SHREDDING SYSTEMS FOR MIXED MUNICIPAL AND INDUSTRIAL SOLID WASTES

W. D. ROBINSON
Hammermills, Inc.
Cedar Rapids, Iowa

ABSTRACT

Shredding is probably the least understood technology involved in solid waste disposal and resource recovery concepts.

This paper is the author's critique of current shredder design criteria, operating characteristics and performance data based upon observation of shredding plants presently operating in the U.S.A. and abroad. It examines weakness in equipment and system design, and suggests means for improvement along with areas for further investigation to better understand and improve performance of such systems.

INTRODUCTION

Shredding system equipment selection and system design to date has been by imprecise methods including questionable input from manufacturers. Result: Shredding systems that do not perform as predicted with a disappointing realization that the task is more difficult than expected.

There is no established body of knowledge, and the value of recent published information is dubious. It is mostly:

1. Survey of manufacturers' recommendations by government and independent research organizations.

2. Pilot scale data, usually questionable because mixed solid waste does not lend itself to simulation or scaling-up.

There is a distressing dearth of thorough and forthright reports of performance and operating experience from mixed solid waste shredding plants. When and if such reports become available, the performance of future installations can be notably improved.

SHREDDERS, GENERAL

Unlike most other rotating equipment (pumps, fans, turbines, etc.), there is very little design criteria for predictable performance of mixed solid waste shredders. Size, style and power selection is on an empirical basis and this is not likely to change in view of the infinite types and combinations of input material. However, with an increasing backlog of experience, we can certainly expect to approach optimal results on this generally empirical basis.

SHREDDER DESIGN

There are three basic types of horizontal shaft swing hammer type shredders, Fig. (1).

A. Topfeed, single direction rotor rotation.
B. Topfeed, reversible rotor rotation.

Although they do not process truly mixed solid waste (as defined by E.P.A.), excellent reports from the Cringle Dock Plant of the Greater London (England) Council; City of Madison, Wisconsin pulverizing station; and the City of Tacoma, Washington bulky trash shredder are exceptions.

1 This paper will focus principally on the horizontal shaft swing hammer type hammermill which will also be referred to as "mill".
C. Controlled feed, single direction rotor rotation.

Each has operating characteristics peculiar to its style.

Topfeed, single direction rotation (Fig. 1A).
Mills of this type have the in-feed material entering by free fall over an inclined feed chute at an angle between \(60^\circ\) and vertical. Shortcomings are:

1. Difficulty penetrating the hammer circle with a resulting rejection tendency which contributes to an increased retention time — hammer wear syndrome.

2. Shrapnel throwback requiring effective containment hoods and curtains with restrictive movement of personnel.

3. High shock loads on mill and motor rotors, bearings, couplings and alignments.

4. Uneven discharge flow rates — slugs or surges which can reduce the efficiency of downstream separation systems.

5. Adverse rotor windage conditions which can impede material discharge and aggravate housekeeping problems.

Rotor windage is being recognized as a significant factor contributing to the less-than-expected performance of many existing topfeed shredder installations.

Test sampling by the author along with similar test data by others indicates rotor windage air flow patterns as shown by Fig. (2). Here again, it appears that empirical methods must prevail since shredder configurations and the vagaries of operation preclude application of fan laws in the effort to understand rotor windage vectors.

This test data was obtained by traversing above and below the shredder rotor with pitot tube and anemometer equipment. Although strictly cursory, the apparent results appear significant and suggest that more thorough investigation is warranted.

\[\text{Mr. Brodie Crawford, National Center for Resource Recovery, Washington, D.C.}\]
Fig. (2) shows evidence of wind-shear vectors and that the counterflow (upward) vectors increase as the shredder throat widens.

Readings taken with material being processed through the mill indicate that these adverse or counterflow (upward) windage vectors increase with load. This is very likely caused by material in and above the discharge grate openings impeding escape of the windage downward vector through these openings.

