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Measuring and Use of Wear Properties for Predicting Life of Bulk Materials Handling Equipment

Jerry R. Johanson and T. Anthony Royal, USA

Summary

Abrasive wear of equipment is an important problem when storing and handling bulk granular solids. A long-needed wear tester has been developed by Jenike & Johanson, Inc. to measure the wear rate due to solids sliding on a wearing surface as would occur during bulk solids flow. A dimensionless wear ratio is defined and measured as a function of the solids normal stress applied to the wearing surface for a particular wear material and bulk solid of a certain particle size distribution, moisture content and temperature. Incorporating the principles of bulk solids flow, a method is presented to predict in absolute quantitative terms the wear which will occur at any location within a bin, hopper, feeder, chute, or other bulk solids handling equipment. In the design or modification of such equipment, this serves as an invaluable aid to the engineer when faced with questions of placement, quantity, type, and service life of wear resistant materials.

1. Introduction

The ever increasing economic impetus for development of today's new pilot-scale processes invariably leads to large continuous processing plants rather than less economical batch plants. Since most of these processes involve handling bulk solids, the problem of wear in bins, hoppers, chutes, conveying screws and other bulk solids handling equipment becomes increasingly important for the following reasons:

- Significant cost savings can result from the proper choice of wear material because of decreased maintenance requirements and the avoidance of using expensive wear-resistant materials when not needed.
- A continuous process dictates that the reliability of the entire plant be dependent on almost every individual component station. Unexpected downtime of one component from wear can force the whole plant to cease operation — a very expensive incident.
- The magnitude of cumulative wear increases with the continuous use of equipment.
- Wear skyrockets as the size of bulk solids handling equipment increases from pilot scale to commercial scale. Quite often the expected wear for a commercial plant is extrapolated from pilot plant operating experience with disastrous results. The wear rate of a bulk solid sliding against a wear material increases with greater bulk solids pressure on the wear surface, and since the solids pressure rises with both increased height and diameter of equipment, the wear rate associated with commercial equipment can be many times that of pilot scale. The rate of wear also increases with the larger solids velocities generally associated with commercial scale equipment.

In storing and handling abrasive bulk granular solids, wear is a problem which cannot be overlooked. It is important that wearing surfaces be hard and thick enough, and, if necessary, replaceable, so that maintenance is minimized or even avoided. Besides establishing maintenance schedules, the ability to predict wear allows various economic alternatives to be evaluated, e.g. hopper liners or wall coatings necessary to promote mass flow. Prediction of the wear rate of a given wear material subjected to the sliding of a given bulk granular solid is necessary for the selection and placement of appropriate wear resistant material in solids handling equipment. This can be accomplished as a two-step process. The first step is both empirical and theoretical and determines the bulk solids velocity and pressure at the wear surface by application of solids flow theory to the equipment geometry and experimentally measured flow properties of the bulk solid. The second step is empirical and establishes the wear rate as a function of bulk solids pressure at the surface of a given wear material by measurement of wear in a suitable wear testing apparatus. Since the solids pressure and velocity distribution can be computed and wear at various solids pressures measured, the wear rate at any location in the equipment can be predicted.

Dr. J. R. Johanson, President and T. A. Royal, Research Engineer, Jenike & Johanson, Inc., 2 Executive Park Drive, North Billerica, MA 01862, USA

2. Previous Work

In the past, wear testing has focused on two-body abrasive wear that occurs between a wearing surface and a fixed matrix of abrasive particles (e.g. sandpaper) [1—4]. This is generally associated with material removal processes such as sanding and grinding. The majority of two-body abrasive wear tests involve moving a weighted pin across abrasive paper on a spiral or helical path and measuring the weight loss of the pin.

Although voluminous literature exists describing investigations of two-body abrasive wear, very little research has been directed toward three-body abrasive wear, defined as wear due to loose abrasive particles sliding on a wearing surface. This is surprising, since most abrasive wear problems occurring in industry and agriculture are generally three-body abrasive wear. Three-body abrasive wear has been further classified as closed or open by Misra and Finnie [5]. Closed three-body abrasive wear pertains to wear due to loose abrasive particles being trapped between two closely spaced rolling or sliding wear surfaces, and is important in premature bearing failure caused by contaminant particles. It has been found by Rabinowicz [6] that three-body abrasive wear is approximately a magnitude less severe than two-body abrasive wear. This is believed to be the consequence of the added freedom of particles to roll along the wearing surface in three-body wear.

