

# EXPERIENCES WITH A NUMERICAL METHOD OF CALCULATING SLURRY PUMP CASING WEAR

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## ABSTRACT

A numerical method computer program has been developed by the authors that enables calculation of the wear inside a slurry pump casing. The capabilities of the program are presented along with a description of how it models the solids in suspension and calculates where and at what rate they erode the shell, due to different impact and sliding wear mechanisms.

Solutions from the program are used to show how wear varies with the design of the pump, its application in terms of flow and rotational speed and with the particulars such as size, concentration, type, etc., of the slurry. Comparisons of numerical solutions with shells worn out in the field are made to demonstrate how well the method works and where it encounters limitations.

## INTRODUCTION

A numerical method of calculating slurry pump casing wear due to a two phase mixture of particles and liquid has now been used to achieve a better understanding of the

factors influencing wear, to improve new design pump wear performance and predict (and better apply) the wear performance of existing slurry pumps.

Results derived from the use of this tool are presented, and they show calculated effects on the casing wear distribution of various pump casing designs, slurry particulars and application parameters, and comparing these where possible with measured wear performance.

## DESCRIPTION OF THE METHOD

The casing wear calculating method previously described in references [1, 2, 3, 4, 5, 6] is a two dimensional finite element model solving for the particle concentration and velocity distributions inside a pump shell. This solution is then used to predict the wear distribution around the entire volute using an energy dissipation approach which can be empirically correlated to laboratory wear rate experiments. Three types of wear mechanisms are considered: 1) wear due to the normal component of impact velocity, 2) wear due to the tangential component of impact velocity (effect of turbulent motion), and, 3) wear due to the formation of a sliding bed at the wall which supports a given fraction of the material above the wall. The program, being two dimensional, assumes the casing cross-section to be rectangular. This assumption suffices for the majority of slurry pump designs.

Two modes of operation exist. There is the calculation of instantaneous "wear rates" and also a cumulative "wear-out mode." The former gives the initial rate of wear in inches per hour at each node along the casing wall. The latter accounts for the change of shape of the casing as it wears by running the wear rate method iteratively and altering the casing geometry between each iteration in accordance with the calculated wear rate taken over a given time period. The result is a plot of the wear depth distribution through the casing belly vs time.

Particle size and shape, concentration, flow rate, rotational speed, and casing geometry all affect the final solutions as well as internally calculated velocity dampening due to concentration gradients, recirculation flow rate between the cutwater and impeller and supported loads of a sliding bed on the casing wall. The entire method has been implemented in FORTRAN on an Apollo DN660 with terminal input and full plotted output.

While not three dimensional, the program provides a valuable insight into three dimensional effects through calculation of velocity distributions and the recirculating flow past the cutwater. Another output from the solution comes in the form of a casing head loss, value which includes the effect of the slurry and can be used in performance estimation.

## CALCULATED WEAR RESULTS

As described previously, the first version of the program provides initial wear rate information in terms of a given pump geometry, set of operating conditions and slurry particulars. The effect of these is studied in the following five subsections relating to the individual types.

The sixth study uses the program version SHELLWORN to calculate the effect of wear as a shell is eroded and the shape of the shell changes, and the seventh uses the wear out mode to determine shell life for given turndown cases.

Because there are in fact only limited empirical wear rate indices available for use with this program, and for the sake of the comparison (unless otherwise noted), constant wear rate indices for an assumed sand water mixture on a NiHard

4 metal wear surface have been used throughout the examples.

## EFFECT OF PARTICLE SIZE

The calculated effect of particle size is shown in Figure 1 on a study carried out on an 8 in (0.2 m) discharge, 10 in (0.25 m) suction, 32 in (0.81 m) diameter GIW 8x10LSA32 pump utilizing a semi-volute shell and 4 vane ME impeller operating at 450 cpm at the best efficiency point (BEP) flow on a slurry of 24 percent by volume concentration.

Wear rate is very sensitive to particle size. Larger particles transmit much greater energy upon impact with the casing wall. Also, finer particles will often form a sliding bed at the wall which protects it from impact; wear due to sliding friction is increased in this way, but the overall wear is often reduced.

The contour plot on Figure 1 shows wear as having a strong dependence on particle size with calculated increases of as much as 20 times with a particle size change of 100 microns to 1000 microns.

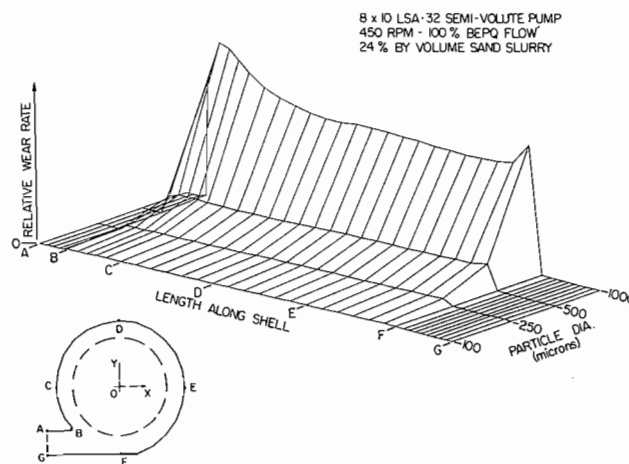


Figure 1. Calculated Wear Rate vs Particle Size.