For primary shredding to a nominal 15 cm (6") topsize through say 20 x 25 cm (8" x 10") grate openings, the adverse upward air flow effect can be reduced by a narrower throat, but the risk of feed material bridgeing in the feed chute increases.

The effect of rotor windage in secondary shredding is considerably greater due to the much lower bulk density and smaller particle size of the infeed material, and air sweeping the secondary (inducing the shredded material out of the mill with an air stream) may be required for most applications.

*Topfeed, reversible rotor rotation (Fig. 1B).* Mills of this type have the infeed entering by free fall vertically downward into the hammer area and with the option of reversing the direction of rotor rotation. Its cross section profile is essentially symmetrical and was developed originally for uniform, friable material such as coal, rock, etc.

The principal advantage claimed for this type shredder is reverse rotor rotation, requiring less frequent manual turning of hammers.

The rejection tendency and adverse rotor windage factors described previously also apply to this type shredder but with greater severity because:

1. More material tends to fall in the upward path of the hammers.
2. The entry throat is wider.

Furthermore, when material already bouncing off the hammers is suddenly forced into the hammer paths along with incoming feed, shock loads are increased notably.

The theory that rejected material impinging on incoming material provides an initial reduction by attrition in mid-air doesn't seem to be confirmed in practice. It isn't very effective even with friable material.

Hammer wear advantages claimed for reverse rotor rotation and adjustable breaker plates seem dubious because:

1. Downtime and labor for manually turning hammers in the single direction mill is not significant.
2. Hammer life is equivalent.
3. Self-sharpening hammers is a myth. There is no way to compensate for rounding-off of hammer and grate edges.

The adjustable breaker plate feature which provides clearance adjustment between hammers and breaker plate, purportedly compensating for hammer wear, seems ineffective in the solid waste application. In any topfeed free fall entry mill, hammer sharpness is much more critical regarding product size, fragmentizing ability, and grabbing hold of certain material (with smooth round surfaces) which otherwise bounces above the rotor.

Equally ineffective are rejection devices (for difficult or non-reducible material) such as kick-out panels, traps, skip ejection or ballistic trajectory escapes. They are:
1. Overly selective, or
2. Reject too much material, or
3. Don’t function at all.

Reversible mills usually have no more than 100° grate arc which can reduce capacity.

**Controlled feed, single direction (Fig. 1C).** The controlled or force feed type mill has the infeed entering by gravity down a feed chute at a minimum angle of 45° assisted as required by a crawler apron compression feeder directly above the feed chute at the mill throat. It directs (grips and forces) the material into the hammer paths tangentially. Advantages claimed for it are:
1. Minimal rejection, retention time, and shrapnel backfire.
2. Significantly lower shock and power consumption peaks.
3. Uniform input and output flow rates because it can work from an inventory in the feed chute, compensating for irregularities in delivery to the mill from the feed conveying system.
4. Minimal adverse rotor windage, — it tends more to force the shredded material out the bottom instead of back out the feed end.

Data supporting these claims is not yet available because installations in the U.S.A. with this type shredder in mixed solid waste service will not be operational until late 1975. However, data from similar installations operating abroad indicate these projections to be essentially correct.

In bulky trash service, the controlled (force feed) device is operating successfully in several U.S. installations.

Altogether, the controlled feed concept appears to be a promising development in the effort to improve the performance of shredding systems.

