14.573

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DOE/MC/14141-1158
(DE82009994)

STATE-OF-THE-ART REVIEW AND REPORT ON
CRITICAL ASPECTS AND SCALE-UP CONSIDERATIONS
IN THE DESIGN OF FLUIDIZED-BED REACTORS

Final Report on Phase I

Work Performed Under Contract No. AC21-80MC14141

Frederick A. Zenz, Inc.
Garrison, New York

TECHNICAL INFORMATION CENTER
UNITED STATES DEPARTMENT OF ENERGY

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**Final Report on Phase I
Of Contract Number DE-AC-21-80 MC 14141**

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Distributors and Plenums in Fluidized Beds

The details of the means for homogeneously distributing fluidizing media into a bed of solids (or in general into any large vessel containing a "bed" of liquid or gas) is one of the most important aspects in the design of fluidized bed reactors. It is frequently the only available physical control, or influence, able to be exerted on the system's fluid dynamics. Once past the grid ports the fluid traverses the bed via defined, though calculable, paths from which it cannot be diverted easily, if at all. The directly analogous, quantitatively as well as qualitatively equal, situation arises in the design of homogeneously bubbling perforated, valve or bubble cap distillation and absorption tower trays. Just as vapors must be distributed across a tray to insure all holes, valves, or caps are bubbling, so must gas be distributed over a large diameter vessel to insure that all grid ports are bubbling. The analogy is even more distinct with the realization that a conventional 26 tray column has been used industrially with an adsorbent powder feed in place of liquid; essentially 26 shallow beds spilling from one to another through downcomers identical to the path once provided for liquid downflow.

Generic Forms of Distributors

Within a nearly infinite variety of mechanical renditions

grids can be categorized into three generic forms according to the direction of gas entry into the bed; whether directed vertically upward, horizontally, or vertically downward. There is no single form or single mechanical design that is universally best; each must be custom designed for its specific application and might suffice for only the one for which it was intended.

As illustrated in Figure 1(a) vertical upflow through a

Figure 1. Generic Forms of Grid Design

flat grid plate affords the same clean simplicity as the perforated distillation tray. By the same token it also suffers the "sifting" or weepage of solids down through the holes if the gas throughput is insufficient to maintain all holes blown open and bubbling homogeneously.

In shallow bed (gas-liquid) distillation practice weepage

at high gas turndown is averted by affecting a pigtail seal or vapor trap at each port, as illustrated in Figure 2(a). Rotation through 360° (Fig. 2b), extraneous metal deleted (Fig. 2c), and edges serrated or perforated (Fig. 2d and 1b),

Figure 2. Development of the Bubble Cap Seal

results in the conventional bubble cap with gas distributed in a horizontal direction. The cap still acts as a non-weeping seal when submerged in a bed of solids deeper than its riser height because unaerated solids exhibit an angle of repose limit to lateral (and hence upward) flow and therefore cannot rise through the annulus from the lip of the cap to overflow into the riser.

Vertically downward directed entry, as in Figure 1(c), follows the elementary principles of sparger or "blowing manifold" design. It should again afford the advantages of eliminating solids weepage upon high gas turndown or even com-

plete shut down. As detailed on subsequent pages innumerable mechanical permutations and combinations of these 3 generic forms have been employed in industrial units each having advantages specific to the given process or operation.

The Plenum Chamber or Wind Box

Function of the Plenum Chamber

In the most general sense the plenum above or below the ports of a distributor is in effect merely a header allowing feed gas access to a series of parallel ports. It can take the form of a typical perforated pipe sparger as in Figures 3(a) and (b) or if the area to be covered is too large to justify a multiplicity of individual spargers then they may be

Figure 3. Sparger and Grid Plate Plenums

blended into one large chamber as in Figure 3(c) which leads to the conventional "grid plate" configuration where in effect the body of the header becomes the bottom head of the vessel. In the arrangement of Figure 3(c) the top of the sparger usually becomes a flat plate, and since it bridges the vessel area it must be sealed to the vessel wall to avert gas leakage around its periphery. Obviously, such a flat plate requires special mechanical design considerations regarding warpage, bed support "strength", thermal expansion, etc., which inserted spargers such as in Figure 3 (a) and (b) avert. The choice of design is governed in some instances by cost or convenience, and in others by process limitations since the spargers of Figures 3 (a) and (b) cannot avoid the accumulation of static solids below their level in the vessel, which in some applications would be intolerable. By the same token, a large plenum as in Figure 3 (c) can represent too long a holdup time, intolerable for feed gas mixtures in or near explosive concentration ranges.

Design Criteria for Sparger Plenums

The plenum of a sparger such as in Figures 3 (a) and (b) represents a classic "blowing" manifold for which well developed and verified design criteria are available as exemplified in Figure 4 and detailed in Table 1. For other than 90° changes in flow direction, as in the branched manifold grid illustrated

Figure 4. Flow Distribution from Blowing Manifolds

in Figure 5, the flow-pressure drop relationships are given in Figure 6.

