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## Article

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Development Engineering

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*Suggested Citation:* Ottosson, Hans J.; Mattson, Christopher A.; Johnson, Oliver K.; Irot, Immy (2022) : Use of simulation and wear prediction to explore design improvements to the cup seal in the India Mark II/III hand pump system, Development Engineering, ISSN 2352-7285, Elsevier, Amsterdam, Vol. 7, pp. 1-9,  
<https://doi.org/10.1016/j.deveng.2022.100092>

This Version is available at:

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# Use of simulation and wear prediction to explore design improvements to the cup seal in the India Mark II/III hand pump system

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## ARTICLE INFO

### Keywords:

Engineering for global development  
Design for the developing world  
Robustness  
Wear  
Optimization  
Design exploration  
Design-of-experiment  
India Mark II/III hand pump

## ABSTRACT

The India Mark II/III hand pump system is now over 40 years old and has become a staple in how water is retrieved in many rural areas across India and Africa. With over 4 million installations, it is estimated that 10% of the world's population is using one on a daily basis. One of the components that fails most frequent is the nitrile cup seal found in the under-ground pump cylinder, often causing the pump to become derelict for undesirably long periods of time. This paper's focus is on this cup seal and how the wear and life of the seal can be predicted. A test rig was created to evaluate and test cup seals. From this, seal performance and the nitrile cup seal wear coefficient were found. A two-dimensional finite element model of the seal and the pump cylinder was then developed and used to calculate the pressure distribution and wear of the seal. Archard's wear law was used to simulate the wear. From this, models for pump performance and seal wear prediction were created. This data was then used to explore possible design improvements to the cup seal. By combining this data with engineering tools such as ANSYS, MATLAB, and JMP, a new seal design was generated that has the potential to last 12% longer than the original cup seal.

## 1. Introduction

Mechanical water hand pumps, with the India Mark II/III hand pump system being the most prevalent (see Fig. 1), are used worldwide to supply clean water to at least 1 billion people (Carter and Lockwood, 2011). But over time, these mechanical hand pumps become dysfunctional, largely due to the degradation of wear items such as seals, bushings, and bearings (see Fig. 2). It is estimated that 15% of India Mark pumps are not supporting the needs of the local communities (Mudgal, 1997). Approximately 70% of these are non-functional due to reparable hardware failures (Klug et al., 2018). The degradation of the nitrile cup seal causes the pump efficiency to worsen (Mudgal, 1997; Klug et al., 2018) and is the most frequent point of failure for this system (see Fig. 2) (Erpf, 2004).

The India Mark II/III hand pump system, with over 4 million installations worldwide (Carter et al., 2010; Mudgal, 1997), is a reciprocating pump system with the piston moving up and down inside the pump cylinder during operation with each stroke of the handle (see Fig. 3). The piston is installed with two cup seals. The nitrile cup seal can be seen in Fig. 4 and the cup seal dimensions can be seen in Fig. 5. When in use, the water pillar in the *rising main* produces a hydrostatic head pressure, acting on the cup seals. The performance of the cup seals determines how much of the water is lifted during each pump stroke.

With each pump servicing between 150 and 250 people (Mudgal, 1997; Ottosson, 2021), approximately 10% of the world's population is affected by India Mark seal decisions. It is rare that a single product affects such a large portion of the world's population and this acts as a great motivation for us to present findings that lay the ground work for re-engineering the cup seal. With a better understanding of the cup seal mechanics and its performance sensitivity to real geometric and material variations, the potential to engineer a more robust and longer lasting seal can be examined.

This study models predicted pump performance, models the wear of the nitrile cup seal, and investigates improvements to the cup seal. To do this, the following was completed:

1. Tested seals in field and laboratory settings
2. Performed seal degradation tests
3. Developed a prediction model for pump performance
4. Developed a wear prediction model for the nitrile cup seal
5. Developed and explored different seal geometry designs
6. Set up design-of-experiments and explored the design space
7. Performed numerical optimizations.

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Fig. 1. India Mark II hand pump. Source: photo by the authors.

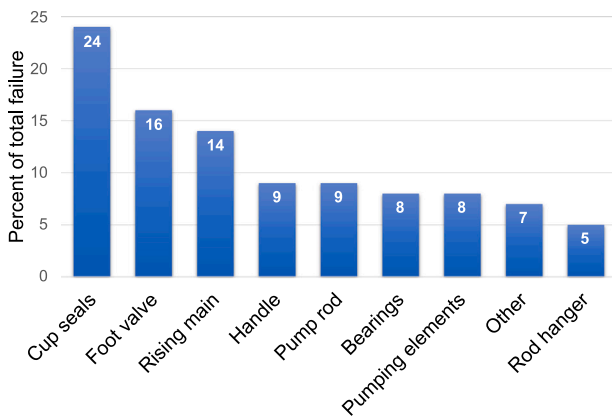


Fig. 2. Points of failure for the India Mark II/III hand pump system (percent of total failures) (Erpf, 2004).

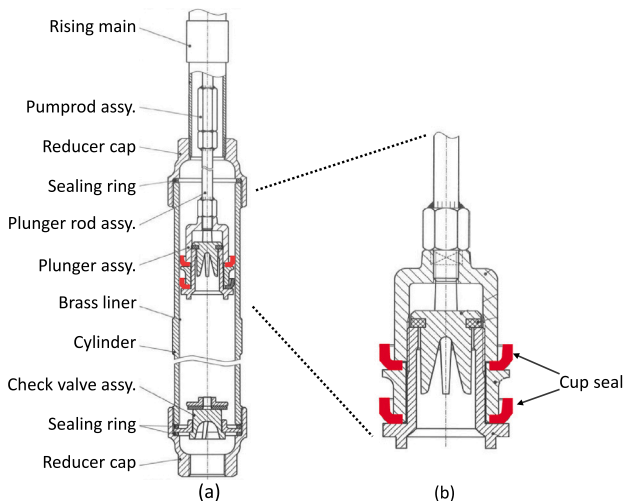


Fig. 3. (a) Schematics of the India Mark II under-ground pump cylinder and (b) piston assembly (cup seals highlighted) (Erpf, 2007).