## EFFECT OF PUMP ROTATIONAL SPEED

A calculated analysis was carried out at four different speeds using the "wear rate" program on an 8 in (0.2 m) discharge, 10 in (0.25 m) suction, 32 in (0.81 m) diameter pump operating on a 12 percent by volume (1.2 SG) slurry.

A slurry of 1000 micron foundry sand was assumed in the calculated analysis and speeds of 450, 600, 750 and 900 cpm considered with the flow equivalent to the (BEP) operating flow.

The results illustrated in Figure 2 show calculated wear as the vertical axis against length along the shell centerline and pump cpm. Taking the wear results at given locations along the shell and plotting these against the different cpm's, we find that the variation is approximately the same along the shell outline and approximates to a power relation with the wear varying at slightly less than the cube of the cpm.

## EFFECT OF SLURRY CONCENTRATION

A calculated initial wear rate analysis was carried out on a 6 in discharge, 8 in (0.2 m) suction, 19.5 in (0.50 m) diameter impeller pump at a number of different slurry concentrations.

The slurry chosen was a 270 micron 2.65 solids specific gravity (SG) foundry sand. The pump was operated with an

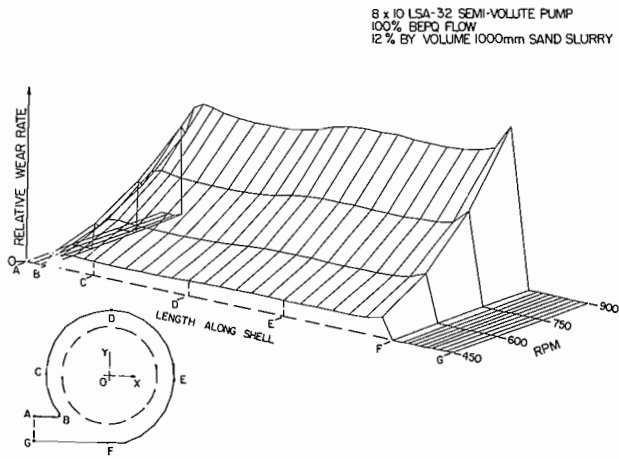


Figure 2. Calculated Wear Rate vs CPM.

impeller peripheral speed equivalent to 6000 surface feet per minute (30.5 m/s) and a flow equal to the BEP flow of the pump.

The results are shown in Figure 3, with the wear as the vertical axis against length along the shell centerline and slurry concentration (from one percent through to 39 percent by volume) as the other two axes.

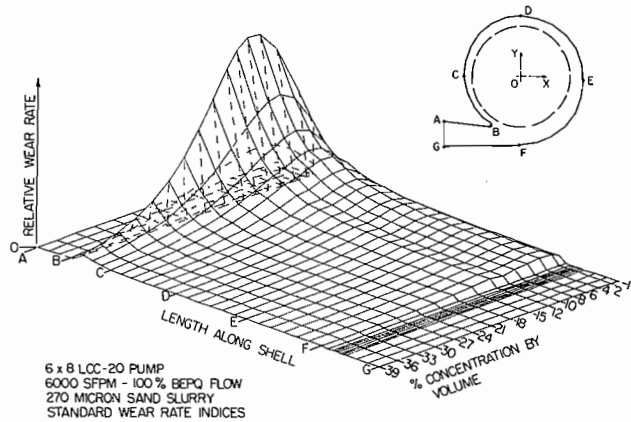


Figure 3. Calculated Wear Rate vs Concentration.

This contour plot shows the highest wear rate in the tongue area. The magnitude of this wear is seen to increase with increasing average concentration as would be expected, but then to decrease at concentrations above 12 percent by volume. This is due to the change in the relative wear contribution of the sliding and impact types of wear as the concentration increases. The effect of the higher concentrations of slurry nearer the wall (for this set of conditions) is to damp the effect of the impact wear enough to reduce the total wear, even though a heavier sliding bed is present.

EFFECT OF FLOW

The calculated effect of flow was obtained by running eleven different analyses at flows between 30 percent and 130 percent of the best efficiency point flow (BEPQ) on an 8 in (0.2 m) discharge, 10 in (0.25 m) suction, 32 in (0.81 m) diameter pump operating at 600 cpm pumping a 300 micron foundry sand of 12 percent by volume (1.2 SG) concentration.