TABLE 1—Equations for Pressure Loss in 90° Pipe Branches

Elbows
$\Delta P_{1-2} = 0.000135 \rho [1.2 V_2^2]$
$\Delta P_{1-2} = 0.000135 \rho [1.8 V_2^2 - 0.368 V_1 V_2]$
Split Flow
$\Delta P_{1-2} = 0.000135 \rho [1.36 V_2^2 - 0.64 V_1^2 - 0.72 V_1 V_2]$
$\Delta P_{1-2} = 0.000135 \rho [1.8 V_3^2 - 0.368 V_1 V_3]$
$\Delta P_{3-1} = 0.000135 \rho [1.8 V_1^2 - 0.368 V_1 V_3]$
Join Flow
$\Delta P_{1-2} = 0.000135 \rho \left[2 V_2^2 - 0.05 V_1^2 - 2 V_2 \left(0.205 V_3 \frac{Q_3}{Q_2} + V_1 \frac{Q_1}{Q_2} \right) \right]$
$\Delta P_{1-3} = 0.000135 \rho \left[2 V_3^2 - 0.4 V_1^2 - 0.41 V_3 \left(V_1 \frac{Q_1}{Q_3} + V_2 \frac{Q_2}{Q_3} \right) \right]$
$\Delta P_{3-1} = 0.000135 \rho \left[2 V_1^2 - 0.4 V_3^2 - 2 V_1 \left(0.205 V_3 \frac{Q_3}{Q_1} + V_2 \frac{Q_2}{Q_1} \right) \right]$

Figure 5.
Branched Manifold
Grid

In the case of a large plenum below a conventional flat grid plate where gas enters through a relatively small pipe at high velocity, various criteria have been proposed to circumvent preferential gas flows to grid holes located within "line of sight" of the entering gas as shown in Figure 7.

These criteria can be shown to relate to the more fundamental Figure 4 by rewriting the expression in Figure 7 in the form:

$$\frac{\rho V^2}{\Delta P} = 29/100 \left[1 - \frac{\pi}{4} \frac{D_o^2}{HD} \right]^2$$

If $D_o \leq D/100$, then $H = 18D_o$, in which case:

$$\frac{D_o^2}{HD} = \frac{D_o^2}{18D_o(>100D_o)} = \frac{1}{>1800} \longrightarrow 0$$

For Elbows, Y's, and Branches :

$$2gh = C_0 V_1^2 + (2C - C_0) V_2^2 - 2C V_1 V_2 \cos \alpha$$

For Converging or Joining Flow :

$$2gh_{1-2} = C' V_1^2 + V_2^2 - 2V_2 \left(\frac{V_1 Q_1}{Q_2} \cos \beta' + \frac{V_3 Q_3}{Q_2} \cos \gamma' \right)$$

where $h = \Delta P / \rho$

and subscripts 1 and 3 refer to upstream
and subscript 2 refers to downstream

Figure 6. Coefficients for Mixing Losses in Branching and Joining Flow

For horizontal entry:

$$\Delta P_{exp} = \left[1 - \frac{0.785 D_{noz}^2}{H_{vj} D} \right]^2 \left[\frac{\rho V_{noz}^2}{2g_c} \right]$$

$D_{noz} > D/100$, then $H = 0.2 D + 0.5 D_{noz}$

and if $D_{noz} < D/100$, then $H = 18 D_{noz}$

For vertical entry:

$$\Delta P_{exp} = \left[1 - \frac{0.785 D_{noz}^2}{D^2} \right]^2 \left[\frac{\rho V_{noz}^2}{2g_c} \right]$$

$H = 3(D - D_{noz})$, when $D_{noz} > D/36$

and $H = 100 D_{noz}$, when $D_{noz} < D/36$

Figure 7. For Good Distribution $(P - P_e) > 100 \Delta P_{exp}$

or

$$\frac{\rho V^2}{\Delta P} = 2g/100 \left[1 - \frac{\pi}{4} (\rightarrow 0) \right] = 0.644$$

and

$$\frac{n d^2}{D^2} = \frac{A_{Total \text{ of holes}}}{\pi D_o^2 / 4}$$

so that

$$\frac{n d^2}{D^2} \ll 0.01 ; \text{ or } \frac{0.21}{D^2}$$

If $D_o \geq D/100$, then $H = 0.2D + 0.5 D_o$

in which case

$$\begin{aligned}\frac{D_o^2}{HD} &= \frac{D_o^2}{[0.2D + 0.5D_o][<100D_o]} \\ &= \frac{D_o^2}{(<20D_o + 0.5D_o)(<100D_o)} \\ &\rightarrow 2\end{aligned}$$

or

$$\begin{aligned}\frac{\rho V^2}{\Delta P} &= 2g/100 \left[1 - \frac{\pi}{4} (\rightarrow 2) \right]^2 \\ &= 2\end{aligned}$$

and

$$\frac{\eta d^2}{D^2} \approx \frac{0.37}{D^2}$$

Note that the values 0.644 and 2 represent such small values of the abscissa of Figure 4 that distribution would certainly be excellent under the criteria in Figure 7.