### OTHER DESIGN AND OPERATING FACTORS

A minimum height of free fall (Fig. 3) required for infeed material to penetrate the hammer circle of topfeed machines can be calculated theoretically with certain assumptions. The calculation is:

\[
V = \frac{N \times P \times RPM}{3000*} = \text{M/Sec}
\]

(1)

\[
V = \frac{N \times P \times RPM}{360*} = \text{Ft/Sec}
\]

(English units)

\[
H = \frac{V^2}{2G}
\]

(2)

Where: 
- \( V \) = Velocity, M/Sec (Ft/Sec)
- \( N \) = Number of rows of hammers
- \( P \) = Desired penetration into hammer circle, CM. (in.)
- \( H \) = Height of free fall above hammer circle, M (ft.)
- \( G \) = Gravity acceleration, 9.8 M/Sec² (32.2 ft./Sec²)

* = Constants evolved from the assumptions

“\( P \)” values between 5 cm and 6 cm (2” to 2.75”) usually result in adequate penetration without unreasonably high or ineffectively short feed chutes. This calculation does not apply to the force feed style mill.

**Example:** 4 Rows of hammers rotating @ 900 RPM and selecting penetration of 5 CM (2 in.)

\[
V = \frac{4 \times 5 \times 900}{3000} = 6 \text{ M/Sec (20 ft/sec)}
\]

(3)

\[
H = \frac{6^2}{2 \times 9.8} = 1.8 \text{m (6 ft)}
\]

(4)

The principal assumptions in this calculation are that on the average, material enters the mill directly over the rotor centerline and that it reaches the hammer circle midway between two hammer rows. Obviously, this does not occur completely in practice. Still, it is about the only method to calculate a reasonable free fall drop height.
Theoretically, if the hammer circle penetration “P” (the point of impact on the hammers) corresponds to the center of percussion of the hammer, shock forces would be minimized. Again, it doesn’t happen that way in practice. Furthermore, inordinately high values for “H” would result.

Theorizing further, if the rotor rotation is at infinite speed, the rotor would act as a cylinder and nothing could penetrate the hammer paths. Again, it is obvious this doesn’t happen, but these concepts serve to stimulate thought regarding the phenomena of rejection tendency, penetrating the hammer paths etc., and strongly suggests pursuit of a better way to feed shredders.

Better understanding of what happens inside a refuse shredder is imperative in order to improve performance.

It must be considered a transitory expansion chamber which aerates the material during the shredding process. Resistance to discharge must be minimal along with maximum discharge area in order for this lighter shredded product, much of it in turbulent suspension, to pass through the discharge openings.

After entering the mill, raw material which penetrates the hammer circle is carried around to the grate section where much of it is extruded through and out of the forward grates in an attrition process. The remainder continues around, some of it finding random escape through rear grates and the rest of it remaining inside in an orbital inventory until a transient force (incoming material and/or higher mass particles) force it out. This seems to be confirmed by the fact that forward or inlet grate sections tend to wear faster than in the rear.

It is also interesting that rotor discs (or rotor arms) do not seem to wear very much, indicating that the material tends to stay out toward the hammers.
With household refuse, shredder upper limit capacity seems to depend more on these factors along with rotor windage rather than power and/or RPM. Product particle size, however, seems to depend more on RPM and power, and this appears to be confirmed by recent pilot scale research. Also, power and RPM are critical if bulky trash is included.

When shredding at rates upwards of 50 percent of rated capacity, there is a significant and often critical inventory of material remaining in the mill continuously and this is the result of the aforementioned factors establishing the upper capacity limit. Anyone who has ever operated a refuse shredder is aware of the difficult and time consuming task of clearing the mill if it stalls at such times.

**SHREDDER POWER REQUIREMENTS**

There are two power indices to be considered:
1. Installed power.
2. Specific power consumption, Kwh/ton.

**INSTALLED SHREDDER POWER**

The requirement is estimated empirically since there is no precise calculation yet available. Each manufacturer uses his own criteria and the estimates can range from 7.5 kw/ton/hr (10 HP/ton/hr) to 25 kw/ton/HR (30 HP/ton/hr). The selection should depend upon careful examination of the types of feed material to be expected. Usually, the most difficult to shred items, feeding pattern, and product size establish this judgment. The objective is to select sufficient power to:
1. Overcome choke feed conditions without stalling the mill.
2. Minimize interruption of infeed while the mill digests its load.