The contour plot shown as Figure 4 reveals a fairly even wear distribution for this particular shell design and set of conditions at flows near 100 percent of BEPQ. At lower flows, the wear along the shell centerline increases by up to 100 percent in the tongue or cutwater accompanied by varying decreases down to only five or ten percent at areas adjacent the branch.

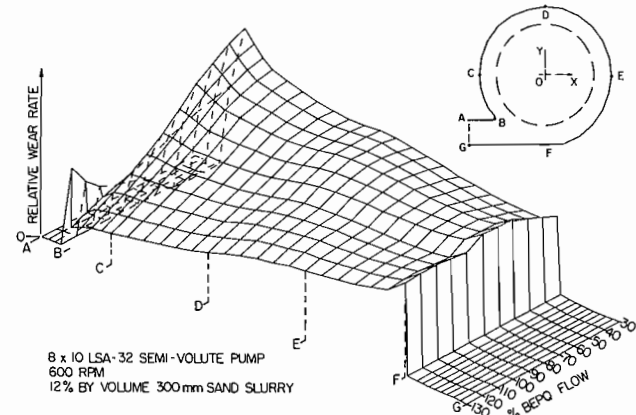


Figure 4. Calculated Wear Rate vs Percent BEPQ.

At flows greater than the BEPQ, wear for the example shown in Figure 4 is approximately constant adjacent the branch area and decreases towards the tongue tip where a localized peak is seen to have developed. The changes in wear distribution seen by varying the BEPQ are largely the result of variations in the amount of flow recirculated within the pump.

EFFECT OF SHELL SHAPE

The calculated effect of a change in pump geometry in the form of a tongue cutback for a 6 in discharge, 8 in (0.2 m) suction, 19.5 in (0.50 m) diameter impeller pump is shown in two different contour plots (Figures 3 and 6).

The first contour plot is the same plot described in the evaluation of concentration earlier, with wear on the vertical axis and length along the shell and concentration on the other two axes.

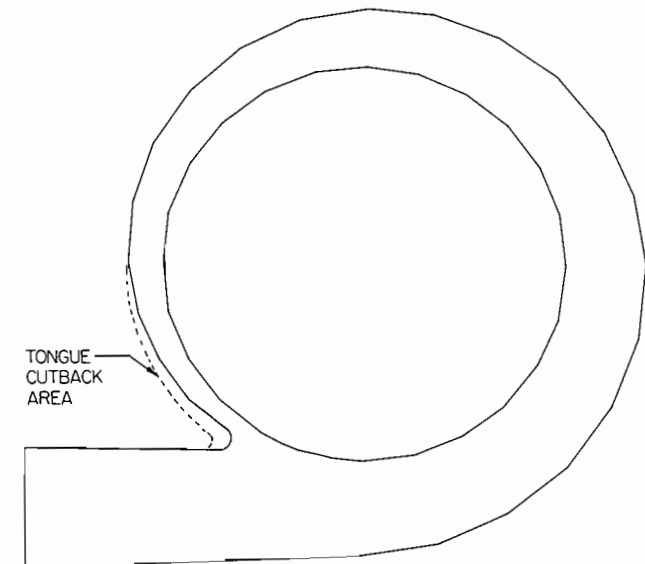


Figure 5. 6 x 8 Pump Shell Outline.

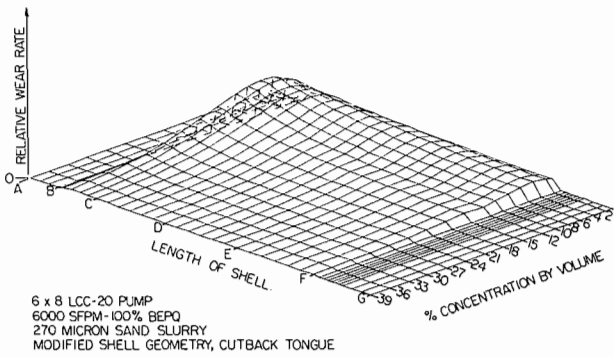


Figure 6. Calculated Wear Rate vs Concentration with Tongue Shape Change.

The second contour plot, Figure 6, using the same scale and set of axes is for a modified shape of shell as shown in Figure 5. It is seen that the wear is reduced overall and that the maximum wear in the tongue area is reduced by more than 100 percent. Identical operating conditions (as reported earlier) were maintained in this comparison.

**EFFECT OF CHANGE OF SHELL SHAPE WITH WEAR**

The “wear out” version of the program previously described provides a unique method of looking at the effect of wear as the shell changes shape through wear.

The shell outline and mesh used for a wear out solution on an 18 in (0.46 m) branch, 44 in (1.12 m) diameter pump are shown in Figure 7.