Plenum Inlet Configurations

With a bed of fluidized solids above grids which permit particles to weep into the plenum either under reduced gas throughputs, or at shutdown, it is more conventional to introduce gas into the plenum in such a manner as illustrated in Figure 8. In all instances a blind flange or valve is provided

Figure 8. Common Plenum Inlet Arrangements

for solids drainage and the inlet pipe directly or indirectly protected from solids backflow, particularly if coming directly from the discharge of a compressor. Numerous non-weeping grid designs are depicted in subsequent illustrations though even such will weep if the bed solids have long deaeration times or gas flows are reduced abruptly.

It is not uncommon to introduce solids, as for example recirculating catalyst, along with the reactant feed gases, in which case arrangements as in Figures 8 (c) (d) (e) or (f), where the feed flow enters at the lowest point, are the preferred in order to minimize solids accumulation in regions inaccessible to agitation or reentrainment. Where such is unavoidable special "plenum-sweeping" gas inlets have been employed.

Where not otherwise incompatible, it is desirable to provide access to the grid through the plenum. This is necessary under arrangements such as in Figures 8 (a) (b) (d) (e) or (f), in instances where the plenum or bottom side of the grid requires lining, and where structural members support the grid from the vessel head. Though the grid might be provided with a manway it is in any event convenient to have independent access from below even if only through a Tee in the feed gas piping.

Packed Plenums

Packing the plenum to reduce its volumetric gas content and thereby its harmonic response to feed gas compressor surges is rarely practiced. It is not particularly effective without a substantial volumetric decrease, in which case possible pluggage and the resulting maldistribution from weeping solids represent far more undesirable and formidable consequences. Increasing grid pressure drop relative to that through the bed is far more effective in dampening compressor surges.

Bundled Reactors

There are several instances in industrial practice in which what outwardly appears to be one large reactor, is actually a "bundle" of small diameter reactors operating in parallel. This can result from a simplistic form of scaleup. If for example the capacity of a commercial plant is desired to be 50 times that of say a 10" i.d. pilot plant then there can be little risk in simply building 50 identical pilot plant reactors and housing them in one surrounding vessel shell. Each internal parallel flow reactor is provided with its own independently controlled feed gas flow and therefore in effect with its own plenum. This avoids uncertainties in gas distribution which would accompany any shifts and inequalities in bed inventories. In instances where substantial changes in capacity

are desired or anticipated, it is not uncommon to build several reactors and operate any number in parallel following market demands. A multi-orifice grid for a large reactor can generally not be designed to operate effectively or economically over more than a $\pm 30\%$ variation from a "normal" design load. Fluidized bed combustion represents such a parallel multi-reactor arrangement necessary to follow wide extremes in load, hence beds and plenums must be compartmentalized. Though flow between beds can be sealed by the angle of repose of the slumped solids, the plenums must be individually gas tight and valved.

Port Characteristics

The versatility, innovations, and occasional fallacies encountered in industrial grid designs can best be appreciated and assimilated by examining the details of some examples encountered in past practice.

Pipe Grids

A common form of pipe grid (basically an elaborate sparger) is illustrated in Figure 9. Gas enters through a vertical pipe which Tees into a horizontal header from which smaller pipes emanate to fill the cross section of a cylindrical vessel. The smaller pipes would normally be perforated along their bot-

Figure 9. Simple Sparger Pipe Grid

toms thus introducing gas into the bed in a vertically downward direction. The holes or ports in these pipes are generally staggered rather than in straight line orientation. The stagger is not necessarily at 90° as shown, but should be sufficient to bias the gases leaving the ports to one or the other side of the pipe. If the perforations were in a straight line there would exist the possibility, if not the probability, that all the gas from two adjacent pipes would head up through the same space between the two adjacent pipes. This would circumvent the very object of providing individual ports to create individual bubbles, would leave greater areas unaerated, and would promote excessive bubble size immediately at the grid level. Staggering of the ports might be necessary in any event when the desired number of ports is so large that if placed in a single line there would be insufficient metal between the holes to maintain the structural integrity of the pipe.

Figures 10 (a) and (b) illustrate plan views of the so-called "Wagon Wheel" form of sparger pipe grid in which a greater

Figure 10 "Wagon Wheel" and Multilevel Grids

symmetry is intended to minimize minor inequalities among the manifold pipes of Figure 9, as a consequence of their varied length and number of ports.

Multilevel Level Grids

Pipe grids are particularly adaptable to multi-level location whether in a conical bottomed vessel, as illustrated in Figure 10(c), or in any situation in which multilevel injection of the same, or different reactants, is desired. In the former case, Figure 10 (c) is preferable to 10 (d) since the varied back pressure at the 2 levels in 10 (d) makes it difficult to maintain the desired split in gas flow between the two rings particularly if the quantity of gas flow or the bed level varies significantly. Figure 11(a) exemplifies the more conventional two-level introduction of reactants which if premixed might present a hazard or incur undesirable premature reactions. Obviously R_1 , might also represent a mixture intended to undergo partial reaction prior to meeting a third reactant in stream R_2 .