This paper is part of a larger study to understand – mechanically and socially – how hand pumps (as a whole system) perform, degrade, and ultimately meet human needs. The findings presented here follow a robustness analysis of the India Mark II/III hand pump system done



Fig. 4. New, off-the-shelf India Mark II/III cup seal.

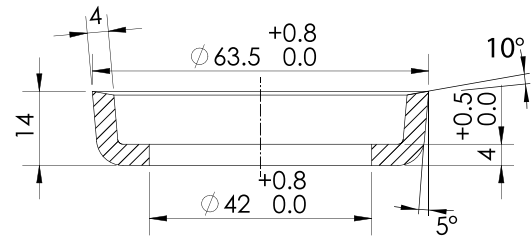


Fig. 5. Specified cup seal dimensions (mm) (Erpf, 2007).

by the authors (Ottosson et al., 2021) with a focus on the wear of the cup seal. The findings of the present paper will be incorporated into our larger research where field sensor data were used to capture pump usage scenarios, and machine learning techniques to begin mapping engineering design parameters, such as those of the cup seal, to the social impacts of an engineered product (Ottosson et al., 2020; Stringham et al., 2020).

In the remainder of this paper a seal performance model is presented, followed by a wear prediction model. The geometric improvements to the cup seal is then discussed to show how the models facilitate design. Lastly, the changes in the cup seal design is looked at in order to see how they can affect pump users and seal manufacturers financially.

### 1.1. Approach and limitations of the current study

While there are many sources of failure, this paper focuses only on the eventual failure of the cup seal and geometric improvements to it. The focus on the cup seals is based on literature research (Fagan et al., 2015; Erpf, 2004; Ottosson, 2021), our interviews with hand pump technicians and suppliers in Uganda, and on collaborations with an NGO providing access to clean water to people in more than 40 countries (Mattson et al., 2017).

The primary limitation of this study is that the experimentation of dynamic performance of the cup seal is limited to a laboratory setting with a pump depth of 0.6 m. In a non-laboratory setting, a greater borehole depth increases the hydrostatic pressure on the cup seal. This hydrostatic pressure is caused by the column of water in the rising main). An important implication of this limitation is that cup seal experiencing greater hydrostatic pressure may behave differently in a dynamic setting. This paper did not consider manufacturing variability or its affect on the identified solution. This would be an important part of future work.

## 2. Performance prediction of current cup seal design

To be able to predict the performance of the current cup seal design, seals were purchased in Uganda by the authors. A dynamic test rig was designed and built to test the performance of each seal in terms of water output (see Sections 2.1 and 2.2). A seal degradation test was then performed to find how well the cup seals behaved as their geometric values changed (see Section 2.3). The data was then analyzed and used to create a model to predict water output with cup seal geometry data as input (see Section 2.4).

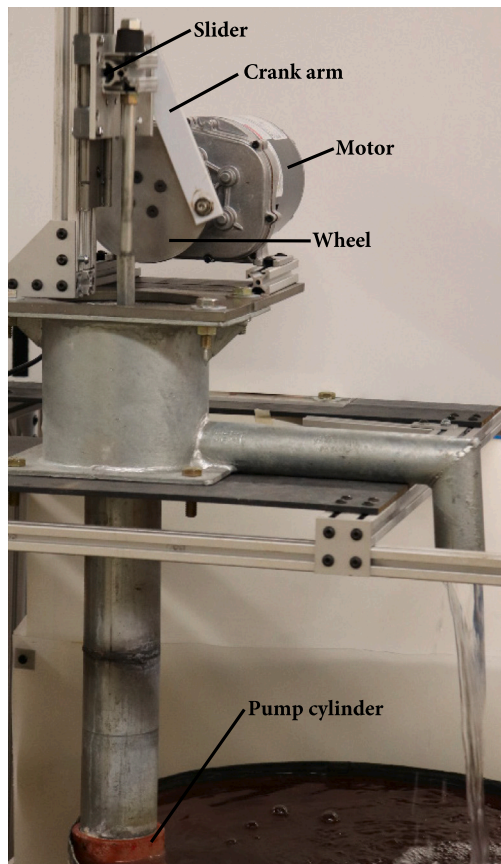


Fig. 6. Dynamic test rig for determining cup seal performance complete with a crank-slider mechanism, (motor, wheel, crank arm, and a slider).

### 2.1. Dynamic pump test rig

To measure how each seal performed off-the-shelf and over time, a dynamic test rig was created (see Fig. 6). The reciprocating movement of the pump was done by a crank-slider mechanism as seen in Fig. 6, simulating the up-and-down linear pump motion done by a human. Consistent with hand pump usage, the mechanism operated at a speed of 42 full pump strokes per minute.

For the test, a seal was inserted in the lower placement in the piston assembly (see Fig. 3) and tested alone. The test setup was created to match a volumetric discharge test done in the field at the time of cup seal/pump cylinder installation (Erpf, 2007). A scale was used to measure the water output (Mango Spot portable scale).