When the wearing surfaces are far apart or only one wearing surface exists, the wear is classified as open three-body abrasive wear. Wear from the flow of bulk solids falls into this category. To the best of our knowledge, the device shown in Fig. 1, which was developed by Misra and Finnie at the University of California at Berkeley [5, 7], represents the best

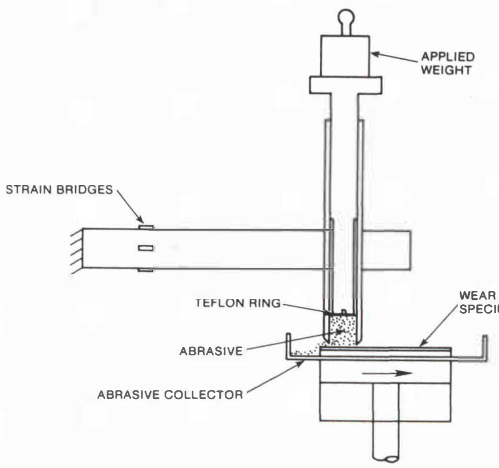


Fig. 1: Low-stress open three-body abrasive wear tester of Misra and Finnie

open three-body abrasive wear tester of others. However, it has limitations which restrict its use for testing wear due to bulk solids flow:

— The constant gap between the wear specimen and tube limits the size of the abrasive particles to only several hundred microns in diameter.
— The flow rate of abrasive particles can only be controlled by adjusting the gap which affects the pressure distribution applied to the wearing specimen.
— A non-uniform stress distribution acts on the wearing surface because particles are caught in the gap between the wearing surface and the load application tube causing high stress concentrations. This masks the measurement of wear rate as a function of applied pressure.
— The applied load is not constant, but is dependent on the level of abrasive in the tube.

Other methods of abrasive wear testing [8—11] such as use of rotating discs in sand slurries, use of ball mills, jaw crushers, lapping machines, and rubber wheel abrasive testers do not simulate the proper conditions of wear resulting from bulk solids flow and provide only a crude relative comparison of wear to a reference wear material [12, 13]. Consequently, the prediction of absolute quantitative wear rates is impossible when employing these methods.

3. Wear Tester Apparatus

A wear tester is suitable if it can simulate the conditions of wear consequential to the flow of a bulk solid. The requirements of a tester are therefore:

— A fresh supply of bulk solid must continually be presented to the wear surface as occurs in reality. The ever new bulk solid particles at the wear surface, being irregular in shape, cause stress concentrations above the elastic limit at points of contact with the wear surface and is the mechanism of abrasive wear. Failure to refresh the abrasive bulk solid at the wear surface leads to erroneous results — the particles themselves can wear and become rounded reducing stress concentrations in the wear surface material and hence, the consequential abrasive wear.
— The solids pressure exerted on the wear sample surface must be continuously controlled and measured. Because of the desire to express wear as a function of solids pressure, the pressure distribution should be as uniform as possible.
— The relative velocity between the bulk solid and the wear material must be controlled and measured.
— The shear stress exerted on the wear material should be continuously measured so that any changes in the coefficient of friction between the bulk solid and the wear surface can be observed as it is worn.