Figure 11. Conventional Two - Level Reactant Feeds

Figure 11 (b) represents a combination of practices which have been found suitable in some instances and have created problems in others. As shown, it suggests a direct gas fired dryer, heater, stress relieving, waste gas incineration or similar application. Placing a layer of coarse inert stone, gravel, or so-called "grog" atop a flat perforated plate grid is often intended to protect the metal plate from the higher bed temperature, prevent finer bed solids from sifting through the grid (by acting like a filtering bed), more uniformly disperse and distribute the fluidizing media, and in this instance also protect the fuel gas nozzles from breakage if a heavy object were to fall through the bed from above. To varying degrees some of these intents are more myth than reality. Though the coarse grog may act like a filter to prevent finer bed solids from sifting through the grid (even with zero gas flow) it will not enhance the gas dispersion since the gas velocity through the usual 40% void interstices of coarse grog would be

only about 2 1/2 times its superficial velocity through the fluidized bed and this would be insufficient to guarantee all the grog interstices passing gas. In fact if the grog is too deep it can destroy the distribution accomplished by the higher pressure drop perforated grid plate. At best the grog layer (if shallow) might not worsen the distribution. The deeper the layer the more bed fines will sift into the grog interstices and cause permanent blockages leaving open only sufficient channels to satisfy the required distributing pressure drop to overcome the random local bed depth gradients. The number of these channels might well be less than the number of grid holes.

With the intent of creating a more uniform "layer" of flame across the vessel there have been attempts to design with the flows reversed from that shown in Figure 11 (b); the fuel gas introduced in the plenum below the perforated plate grid and the air blown into the vertical pipes. Unfortunately the low gas velocity through the grog interstices coupled with air backflow into the grog via the interstices of the sifting bed of fines caused backflashing within the grog bed throwing it about, uncovering the grid, destroying its intended function and creating a hazard in the plenum. In any such reactive system the least hazardous gas should represent the purge from below, allowing the rapid reaction (even explosive) to be dissipated within the cushioning bed.

If a desirable coarse grog is still fine enough to fall through the grid plate holes then obviously the perforations can be replaced with pipe capped tuyeres; if fuel gas dissipation must be improved then a greater number of smaller gas pipes must be provided. In any event the grog particles must be heavier than the volume of fluidized bed they displace lest the grog simply "floats" to the bed surface.

Tuyeres and Caps

Tuyere designs range from simple drilled pipe caps threaded onto nipples, as in Figure 12(a), to ornate variations with specific and sometimes very necessary modifications. The Tuyere

Figure 12. Pipe Caps and Tuyeres

in Figure 12(b) has a peaked top presumably intended to avoid accumulation of unaerated solids which in the case of 12 (a)

form a mound atop the cap sloped to the solid's angle of internal friction. Apparently the designers of the Tuyere in Figure 12 (b) felt 45° was steep enough or had the less likely good fortune of working with a bed of highly fluid-like free flowing solids exhibiting so inordinately shallow an angle of shear. The 2 to 4 gas ports drilled into the Tuyere represent long or high L/D holes directing gas downwardly toward the bed support plate. The downward direction was intended to create some degree of sweeping action on the plate between the tuyeres to again avoid stagnant zones of unaerated solids. The length of the holes was intended to make certain that when the fluidizing gas flow ceased the slumped solids would surely "seal" by their angle of repose within the holes and not overflow into the riser and fall to the plenum. Depending on the combination of sizes of the openings and number of ports the distribution-controlling restriction to gas flow exists either in the riser or the downsloped ports.

In Figure 12 (c) the four $1/4$ " pipe "holes" represent the restriction rather than the $1\ 1/4$ " pipe nozzle. This guarantees that all 4 ports will be bubbling. The tuyere is again peaked to minimize pileup of stagnant solids, the holes are again directed down towards the grid floor to minimize stagnant regions between the tuyeres, and the holes are extra long to assure an angle of repose seal when the gas flow is halted. The holes

in these instances are small diameter tubes, threaded into the tuyere, so that they may be replaced with smaller diameters in the event that stable operation is desired at a lower gas throughput where the present holes would not provide sufficient gas distributing pressure drop.

Figure 13 represents a tuyere more closely resembling a conventional bubble cap. The carbon steel grid plate is pro-

Figure 13. Bubble Cap Tuyere

tected by a thick layer of castable refractory so that only the cap and riser need be of a more strategic metal. In this instance the gas-distributing resistance lies again in the re-

placeable bushing at the base of the riser and not at the gas-solids interface.

Figure 14. Variants in Cap Design

The cap in Figure 14 (a) resembles more closely a conventional bubble cap. Obviously the flow distributing restriction again appears in the base of the riser thus assuring distribution of gas to all caps, but not necessarily assuring uniform bubbling all around the cap periphery at the gas-solids interface. Serrating the cap skirt as commonly practiced in gas-liquid applications would have been equally desirable in this instance to overcome inevitable out-of-levelness and force at least some gas to issue from all around the cap periphery. By contrast the cap in Figure 14 (b) has its limiting restrictions at the gas-solids interface guaranteeing that all of the 1/8" holes around its periphery will be bubbling. In view of the

flat topped caps solids stagnation was not a significant consideration in either 14 (a) or (b).