About the only power formulae adaptable to refuse shredders is applicable to just about any rotating equipment, and the principal use is in determining the power required to bring a rotor up to operating speed from zero in a pre-selected time:

\[
\text{Torque} = \frac{WK^2 \times RPM}{9.6 \times g \times t} \quad \text{(S.I. or English)} \quad (5)
\]

\[
KW = \frac{\text{Torque} \times 2\pi \times RPM}{6082} \quad \text{(using S.I. units)} \quad (6)
\]

\[
HP = \frac{\text{Torque} \times 2\pi \times RPM}{33000} \quad \text{(using English units)} \quad (7)
\]

Where:
- \(WK^2\) = Rotor inertia, KgM² (LB-FT²), from shredder manufacturer.
- \(t\) = Startup time, seconds
- \(g\) = Gravity, 9.8M/SEC² (32.2 FT/SEC²)

Although this calculation alone is not adequate in selecting installed primary shredding power, it appears to produce a reasonable determination if qualified by certain shredder design and operating criteria as follows:

<table>
<thead>
<tr>
<th>Width/Dia. Ratio</th>
<th>&gt; 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hammer Weight</td>
<td>&gt; 65 Kg (145 lb.)</td>
</tr>
<tr>
<td>Rotor WK²</td>
<td>&gt; 1475 KgM² (35,000lb.ft²)</td>
</tr>
<tr>
<td>Starting Time (t)</td>
<td>= 30 Sec.*</td>
</tr>
<tr>
<td>Hammer Tip Speed</td>
<td>&gt; 70 M/Sec (14,000 ft./min.)</td>
</tr>
<tr>
<td>Rows of Hammers</td>
<td>= 4</td>
</tr>
<tr>
<td>Grates</td>
<td>= Say 20 cm x 25 cm (8&quot; x 10&quot;), 180° grate arc</td>
</tr>
<tr>
<td>Product Size</td>
<td>= Nominal 15 CM (6&quot;) topsize</td>
</tr>
<tr>
<td>Feed Material</td>
<td>= Raw mixed solid waste with little or no heavy oversize</td>
</tr>
</tbody>
</table>

On this basis, a primary (or first stage) shredding rate of approximately 1 ton/15 kw/hr (1 ton/20 HP/hr) can be expected, assuming that other system design criteria are compatible, i.e. optimal feed and discharge systems.

Table I has been assembled using this method. The shredder sizes are well within standard ranges offered by most horizontal shaft shredder manufacturers.

**SPECIFIC POWER CONSUMPTION**

Shredders are most often driven by electric motors and prediction of specific power consumption and demand peaks should not be difficult when

---

*Trezek and Savage, Department of Mechanical Engineering, University of California, Berkeley, California.
and if sufficient measured data is forthcoming from existing and future installations. Shredding mostly household wastes to a nominal 6” topsize, reasonable data is already appearing from a few plants as follows:

Running empty, power consumption appears to be about 25 percent of motor nameplate rating where shredder design and installed horsepower fail within the range of Table I.

Under load, specific power consumption of 6 to 10 kWh/ton is indicated. For bulky trash only, 15 to 20 kWh/ton is expected.

Cyclic torque swings from 20 percent to over 200 percent of nameplate torque in a second or less is not uncommon, especially with top feed mills, and the motor should have 250 percent breakdown torque @ 100 percent rated voltage.

Under full load or choke feed conditions, the shredder has up to 2½ times the motor nameplate torque available for 10 to 15 seconds. Other motor criteria are: 3 to 5 percent slip, and a 1.15 service factor.

The experience to date with squirrel cage motors with this specification has been excellent. Likewise, the experience with direct connected flexible couplings and belt drives, with the choice often depending on space availability.

Conversely, the experience with fluid clutches in high shock service has been poor and the slip ring motor (at much higher cost) is preferred where concern over start-up power draw is critical (a rare occurrence and with very large motors only — upwards of 1500 kw (2000 HP).