### 2.2. Error analysis for the pump test rig

The dynamic test setup was analyzed by performing two studies with one randomly selected cup seal. The first study was to find the variation in output due to the mechanical pump system and the second was to find the variation in scale readings for the same water output. Each study was repeated 30 times for a single seal.

In the first study, the results of the output ranged from 15.90 to 16.03 liters with an average of 16.00 liters and a standard deviation of 0.02331. For the second test, a set of calibrated weights were used to ensure that the scale was accurate. A combined mass within the range of normal water output for one minute was chosen (13.66 kg) to find the variation of measurements for the scale. Variation was not detected for the scale at a resolution of  $\pm 0.01$  kg.

These tests show that there are minimal or no variation in pump output due to the test rig, drawing the conclusion that the uncertainty in the pump output is negligible.

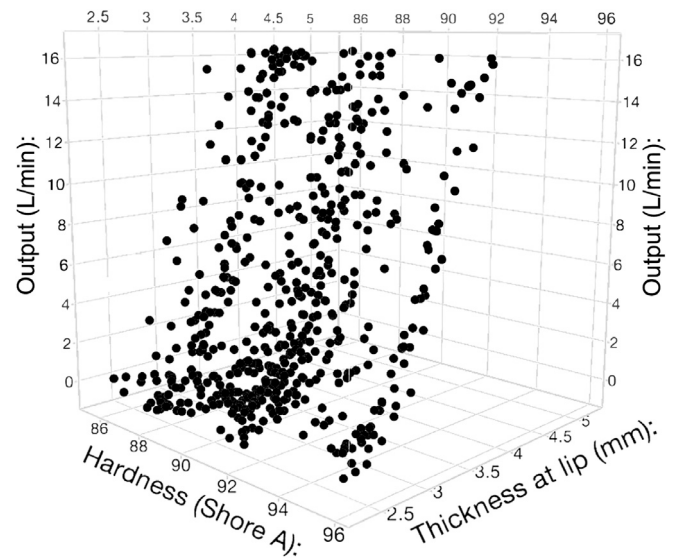


Fig. 7. Results from the seal degradation test.

### 2.3. Seal degradation test

In order to better understand how the seals perform over time, a seal degradation test was created where a seal was worn, measured, and tested repeatedly using the dynamic test rig until it failed to lift the minimum amount of required water. The seal was considered non-functional when the output was less than one liter/minute (similar to when a pump is abandoned in the field (Ottosson, 2021)). Fig. 7 shows a 3D plot of the seal degradation test results where the marks on the graph represent the seal measurements and pump outputs for each seal during the test. Fifty-five seals were used for this test.

### 2.4. Seal performance model

A pump performance model, with the cup seal parameters as input, was derived from the collected data using JMP Pro 15 statistical software (SAS Institute Inc, 2022). Only terms with p-values less than 0.05 were included. Volume was also excluded since it would be hard to measure it in the field. The predicted water output is found by entering the seal outer diameter, wall thickness, and weight values (see Eq. (1)). A plot showing predicted versus actual output can be seen in Fig. 8.

$$O_p = -328.18 + 1.3434 \cdot T_w + 5.2627 \cdot D_o + 51.088 \cdot T_w \cdot W - 655.37 \cdot D_o \cdot W + 4,365.2 \cdot D_o^2 \quad (1)$$

where

$O_p$  is the projected water output (l/min)

$T_w$  is the seal wall thickness (mm)

$D_o$  is the seal outer diameter (mm)

$W$  is the seal weight (g).

### 3. Wear prediction

To develop a wear model for the cup seal, Archard's wear equation was used (Eq. (2)) (Archard, 1953). Archard's wear equation is often used as a universal wear volume predictor (Andersson et al., 2011; Schmidt et al., 2018). It works well for polymer-metal contacts (Ramesh et al., 2019), as is the case for the India Mark II/III hand pump.

$$V = K \cdot F_N \cdot s \quad (2)$$

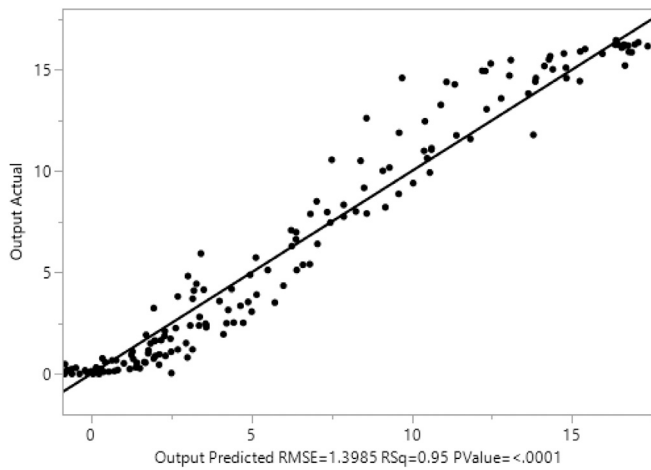


Fig. 8. Predicted output versus actual output (l/min).

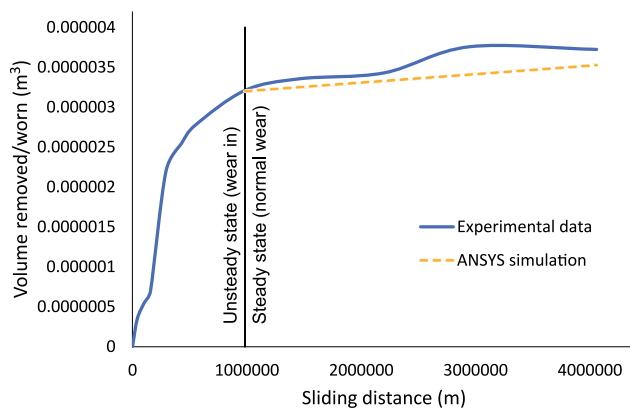


Fig. 9. Wear curve showing the wear experiment and ANSYS results.

where

- $V$  is the wear volume ( $m^3$ )
- $K$  is the wear coefficient ( $m^2/N$ )
- $F_N$  is the normal load (N)
- $s$  is the sliding distance (m).