In contrast, 14 (c) represents a design intended specifically to circumvent stagnation. In this configuration gas entered the bed horizontally, through 0.12" peripheral slot clearances between the grid plate and 2 1/2" diameter discs, centered over 1 7/8" diameter grid perforations. The clearance was maintained by drawing three attached stiff "outriggers", extending 0.12" past the disc, down onto the grid plate via a threaded 4 legged "spider" from below each perforation. Rigidity of the disc, to insure its lying in a plane parallel to the perforated plate, was provided by an attached steep cone which also prevented accumulation of stagnant solids atop the discs. The limiting area restriction in the clearance between the discs and the plate insured high velocity sweeping flows of gas across the plate in the spaces between the discs.

Inserts, Filtered, and Tortuous Perforations

Figure 15 (a) represents a simple abrasion resistant material insert suitable for instances in which circulated catalyst or other treated solids may enter from beneath the plate suspended in the bed-fluidizing feed gas stream. Figure 15 (b) represents an insert providing a tortuous path wherein the bed

Figure 15. Check Valves and Inserts

solids may seal against weepage by their angle of repose, when fluidizing gas is curtailed or completely halted. This is not a desirable practice since it makes it difficult to reopen all ports after a shutdown unless the grid is designed to operate well above the conventional minimum necessary pressure drop, when the bed is fluidized at the normal operating superficial velocity.

Figures 15 (c) and (d) represent attempts to incorporate weep limiting check valve functions similar to valve trays in distillation columns. The ball-in-a-cage arrangement of Figure 15 (d) was eventually abandoned due to wear as the ball kept moving around continuously. The loose nut-and-bolt illustrated in Figure 15 (c) is a common valve device, but again one employed in small units, with shallow beds, operating under mild conditions where wear is either minimal or replacement

relatively inexpensive. They are not found in reactors expected to operate continuously for several years without a shutdown.

Dished grids such as in Figure 15 (e) may be provided with inserts all terminating at the same level to overcome otherwise different bed depth resistances at the center versus at the walls.

Instances in which the bed is deep, the particles heavy, the fluidizing gas of low density, and the operating superficial velocity relatively low, it may be found that providing homogeneous gas distribution at the grid will necessitate a very small percent of open hole area. A design permitting 1/8" holes no closer than 12" apart would appear to have a substantial unaerated solids inventory at the grid level. To spread the gas out over a wider area, porous sintered metal or similar materials (fritted glass, compressed matts of woven wire mesh, etc.) are inserted with greatly enlarged holes as represented in Figure 15 (f). Small cups of coarse inerts are hung below each such insert to act as granular bed filters and delay their inevitable pluggage by contaminants in the fluidizing gas.

Penetrated Grids

In instances where exothermic or endothermic reactions require the transfer of substantial amounts of heat greater than able to be reasonably passed through the vessel shell or

carried in the enthalpy difference of the fluidizing media it is common practice to submerge vertical tubular boilers directly into the bubbling beds. Unless fabricated in the costly and cumbersome form of concentric bayonet tubes hung from the vessel head it is inevitably necessary to provide some form of headers or return bends as near to the grid level as possible. To maintain uniformity of flow paths or uniformity of cross section at all bed levels, it would be desirable to locate such headers or bends in the plenum below the grid. Since erosion considerations will not permit slipping the tubes through loosely fitting holes in the grid and since thermal expansion and possible grid buckling rule out welding the tubes to the grid plate, alternatives conceived to overcome both of these limitations are illustrated in Figure 16. In the arrangement of Figure 16 (a) the

Figure 16. Tubular Grid Penetration Designs

"floating" caps, to which the tubes are welded, are provided with internal fins in the annulus between cap and riser to prevent lateral movement which would erode the tubes and distort

the uniform emission of gas around the cap periphery. In the concept of Figure 16 (b) the grid plate has in effect a false bottom of unfluidized grog in which the bends are buried while the fluidizing medium enters the bed above this level. As shown, the tubes would likely be eroded rapidly by the gas jets issuing from the cap ports. In practice the caps would either be of the form shown in 16 (a) wherein the gas enters downwardly directed, away from the tubes, or else the tubes would be situated far enough from the cap jets to avoid impingement. Neither arrangement has been employed in industrial scale petrochemical reactors.

Sloped Grids

Other than the slope accompanying a standard dished head grid, such as in Figure 15 (e), there are instances in which grids are sloped to intentionally create a low point for solids withdrawal. This is particularly necessary in instances where undesirable agglomerates might form or where the bed solids represent the reaction product to be withdrawn, as for example in the gas phase polymerization of olefins. The more symmetrical slope to a central drawoff as illustrated in Figures 17 (a) and (b) is more common than the tilted grid which involves an elliptical pattern. Figure 17 (a) is simply a perforated cone; Figure 17 (b) represents a series of concentric overlapping cones in which the solids flow to their angle of repose when

Figure 16. Sloped and Rotating Grids

the fluidizing medium ceases and thus limit bed weepage into the plenum. Horizontal plates below each hole in Figure 17 (a) would serve the same purpose. Grids such as in Figure 17 (a) are in quite common usage. Since the pressure drop from plenum to bed surface is the same no matter what path the gas follows it is obvious that adjustments must be made in the size and/or number of holes in order to distribute the gas uniformly over the bed cross section. The deeper bed in the center of the vessel represents a greater restriction to flow than the progressively shallower depths approaching the vessel wall.