### OTHER SHREDDER DRIVE OPTIONS

1. Diesel engine
2. Steam turbine

Electric motor drives are usually preferred because of simplicity, lower installed cost, and maintenance. However, the diesel engine and steam turbine are viable alternatives with significantly lower operating costs in most locations.

Typical torque-speed curves for electric, diesel and steam turbines are shown in Fig. (4). Compared with the electric motor, the diesel engine and steam turbine will tolerate a greater reduction in shredder rotor speed before stalling under full load and choke feed conditions. Nevertheless, in practice, the electric motor with 250 percent breakdown torque has an excellent performance record and the selection should depend more upon other factors, — availability and economics of energy supply, installed costs etc.

The experience with diesel drives in many shredder installations has been excellent, and the steam turbine experience has been excellent in at least one installation (in severe bulky trash service).
## TABLE II

<table>
<thead>
<tr>
<th>PLANT</th>
<th>MILL TYPE</th>
<th>DIA.* X WIDTH</th>
<th>INSTALLED POWER</th>
<th>DESIGN CAPACITY TON/HR</th>
<th>OBSERVED CAPACITY TON/HR AVG.</th>
<th>PRODUCT NOMINAL TOSIZE</th>
<th>INFEED MATERIAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Topfeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>600 KW (800 HP)</td>
<td>40</td>
<td>25</td>
<td>15 cm (6&quot;)</td>
<td>Mixed</td>
</tr>
<tr>
<td>B</td>
<td>Topfeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>600 KW (800 HP)</td>
<td>30</td>
<td>15</td>
<td>20 cm (8&quot;)</td>
<td>Bulky</td>
</tr>
<tr>
<td>C+</td>
<td>Topfeed</td>
<td>1.78M X 2.28M (70&quot; X 90&quot;)</td>
<td>300 KW (400 HP)</td>
<td>40</td>
<td>30</td>
<td>10 cm (4&quot;)</td>
<td>Sorted</td>
</tr>
<tr>
<td>D+</td>
<td>Forcefeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>600 KW (800 HP)</td>
<td>50</td>
<td>40</td>
<td>15 cm (6&quot;)</td>
<td>R I C</td>
</tr>
<tr>
<td>E</td>
<td>Forcefeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>600 KW (800 HP)</td>
<td>25</td>
<td>20</td>
<td>15 cm (6&quot;)</td>
<td>R I C</td>
</tr>
<tr>
<td>F</td>
<td>Topfeed</td>
<td>1.5M X 2.5M (60&quot; X 100&quot;)</td>
<td>375 KW (500 HP)</td>
<td>50</td>
<td>35</td>
<td>15 cm (6&quot;)</td>
<td>Sorted</td>
</tr>
<tr>
<td>G</td>
<td>Topfeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>675 KW (900 HP)</td>
<td>40</td>
<td>20</td>
<td>20 cm (8&quot;)</td>
<td>R I C</td>
</tr>
<tr>
<td>H</td>
<td>Topfeed</td>
<td>1.5M X 2.0M (60&quot; X 80&quot;)</td>
<td>600 KW (800 HP)</td>
<td>25</td>
<td>15</td>
<td>20 cm (8&quot;)</td>
<td>R</td>
</tr>
<tr>
<td>I</td>
<td>Topfeed</td>
<td>2.28M X 2.38M (90&quot; X 94&quot;)</td>
<td>750 KW (1000 HP)</td>
<td>60</td>
<td>45</td>
<td>10 cm (4&quot;)</td>
<td>R I C</td>
</tr>
<tr>
<td>J+</td>
<td>Forcefeed</td>
<td>1.8M X 2.5M (72&quot; X 100&quot;)</td>
<td>1500 KW (2000 HP)</td>
<td>50</td>
<td>50</td>
<td>15 cm (6&quot;)</td>
<td>Bulky</td>
</tr>
<tr>
<td>K</td>
<td>Topfeed</td>
<td>1.27M X 1.5M (60&quot; X 60&quot;)</td>
<td>375 KW (500 HP)</td>
<td>40</td>
<td>35</td>
<td>10 cm (4&quot;)</td>
<td>Sorted</td>
</tr>
</tbody>
</table>