### 3.1. Wear coefficient

To find the wear coefficient ( $K$ ), a long-term wear experiment was performed with a new cup seal placed in the lower position in the piston (see Fig. 3) with the intention to find the change of wall thickness due to wear over time (Sarkar et al., 2013; Stankovic et al., 2019). Measurements of the cup seal was performed throughout the experiment by removing the cup seal and measuring its lip thickness, weight, and volume. The test was done over a period of a year in the same test rig that was used for the dynamic tests.

The wear results can be seen in Fig. 9. The data corresponds well with the theoretical wear curve (unsteady/steady state of wear) (Zhang, 2004). Eq. (2) was used to calculate the wear coefficient. The value of  $F_N$  was calculated through the use of the test rig. The value of  $K$  was found to be  $2.41E-8 m^2/N$  and in line with tribology literature (see Table 1) (Bhushan, 2013; Stankovic et al., 2019).

### 3.2. Wear prediction model

ANSYS (ANSYS Inc, 2022) was used to create a wear prediction model for the cup seal wear. A 2D model was created in ANSYS with asymmetric contact (only simulates the wear of the seal). Nitrile

Table 1

Typical values of wear coefficient ( $K$ ) for non-metal on metal under different degrees of lubrication Bhushan (2013).

Condition	$K$ ( $m^2/N$ )
No or poor lubrication	$1.5E-6$
Average lubrication	$3E-7$
Excellent lubrication	$3E-8$

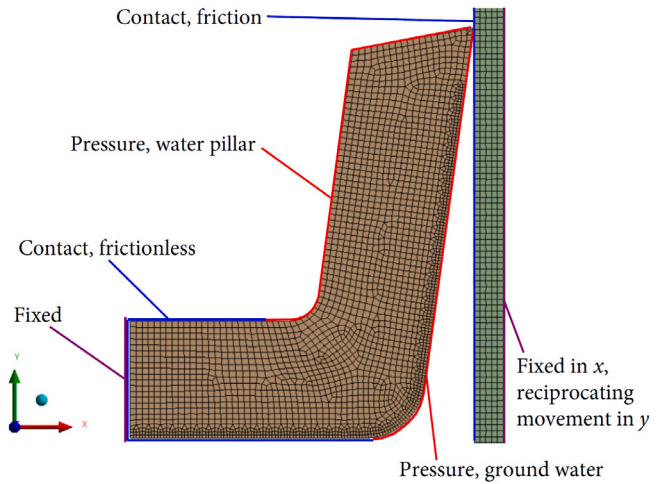


Fig. 10. Model for the ANSYS simulation with the cup seal on the left and pump cylinder on the right.

Rubber/Acrylonitrile butadiene copolymer (NBR), a Mooney–Rivlin 2-parameter material supplied with ANSYS was used for the simulations. This material is a nearly incompressive hyper-elastic material often used in rubber analyses (Agosti et al., 2018; Suraj et al., 2019; Sokmen and Karatas, 2020). The density was determined to be  $1420 kg/m^3$  by taking the mean value of the density of 110 nitrile cup seals purchased in Uganda by the authors (Ottosson, 2021). The setup of the model geometry, the mesh, and the constraints and loading of the cup seal can be seen in Fig. 10. For the mesh, the PLANE182 elements were used and the contact was modeled between the seal and pump cylinder by overlaying the surfaces with contact elements CONTA172 and target elements TARGE169.

To validate the mesh, different mesh sizes were tested with the same setup and the results compared. As the results converged, the mesh size was decided (AutoDesk Inc, 2017). Through this process, the mesh element size for the seal was set to 0.2 mm with the refinement option for the outer and lower seal edge (see Fig. 10). The mesh size for the pump cylinder was also set to 0.2 mm. The *Nonlinear Adaptive Region* option in ANSYS was also applied to the seal body.

The model was constrained by preventing movement in the  $x$ -direction of the left side of the seal and the three sides of the seal that are in contact with the piston were set as a “Frictionless” contact type. The piston was fully constrained and the pump cylinder was prevented to move in the  $x$ -direction and set to move in a reciprocating motion in the  $y$ -direction. The contact pair between the seal and the pump cylinder was set to “Frictional” with a frictional coefficient of 0.15 (see Fig. 10). The frictional coefficient was calculated from experiments using the test rig. It falls within the range found for nitrile in literature (Bhushan and Gupta, 1991; Bhushan, 2013; Dong et al., 2015).

A pressure was then applied to the top of the seal to simulate the pillar of water. The deformation of the seal can be seen in Fig. 11. A lower pressure was also applied to the bottom side of the seal, representing the pressure from the ground water. The right, vertical part of the seal in Fig. 11 is where the seal is in contact with the pump cylinder.

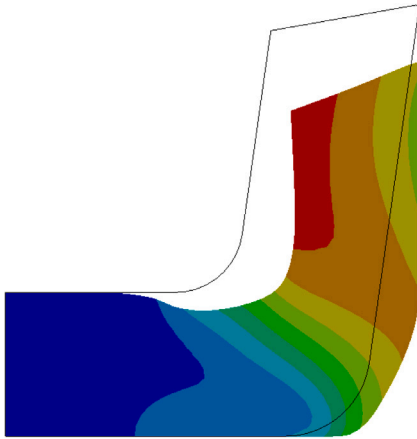


Fig. 11. Seal deformation due to a simulated water depth of 42 m. Right side of the cup seal is in contact with the pump cylinder as shown in Fig. 10).

