In handling difficult slurries, the process can frequently be improved by eliminating the need for seal water in the stuffing box. In the production of water-soluble potash, for example, any additional water introduced in the course of processing the ore must eventually be removed, raising the cost of the already-expensive chemical compound.

Even where dilution of the product with flush water is no special concern, clean water may be hard to come by. And packing may fail quickly from abrasive slurry particles making their way into the stuffing box.

Expeller slurry pump: sealed without water.

By Gary Davidson

For these reasons, a waterless seal would be a very welcome addition to a slurry pump line. Of course, improperly flushed mechanical seals have very limited life expectancy in an abrasive environment. But what about a hydrodynamic *seal in which sealing* pressure *is created by an expeller? The concept is relatively simple, but designing a practical seal to work over a range of heads and capacities is more complicated. In this article, considerations in the development of Worthington's hydrodynamically sealed expeller pump are described.*

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Worthington's research group has been working for several years to perfect a practical system of expeller shaft sealing. The result: a series of hydrodynamically sealed "MX" expeller pumps has been added to our service-proved line of hard-metal slurry pumps. Presently, "MX" slurry pumps are available in sizes from 2½- to 12-inch (60 mm-to-300 mm) discharge with capacities to 10,000 gpm (2300 m^3/h) and heads to 165 feet (50m).

No-water, low-maintenance seal.

A hydrodynamic seal has distinct advantages over conventional slurry pump sealing arrangements. By eliminating the need for a continuous flush water supply to purge the stuffing box of ingressing solids and contaminants, it also eliminates the associated seal water supply system, including pump, motor, controls, valves, piping, and fittings. The clean water which is not used for flushing is available for other process use. And, of course, there is no product dilution, and so no additional dewatering is required at the end of the process.

A second group of advantages relates to maintenance. The hard-metal expeller is designed as an integral part of the pump, with the same reliability and dependability as other components. While mechanical seals do not lend themselves to abrasive slurry service and packing must frequently be adjusted or replaced, an expeller works maintenance-free until severely worn. Maintenance for the pump is therefore limited to occasional impeller axial clearance adjustments or the replacement of worn parts.

Principle of operation.

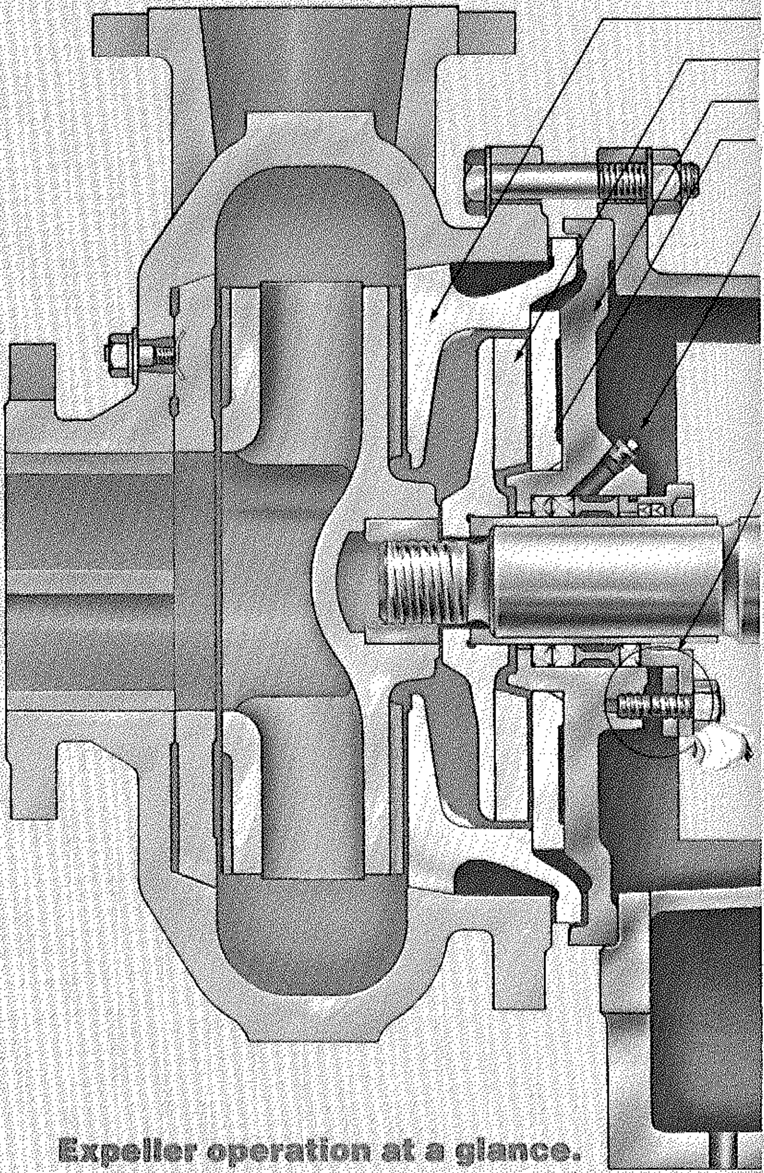
The physics governing expeller operation are straightforward. As the expeller rotates within a confined cavity, the vanes impart forces to the mixture of fluid and solids within the cavity, causing it to rotate about the shaft in a manner similar to a forced vortex. Centrifugal action associated with this rotation flings the mixture outward, but the expeller housing boundaries confine the outward motion.

