. Due to substantial amounts of uncertainty in the calculations, a range of different springs was obtained.

4. PROOF OF CONCEPT

The initial prototype was tested using an Instron 5500 tensile machine, which pulled the device
apart with a linearly increasing force. The clamping force of the prototype was adjusted by changing the initial extension of the springs. The setup for the prototype testing is shown in Figure 10.

The clamping force of the prototype was adjusted by changing the initial extension of the springs. The setup for the prototype testing is shown in Figure 10.

As the spring tension was increased, the clamp began to bend about a thin section created from deep relief cuts. Additionally, the design was not able to accommodate both door models as required, so a second prototype was made to cope with these issues.

The second prototype (Figure 11) was fabricated from 1/16” stainless steel sheet and was made slightly wider to fit on both door models.

The second prototype (Figure 11) was fabricated from 1/16” stainless steel sheet and was made slightly wider to fit on both door models.

Thinner steel was chosen for this prototype for two primary reasons. First, the device had to be bent into shape manually, with the aid of vices, pliers, hammers, and the like. This was difficult with the 3/32” material, so the thinner metal was chosen to ease fabrication. Second, it was desired to know whether thinner steel would suffice for the application. Thus, by fabricating the second prototype out of the 1/16” steel, two questions were answered simultaneously.

The same tests as performed on the first prototype were conducted on the second to verify functionality of the new design. Similar to the tests on the first, the second prototype began to bend at the thin sections as spring tension was increased.

5. Results

The results from the first prototype tests are listed in Table 2. The data collected indicate that in general, maximum separating force increased with increasing spring rate. However, the results presented do not accurately reflect the true performance of the device. Inspection of the steel sphere after the first tests revealed sharp gashes on the surface near the contact points of the clamp. From this, in addition to the unrealistic results obtained, it was surmised that the clamp was catching on the gashes during the test, thereby giving rise to the erroneously large separating forces.

Results from second prototype tests indicated separating forces much lower than those obtained previously. This was due primarily to the precautions taken when preparing the assembly. The sphere was turned in such a way that the gashes would not catch on the clamp. Also, the inside of the clip surface was made smooth to ensure no extraneous grip. The results from the second prototype tests are listed in Table 3.

<table>
<thead>
<tr>
<th>Spring No.</th>
<th>Tension Position</th>
<th>Rate (lbf/in)</th>
<th>Force (lbf)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>15</td>
<td>111</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>15</td>
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<tr>
<td>2</td>
<td>2</td>
<td>17</td>
<td>365</td>
</tr>
</tbody>
</table>
The results from the second prototype tests reveal that the device is very sensitive to defects between contacting surfaces. Additionally, the thinner steel used was not adequate for the application, as the clamp easily bent under an applied force.

### 6. CONCLUSIONS AND RECOMMENDATIONS

This report has shown the development and proof of concept for a new, self-repairing industrial door breakaway mechanism. Due to temporal and manufacturing limitations, a sufficient number of prototypes could not be fabricated and tested. The results presented in this document indicate that the basic concept is viable, but that further work must be done to improve device performance. In particular, it is recommended that the device be made of steel no less than 3/32”; that relief cuts be made with consideration to overall part thickness; that the sphere component be made from a hardened steel to prevent hazardous wear; and that a cover for the device be included to protect the device from environmental conditions.

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