### 3.3. Wear results for different depths

The ANSYS wear simulation was extended to find the seal wear over time. The wear experiment and ANSYS results can be seen in Fig. 9. Agreement was observed between the steady-state segment of the experimental and simulated results (no wear-in regime was done in ANSYS).

After finding correspondence between the seal degradation test and the FEA analysis for a depth of 0.6 m, a wear analysis was performed for a simulated depth of 42 m (the average well depth in Uganda (Ottosson, 2021)). Two used nitrile seals acquired in Uganda that had previously been replaced due to wear were measured and the average worn-off material was found to be  $1.324 \times 10^{-6} \text{ m}^3$ . The wear rate from the ANSYS simulation for a depth of 42 m was used and it was determined that it would take 12.4 months for the ANSYS model to reach the same amount of wear. This corresponds well with findings during field studies in Uganda and with what is found in literature where the cup seal is to be replaced each year (SKAT, 2008; Action Against Hunger, 2010).

## 4. Potential geometric improvements to the cup seal

Using an ANSYS wear model for predicting cup seal wear life is a useful tool in evaluating different geometric variations (Podra and Andersson, 1999; Hannifin, 2014). Design parameters can quickly be changed in ANSYS, and after each simulation, the different scenarios can be compared. Real-world tests can then be performed with promising geometry.

### 4.1. Seal lip geometry

Since the lip design of a seal impacts performance and wear rates (Hannifin, 2008), FEA software can be used to model and evaluate different geometric designs of the lip. A knife-edge or straight lip design provides good scraping properties but also reduces sealing performance. They are often used in sealing applications where there are high fluid contamination (American High Performance Seals, 2017). See Fig. 12 and Section 4.2 for an FEA comparison between different lip geometries.

### 4.2. FEA comparison between different lip geometries

ANSYS was used to compare the effect different lip geometries could have on wear. The comparison of the results can be found in Table 2 where the original design is used as a baseline. The analyses show different wear rates for the different lip designs, indicating that the way the lip is designed affects the wear and should be considered during a redesign of the seal. The different lip geometries can be seen in Fig. 12. The FEA simulations for this comparison was for a depth of 42 m.

Table 2

ANSYS FEA results at a simulated depth of 42 m for different seal lip designs (numbers less than 1 corresponds to less wear)

Edge design	Seal wear baseline
Original design	1
Knife edge	0.97
Straight edge	1.02
Beveled edge	1.05
Rounded edge	1.05
Optimized design (geometry)	0.88

## 5. Simulation driven product development

This section describes how simulations were used to explore different design options.

### 5.1. Using ANSYS as a design exploration tool

After running the ANSYS simulation for the current cup seal design, the model was parameterized and ANSYS DesignXplorer was used to apply DOE algorithms to the cup seal in order to better understand its design space. A full factorial DOE was performed with seal height, thickness, and top angle as parameters. The DOE table of design points was then solved in batch mode. ANSYS and JMP were used to evaluate the results.

The correlations between seal height, thickness, and top angle in relation to wear were found (see Figs. 13, 14, and 15 where dark blue denotes less wear). These correlations can be used during a redesign of the cup seal.

### 5.2. Seal wear model

A seal wear model with seal parameters as input, was derived from the ANSYS simulation data using JMP. The predicted seal wear is found by entering the seal wall thickness, height, and top angle (see Eq. (3)). A plot showing predicted versus simulated wear can be seen in Fig. 16.

$$\begin{aligned}
 V_p = & 9.707\text{E-}13 - 4.3\text{E-}14 \cdot S_H \\
 & + 1.039\text{E-}13 \cdot T_W + 2.331\text{E-}16 \cdot \alpha \\
 & + 8.257\text{E-}15 \cdot (S_H - 13.6148)^2 \\
 & - 2.7\text{E-}14 \cdot (S_H - 13.6148) \cdot (T_W - 4.03408) \\
 & - 4.09\text{E-}16 \cdot (T_W - 4.03408) \cdot (\alpha + 1.45431) \\
 & - 2.17\text{E-}17 \cdot (\alpha + 1.45431)^2
 \end{aligned} \quad (3)$$

where

$V_p$  is the projected seal wear ( $\text{m}^3$ )

$T_W$  is the seal wall thickness (mm)

$S_H$  is the seal height (mm)

$\alpha$  is the seal top angle ( $^\circ$ ).

### 5.3. Using MATLAB to further optimize and improve the cup seal design

Since optimization is often used to find improved designs (Mattson and Sorensen, 2019), MATLAB (The MathWorks Inc, 2022) was used to minimize the predicted wear equation found from the ANSYS simulations (see Eq. (3)). This gives us the optimization problem seen in Eq. (4) with seal design parameters as inputs (as can be seen in Fig. 17). MATLAB was used for the optimization calculations.

$$\begin{aligned}
 & \text{minimize } V_p \\
 & T_W, S_H, \alpha
 \end{aligned} \quad (4)$$

subject to

$$3.6 < T_W < 4.4$$

$$12 < S_H < 15.5$$

$$-45^\circ < \alpha < 45^\circ$$

where

$V_p$  is the projected seal wear ( $\text{m}^3$ )