*Outside USA

*Hammer Circle Diameter

R = Residential
I = Industrial
C = Commercial
MATERIALS HANDLING—SHREDDER FEED CONVEYING SYSTEM DESIGN

This is critical because topfeed type mills require a steady, uniform feed rate in order to perform with any degree of acceptability, i.e. uninterrupted throughput of most material delivered to the plant without excessive hand picking or segregation. Even with off-site pre-sorting, this places a difficult burden on the materials handling system. The vagaries in material delivery to the receiving conveyor include slugging from trucks, payloader or grapple, followed by avalanche or slug discharge from the receiving conveyor often causing excessive interruption of feed to the shredder while it digests its erratic load. Result: Average hourly production less than expected.

The experience to date with methods to optimize input flow rates is discouraging. These include: differential speeds for receiving and feed conveyers (when they are separate units), steep slope conveyer roll-back, strike off baffles, trapeze type levellers and vibratory pans.

When a feed conveyor system consists of multiple belts, Fig. 2A, the transfer points are usually troublesome and require constant attention.

The most common feed conveyor arrangement (Fig. 5B) has the inclined section discharging material directly into the mill after turning an acute angle. It has several disadvantages:

1. Longer items must extend further out over the feed chute before cresting over and often bridge above the mill throat.
2. Items with uneven weight distribution (freezer, refrigerator etc.) can crest over with a tendency to whip, tumble and bridge. Gravity force vectors explain this.

Opening wide the mill throat to minimize bridging can be a poor trade-off. It increases rejection tendency, shrapnel backfire and adverse rotor windage as discussed previously.

It is suggested that a single continuous belt of the bend-back type (Fig. 5C) with vertical feed hopper sides and vertical skirt boards will prove to be most reliable. The bend-back configuration has a horizontal section at its discharge end.

The same advantages are possible with a combination of inclined feed conveyor section discharging onto a declined vibratory conveyor, thence into the mill. In any case, vertical loading hopper sides and vertical conveyor skirt boards, both in the same vertical plane and joined to form single continuous vertical sidewalls is important if smooth continuous flow is to be expected.

Feed conveyers can be steel or synthetic belts, or steel vibratory pans. The selection depends, of course, upon the service. If inclined steeper than
20°, bulky items can slide back down a belt with or without risers.

Caution must be exercised with vibratory feeders regarding any possible combination of mass, moisture and irregular shapes which may not move. Critical factors are amplitude, frequency, stroke and angle of decline. In selecting vibratory pan conveyers, the possibility of establishing resonance with the natural frequency of substructures and adjacent or nearby structures must be considered. If this occurs, it is usually necessary to change the vibratory mode of the conveyer without rendering the conveyer ineffective, otherwise it should be replaced by a belt.

**SHREDDER DISCHARGE CONVEYERS**

These can also be steel or synthetic belts or steel vibratory pans. Selection for reliable continuity of flow depends upon the types and combinations of shredded material with regard to impact and abrasion as well as mass, moisture and shapes.

Also to be considered are downstream separation systems. If otherwise suitable, the vibratory pan tends to even out the discharge flow, — an advantage downstream. On the other hand, it usually adds a transfer point, whereas a single belt can run directly to the next operation.

Discharge conveyers should be able to withstand the downward force of an explosion in the mill. Otherwise, it should be repairable or replaceable with a minimum of downtime. Explosions in shredders is discussed in “Shredding Plant Hazards and Environment” below.

In general, materials handling systems for solid waste shredders perform best with straight line flow and few transfer points.

Again, the importance of uniform feed flow rates to the shredder is emphasized. It not only improves performance of the shredder, but discharge conveying and downstream separation systems as well.