The wear tester (patent pending) developed by the authors and shown schematically in Fig. 2 consists of a hopper to contain the sample of bulk solid, a screw feeder to withdraw the solids from the hopper and convey it to the wear surface,

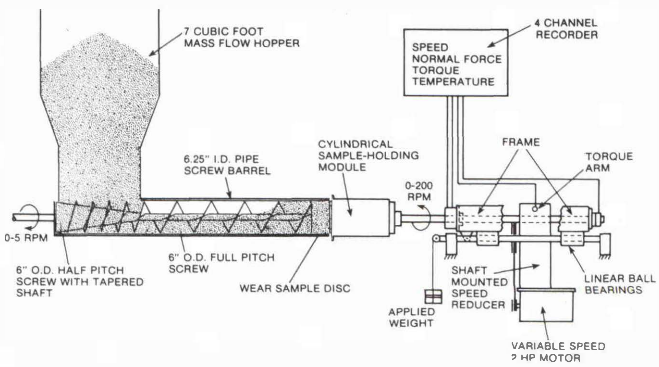


Fig. 2: Schematic of Wear Tester

and a sample-holding module and supporting frame which applies and measures torque, thrust, and number of revolutions. The actual wear sample is a circular disc rotated independently of the screw, so that the relative velocity between the bulk solid and the wear sample can be controlled and maximized, thereby minimizing the time required for a test. Wear is measured as the weight loss of the sample disc for a certain number of revolutions of the disc.

The wear sample disc is mounted on the cylindrical assembly as shown in Fig. 3. The sample disc is of smaller diameter than the inside of the screw barrel so that the increased solids pressure at the barrel edge is not applied to the wear sample, but to the hardened replaceable faceplate ring. The sample disc is machined so that there is a small gap between the disc and the faceplate ring. It is important to keep this gap small to eliminate any increased wear in the vicinity of the gap. This gap is then filled with a flexible abrasion resistant silicone sealant to prevent solid particles from wedging into the gap and affecting the pressure reading from the load cell. The sealant must be flexible for the load cell to accurately measure only the solids pressure on the sample disc.

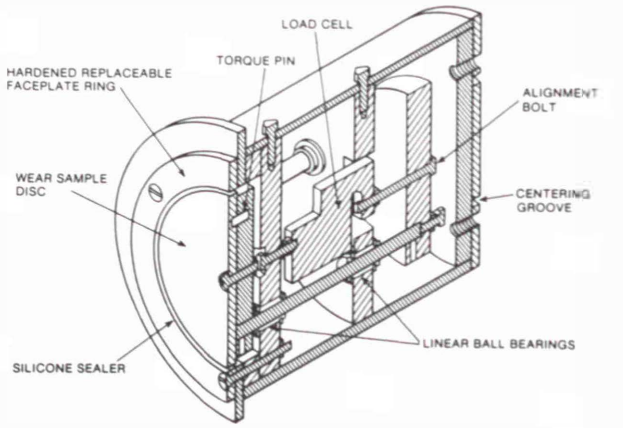


Fig. 3: Cylindrical sample-holding module

The cylindrical sample-holding module is connected to a shaft mounted speed reducer and variable speed motor which are balanced and held by a load cell which continuously measures the total torque applied to the cylindrical sample-holding module. Furthermore, the speed reducer is attached to a frame on linear ball bearings which can freely move in the direction of the screw barrel length. A weight and pulley system is used to push the frame and consequently the cylindrical sample-holding module against the end of the screw barrel. This guarantees a constant solids pressure against the wear surface. An adjustable stop is used to set a minimum gap between the end of the faceplate and the barrel to prevent metal to metal contact during startup. The free horizontal movement of the frame enables the gap to open between the screw barrel and the wear sample faceplate ring, permitting occasional large sized particles to pass. This is a very important feature as it allows a bulk material of virtually any particle size distribution to be tested.

A three inch gap between the end of the screw and the end of the screw barrel smooths out fluctuations in the solids contact pressure due to the end of the screw flight. The speed of rotation of the wear sample disc is adjusted so that its temperature as continuously measured by the thermocouple does not become excessive.

4. Dimensionless Wear Ratio

It has been observed that the total amount of wear (measured as weight loss) is a linear function of the total distance travelled by the bulk solid relative to the wear surface. Hence, the wear rate is a linear function of the magnitude of the velocity of the bulk solid except for very high velocities when inertial forces of individual particles or heat generation becomes significant. Therefore, it is convenient and proper to define a wear ratio *WR* as the dimensionless ratio of unit thickness of material lost per unit distance travel of the bulk solid sliding relative to the wear surface. Multiplying *WR* by the velocity of the bulk solid at the wear surface thus determines the wear rate in loss of thickness per unit time.