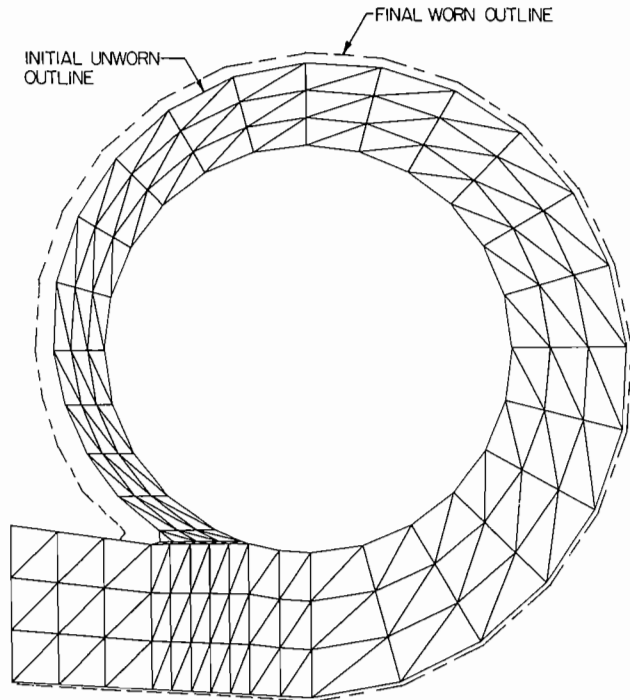


Figure 7. Computational Mesh for Wear Out Mode 6 x 8 Pump.

A wear out plot of wear depth versus wear along the shell centerline is shown in Figure 8, with the time interval between wear rate and shell shape adjustments set at 720 hr.

The results include four iterations and show that at the branch end there is little change in wear rate between the

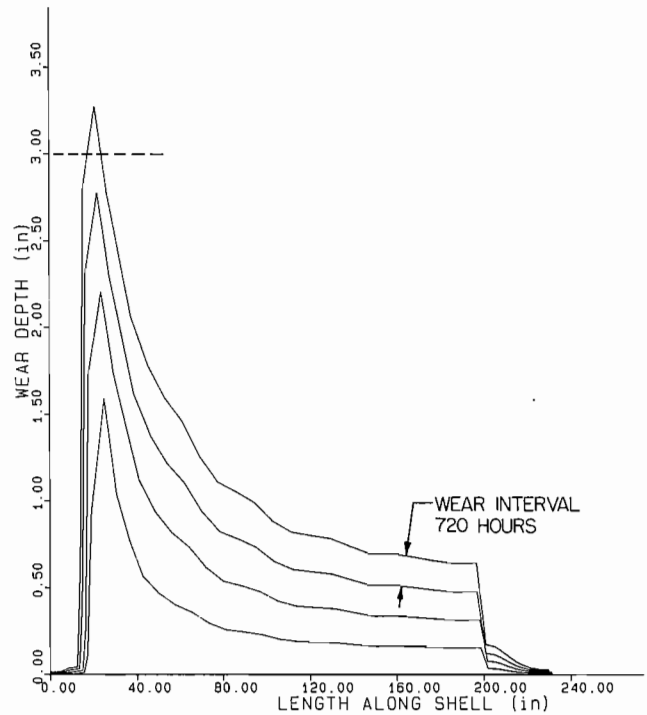


Figure 8. Calculated Wear Depth vs Time 18 x 18 Volute Pump.

first iteration and the last. At the tongue area, however, the wear rate change is much higher and drops off very rapidly as the shell wear is seen to occur.

**EFFECT OF IMPELLER TURNDOWN**

Whether it be the turndown of an existing pump or a different impeller diameter and shell design combination, it is of interest to see how impeller turndown affects wear in terms of given different pump speeds for the same head produced. Using an existing 8 in (0.2 m) discharge, 10 in (0.25 m) suction, 32 in (0.81 m) diameter impeller pump with a full size impeller, and comparing this with the same pump fitted with a 25 in (0.64 m) diameter impeller running faster to produce the same head, the wear out program was used to compare the wear life at two different speeds, at 50 percent, 75 percent and 100 percent of BEPQ flow at two slurry SGs and two sizes of particles.

The results are as shown in Table 1 and shows that in the larger proportion of the cases for the pump chosen the turndown impeller version wore better.

**EFFECT OF DIFFERENT WEAR COEFFICIENTS**

All of the above analyses assumed constant material properties. Rates for the three components of wear shown in each plot are correlated to experimental data through the wear

Table 1. Shell Wear Life in Months 8x10LSA32 Semi-Volute.

