# Wear Analysis of Machinery Components in Buildings

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# Abstract

Buildings incorporate a variety of mechanical systems for the convenience of occupants. These systems may include machinery for heavy lifting. Excessive wear in the machinery components can result in unanticipated maintenance and loss-of-service costs. The Wear Analysis utility in LS-PrePost<sup>®</sup> was used to quantify component wear over the service life for a vertical lift system. Candidate materials for a roller and jamb geometry were modeled. The wear predictions were then used to evaluate the candidate material combinations.

The methodology provided by Jernberg and Borvall was followed for the analysis. Due to the relative low loading rate in the problem (i.e. much less than the loading rate in automotive impact), the contact force history between the roller and plate was calculated using implicit dynamics. This force history was the input into the Wear Analysis utility using the dynain file from *\*INTERFACE\_SPRINGBACK\_LSDYNA*, and Archard's wear law was used to calculate wear depth. The wear law returns wear volume as a function of surface hardness, normal force (as calculated in the implicit analysis), sliding distance and wear coefficient.

The above methodology was applied to three combinations of materials for the roller and jamb plate selected from hardened steel, stainless steel and aluminum. \*CONTACT\_AUTOMATIC\_SURFACE\_TO\_SURFACE\_MORTAR was necessary for the implicit calculation to converge. Static and dynamic coefficients of friction (COFs) were obtained from the CRC Handbook; the decay constant from static to dynamic COF was from Stembalski.

In this paper, the remaining details of the analysis are discussed, including the constitutive models used for the metals. The wear predictions are provided. Finally, recommendations are made for increasing confidence in the calculations and convenience of the calculation method.

### Introduction

Buildings include a variety of mechanical systems for the convenience of occupants. These systems may involve machinery for heavy lifting. Excessive wear in the machinery components can result in unanticipated maintenance and loss-of-service costs.

To control these costs, the Wear Analysis utility in LS-PrePost was used to quantify component wear over the service life for a vertical lift system. The vertical lift system consists of a heavy panel traveling between vertical jambs while being guided by rollers. Candidate materials for the roller and jamb included in the system were modeled. The wear predictions were then used to evaluate the candidate material combinations.

In this paper, the methodology used for the analysis is described. The geometry, constraints and element formulation are detailed. The analysis cases and associated constitutive and contact inputs are then discussed. Finally, the calculated results are presented, and conclusions and recommendations are drawn.

## Methodology

The methodology provided by Jernberg and Borvall was followed for the analysis [1]. Wear calculation is implemented in LS-DYNA<sup>®</sup> in the following form:

$$d = \frac{K(p, v)}{H_s(T_s)} pL$$

Where: d = nodal wear distance in direction of the surface normal

K(p,v) = dimensionless scale factor (can be function of pressure p and sliding velocity v)

 $H(T_s)$  = hardness for the slave side (can be function of temperature)

p = nodal pressure

L = sliding distance

The wear law used can be user-defined or Archard's wear law. For these calculations, Archard's wear law as used, which calculates metal wear for a given loading [2]:

$$Q = \frac{K}{H}WL$$

Where: Q = volume of wear debris

K = dimensionless constant H = hardness of softest contacting surfaces W = normal load L = sliding distance

The wear law returns wear volume as a function of surface hardness, normal force (as calculated in the implicit analysis), sliding distance and wear coefficient.

Due to the relative low loading rate in the problem (i.e. much less than the loading rate in automotive impact), the contact force history between the roller and plate was calculated using implicit dynamics. This force history input into the Wear Analysis utility using the dvnain file from was the \*INTERFACE SPRINGBACK LSDYNA, and Archard's wear law was used to calculate wear depth.

The calculation was run using implicit dynamics with these keywords invoked:

- 1. \*CONTROL\_IMPLICIT\_DYNAMICS
- 2. \*CONTROL\_IMPLICIT\_GENERAL
- 3. \*CONTROL\_IMPLICIT\_SOLUTION
- 4. \*CONTROL\_IMPLICIT\_AUTO
- 5. \*CONTROL\_IMPLICIT\_SOLVER

The controls were set to minimize run time for the implicit calculation. Changes from default controls included introducing numerical damping by setting gamma=0.55 and beta= 0.276 in \*CONTROL\_IMPLICIT\_DYNAMICS.

### **Geometry and Element Formulation**

The model included a 3-inch diameter roller, a 1-inch diameter pin inserted into the roller and 1-inch thick jamb plate, as shown in Figure 1 and Figure 2. The rigid wall shown in Figure 1 was included to represent a wall supporting the jamb plate. The boundary conditions applied to the model are shown in Figure 3.



Figure 1.Wear Model Geometry



Figure 2. Wear Model Geometry (Detail)



Figure 3. Model Constraints

The assumed travel velocity was applied to the end nodes of the pin using \*BOUNDARY\_PRESCRIBED\_MOTION\_SET. The set of end nodes is shown in Figure 4. Similarly, the force applied by the panel to the roller was applied to nodes at the end of the pin using \*LOAD\_NODE\_SET. The force on the nodes (2300 N) was obtained from a separate analysis.