The object of the wear tests is to measure the wear ratio *WR* as a function of the applied solids pressure at the wear surface. From the raw data of the wear tests — namely the total number of revolutions *REV* of the wear sample disc and the measured weight loss ΔW of the sample disc — it is possible to calculate the wear ratio for the applied solids pressure level. It is observed that the wear profile of the sample disc is a linear function of the radial distance from the center of the disc with no wear at the center and maximum thickness worn away at the circumference. This is shown in Fig. 4 for a sample of aluminum after several thousand revolutions at 440 lbs/ft² using an abrasive ore passing a – 6 mesh U.S. Standard sieve. The observed linear profile confirms a uniform solids pressure on the wear disc, as well as the statement above regarding the linear relationship between wear rate and solids velocity. It is not practical during a typical test to wear away enough material so as to be able to measure the change in thickness as was done for Fig. 4.

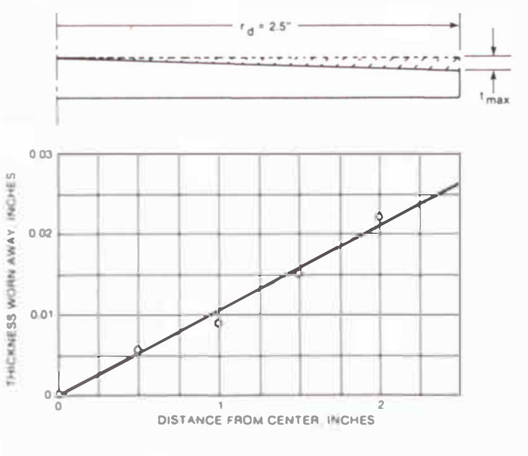


Fig. 4: Wear profile of aluminum plate after wear test

Instead, the thickness change profile is inferred from the weight loss ΔW and the weight density Γ of the wear material by considering the conical shaped volume that remains. The wear ratio is then computed as the thickness lost t_{max} at the outside edge of the sample disc of radius r_d to the total distance travelled in *REV* total revolutions:

$$WR = t_{max} \frac{1}{2\pi r_d REV} = \frac{3\Delta W}{2\pi r_d^2 \Gamma} \frac{1}{2\pi r_d REV}$$

$$= \frac{3\Delta W}{4\pi^2 \Gamma r_d^3 REV}$$

5. Variables Affecting Wear Ratio

Abrasive wear due to bulk solids flow or low stress open three-body abrasive wear is a complicated phenomenon with many mechanisms at work. Several variables can affect the measured wear ratio, a few of which are given below:

— Pressure level.
Fig. 5 shows some typical wear test data. Note that the wear ratio *WR* is essentially linear with solids contact pressure except in the higher pressure range where wear increases at an accelerated rate with pressure. The non-linear behavior at higher pressures is believed by the authors to be caused by the decreased tendency of bulk particles to roll at the wear surface as interparticle friction and interlocking increase with applied pressure. This decreased propensity for rolling increases the amount of material loss as the stationary particles abrade the wearing surface.

— Difference in hardness between bulk solid particles and wear surface material.
Abrasive wear only occurs when the abrasive particle is harder than the wear surface material. The wear ratio decreases with increasing hardness of the wearing surface as illustrated in Fig. 5 for an abrasive ore. Many steels work-harden at the wear surface as wear progresses providing an additional resistance to wear.