**SHREDDING PLANT HAZARDS AND ENVIRONMENT**

Most operational solid waste shredding plants seem to be in compliance with the more common workplace safety requirements promulgated by OSHA, – guard rails, belt guards, containment hoods and curtains, noise levels etc., and extraordinary difficulty in complying henceforth is not
expected. The subject of explosions, however, is of great concern. To date, there have been upwards of a dozen damaging explosions in solid waste shredding plants. Serious damage to plant and equipment has occurred in several episodes but with little information available regarding injury to personnel.

It must be recognized that solid waste shredding plants are vulnerable to explosions, usually from cans or tanks of gasoline, propane, etc., and occasional higher explosives such as dynamite, live ordnance, etc.

Vulnerability depends largely on logistical factors of location and industrialization as well as categories of solid waste delivered.

Concern is high enough that a trend is developing with some plant designers toward maximum front end separation to divert explosives. Otherwise, provision for maximum visual inspection of incoming material is paramount.

With shredding stream capacities upwards of 50 ton/HR daily hourly average, the feasibility of either is uncertain.

The primary explosion in a shredding system is a gas explosion caused by a friction spark and sometimes followed by a more violent dust explosion. Explosive dust mixtures of the type most likely to form in a municipal solid waste shredder require a higher energy level for ignition than available from a friction spark. Explosion suppression systems have proved effective for gas and dust explosions and are being specified frequently. In addition, other means for minimizing building damage and personal injury are:

1. Rigid enforcement of off-limits areas for roving personnel.
2. Complete enclosure protection for the shredder operator.
3. Separate or detached shredder building enclosures with blow-out sidewall and roof panels.
4. Partially open walls and/or roof.
5. Shredder completely outdoors.

The shredder itself must be sturdy enough to withstand most explosive forces encountered from the more common explosive gas and dust mixtures, exceptions being dynamite or equivalent higher energy sources.

It is imperative that sufficient pressure relief area is provided in the shredder and connecting superstructures such as hoods, ducts, — any connected enclosure. Excepting the shredder, this can be by means of hinged flaps, tethered blow-out panels and flexible flaps. Much can be learned in this regard from automobile shredder installations and it is rather surprising that solid waste shredding plant designers do not seem to be aware of this, — not only in connection with explosions, but also the value of operating, maintenance and reliability factors in such installations.

DUST EMISSIONS

Most solid waste shredding systems do not require dust collection if the installation is indoors and ambient dust conditions can be tolerated. Water spray in the mill is effective in containing dust if downstream processes do not preclude any higher than as-received product moisture.

In any event, water spray is recommended if only for containment of fires in the mill. If, however, downstream processing includes air classification, pneumatic conveying, air cleaning etc. involving cyclone separation, then dust collection will inevitably be required. If current industrial process particulates codes apply, and it seems likely they will, simply dry cyclone dust collectors may not suffice and bag filters or wet scrubbers will be required. Again, much can be learned from the experience in scrap auto shredding plants where large quantities of air for product cleaning and dust collection often require wet scrubbers with fan power from 375 KW to 750 KW (500 HP to 1000 HP).

OPERATION AND MAINTENANCE

Total owning and operating costs for mixed solid waste shredding systems appear to be between $6.00 and $10.00 per ton, including installation costs, but not including land, building and ancillary materials handling equipment (payloaders, etc.).

Fixed costs for amortization, depreciation taxes and insurance can represent between 25 percent and 50 percent of the total, depending largely upon public or private ownership.

Although labor is usually included with operating cost factors of power and maintenance, it could realistically be included as a fixed cost item instead.

Power and maintenance costs are essentially proportional to production, but labor is not, with perhaps the exception of the maintenance labor increment. Even so, the labor force usually remains constant regardless of production.