The net result of this action is the creation of a low-pressure area about the shaft. Air fills the region of low pressure, and a concentric mixture-gas interface is formed at some radial diameter within the expeller vanes. This interface is essential, as it forms a distinct barrier between the mixture and the shaft. It also protects the static seal from exposure to abrasives during pump operation.

The interface may form at various radial distances from the shaft, depending on the conditions of operation, since its diameter is determined by a balance of pressures, i.e., the sealing pressure generated by the expeller and the opposing pressure generated within the pump casing. The interface diameter can be determined from a knowledge of expeller geometry, speed of rotation, density of the mixture, and pressure to be sealed.

In Figure 1, the mixture to be sealed has a pressure P_h at the hub of the impeller back shroud. As the mixture floods the smooth face of the rotating expeller, a pressure rise occurs because of viscous pumping action attributable to disk friction. The mixture pressure is increased to a value P_t at the tip radius of the expeller. Pressure P_t is balanced by permitting the mixture to partially flood the vaned side of the expeller to a radial location r_i. Presence of a gas pressure (normally atmospheric), along the shaft on the vaned side of the expeller, completes the pressure balance.

The sum of the interface pressure P_i and the pressure rise developed in the rotating ring of mixture on the vaned side of the expeller must equal the tip pressure P_t in equilibrium conditions. The location of the mixture-gas interface at r_i can



Expeller operation at a glance.

An expeller is a hydrodynamic sealing device consisting of a vaned disk rotating on the pump shaft, which generates a positive sealing head across its vanes. The expeller is positioned in a metal housing located behind the conventional pump impeller, and the front wall of the housing separates the expeller cavity from the pump casing. The action of the rotating expeller vanes forces all liquid and solid materials in the housing away from the shaft, maintaining a dry area about the shaft — essentially a pump seal.

The expeller is the primary sealing device, and as long as the pump is operating, it keeps the shaft dynamically sealed. A packed stuffing box acts as a secondary static seal to prevent leakage when the pump is shut down.

