Failure Analysis and Redesign of a Pressure Powered Pump Mechanism

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ABSTRACT
This paper identifies the field failures of a Pressured Powered Pump (PPP) mechanism and presents the analysis and subsequent corrective action taken to prevent these failures. The Pressure Powered Pump is a mechanism that converts the liquid level within the body of the pump to the position and actuation of two valves on top of the pump. The two valves control motive pressure within the pump, and therefore, control the pumping action. Tracking in-use field data, it was seen that the pump fails at several locations, which, regardless of the location of the failure, can be attributed to an ill-designed crank-slide mechanism inside the pump. The purpose of the project, and thus the subject of this paper, is the analysis of these failures, the identification of the typical mode of failure, and the subsequent corrective action taken to prevent these types of failures.

The PPP project was executed at the University of South Carolina with support from the manufacturer of the pump. To identify and prevent failures of the pump, three phases were undertaken. First, a failure analysis of the existing mechanism and failed components was performed. Second, using the analysis information, a redesign of the pump was undertaken. And third, a new analysis was performed on the redesigned mechanism, which turned out to be a slider crank. The outcome of this work is a pump whose mechanism now operates under reduced forces and stresses; thus, results in a pump with a predictably longer lifecycle and far fewer failures.

INTRODUCTION
As an energy saving practice, many organizations that use steam for process control or heating take considerable effort to recover the subsequent hot condensate. This condensate is returned to the boiler, where it is again heated and pressurized to the system requirements. The energy savings is realized because the temperature of the condensate is considerably higher than that of fresh water supplied at room temperature. Therefore, the boiler needs to provide less energy to convert the condensate to steam.

To produce greater energy savings, the pressure powered pump (PPP) was developed. The advantage of PPP’s is that they use no motors, and therefore, no electrical energy to move condensate. Instead, the steam available in the facility is used to pump the condensate from its source back to the boiler. This is accomplished by a mechanism which alternately exhausts and fills the pump body with steam. The mechanism, shown in Fig. 1, is powered by the buoyancy of the float rising and falling with the condensate as it is pumped.

During the exhaust cycle, air within the pump body exhausts as condensate fills the volume. Condensate flows into the pump body through a check valve, and once the liquid reaches a predetermined level, the mechanism closes the exhaust valve and simultaneously opens the steam inlet valve. For the duration of the inlet cycle, steam pressurizes the pump body and forces the condensate out of the unit through a second check valve. This continues until the condensate reaches its low level within the pump body; then the steam inlet valve closes, the exhaust valve opens, and the cycle repeats. A complete cycle typically requires 20 seconds and pumps approximately eight gallons of condensate.

The pumping application and use of a float powered mechanism require the pushrod and the pushrod actuator to be stationary during the majority of the cycle time. The pushrod only moves when the driving link and pushrod actuator pass their toggle position, and the extension spring pulls the two links together. See Figure 2. The configuration of the pushrod actuator’s tertiary link, combined with the constraints of the mechanism’s casting, limit the pushrod to vertical, linear motion. Although the float travel is a long sweeping are resulting in eight inches of total vertical movement, the pushrod moves only ¼ inch. This slight movement is all that is required to actuate the valve pair.

The pushrod’s direction of travel and that of the float are directly related. When the float is rising (negative y), the pushrod will also suddenly rise as the float nears the top of its path. When the float is descending (positive y), the pushrod will suddenly drop as the float approaches its lowest point. The sudden motion of the pushrod
simultaneously closes one valve and opens the other; therefore, when
the pushrod is rising, the exhaust valve closes and the inlet valve
opens. The valve actuation is opposite for the pushrod descending.
Considering the severe environment, the continuous cyclic
motion of the pump can lead to damaged components. This paper
addresses the modes of failure, the motion of the mechanism, the
resultant forces, and the proposed redesign.

OBSERVED FAILURES AND MECHANISM DRAWBACKS
The modes of failure for pressure powered pumps are as varied as
the applications. However, there are two types of failures that are
most common. One frequent failure is due to wear of the pins.
Excessive wear on the pins can cause misalignment of components
resulting in poor pumping action, and in severe cases, the pins have
fractured. A second and more common failure is a fractured pushrod.

Pushrod failures typically occur near the actuator plate, just below the
steam inlet and exhaust valves.

These two modes of failure are directly related to the forces
generated during the snap-action of the mechanism. The forces on the
pin joints and the pushrod are greater than necessary due to the
constraints of the current design. The pushrod is restricted to vertical
translation by both the cast mechanism support bracket and the valve
actuator disc. However, the pin connecting the bottom of the pushrod
to the pushrod-actuator is required to move through an arc.