Figure 17 (c) represents an inverted form of that in 17 (b) intended to permit introducing more gas into a smaller bed area to maintain coarser segregating solids fluidized rather than stagnantly slumped; solids drawoff would be taken through ports in the vessel wall.

Figure 17 (d) is identical to 17 (c) but has in addition an interlaced set of lipped discs held together via a spider and supported from a central rotating shaft. Its purpose is the same as that of 17 (c) but in addition the continuously rotating elements presumably sweep the passages clean and thereby remove any accretions resulting from adhesive stagnated solids such as associated with polymerization processes. Rotating scrapers or stirrers whether acting on the grid or on the vessel walls, have been used industrially. The form shown in Figure 17 (d) has however not been employed in reactors greater than about 2 feet in diameter. The small clearances usually required with slotted type ports as in Figure 17 are not readily adaptable to high temperature reactors where thermal distortion can radically destroy gas distribution.

Stagnant versus Fully Active Grids

Flat Grids

Flat perforated plates are certainly the simplest and most

common form of grid plate, or gas distributor. On such plates there are inevitably stagnant regions between the perforations where solids might seldom be stirred into the bed mix. Inserts such as in Figure 15 (f) accomplish the objective illustrated in Figure 18, where no filter baskets were employed to protect the high pressure drop mesh insert. Flat bar-type grids, remi-

Figure 18. Mesh Inserts for Reducing Stagnant Areas

niscent of the Shell "Turbogrid" trays in absorption and distillation practice, have generally been abandoned. The clearances required to achieve the necessary gas distributing pressure drop are too small to remain constant under any reasonable temperature gradient, and are otherwise again prone to pluggage. It should be obvious that a round perforation, stretched into a long slot of equal area, becomes a rectangle with one very narrow side.

Non Fouling Grids

There are instances in which no position of the grid may

permit solids stagnation. This is particularly the case with solids which can become "sticky" or "tacky" under the operating temperatures. Figure 19 illustrates the initial stages in the progressive sintering of an entire bed atop flat grid with pipe cap tuyeres. The fluidizing gas finds its way through the bed via "rat holes" or "blow holes" while the overheated solids congeal. Such phenomena are not uncommon, and are best avoided by the use of grid plates which are active over their entire area keeping the solids continually in motion. The rotating concept in Figure 17 (d) represents one attempt to avoid such stagnation though a more common and simpler approach lies in fabricating the grid as a honeycomb of nearly touching cones or hexagons illustrated in Figure 20. Figure 20 (a) represents an elevation view of (b) or (c); (b) is a sketch made from a photograph looking down into a bricklined reactor (the lining and two thermocouple wells are discernable). The flat portion of the grid floor is minimized by the hexagonal conical depressions terminating in orifices in their apexes. The slope of the cones must exceed the solids angle of internal friction in order to remain free of possible accretion. Rather than fabricate half-cones at the periphery, so-called "dead men" were provided to blank the inactive grid floor.

The effectiveness of such conical grids in maintaining complete motion is evidenced by its use in the precise blending of

Figure 19. Cross - Section of the Sintered Buildup Surrounding a Tuyere Illustrating Initiation of Progressive Bed Sintering

Figure 20. Examples of Fully Active Non-Fouling Grids

uranium oxide fuel powders in apparatus such as illustrated in Figure 20 (c). In this application the fluidizing or mixing gas is admitted through a single downpointed nozzle in each cone; the mixed product is withdrawn through valves at the apex or each cone.

Cold Grids

Where operating temperatures are near ambient, the fluidizing gas is assuredly free of dust, mist, or compressor lubricant, absolutely uniform gas distribution is not paramount, beds are shallow and vessel diameters less than 24", such simple materials as fritted glass, sintered metal, porous polyethylene, porous ceramics and woven cotton belting have been successfully applied. If weeping and/or stagnation are tolerable then bars, perforated plates or most any other arrangement are also satisfactory.

Laboratory and Pilot Plant Grids

The choice of a pilot plant grid should to a reasonable degree be tempered by the system's predicted maximum stable bubble diameter. This is usually so large that a vessel less than 6" in diameter will likely be operating as a slugging reactor within a few inches above its grid. A single cone with an orifice in its apex may therefore be as effective as any more

sophisticated design; in scaleup to the industrial bed height it is frequently adequate to ignore the grid and bubble merger regions of the pilot plant and conservatively assume it to have been slugging from the bed surface down to the grid level. In larger diameter pilot plants the grid hole size, layout, and general design characteristics should preferably duplicate what is intended for the industrial scale unit; the pilot plant grid would essentially represent a piece of the industrial grid, in anticipation of minimizing scaleup calculations.