	200 micron sand				400 micron sand				
	BEPQ %	1.2 SG		1.3 SG		1.2 SG		1.3 SG	
		Full	T/D	Full	T/D	Full	T/D	Full	T/D
700 RPM (906 T/D)	50%	4.8	7.4	5.7	7.7	0.30	0.51	0.43	0.69
	75%	5.4	7.9	6.1	8.0	0.39	0.64	0.63	1.0
	100%	5.5	8.0	6.3	8.1	0.49	0.71	0.80	1.0
350 RPM (453 T/D)	50%	20.3	18.5	16.8	13.2	2.0	2.7	2.4	2.6
	75%	20.6	18.5	17.0	13.3	2.4	3.0	3.0	3.0
	100%	20.9	18.6	17.1	13.0	2.9	3.2	3.5	3.0

coefficients. These coefficients vary from material to material. Hard brittle materials are very good for wear modes which involve scratching or cutting at small impact angles. Such wear modes predominate in pumping applications. Wear resistance at high impact angles for hard materials is also good, but depends more on strength than ductility. Ductile materials show their best performance at high impact angles due to their ability to absorb energy through deformation, but usually wear faster under all conditions due to the ease with which deformed material, being raised above the surrounding casing wall, is scraped away [7].

Inasmuch as the sliding and impact wear coefficients used in the numerically based computer program are based on a limited number of tests on a restricted combination of resisting alloys and eroding slurries, it was thought worthwhile to numerically compare the effect of varying some of these.

In a previously presented contour plot (Figure 3), the wear of a 6 × 8 pump was shown against change of concentration. The same analysis carried out with the sliding wear coefficient increased 100 times is shown in Figure 9 for the same pump. This shows a small increase in wear generally but with the greatest increase at the higher concentrations towards the branch area of the shell where the formation of a sliding bed could be expected to be most significant.

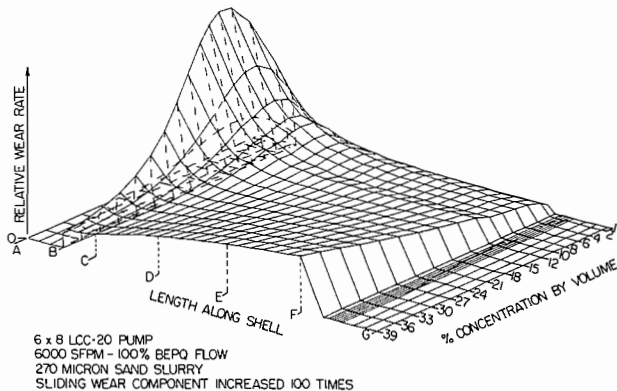


Figure 9. Calculated Wear Rate vs Concentration with Increased Sliding Wear Component.

### LIMITATIONS OF THE METHOD

A standard wear rate solution of a typical shell using the standard mesh comprises approximately 200 elements with 140 nodes. A finer mesh may result in less solution stability under the high concentration gradients seen at the casing wall in an operating centrifugal pump. The standard mesh spreads this gradient over a larger distance and although this may result in less accuracy, it gives more stable and consistent results.

Practical limitations on particle size are affected by concentration, velocities, particles SG and casing geometry. Roughly speaking, under average conditions a size of 1000 micron is the practical limit. Higher concentration, lower velocity and particle size and larger pump size all contribute to stretching this limit. Conversely, unfavorable conditions may reduce this limit to as low as 500 micron.

Although the 2D wear program is not suited to model 3D gouging wear, the 2D flow velocities which it predicts can indicate when gouging is likely. In general, higher velocities between the tongue and impeller at the cutwater will result in increased chance for gouging wear.

Substantial reductions in tongue velocities can be achieved with modest losses in efficiency. Exactly how much the

gouging wear can be reduced in any given application is sometimes difficult to determine and requires reliance on field experience. Erosive wear is, however, often found in scientific literature to be proportional to the third power of the velocity, so that even small changes in velocity can result in large differences in wear rate.

The program assumes a constant D50 particle size neglecting the effect of varying size distributions. In actual practice, a modest percentage of fines may greatly dampen the abrasivity of the entire slurry or conversely, a small percentage of very coarse material may accelerate wear in one location under the action of gravity. Cases are known where pump orientation has affected the wear distribution dramatically. For example, on some ball mill discharge pumps, maximum wear occurs at the bottom of the pump no matter how the shell is oriented due to the action of gravity on ball ships passed through the pump.

All limitations must be kept in mind and weighed for intelligent use of the method.

### WEAR COMPARISON OF A VOLUTE TYPE SHELL

An 18 in (0.46 m) discharge, 18 in (0.46 m) suction 44 in (1.12 m) diameter slurry pump was installed at IMC in Florida, pumping phosphate matrix of 1.28 SG at 12,000 gpm (760 l/sec) while running at 585 cpm on March 3, 1985, and ran until July 2, 1985 when it was removed from service.

Life for the components based on a 60 percent operating factor was 96 days. The wear experienced by the shell is as shown in Figure 10.