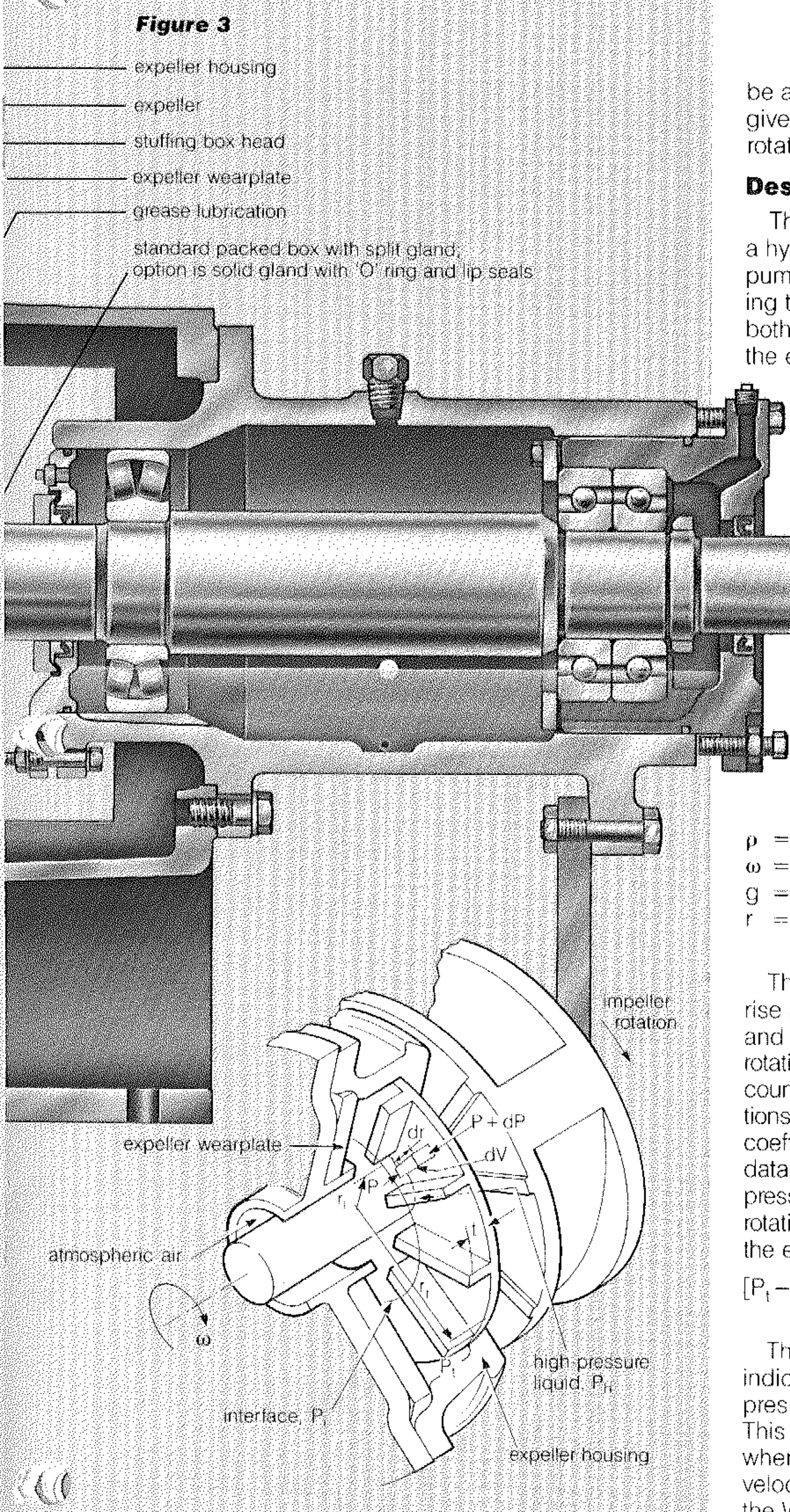


Figure 1 — Expeller seal — cutaway view.

be altered by changing P_h for a given seal geometry and a fixed rotational speed.

Designing a working seal.

The primary task in designing a hydrodynamic seal for a particular pump application is that of evaluating the pressure rise developed on both the smooth and vaned sides of the expeller for a predesignated mixture-gas interface diameter. This is accomplished by analyzing all of the external forces acting on an elemental fluid volume dV, between two consecutive rotating vanes. By equating pressure forces to centrifugal forces exerted on the fluid element, an ordinary differential equation is obtained, the solution of which yields:

$$[P_t - P_i]_{ideal} = \frac{\rho\omega^2}{2g} \times (r_t^2 - r_i^2)$$

where:

ρ = density of fluid or mixture
ω = angular velocity of shaft
g = gravitational constant
r = radial distance from shaft centerline

This is an expression for pressure rise across the vanes for an ideal fluid and neglects the effects due to the rotating inner annulus of air. To account for real fluid effects and deviations from solid body rotation, a vane coefficient based upon experimental data is introduced. The actual pressure rise across the annulus of rotating mixture on the vaned side of the expeller becomes:

$$[P_t - P_i]_{actual} = K_G \frac{\rho\omega^2}{2g} \times (r_t^2 - r_i^2)$$

The vane coefficient K_G, is an indication of vane effectiveness expressed as the amount of "fluid slip." This is often denoted as $(\beta/\omega)^2$ where β is the actual fluid angular velocity in the expeller cavity. For the Worthington expeller, β/ω is very close to the ideal value of unity.