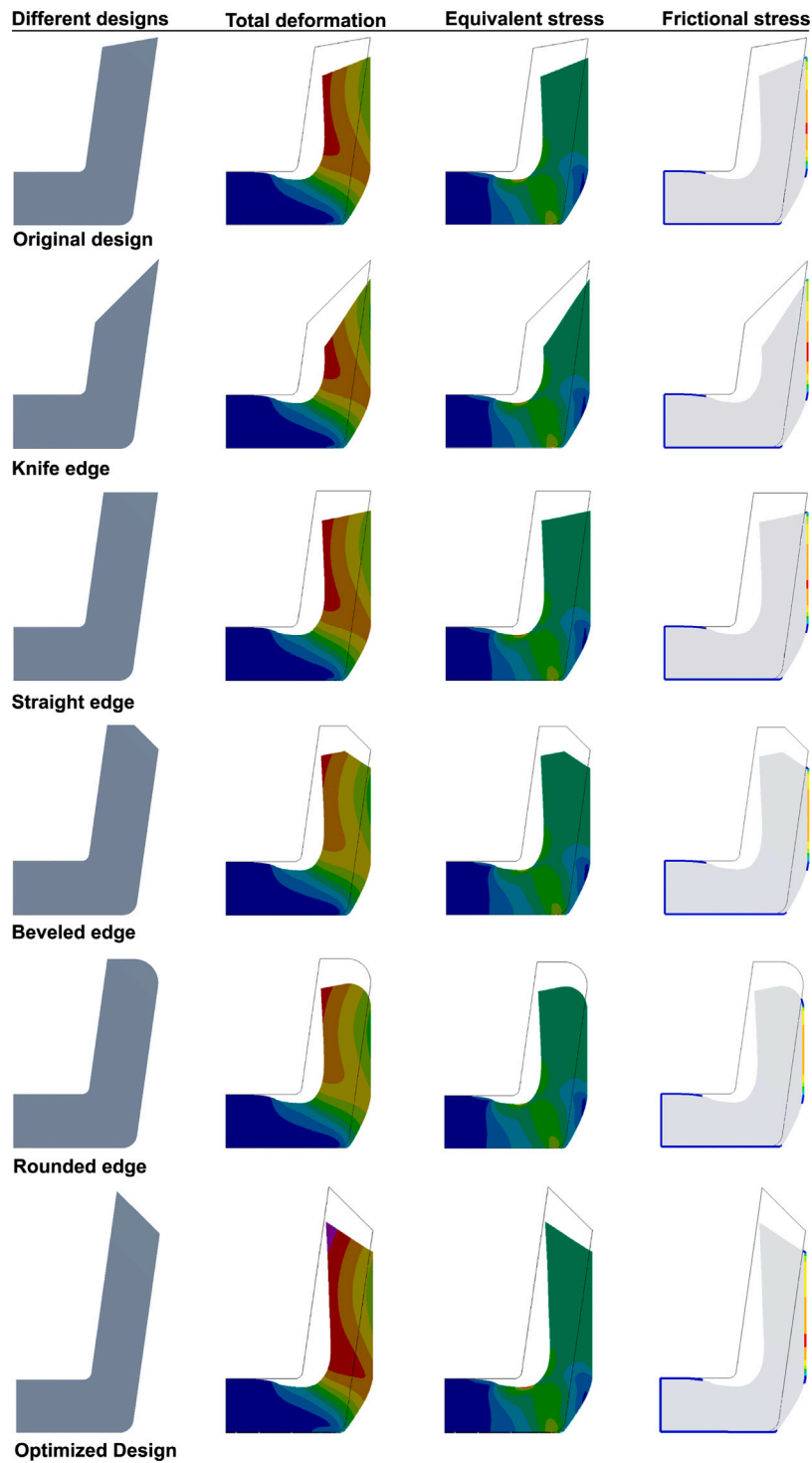


Fig. 12. Different Seal designs used for wear comparison.

$T_W$  is the seal wall thickness (mm)

$S_H$  is the seal height (mm)

$\alpha$  is the seal top angle ( $^\circ$ ).

It is important to recognize what this formulation searches for. It is seeking the values of  $S_H$ ,  $T_W$ , and  $\alpha$  that minimize the wear ( $V_p$ ). This search is bound by the side constraints on  $S_H$ ,  $T_W$ , and  $\alpha$ . These constraints are chosen so that the optimizer searches only within the known functional limits observed through the full factorial experiments. This means that the search will not identify a solution through extrapolation, instead it will use interpolation. Also note that  $V_p$  is

derived from the full factorial experiment where the same hydrostatic pressure is present due to the weight of the column of water. Because of the nature of this formulation, a geometry adequate to produce a seal will result. Notice that a reasonable contact patch is present in both the original and optimal designs (see Fig. 12, Frictional Stress plot).

Since the performance functions have many local minimum points, a Monte Carlo approach was used to find 1,000 initial starting values of the seal parameters to be used in a nonlinear optimization using the Sequential Quadratic Programming (SQP) method, which is executed using MATLAB's `fmincon` function. The results from each run were then

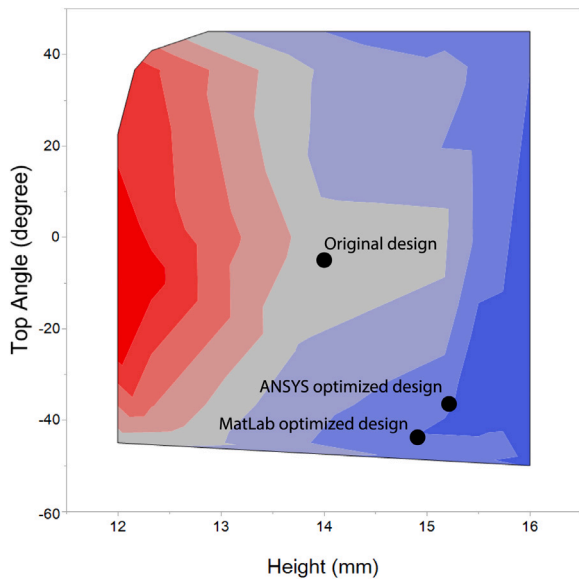


Fig. 13. How wear is affected by seal height and top angle (dark blue denotes less wear). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