Generally, the wear was reasonably even all over considering the volute type of shell. Some gouging was evident in the

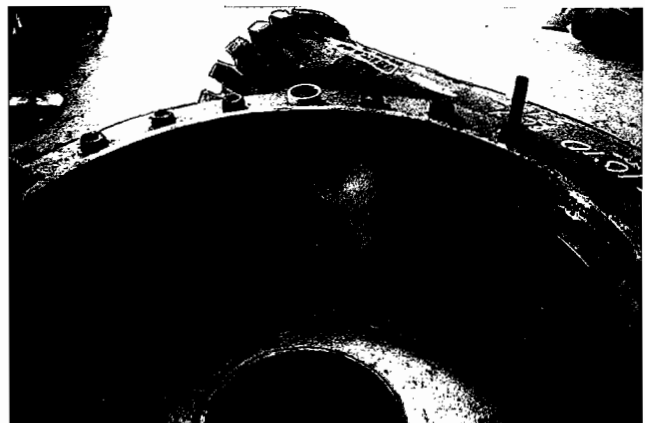
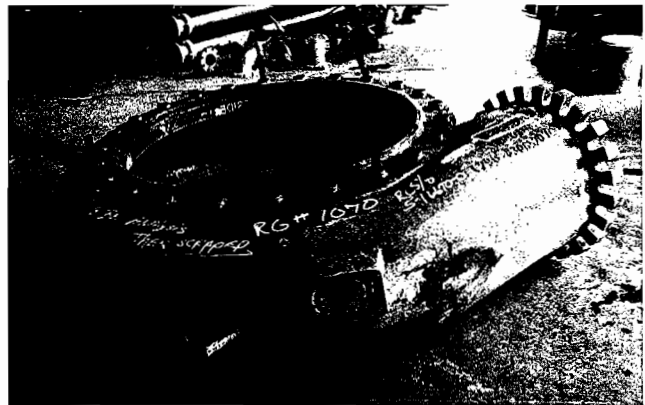


Figure 10. Photograph of Worn 18 × 18 Volute Type Shell (1846D).

sidewall adjacent the tongue but had not yet holed through. Measurements show that the tip of the tongue had worn back 3 in (0.076 m) from the original drawing dimensions and the belly thickness was down to 1/2 in (0.13 mm) in the area just after the tongue.

The shell actually cracked in service and was removed. Some sort of pressure spike presumably caused the breakage. Regardless, the thickness to which the shell wore is representative of maximum casing life.

Previous to the failure of this shell, a "wear-out" mode calculation was performed to estimate its life. Wear out analysis was first performed on a different shell in the same service for which the wear life was known and a particle size was found which best modelled the phosphate slurry abrasivity. This particle was then applied to the analysis of the subject shell (Figure 8) and the result was an estimated life of about three months. This compared well with the actual life.

A plot comparing the computed and actual wear distributions is shown in Figure 11.

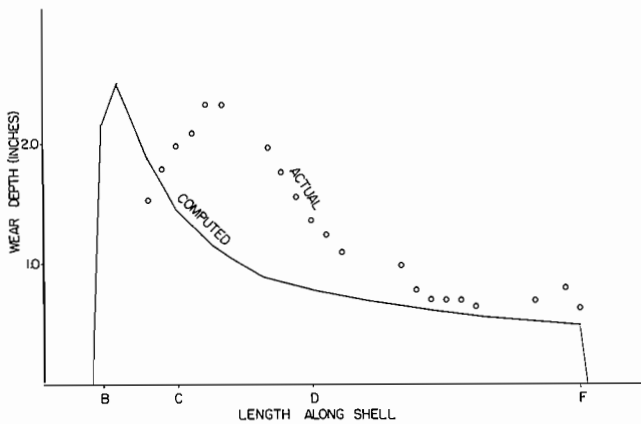


Figure 11. Actual vs Calculated Wear 44 In Volute Pump.

**WEAR COMPARISON OF A SEMI VOLUTE SHELL**

An 18 in (0.46 m) discharge, 20 in (0.508 m) suction, 48 in (1.219 m) diameter impeller pump was installed on phosphate tailings service at W. R. Grace in Florida. Operating at 18,000 gpm (1136 l/sec) and 505 cpm with a 1.3 slurry SG and particle D50 of 420 micron, this pump ran intermittently for nine months. The shell then failed under pressure by a circumferential crack in the casing belly. This crack split the casing in two and allowed direct thickness measurements to

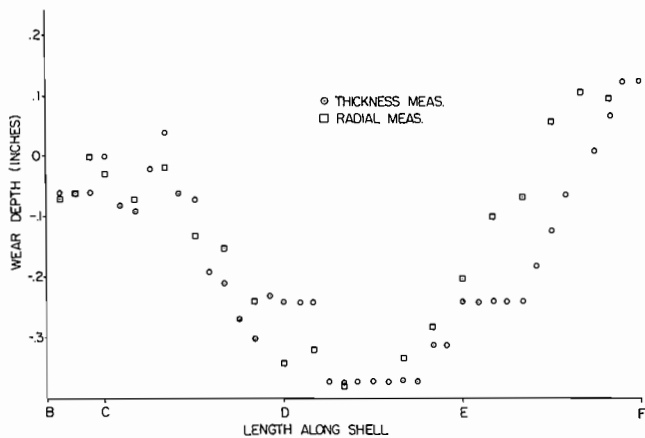


Figure 12. Measured Wear of 44 In Semi-volute Pump.

be made over the entire volute. Wear was very light and even with no gouging.