Similarly, the pressure rise across

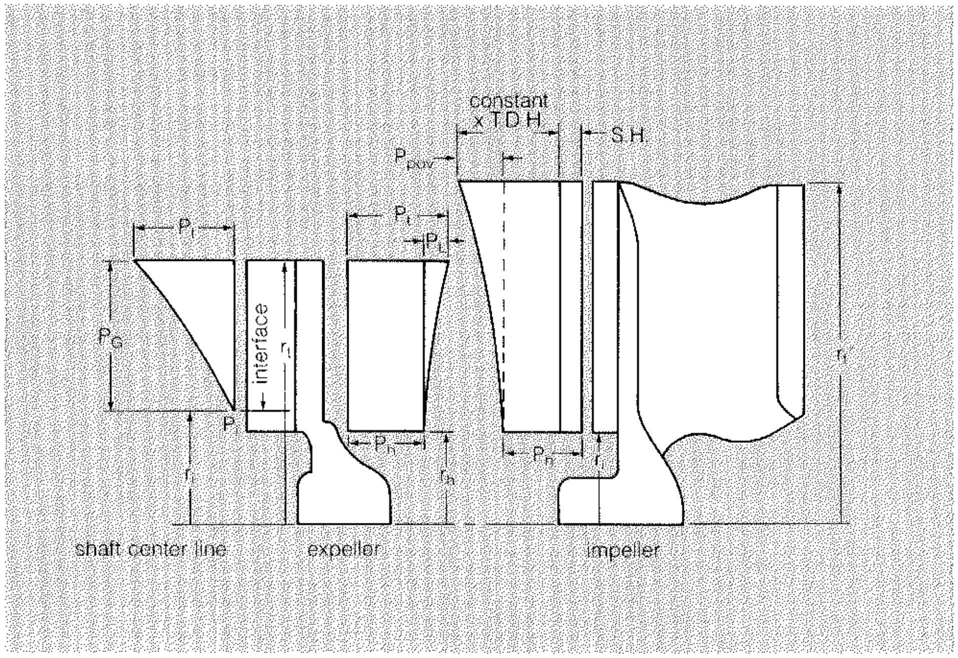


Figure 2 — Distribution of sealing pressure developed by impeller and expeller vanes.

the flooded smooth side of the expeller may be expressed as:

$$[P_t - P_h]_{\substack{\text{smooth}\\\text{disk}}} = K_L \frac{\rho\omega^2}{2g} \times (r_t^2 - r_h^2)$$

Once again, K_L is determined experimentally and is considerably less than K_G.

The total pressure P_E, generated by the expeller is the difference between the pressure generated on the vaned side and the smooth side.

Expeller power requirements can be calculated from shaft speed, torque (via a torque coefficient related to the height of the vane) and the ratio of interface radius to expeller tip radius.

Optimizing the seal.

An important question for design of the specific pumps is, "To what outer vane diameter should the expeller be restricted?" The pressure ranges the expeller must seal against need to be clearly established in order to keep the interface within a realistic range of radii — especially since the energy needs of the expeller are proportional to the fifth power of tip radius.

Figure 2 illustrates the distribution of pressures arising from the impeller and expeller vanes. In a pump, the expeller must seal against the pressure at the hub of the back shroud of the impeller. This pressure is transmitted through the mixture, along the shaft, to the smooth face of the expeller.

In conventionally sealed pumps, this would be called the stuffing box pressure (S.B.P.). Its magnitude depends on several factors, such as the suction head (S.H.), the total developed head (T.D.H.), which is a function of speed and capacity, and the effectiveness of the back shroud pump-out vanes at generating a sealing pressure (P_{pov}).