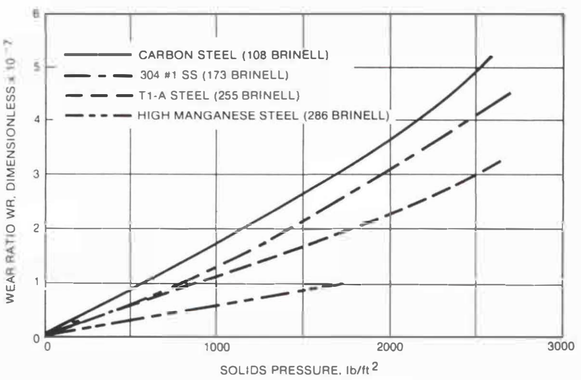


Fig. 5: Measured wear ratios of abrasive ore on various steels

— Particle size distribution/packing density.
It has been found that the particle size distribution significantly affects wear. For example, Fig. 6 gives results of wear tests of an abrasive ore on carbon steel for three different particle size distributions: a − 1/8 inch particle size fraction, a + 1/8 inch fraction, and a composite mixture of the two fractions. The tests on the + 1/8 inch fraction yielded the least amount of wear, while the composite mixture exhibited the most wear. The + 1/8 inch fraction exhibits less wear than the − 1/8 inch fraction — the exact opposite size effect that occurs in two-body abrasive wear [14, 15]. Several plausible explanations for these phenomena exist: First, for the three-body abrasive wear of the small particles, clogging, of the voids between abrasive particles is not as significant as with two-body abrasive wear because the particles are in motion relative to one another and are continually being replenished at the surface. Second, and most important is the decreased rolling of smaller particles compared to larger particles because of increased cohesiveness of the smaller fraction. Third, is the infiltration of smaller particles in the voids between larger particles at the surface, preventing the larger particles from rolling. Consistent with this reasoning, it has been found that the particle size distribution giving the highest packing density usually exhibits the most wear.

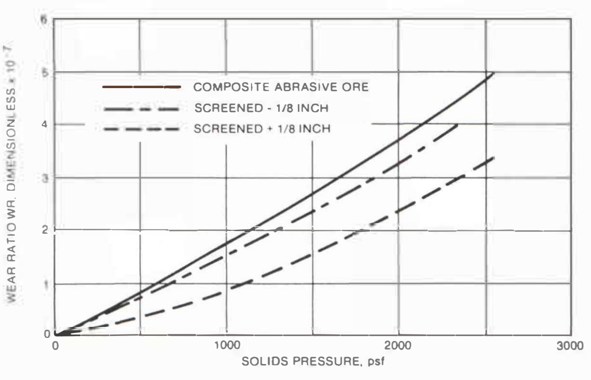


Fig. 6: Measured wear ratios for different particle size distribution of abrasive on carbon steel

— Surface moisture.
Moisture on the bulk particle surfaces generally increases the wear because of the increased cohesion of the bulk solid and hence, less propensity for particles to roll. Additionally, the moisture may act as a lubricant for the cutting of the wear surface by the abrasive particles [6, 16].

— Particle shape.
Rounded particles cause less wear than angular particles for several reasons. The stress concentrations at the wear surface are considerably less for rounded particles because of the increased area of contact. The rounded particles are more free to roll as particle interlocking does not take place. The rake angle for angular particles is more optimum for increased cutting of the wear surface.

— Temperature.
Temperature influences the hardness of the bulk solid and the wear surface and consequently affects the wear.

6. Use of the Wear Results

The remainder of this paper deals with the use of wear test data to quantitatively predict wear rates. There are two quantities to be calculated in any given problem: first, the actual solids velocity at the wearing surface in terms of the average solids flow rate, geometry and dynamics of the equipment under investigation; second, the solids contact pressure at the wear surface.

The following example for wear of mass flow hopper walls under certain specific conditions illustrates some of the factors and complications that accompany these calculations.

6.1 Determining Solids Velocity at Hopper Walls

In order to compute the wear rate in mass flow hoppers, the solids velocity at the wall must be calculated. In general, the wall velocity is slower than the average velocity within a converging flow channel and the maximum velocity occurs in the center of the channel. The average velocity at any cross-section of a hopper can trivially be computed as the bulk solids weight flow rate divided by both the solids bulk density and cross-sectional area. It is possible to relate the solids velocity at the wall to the average solids velocity by assumption of a radial velocity field [17, 18] in conjunction with the following formulae:

Cone:

$$\frac{V}{V_a} = \frac{\sin^2\theta'}{2\cos\theta' \left\{ (V_o/V) \ln \sec\theta' \quad \frac{[(V_o/V) - 1]}{\theta'} \int_0^{\theta'} \epsilon \tan\epsilon \, d\epsilon \right\}}$$

Wedge:

$$\frac{V}{V_a} = \frac{2\sin\theta'}{\theta' \, [(V_o/V) + 1]}$$

where:

V = solids velocity at the wall in the direction parallel to the wall

V_a = average solids velocity

V/V_o = ratio of velocity at the wall to the velocity at the center of the hopper channel for a radial velocity field being a function of wall hopper angle θ', wall friction angle ϕ', and effective angle of internal friction δ. Plots taken from Jenike's work [18] for conical channels and plane flow channels for $\delta = 50°$ are given in Fig. 7.