Using Gruebler’s equation (Norton [1]), the mechanism has zero
degrees-of-freedom. See Eq. (1) and the kinematic skeleton of the
mechanism, Fig. 3.

\[ D.O.F. = 3 \cdot (L - 1) - 2 \cdot J_1 - J_2 \]  (1)

The three full joints \( J_1 = 3 \) are the two revolute joints \( R_1 \) and \( R_2 \),
and the single prismatic joint \( P_1 \). Motion is achieved, however, due to
imperfect constraints on the pushrod’s linear translation and the slight
bending of the pushrod during each cycle. These incompatible
requirements of linear motion and rotational motion cause excessive
force to be exerted in the x-direction where the pushrod actuator and
pushrod are connected through the revolute joint, \( R_2 \). It is this force
that increases the wear on the pins and causes the cyclic bending of the
pushrod. Another force due to the impact of the exhaust valve
compounds the bending stresses induced on the pushrod. This impact
transmits a bending moment through the actuator disc (see Fig. 1) and
further leads to premature pushrod failure.

Both of these described failures have been addressed through the
iterative design process with varying degrees of success. However,
the redesigns have focused on heat treatment or strength of material
issues rather than addressing the zero degree-of-freedom nature
inherent to the current mechanism.
ANALYSIS OF CURRENT MECHANISM

Velocity and acceleration measurements were taken to understand the magnitude of the forces generated during the sudden snapping movement of the mechanism. A lightweight, small aluminum plate was attached to the pushrod to be used as an instrumentation platform. Velocity and acceleration curves were established for both the ascending and descending motion of the pushrod using a Linear Variable Transformer (LVT) and an accelerometer attached to the aluminum plate.

The velocities both ascending and descending neared 25 in/s, but took place over a short time period on the order of .025 seconds (see Fig. 4). Approximating the acceleration as linear results in an 960 in/s² acceleration, which corresponds with the accelerometer measured acceleration of 3 g’s. The curves also demonstrated the pushrod’s severe deceleration due to the seating of the exhaust valve. The deceleration approached values of 60 g’s and resulted in the pushrod “ringing.” Therefore, each stroke of the mechanism results in several changes in direction and added bending moments.

The acceleration data generated during testing had considerable noise due to the sensitivity of the accelerometer. Therefore, the direct acceleration data was used only as a check for the mathematically predicted acceleration.

A third-order polynomial curve was fit to the velocity data during the initial movement of the pushrod providing a velocity equation for the pushrod. Acceleration values for each time step of .002 seconds were established by differentiating the velocity equation with respect to time. The position of the pushrod at each time step was also predicted by integrating the velocity equation with respect to time.

\[
v(t) = 2 \cdot 10^6 \cdot t^3 - 33210 \cdot t^2 + 528.08 \cdot t \quad (2)
\]

\[
a(t) = 6 \cdot 10^6 \cdot t^2 - 66420 \cdot t + 528.08 \quad (3)
\]

\[
y(t) = 0.5 \cdot 10^6 \cdot t^4 - 11070 \cdot t^3 + 264.04 \cdot t^2 \quad (4)
\]

Equations (2), (3), and (4) are curves based on the velocity data obtained during one data run (see Fig. 5, 6, 7). Together, they describe the position, velocity, and acceleration of the revolute joint, \( R_2 \) connecting the pushrod and pushrod actuator during the initial movement of the mechanism, or approximately the first .026 seconds.
Numerical results of Eq. (2), (3), and (4) were obtained for .002 second time steps, and this data was used to obtain radial and tangential accelerations of the pushrod actuator; see Eq. (5) and (6). Tangential acceleration was then used to find the angular accelerations at each time step.

\[ a_r = a(t) \cdot \cos(\theta) \]  
\[ a_r = a(t) \cdot \sin(\theta) \]  

An external force due to the bending of the pushrod was calculated based on the geometric parameters of the pushrod and the casting. For calculating the displacement of the pushrod, it was assumed that the interface between the pushrod and casting was a perfect prismatic joint. Therefore, the pushrod is limited to vertical translation. With this assumption, the bottom of the pushrod, where it is pinned to the pushrod actuator, must deflect in the x-direction to compensate for the rotational motion of the pushrod actuator. This deflection varies between 0 and .029 inches. From this information, an external force in the x-direction was calculated using Eq. 7.

\[ F_{Ax} = \frac{\delta \cdot 3 \cdot E \cdot I}{L^3} \]  

The total external force had a maximum magnitude of 129 lb, but like the other values associated with the analysis, it varied with each time step.

The external force and the computation of the acceleration data at each time step were used to calculate mechanism forces for the following free body diagram, Fig. 8, and system of six equations and six unknowns; see Eq. (8) - (13).

\[ F_{12x} + F_{22x} = m_2 \cdot a_{2Gx} - F_{Ax} - F_{s2x} \]  
\[ F_{12y} + F_{22y} = m_2 \cdot a_{2Gy} - F_{Ay} - F_{s2y} \]  
\[ R_{12x} \cdot F_{12y} - R_{12y} \cdot F_{12x} = \]  
\[ I_{pin} \cdot \alpha - R_{s2x} \cdot F_{s2y} + R_{s2y} \cdot F_{s2x} \]  

As anticipated, the forces associated with the current mechanism reach very high values due to the zero degrees-of-freedom constraint and the high magnitude of acceleration. The variation in force throughout the motion closely follows the trend of the mechanism’s acceleration, as expected.