A value for P_h can be determined from an equation of the following form:

$$P_h = (\text{constant} \times \text{T.D.H.} + \text{S.H.}) \times \rho - P_{pov}$$

where the constant is a fraction which can be determined experimentally for any pump geometry.

This means that when all other parameters remain constant:

1. As capacity is decreased at a fixed operating speed, the pressure which the expeller must seal against is increased.
2. As suction pressure is increased, the total pressure which the expeller must seal against is increased.
3. As pump-out vane effectiveness deteriorates due to vane wear and impeller axial clearance adjustment, the expeller must seal against a greater pressure.

These were some of the parameters requiring definition, to assure a satisfactory seal against all stuffing box pressures which might be met in service. The resulting optimized expeller to impeller diameter ratio results in a pump line that maintains a functional seal over a wide pressure range—even under high-suction pressures—with lowest possible expenditure of energy.

Part of a proven pump line.

The expeller pump was designed to use the greatest number of existing pump parts, to take advantage of the well-proved reliability and efficiency of Worthington's slurry pump line, and to maintain interchangeability. As a result, the Worthington expeller package can readily be retrofitted in the field and served from the same basic spare parts inventory.

Figure 3 is a sectional view of the expeller slurry pump. The expeller fits over the shaft and is mounted on the end of the shaft sleeve. The impeller is then threaded onto the end of the shaft, securing the expeller firmly in place.

The expeller housing is a single piece casting which mates with the easily replaceable expeller wearplate. The housing fits over the expeller and into the stuffing box head. The stuffing box head, which can house an optional grease-lubricated static seal system, fits into the bearing frame flange, and the casing bolts tightly clamp the entire assembly in place.

The entire liquid end, bearing

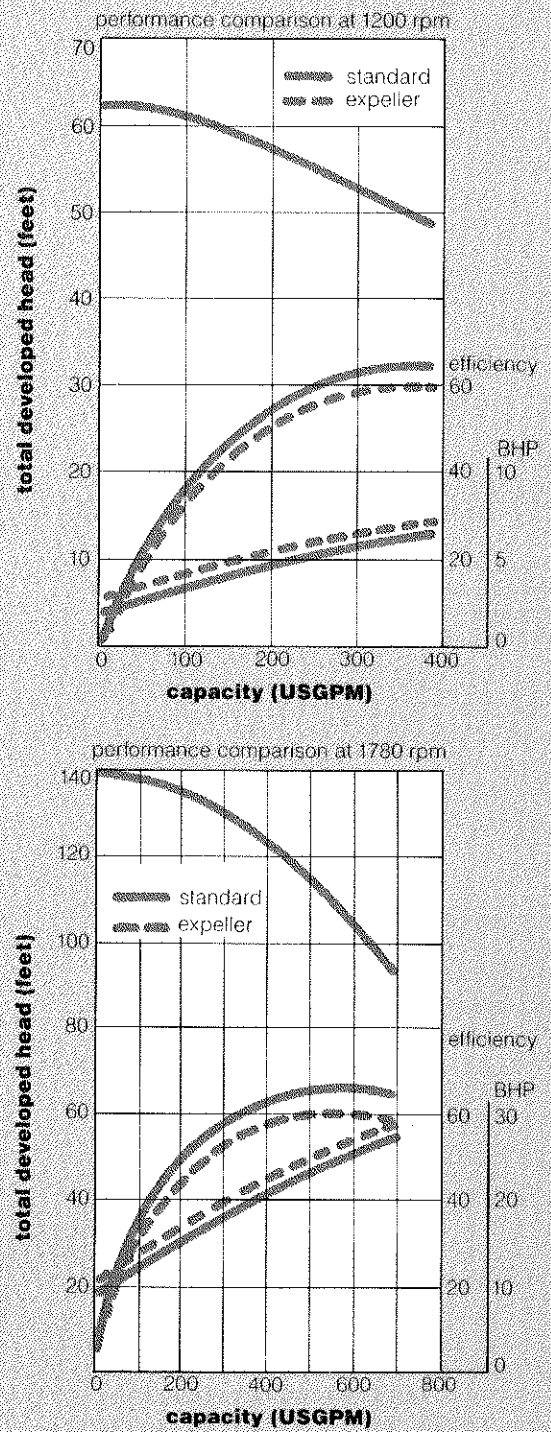


Figure 4 — Effect of expeller unit on 3-inch (76mm) slurry pump performance.