Figure 4. Pin End Nodes with Prescribed Velocity

The characteristic mesh size was 1 mm. All elements in the model were elform = -2 in LS-DYNA, 8-point hexahedron intended for elements with poor aspect ratios, accurate formulation. Elform -2 was used to account for the fact that meshing the roller inevitably introduced elements with poor aspect ratios near its interface with the pin, as highlighted in Figure 5.



Figure 5. Element Aspect Ratio at Pin-Roller Interface

# **Constitutive Models**

The above methodology was applied to three combinations of materials for the roller and jamb plate selected from hardened steel, stainless steel and aluminum. Damage to the roller, whether mechanical wear or corrosion, was assumed to be preferred to damage to the jamb because roller damage is more localized and can be repaired with less effort. Consequently, candidate materials for the roller and jamb plate were selected with two objectives: (1) minimize wear of the roller by ensuring the roller wore preferentially to the jamb plate and (2) maximize galvanic compatibility as possible or have the roller be anodic (or active) to the jamb so that the roller corrodes preferentially to the jamb. These considerations resulted in three cases:

- 1. Case 1 hardened steel roller and stainless steel jamb plate
- 2. Case 2 stainless steel roller and hardened steel jamb plate
- 3. Case 3 aluminum roller and hardened steel jamb plate

The constants for Case 1 through 3 are shown, respectively, in Table 1 through Table 3. For modeling purposes, the pin through the roller was assumed to be A36 steel. Material input constants, specific to each metal, were obtained from [3] and [4].

Table 1. Case 1 Constitutive Models by Part	
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Material	Part	No.	Keyword	Input Data Source(s)
Steel Rc 55-60	Roller			Atlas of Strass Strain
304 SS	Jamb Plate	018	*MAT_POWER_LAW_PLASTICITY	Auas of Stress-Strain
A36	Pin			

Table 2.	Case 2	Constitutive	Models	by Part
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Material	Part	No.	Keyword	Input Data Source(s)
Steel Rc 55-60	Jamb Plate			Atlas of Strass Strain
304 SS	Roller	018	*MAT_POWER_LAW_PLASTICITY	Atlas Of Stress-Strain
A36	Pin			

Material	Part	No.	Keyword	Input Data Source(s)
Steel Rc 55-60	Jamb Plate			Atlas of Strass Strain
Al 6061-T6	Roller	018	*MAT_POWER_LAW_PLASTICITY	Allas of Stress-Strain
A36	Pin			Curves [5] [4]

Table 3. Case 3 Constitutive Models by Part

### Contacts

<u>Friction:</u> \*CONTACT\_AUTOMATIC\_SURFACE\_TO\_SURFACE\_MORTAR was used between the roller and the pin and between the roller and the jamb plate. The static ( $f_s$ ) COF and dynamic ( $f_d$ ) COF were set to zero for the roller-pin contact to represent perfect lubrication. For the roller-jamb-plate contact,  $f_s$  and  $f_d$  were 0.74 and 0.57, respectively, when the contact was steel-on-steel. For aluminum-on-steel contacts,  $f_s$  was 0.61, and  $f_d$  was 0.47 [5].

The coefficient of friction was defined as a function of sliding speed with a curve-fitted exponential decay coefficient (DC) of 8 as shown Figure 6 for steel. DC = 8 was used for all contacts. Test data was used to determine the exponential decay coefficient from Stembalski [6]. The friction inputs are summarized in Table 4.



Figure 6. Steel-on-Steel Coefficient of Friction Function

Table 4. Contact Inputs for COF

Contact Type	<b>f</b> s	f <sub>d</sub>	DC	Source
Steel-on-Steel	0.74	0.57	0	CRC Handbook [5],
Aluminum-on-Steel	0.61	0.47	ŏ	Stembalski [6]

<u>Wear:</u> The parameters for Archard's law are input using \*CONTACT\_ADD\_WEAR. The inputs for Cases 1 through 3 are provided in Table 5 through Table 7, respectively.

For Case 1, the dimensionless scale factor k was obtained from Archard [2]. For the unit system used in this analysis, LS-DYNA accepts hardness in units of MPa. The Vickers hardness scale can be converted to MPa by multiplying the Vickers number by 9.807 m/s<sup>2</sup> (acceleration due to gravity). The Vickers hardness of 304 stainless steel is 129 [7], [8], which converts to 1265 MPa. A Rockwell C hardness of 55-60 for the hardened steel roller converts to 694 Vickers hardness [9], equivalent to 6806 MPa.

Variable	Description	Material	Value
p1	Dimensionless scale factor k	Hardened tool steel on stainless steel	1.7E-05
p2	Slave surface hardness parameter H <sub>s</sub>	Hardened steel (Rc 55-60) roller	6806 MPa
р3	Master surface hardness parameter H <sub>m</sub>	304 stainless steel jamb plate	1265 MPa

Table 5. Case I wear inputs	Table 5.	Case 1	Wear	Inputs
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Case 2 was identical to Case 1 except the roller was set to be 304 SS and the jamb plate hardened steel. The associated inputs are listed in Table 6.