Direct thickness measurements were made which showed about 1/2 in (13 mm) difference between the most and least worn areas. These thickness measurements showed most of the casing to be thicker than the nominal 3-1/2 in (90 mm), making determination of actual wear difficult since the initial thickness is unknown. Given that the initial thickness, whatever its value, was constant, these measurements can still have relative value in determining the distribution of wear around the volute. As a check on the direct thickness measurements, the hydraulics were measured from the pump center and compared to the nominal to obtain a second set of wear depths. Agreement between both methods was good (see Figure 12).

The computed wear program solution for this case is shown in Figure 13. Although no comment can be made about the predicted time of wear due to uncertainty in the initial thickness of the casing and in particle size, the distribution of wear predicted by the program is very similar to that measured.

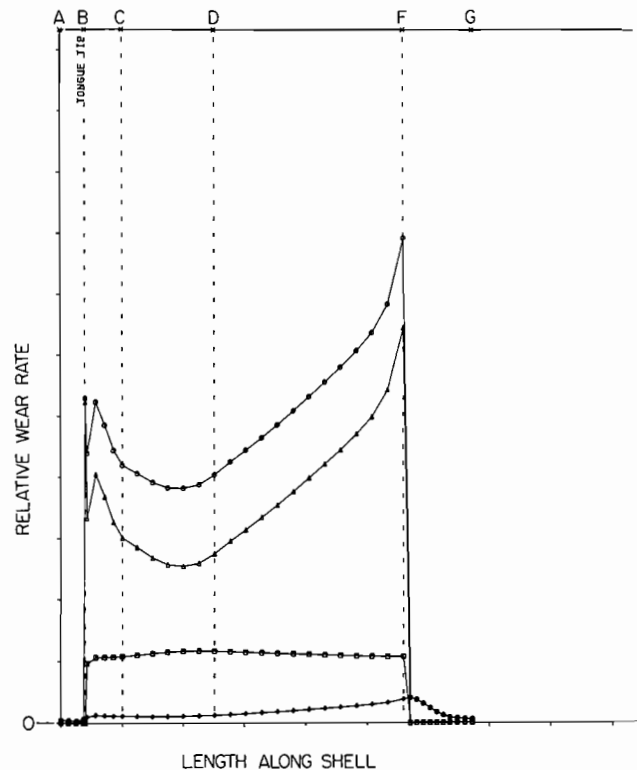


Figure 13. Calculated Wear of 44 In Semi-volute Pump.

**WEAR COMPARISON OF ANNULAR TYPE SHELL**

An 18 in (0.46 m) branch, 39 in (0.99 m) diameter slurry pump operating at IMC in Florida pumping phosphate matrix was returned to the lab for measurement of wear after being in service for 300 days.

The subject 16 in (0.406 m) wide (1590D drg. no.) shell was measured for wear at ten degree intervals along the casing centerline and the results plotted as shown in Figure 14.

A "wear out" analysis solution was also run for the pump at the reported conditions of 590 cpm, 12000 USGPM (760 l/sec) flow, 1.3 SG, and with a properly sized foundry sand found to best simulate the abrasivity of the phosphate matrix. This solution is plotted in Figure 15.

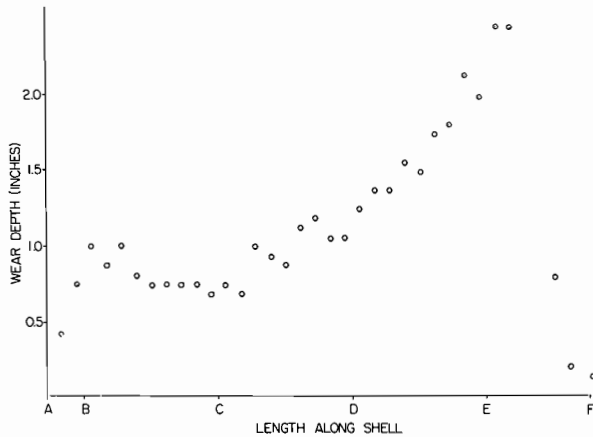


Figure 14. Measured Wear of 39 In Annular Pump.