θ' = wall hopper half angle as measured from the vertical

6.2 Determining Solids Pressures at Hopper Walls

In general the pressure on the hopper walls during mass flow is of the form shown in Fig. 8. In the lower region the pressure is essentially a linear function of the width or diameter of the hopper or the distance from the projected apex. At the transition between the hopper and the vertical wall, a significant overpressure occurs. The magnitude of this overpressure is a function of the height of bulk solid in the cylinder. The method of calculation of these pressures is outlined in many references [20—23]. For the sake of comparison consider the pressures on three hoppers with the same top width or diameter, outlet dimension and solids height above the top of the hopper as depicted in Fig. 8 through 10.

6.3 Prediction of Wear Patterns at Hopper Walls

For our example let us assume that the wear ratio is almost linear with applied solids pressure, having a slope of 1.0×10^{-10} per ft²/lb. Let us also assume a solids bulk density $\gamma = 100$ lbs/ft³, a wall friction angle $\phi' = 15$ degrees (rather low), an effective angle of internal friction $\delta = 50$ degrees, and a weight flow rate $Q = 140$ lbs/ft³. Although Figs. 8—10 are based on these detailed material properties, the following discussion has been generalized.

Since the wall solids velocity is proportional to the average solids velocity, the solids velocity is inversely proportional to the cross-sectional area of the hopper. Using this information in conjunction with the pressure in bins, the likely wear patterns can be deduced for the various hopper configurations.

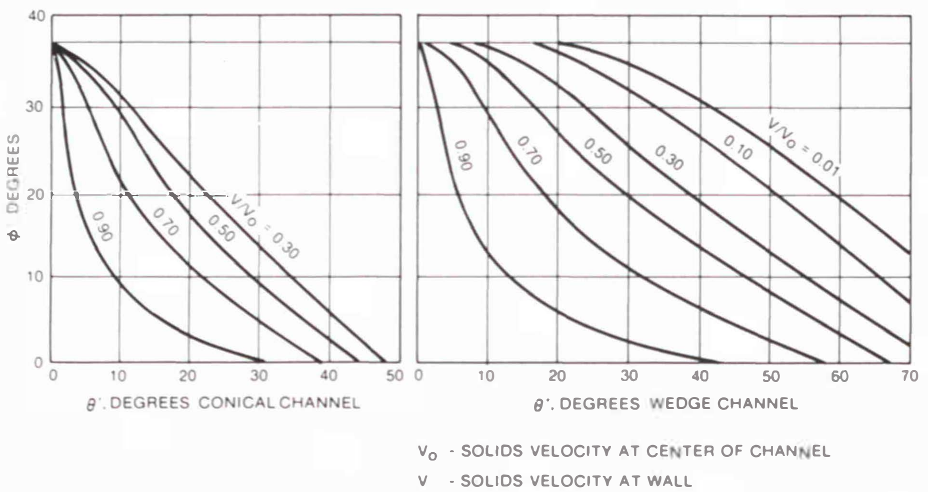
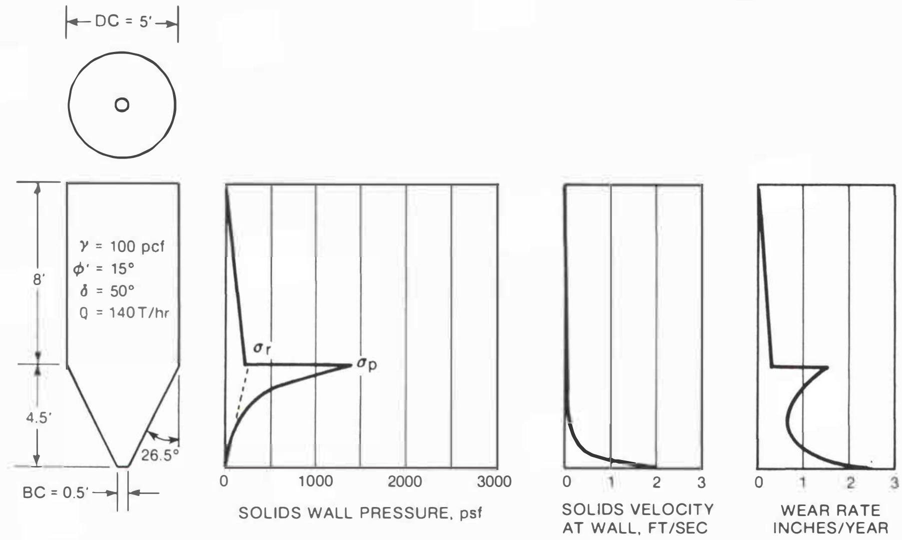