"Thick" Grids

It is commonly the practice to support thin gauge, flat, punched grid plates on I beams, much the same way as large diameter distillation or absorption towers. However, there have been instances in which it was judged expedient instead to employ thick self-supporting drilled plates. If a thick drilled plate is employed in a unit which is likely to be shut down in a manner in which the bed is allowed to drain into the plenum it may be impossible to reestablish homogeneous gas distribution with all holes blowing. As illustrated in Figure 21, upon start-up, after a shut down, the first increment of incoming gas will blow one of the holes clear. Succeeding increments will then face the choice of joining the earlier gas so as to blow through an already opened hole at a higher velocity or blow open another

Figure 21. Illustration of Potential Maldistribution on Startup
Following a Shutdown with a "Thick" Grid

hole. These succeeding increments see two types of holes—clear holes, and holes filled with short fixed beds of high resistance. The clear holes (despite the higher velocity) represent the path of lower resistance and hence when all the feed gas is again being fed to the unit some holes will still be filled with fixed bed plugs. The obvious solution is to countersink the holes as illustrated in Figure 22 so that

β = Solids' Angle of Repose

Figure 22. Countersunk Thick-Grid Ports

the incoming gas never faces two kinds of holes with different resistances. There have been instances in which such maldistribution as illustrated in Figure 21 (e) has been so severe as to reduce a previous conversion of 96% (Figure 21 (a)) to as low as 70%.

Shrouded Grids

Shrouds are generally placed around grid holes to reduce the velocity at the gas-solids interface, and hence the accompanying particle attrition. Shrouds on sparger type grids con-

sist simply of short pipes centered over the grid holes as illustrated in Figure 23 (a); on flat grids they appear in the inverted nipple form shown in 23 (b). To be effective, shrouds must be long enough to "contain" the expanding (11° included

Figure 23. Shroud Configurations

angle) gas jet leaving the grid orifice. This minimum length is defined in Figure 24; it would obviously be prudent to add an inch or two in practice.

$$L_{\min.} = \frac{D - d}{2 \tan 5.5^\circ}$$

Figure 24. Minimum Shroud Length

Grid Pressure Drop

As evidenced in Figure 3, whatever the form of the grid, it represents a multiple port distributor and hence must incorporate sufficient pressure drop to force an equal flow through each port. Maldistribution as illustrated in Figure 25 results in deeper penetration jets, larger bubbles, lower gas residence time, poorer conversion, higher entrainment, higher downstream pressure drop, plugged internal cyclone diplegs, and bed weepage through unaerated grid holes.

Figure 25. Illustration of Poor and Proper Grid Gas Distribution

Analogy to Distillation Trays

Some 40 years ago James Davies of The Texas Co. (now Texaco, Inc.) realized that a distillation tray is simply a means for

distributing vapors, from the plenum below, into a shallow bed of liquid flowing over rows of protruding bubble caps. The resistance the caps impose is reflected in a liquid gradient and hence the entering vapors would favor the less submerged (lower resistance) caps to the point where some rows would not only fail to bubble, but in some instances even begin to weep, as illustrated in Figure 26 (a). To force gas into the non-bubbling

Figure 26. Illustration of Poor and Properly Bubbling Trays

caps requires placing a resistance in the risers sufficient to overcome the incrementally greater resistance imposed by the deeper liquid pool. This necessary resistance was found to be 2.5 times the gradient. Despite the analogy to Figure 25 and

the fact that bubble caps are a common form of distributor the equivalent of a gradient is not immediately obvious in a fluidized bed. However, if one were to consider the downcomer of a distillation tray to consist of a rubber pipe moving randomly over the tray below creating randomly distributed gradients then one sees a more distinct similarity to the locally fluctuating bed heights as large bubbles burst at the surface of a fluidized bed of solids. The criteria for the tray or distribution grid that:

$$\Delta P_{Distr.} \geq 2.5 \Delta$$

might then better be expressed as a function of the maximum local depth:

$$\frac{\Delta P_{Distr.}}{\Delta P_{Bed}} \geq \frac{2.5 \Delta}{\Delta + L}$$

which for typical figures measured by Davies yields:

$$\frac{\Delta P_{Distr.}}{\Delta P_{Bed}} \geq \frac{2.5 (0.5)}{0.5 + 3.5}$$

or a grid pressure drop equal to at least 30% of the pressure drop through the bed (of fluidized solids or of liquid) above.

$$\frac{\Delta P_{Distr.}}{\Delta P_{Bed}} \geq 0.30$$

The Pressure Drop Equation

Apart from the analogy to bubble cap trays in distillation columns numerous experimental investigations have been carried out in industrial laboratories demonstrating in all instances quite conclusively the 30% of bed criterion. Historically it is interesting to note that some 40 years ago it was the practice to specify grid pressure drops on the basis of the following empirical equation, the source of which is unknown:

$$\begin{array}{l} \text{Minimum} \\ \text{Grid } \Delta P, \text{ psi} \end{array} = \frac{0.165(\text{tons above grid})}{\text{vessel diameter, feet}} + 0.00058 (\text{grid dish depth, in.}) \times (\text{bed density, lbs/cu.ft.})$$

This equation can be rewritten as:

$$\begin{aligned} \Delta P_{\text{psi}} &= \frac{0.165}{D} \left(\frac{\pi D^2 L \rho_B}{4 \times 2000} \right) + 0.00058 (D_d) (12) \rho_B \\ &= 0.00934D \left(\frac{L \rho_B}{144} \right) + \frac{D_d \rho_B}{144} \end{aligned} \quad (1)$$

where: D = vessel diameter, feet

D_d = dished head depth, feet

ρ_B = bed density, lbs/cu.ft.