#### Table 6. Case 2 Wear Inputs

Variable	Description	Material	Value
p1	Dimensionless scale factor k	Hardened tool steel on stainless steel	1.7E-05
p2	Slave surface hardness parameter H <sub>s</sub>	Stainless steel roller	1265 MPa
р3	Master surface hardness parameter H <sub>m</sub>	Hardened steel jamb plate	6806 MPa

Case 3 required calculation of the wear coefficient k for aluminum on steel. This value was calculated to be 6.7E-04 from data in published Pramanik [10]. The Vickers hardness of aluminum is 107 [7], [8] which results in a hardness input of 1049 MPa. These inputs are listed in Table 7.

#### Table 7. Case 3 Wear Inputs

Variable	Description	Material	Value
p1	Dimensionless scale factor k	Aluminum on steel	6.7E-04
p2	Slave surface hardness parameter H <sub>s</sub>	Aluminum roller	1049 MPa
р3	Master surface hardness parameter H <sub>m</sub>	Hardened steel jamb plate	6806 MPa

### **Calculated Results**

The results for the wear calculation for Case 1 through 3 were as follows.

### Case 1: Hardened Steel Roller and 304 Stainless Steel Jamb Plate

Figure 7 shows the Case 1 wear depth contour from LS-PrePost for the stainless steel jamb plate, and Figure 8 shows the wear depth for the hardened steel roller. The contour renders wear depth over the service life of the machinery; it is set to a wear depth of 1.0e-03 mm maximum to show wear distribution. The figures also report the magnitudes of maximum wear over the service life: 0.073 mm (2.9 mil) on the plate and 0.024 mm (0.94 mil) on the roller. This result is consistent with the fact that the roller has higher hardness.



Figure 7. Case 1 Wear Depth - Stainless Steel Jamb Plate



Figure 8. Case 1 Wear Depth - Hardened Steel Roller

Case 2: 304 Stainless Steel Roller and Hardened Steel Jamb Plate

The results for Case 2 are shown in Figure 9 and Figure 10. For comparison with the previous case, the maximum in the contour is set to 1.0e-3 mm. As shown in the figures, the maximum wear in the plate was predicted to be 0.011 mm (0.43 mil), whereas the maximum wear in the roller was 0.083 mm (3.3 mil). The wear on the plate in Case 2 is noticeably less than for Case 1. The reverse holds for the roller: roller wear in Case 2 is appreciably more than in Case 1.



Figure 9. Case 2 Wear Depth - Hardened Steel Jamb Plate



Figure 10. Case 2 Wear Depth - Stainless Steel Roller

Case 3: Aluminum Roller and Hardened Steel Jamb Plate

Figure 11 shows the wear depth contour for the hardened steel jamb plate, and Figure 12 shows the wear depth for the aluminum roller. The maximum wear depth in the plate is 1.5 mm (57 mil). The wear in the roller is widely distributed and large, with a maximum of 6.6 mm (260 mil). This level of wear represents failure of the roller, which has a bearing surface thickness of 6 mm (240 mil). This amount of wear would significantly degrade the operation of the roller before reaching panel service life.

The root cause of the large wear depth for the aluminum roller is the fact that its wear coefficient k against steel is nearly 40 times larger than the wear coefficient of stainless steel against steel, i.e. 6.7E-04 for aluminum versus 1.7E-05 for stainless steel. The pairing of aluminum and hardened steel was considered to minimize galvanic difference, but the extensive mechanical wear leaves this paring inadmissible.



Figure 11. Case 3 Wear Depth – Hardened Steel Jamb Plate



Figure 12. Case 3 Wear Depth – Aluminum Roller

### **Conclusions and Recommendations**

From these results, several recommendations can be made. First, Case 2 is the best option of the three cases evaluated. Case 2 localizes mechanical wear to the stainless-steel roller, which is easier to replace than the hardened steel jamb plate. The stainless-steel roller is noble to the hardened steel in the galvanic series, but under near-dry operating conditions, such that the only source of moisture is intermittent condensation, the average galvanic current is expected to be low. Adding insulator is expected to further reduce the galvanic current and associated degradation rate due to corrosion.

The basis for this calculation is Archard's law. The combined LS-DYNA and LSPP wear solver applied Archard's law on a per element basis, over the elements in the contact interface. For validation beyond Archard's law, a validation experiment is required. Such an experiment would include wear history for comparison against calculation.

A limitation to this approach is the run times required to calculate meaningful results. The calculations were run using MPP R11.1.0 (revision 139588) and running one analysis case required 1300 core hours. For this approach to be practical for routine design, the time required to reach the implicit solution must be decreased.

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For future work, it is recommended that means of collecting operational data on machinery in buildings be part of mechanical design. Instrumentation for gathering data might include load cells in the rollers, accelerometers in the panel and laser scans of wearing components. The FEA model of the machinery can serve as a framework for gathering operational data on facility hardware. Once the data is gathered, it can feed back into the model to increase the model's accuracy. In this way, the model can serve as a digital twin of the panel to optimize the panel over its service life.

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