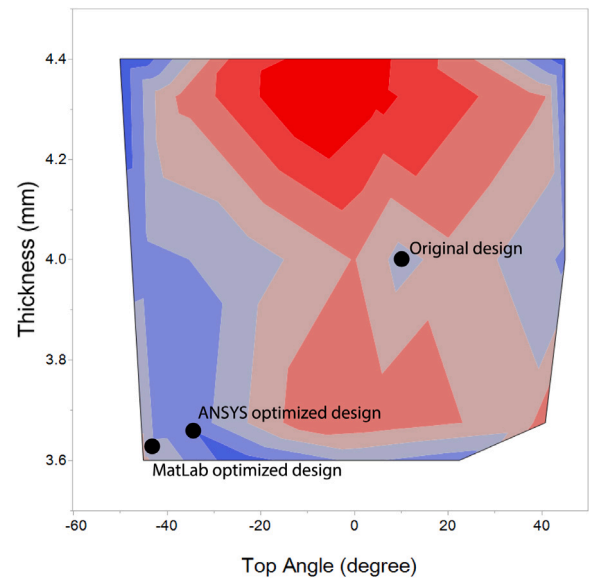


Fig. 15. How wear is affected by seal thickness and top angle (dark blue denotes less wear). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

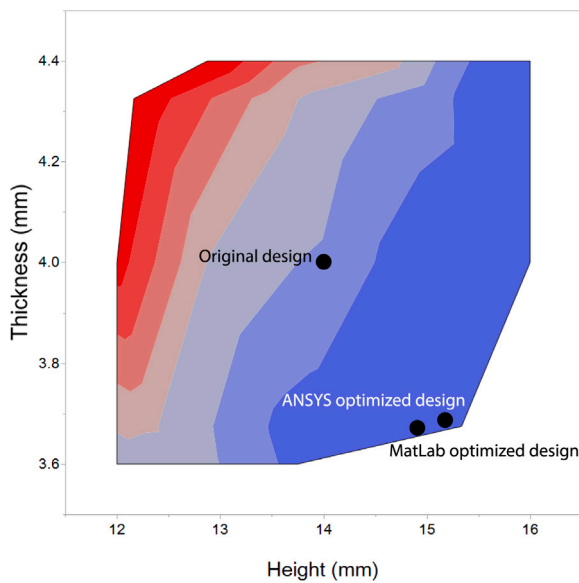


Fig. 14. How wear is affected by seal height and thickness (dark blue denotes less wear). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

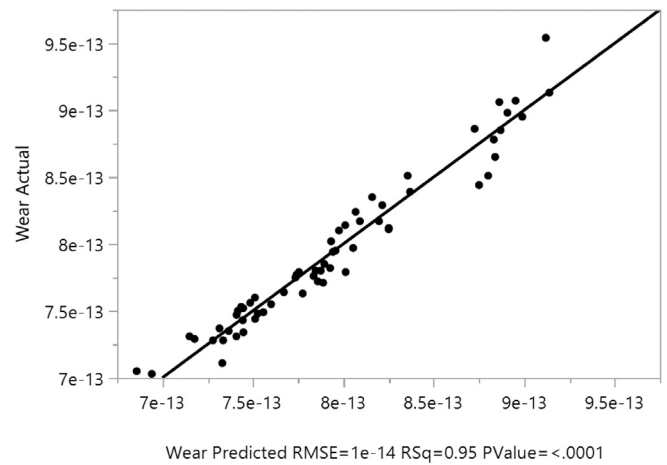


Fig. 16. Predicted wear versus simulated wear ( $m^3$ ).

compared and values for the seal parameters were derived for minimal wear.

The optimized design can be seen in Fig. 18. The seal height increased to 14.9 mm and the seal wall thickness decreased to 3.63 mm. The top of the seal is now leaning outwards instead of inwards at an angle of  $-44.8^\circ$ . The wear simulation results for this improved design can be seen in Table 2. The optimized design performed better in wear when compared to the original design by 12%.

Important to note is that once an optimized design has been determined through the use of analyses, a physical prototype should be made and tested to ensure that all product requirements can be met. Also, to have a lip design with this angle is traditionally not advised in high pressure machinery due to possible collection of debris at the lip (K. Paulson, personal communication, May 21, 2021). Future research will

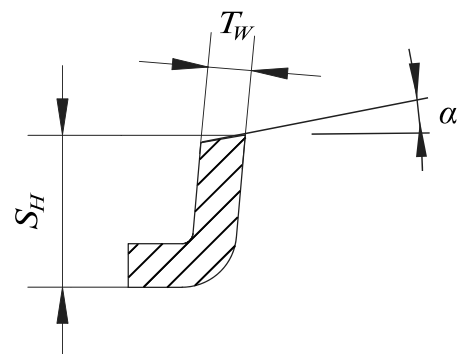


Fig. 17. Seal parameters used in the MATLAB optimization.

determine if this will affect the low speed application of the cup seal negatively. Our suggestion would be to make a seal with a height of 15 mm, a thickness of 3.6 mm, and an angle of  $-45^\circ$  due to variation in manufacturing, and then run physical tests (the initial dimensions were: height of 14 mm, thickness of 4 mm, and angle of  $10^\circ$ ).