RECOMMENDATION FOR REDESIGN

The magnitude of force exhibited during a single stroke of the mechanism is compounded over time and is a significant contributor to pin wear and pushrod fracture. These forces lead to the first recommendation for the mechanism’s redesign; convert the existing three link, three full joint design to a crank slide, one-degree-of-freedom mechanism. The simplest and least costly design change would be to split the pushrod into an upper and lower link (see Fig. 9). By pinning two links together near the cast slot, the upper pushrod could be limited to the required vertical translation, while the lower pushrod would be free to travel with the rotational motion of the pushrod actuator. This design change could be implemented without changes to any components except the pushrod.

Although this design recommendation would significantly reduce the forces associated with the mechanism, it does not address the bending moment applied to the pushrod due to the seating of the exhaust valve. The exhaust valve is offset from the pushrod by 0.6 inches, but is attached via the actuator disc. This disc can be considered rigidly attached to the pushrod and is used to transfer the motion of the pushrod into the actuation of the valves.

Each data run for the pushrod ascending indicates impact between the exhaust valve head and valve seat occurs at a velocity over 20 in/s. This impact results in an essentially instantaneous stop and reversal of
the motion with measured decelerations reaching 60 g’s or 23,000 in/s². The resultant force of the impact is transferred to the pushrod creating a bending moment perpendicular to the bending moment created by the motion of the mechanism; see Fig. 1. The location of this bending moment on the current mechanism and the observed failure point on the pushrod coincide.

This bending moment can be eliminated through a second design change. Simply aligning the exhaust valve with the pushrod will allow for the forces to be transferred into the pushrod as a direct compressive force. An actuator disc would still be required to actuate the inlet valve, but the forces associated with the inlet valve are considerably less than those produced by the seating of the exhaust valve. Unfortunately, this redesign recommendation cannot be accomplished without changes to the pump-mechanism-cover that houses the valves. This change may require a new pattern for the rough cast cover unless machining changes could reroute the exhaust valve to a centered position.

A design change to the exhaust valve head would allow it to be threaded directly on to the upper pushrod. Doing so allows for the needed adjustment that is currently achieved with the actuator plate and two supporting nuts; see Fig. 1. The redesigned mechanism would utilize the bottom of the exhaust valve in place of the upper adjustment nut to secure the actuator plate.

ANALYSIS OF REDESIGN

The recommended redesign is a slider-crank linkage. Therefore, the forces associated with the mechanism can be analyzed using accepted dynamic force analysis procedures. Having solved the kinematic analysis to establish position, velocity, and acceleration of each link, the dynamic calculations follow by summing the forces and torques of each link. This approach results in eight equations and eight unknowns; see Eq. (14) - (21).

\[
\begin{align*}
F_{12x} + F_{32x} &= m_2 \cdot a_{pint} - F_{s2x} \quad (14) \\
F_{12y} + F_{32y} &= m_2 \cdot a_{piny} - F_{s2y} \quad (15) \\
-F_{32x} + F_{43x} &= m_3 \cdot a_{G3x} \quad (16) \\
-F_{32y} + F_{43y} &= m_3 \cdot a_{G3y} \quad (17) \\
R_{32x}^y \cdot F_{32x} - R_{23x} \cdot F_{32y} &= 0 \quad (18) \\
R_{43x} \cdot F_{43y} - R_{43y} \cdot F_{43x} &= I_{G3} \cdot \alpha_3 \quad (19) \\
-F_{43x} + F_{14x} &= 0 \quad (20) \\
-F_{43y} + F_{14y} &= m_4 \cdot a_{G4y} \quad (21)
\end{align*}
\]

The forces at the pins for the redesign are lower than the pin forces found for the original mechanism; see Fig. 10 – 12. Furthermore, there is no external force applied to the mechanism due to the bending of the pushrod, and there is no bending moment applied to the pushrod due to the seating of the exhaust valve.

![Figure 9 Two Piece Pushrod](image)

![Figure 10 Force between Ground and Pushrod Actuator](image)
CONCLUSIONS
The results of the existing mechanism’s analysis compared with those of the redesign clearly show a reduction in forces experienced at the pin locations. The redesigned slider-crank does not require the pushrod to compensate for the constraints of the mechanism by bending during each cycle. Instead, it allows for the free movement of all the links while maintaining the needed linear valve actuation. The combination of lower pin forces, the added degree-of-freedom, and the removal of the bending moment due to the exhaust valve results in a pump mechanism that will experience less frequent failures. The life-cycle of the mechanism is therefore increased.

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REFERENCES