Fig. 7: Radial velocity field solution for $\delta = 50°$

From these formulae and plots it can be seen that the ratio of velocity at the wall to the average velocity becomes smaller as the wall becomes flatter or the wall friction increases. If the walls are vertical, then for many situations the velocity profile is uniform (sometimes referred to as plug flow). However, the velocity profile in a vertical section is influenced by the hopper or feeder that is below it. For instance in a bin such as given in Figs. 8 or 9, the velocity profile in the vertical section is uniform provided the level of material in the vertical section is above a certain minimum level as given by Johanson [19].

— *Conical Hopper*, Fig. 8.
The solids pressures in the lower part of the hopper are given by a radial stress field [24] and are proportional to the hopper diameter and approach zero at the projected apex of the cone. At the transition between the cone and cylinder, the overpressure [22, 23] imposed by the cylinder causes a large solids pressure peak. This peak can be up to five times the value of the extrapolated radial stress at the transition, as indicated. Because of the solids pressure peak, there is high wear at the transition. Also, due to high solids velocity at the outlet of the cone there is

Fig. 8: Conical hopper wear pattern



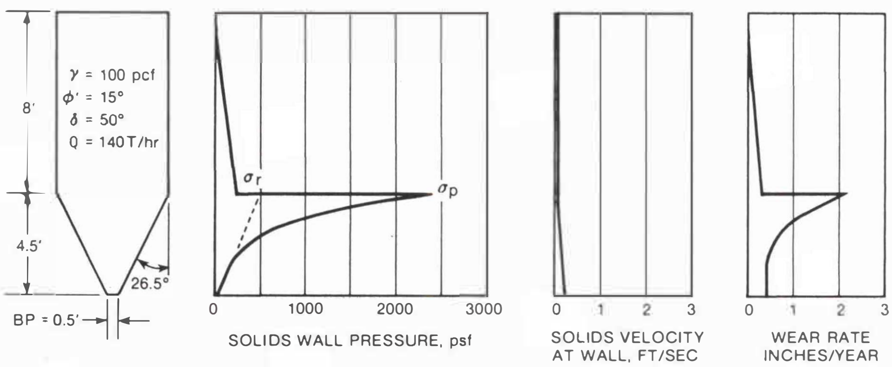
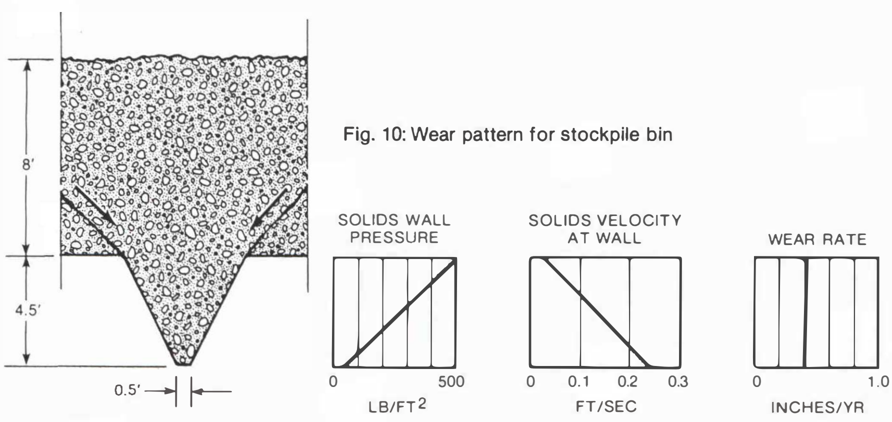