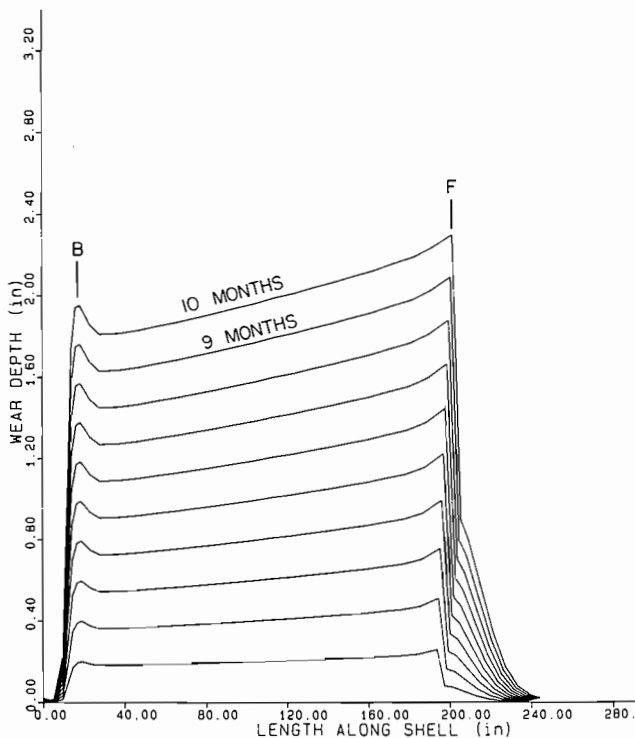


Figure 15. Calculated Wear of 39 In Annular Pump.

When considering the various uncertainties in this analysis such as casting irregularities, variations in duty conditions, and the approximate nature of the finite element program solution; the actual and calculated results are encouragingly similar. Note that the program not only predicted the general shape of the wear curve, but also the small peaks in wear just past the tongue and before the discharge (points B and F in the figures).

**WEAR COMPARISON OF A LARGE DREDGE PUMP SHELL**

A 30 in (0.762 m) diameter branch, 80 in (2.03 m) diameter impeller dredge pump reported as operating at 360 cpm passing a flow of 45,000 gpm (2840 l/sec) of slurry of 400 micron D50 size particle at a concentration of 1.1 SG was numerically modelled to determine the calculated wear rate.

Having received measurements of actual wear on the subject dredge pump, these were then compared with the original analysis.

The results are seen plotted on a relative scale in Figure 16. The analysis predicted both the area of maximum wear and the trend of decreasing wear around the volute towards the discharge.

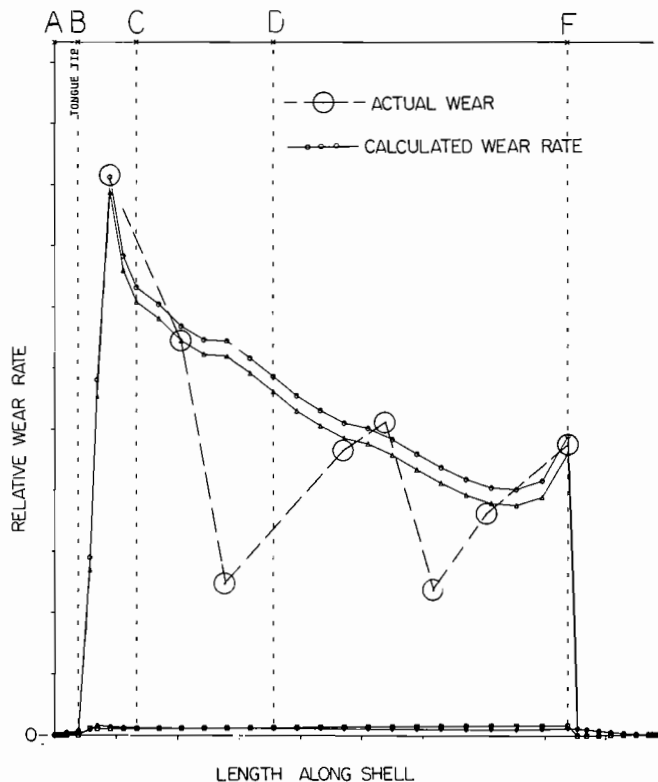


Figure 16. Actual vs Calculated Wear 80 In Dredge Pump.

The actual wear pattern does, however, show large fluctuations not predicted by the analysis. Recall that the analysis represents a prediction of wear based on the original design hydraulics. Under field conditions wear itself affects the hydraulics and can be expected to alter wear rates. Turbulence caused by gouging wear can lead to the uneven wear as seen in this case. Also, the effects of recirculatory "secondary flow" (in a cross section perpendicular to present analysis) may be important.

**CONCLUSIONS**

While it is admitted that some of the individual studies are of a limited scope, the calculated wear for the different slurries, pump designs and application variables provide an insight not previously possible.

Though in most part restricted to individual cases, the field wear measurements collected to this stage show good agreement with the calculated results and lend credence to the general method. A pleasing feature of the results is that while three dimensional effects are obviously present, and in some cases have caused significant local wear, they have not greatly distorted the results or the usefulness of the method.

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