A principal manufacturer of explosion suppression systems does not necessarily agree with this.
Most shredder manufacturers will suggest that 4 to 6 men are required to operate a single shredding line (shredder with feed and discharge conveyors) and this seems reasonable considering that the total work force in a one or two line shredding station is usually between 12 and 16 men, including supervision, for one shift operation and with second shift maintenance and clean-up. This would be a plant where the principal function is shredding, with truck loading for product removal. Exceptions are where a shredder for bulky trash only is ancillary in an incinerator facility where labor chargeable to shredding would be considerably less. Likewise, a single mill operating outdoors at a landfill.

Electric power costs have been found to be between $.25 and $.75/ton depending considerably upon demand peaks and demand charge. An average of $.50/ton has been most common.

Shredder hammer wear is the largest single maintenance item in the system. In some plants, daily welding build-up of spare hammers is required along with tipping or hard facing of hammers in the mill after the operating shift. In others, hammers are re-tipped in place less frequently with correspondingly less frequent sideline build-up of spares. All of this is required to insure the desired product size and the ability of the hammers to grip and pull material into the mill.

These hammer maintenance patterns are determined mainly by the character of the infeed, desired product size, hammer shape and hardness. Retipping in place after 600 to 1,000 tons is common. Most solid waste shredders presently operating employ alloy steel hammers with Brinell hardness in the range of 250 to 300 and fairly high manganese alloys are not uncommon. Since manganese wears very quickly until work hardened, it is hardly suitable for the solid waste application.

A promising development is the use of harder hammers, Brinell up to 500, which appear to be more abrasion resistant with ample shock resistance. The purpose is to eliminate the frequent (often daily) welding, discarding the stubs after each side wears to the critical roundness. The replacement cost and simpler maintenance pattern vs welding, labor, material and nuisance cost factors should easily favor the discard hammer approach. The steel scrap shredding industry has found it to be a major improvement and it could prove to be a major improvement in solid waste shredding.

On the basis of discard hammer life of 10,000 to 12,000 tons, hammer maintenance cost can be estimated at $.50/ton.

**CONCLUSIONS**

Performance data: average hourly production, power consumption, labor, costs, etc., are not available from most solid waste shredding plants.

Table II, therefore, has been assembled from data acquired in an unsolicited survey by observing operations and interrogation of operating personnel in a rough appraisal of the recent state of the art. Although certain installations provided carefully collected data, it should be remembered that the remainder is the judgment of one experienced observer without benefit of cumulative measured data which in most cases has never been gathered by the operators. For these reasons, the installations are not identified.

In any event, it has been apparent to this author that regardless of shredder size, horsepower etc., few shredding systems can average much more than 30 ton/hr without unexpected or undesirable concessions in:

1. Types of infeed material (little or no oversize).
2. Output product size.
3. Additional labor (welders, sorters, etc.).
4. Operating costs.

Many resource recovery entrepreneurs offering plants to process upwards of 1,000 ton/day seem preoccupied with recovery techniques while assuming average hourly shredding capacities upwards of 50 ton/hr per stream and often with unsorted raw material.

Such capacities are very likely possible but with larger machines and design features not usually called out.

**KEY WORDS**

<table>
<thead>
<tr>
<th>Horizontal Shaft Shredders</th>
<th>Calculations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Styles</td>
<td>Power Requirements</td>
</tr>
<tr>
<td>Shortcomings</td>
<td>Feed Height</td>
</tr>
<tr>
<td>Shredding Theory</td>
<td>Shredder Drives</td>
</tr>
<tr>
<td>Plant Performance Evaluation</td>
<td>Electric Motor</td>
</tr>
<tr>
<td></td>
<td>Diesel</td>
</tr>
<tr>
<td></td>
<td>Steam Turbine</td>
</tr>
</tbody>
</table>

**Power Consumption**

<table>
<thead>
<tr>
<th>Operating Costs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Labor Requirement</td>
</tr>
</tbody>
</table>

**Conveyor Systems**

**Explosion Protection**

260