Fig. 9: Wedge hopper wear pattern

Fig. 10: Wear pattern for stockpile bin



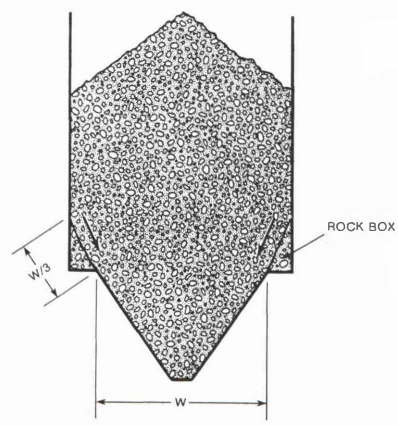


Fig. 11: Use of rock box

high wear at the bottom as well. Which of these two regions is subject to the greatest wear depends on the ratio of the cone outlet diameter to the diameter of the cylinder. Wear is equal in these regions when the ratio BC/DC is equal to the ratio σ_r/σ_p. In the lower region of the cone where stress is radial, the wear rate is inversely proportional to the diameter of the channel as indicated.

— *Wedge Hopper,* Fig. 9.
Similar to the conical hopper, the solids pressure against the wall is linear in the lower part of the hopper with a large peak occurring at the transition. Since the solids velocity and pressure are directly proportional to the width of the flow channel, the wear rate is constant in the lower part of the hopper and reaches a maximum at the pressure peak as indicated. The wear on the vertical end walls of the hopper is approximately one quarter of that on the sloping side walls at the corresponding level.

— *Wedge Hopper Under a Pile,* Fig. 10.
The basic difference between the wedge hopper with the vertical cylinder and the wedge hopper under a pile is that the overpressure acts on a dead region of the pile and not on the hopper walls. This significantly reduces the wear in the hopper, and in fact makes it essentially uniform

throughout. This suggests the possible use of expanded flow bins or rock boxes at the top of a wedge hopper (see Fig. 11) as a means of controlling wear in hoppers at the transition between the cylinder and the hopper. The required size of this ledge is a function of the effective angle of internal friction of the bulk solid and must be large enough to produce a dead region of slant height at least 1/3 the width of the hopper at the transition to eliminate the otherwise high wear area at the top of the hopper.

7. Conclusions

It has been shown that the most severe wear areas in a conical hopper occur at the outlet and at the transition point; for a wedge hopper wear is uniform except for higher wear at the transition. For either type of hopper, if the level of material in the vertical section is low enough or if a rock box is employed (so that the pressure peak at the transition point does not occur) then wear ceases to be critical at the transition. When handling very abrasive materials, it is desirable to use a wedge hopper for two reasons:

- Wear is uniform within the hopper, making efficient use of wear material. Maintenance is only required when the entire hopper liner needs replacement.
- Hopper wear resistant liners are easily fabricated from plate stock with no bending required. This is important because many wear resistant materials are quite brittle and difficult to form.

Special coatings such as teflon and various epoxies are often applied to existing conical bin walls to lower the wall friction, so as to promote mass flow in an otherwise funnel flow bin. Generally speaking, the most critical region for attaining low wall friction is in the vicinity of the outlet where solids pressures are low and the wall friction angle higher than regions further up in the hopper. Unfortunately, as shown the outlet is subject to the most wear, making it necessary in many instances to apply more coating near the outlet. Otherwise, the bin can revert back to a funnel flow pattern if the coating near the outlet wears through.

Although this paper has only cited examples of predicting wear in mass flow bins and hoppers, the general method applies equally well to chutes, screw feeders, and other similar bulk solids handling equipment where the solids velocity and pressure at the wearing surface can be estimated with reasonable accuracy.

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