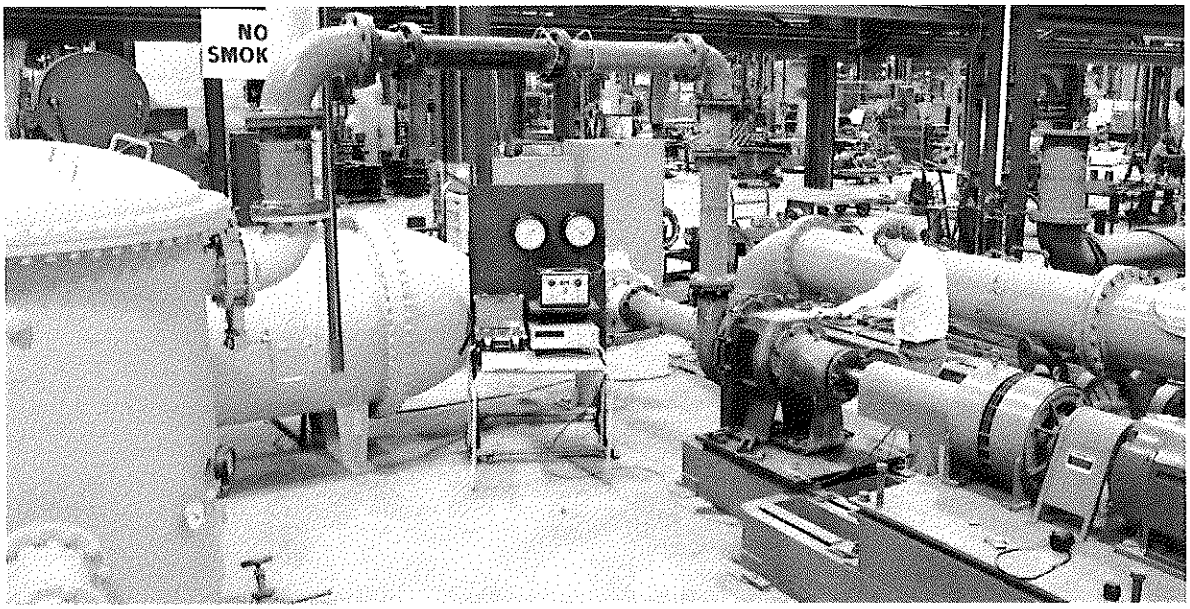


Figure 5— New expeller pump undergoes tests in the lab at Brantford.

frame, and bearing cartridge assembly are the same as the standard, service-proved type M slurry pump. The expeller package is very easily installed in the pump. Each standard bearing frame has the same size expeller for its various liquid-end options, and the greatest sealing pressure is obtained by selecting the smallest liquid end of that frame size.

Wear-resisting materials.

All expeller components subject to wear are made from hard metal ranging from 500 to 600 Brinell to provide the greatest resistance to abrasive wear which results from low-angle particle impingement. The material has been proved in conventional slurry pumps for many years.

The expeller parts have sufficient wear allowance to ensure a service life comparable with or better than the liquid-end parts to avoid need for premature disassembly.

Efficiencies stay high.

Of course, you don't get something for nothing. While saving on flush water and associated pumping costs, plus dewatering and its energy costs, an expeller pump is slightly reduced in overall efficiency because of the small amount of additional power required to maintain an effective seal. As Figure 4 illustrates, the addition of the expeller has no effect on the head-capacity curve. Expeller power consumption is minimized by optimizing vane shape, vane height, number of vanes, and expeller diameter. The small debit to efficiency should quickly be reclaimed in overall maintenance savings, reduced downtime, and elimination of flush water supply and associated equipment.

Process applications.

Worthington's hydrodynamically sealed slurry pump offers the best of both worlds: a new and improved seal system, combined with proved dependability of our existing successful slurry pump line. The expeller pump should be considered for any application where abrasive particles cause frequent packing failure, where clean water supply is limited, and where dilution of the pumped mixture cannot be tolerated. It should find many applications in the potash, coal, and copper processing industries.