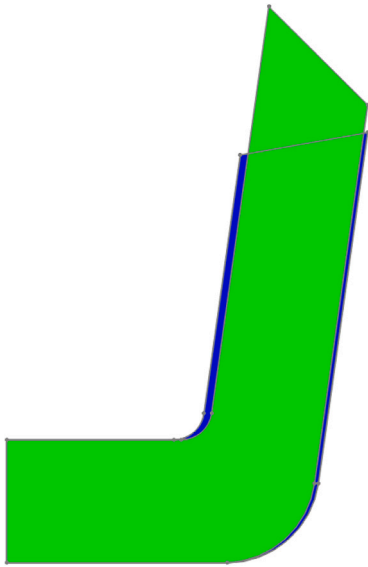


Fig. 18. Profile of optimized seal design overlaid on the original design (optimized design in green color). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

#### 5.4. Material considerations

It was found through the testing of the 110 nitrile cup seals purchased by the authors in Uganda, that hardness affected the cup seal performance (they varied from 75 to 97 Shore A) (Ottosson, 2021). This indicates that even if the base material is the same, additives and/or environmental parameters affect pump performance.

As mentioned earlier, many new materials for seals are now commonplace. Some of these materials are potential candidates for the cup seal. To evaluate a potential material for the cup seal, a wear test should be performed to find the wear rate coefficient for that material. This coefficient can then be used in ANSYS or other FEA software to calculate wear rates. In this way, different materials can be tested and compared for the cup seal. Materials that are promising can then be used to make cup seals for field testing and evaluated against the current nitrile cup seal.

Some additional materials to consider are those being used for similar applications mentioned in handbooks and design guides, such as (Parker, 2018; Hannifin, 2008; Parker, 2011):

- Hydrogenated Nitrile (HNBR)
- Polyurethane (AU, EU)
- Carboxylated Nitrile (XNBR)
- Propylene
- Urethane

By finding the wear rate coefficient for different materials and then using them in FEA software, the materials can be ranked and expensive testing of unsuitable materials can be avoided, leading to lower development cost and faster implementation.

#### 6. Financial analysis

For cup seal manufacturers, to implement a new cup seal design would require new tooling. With an estimated tooling cost of USD 4,000 and a volume of 500,000 cup seals, the added tooling cost per seal would be USD 0.008. With the new seal volume, there would also be an increase in material cost of USD 0.004 per seal (Statista, 2020). With the current price of USD 0.50 per cup seal (Ottosson, 2021), the added

material and manufacturing costs, if considering the same markup as for the original design, the new cup seal price would be USD 0.59.

Servicing the cup seals for an India Mark hand pump in Jinja, Uganda costs approximately USD 56.50 including materials (Ottosson, 2021). If this is done according to the suggested maintenance interval of one year (SKAT, 2008), the monthly cost would be USD 4.71. A 12% increase for the cup seal life could extend the service interval to approximately 13.5 months. If the change in price of the cup seal reflected the added material and manufacturing cost as mentioned earlier, the price would only increase by USD 0.09 per seal but the monthly cost of service would decrease to USD 4.20 (Statista, 2020). A saving of USD 6.05 per year, with over 4 million installations of the India Mark hand pump system (Mudgal, 1997; Carter et al., 2010) corresponds to USD 24.2 million per year saved in servicing cost in the weakest economy system in the world.

#### 7. Conclusions

In this paper, we have looked at the cup seal found in the India Mark II/III hand pump system, a system that is used on a daily basis by approximately 10% of the world's population. We created a performance model for this pump system with the cup seal parameters as input. Then a long-term wear experiment was performed in order to find the wear coefficient for the nitrile cup seal. We used this coefficient in ANSYS and created a wear prediction model. ANSYS DesignXplorer, MATLAB, and JMP were then used to derive a wear optimized design, based on the original seal design, that has the potential to last 12% longer and has the potential to help the economy in the developing world.

We believe that wear can be reduced further when material parameters are added to the optimization (the wear rate for each material of interest must first be found through wear tests). Other, different design improvements to the cup seal were also discussed.

This model can be used to explore further enhancements to the cup seal by changing geometric and material parameters in ANSYS. This allows for an inexpensive way to explore these parameters in order to increase the life of the cup seal. After validating these improvement in the FEA model, physical cup seals can be manufactured and tested in the test rig. By using the model in this paper, engineers can find and computationally test new designs with decreased hardware and resource usage and with an increase in design iterations, leading to more optimized and cost-effective designs.

Going forward, we suggest performing wear tests of materials of interest and after finding their wear coefficients, adding the different material parameters to the optimization analysis. We believe that combining the material parameters to the geometric parameters will further decrease the cup seal wear. Physical prototypes can then be manufactured and tested.

#### Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

#### Acknowledgments

This material is based upon work supported by the National Science Foundation, United States under Grant Nos. 1632740 and 1761505. Any opinions, findings, conclusions or recommendations expressed in this material are those of the author(s) and do not necessarily reflect the views of the National Science Foundation. The authors wish to acknowledge Thomas Naylor and James Mattson for their help in collecting data in Uganda and for helping to build the test rigs. We also would like to and thank the Ira A. Fulton College of Engineering and Technology and the Marriott School of Management at Brigham Young University for funding of this research. Lastly, the authors wish to thank Gabrielle Johnson, Heather Leany, Chris Mabey, and Phillip Stevenson in the Design Exploration Research Group for assisting with the seal leak tests.

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