L = bed depth, feet

The term $(0.00934D)$ represents in effect the per cent of bed ΔP . For a 30 foot diameter reactor equation (1) would call for a grid ΔP 28% that of the bed and in addition the maximum depth of bed in the dish if the grid happened to be simply a perforated dished head. For a 40 foot diameter reactor the 28% becomes

37%. These figures conservatively agree with the 30% criteria accepted at the present day. Equation (1) was never intended for small vessels and could lead to poor design in say a 10 foot diameter vessel where it would specify a grid with 9% of bed ΔP as adequate.

The inadequacy of equation (1) in the general case was dramatically demonstrated in the design of the ill-fated Hydrocol plant in the mid 1940's. This 17 foot diameter vessel bore a 14 foot deep bed of about 150 pounds/cu. ft. density, so that equation (1) predicted a required grid pressure drop a little over 2 psi. This appeared quite reasonable to the M.W. Kellogg engineers whose experience lay in FCC design (and who had left to join P.C. Keith, their former vice president of engineering, in the formation of Hydrocarbon Research, Inc.). A typical 20 foot diameter FCC reactor with a 35 foot deep bed of 40 lbs/cu. ft. catalyst also called for about a 2 psi pressure drop and hence 2 psi was considered nearly a universal criterion in itself. It became dramatically evident in the Hydrocol plant startup operations that this criterion was inadequate. Again, laboratory investigations in Plexiglas equipment demonstrated that the required distributing pressure drop is simply correlated as 30% of that through the bed above. As an additional safety factor it was suggested that 40% would be a universally "safe" criterion.

Operating under conditions wherein the grid pressure drop exceeds 30% of the bed pressure drop, simply means that for that particular gas throughput the grid is more restricted than necessary, jet penetrations will be deeper and initial bubble sizes larger. Operating under conditions wherein the grid pressure drop is somewhat less than 30% of the bed pressure drop, simply means that for that particular gas throughput the grid is not sufficiently restricted and that some holes are likely not blowing and quite possibly weeping. There is also the possibility that the grid holes may have become rounded or otherwise distorted so that their effective orifice coefficient may have changed. In any event it is seldom disastrous to operate at conditions corresponding to a predicted grid pressure drop as low as even 20% of the bed above; however, below this figure loss in performance becomes significant.

It is not uncommon to find operators taking measurements of grid pressure drop during operation and, if meeting the 30% criterion, to then assume feed distribution must be adequate. All too frequently such a measurement can be misleading. If the gas fed to reactor were not sufficient (or the grid too open) to provide the necessary stabilizing grid pressure drop, all holes would not bubble and hence some would drain solids into the plenum below. The feed gases would then pick up these solids and carry them back up through those grid holes

passing gas. Therefore, the measured pressure drop across the grid could easily reach (and would in fact seek) the level of 30% of bed pressure drop simply because all the holes are not active, and those which are active handle a cocurrent gas plus solids stream. The number of holes actually blowing will be the number required to satisfy the 30% of bed ΔP distribution criterion.

The grid pressure drop is calculable from the simple orifice equation:

$$V_o = 0.8 \sqrt{2g \Delta P / \rho} \quad (2)$$

where

V_o = velocity through grid hole, ft/sec

g = 32.2 ft/sec x sec

ΔP = grid pressure drop, lbs/sq.ft.

ρ = density of gas or gas plus solids suspension
flowing through the grid holes, lbs/cu.ft.

The coefficient of 0.8 is based on experiments conducted with fluidized beds and sieve trays. This coefficient was consistently obtained when a fluidized bed of solids or an aerated pool of liquid rested on the downstream side of a perforated plate.

Downflow (Sparger) Grids

Equation (2) pertains as well to downflow or sparger type grids in which the criterion for distribution can be decreased from 30% to as low as 6%, based on experimental observations, though 10% is the usual minimum for design purposes. In relatively low pressure high volume situations, such as FCC regenerators where horsepower is a premium, downflow grids such as illustrated in Figure 10 are preferred. The lower pressure drop criterion for downflow grids is attributed to the lessened opportunity for backflow of bed solids into the header much the same as the analogy to placing an empty bottle right side up or upside down into a tank of water. As obvious as this might appear and as well as confirmed experimentally, inevitably small amounts of bed solids creep up into a port to be carried the length of the header and blown out through another port further downstream. This causes substantial erosion, as illustrated in Figure 27. Increasing port velocities to prevent the solids backflow also increases the pressure drop, and approaches again the "30% of bed pressure drop" as a design criterion.

Supports and Seals

Grids are most commonly supported on rings welded to the vessel shell similar to the support provided for distillation

Figure 27. Illustrations of Sparger Erosion due to Solids In - Leakage

trays. In large vessels 20 to 60 feet in diameter the grid may be fabricated in sections fitting through a manway and welded together in place supported on steel I beams which span the vessel. If the feed gases are cool, and the grid plate covered with refractory, thermal expansion might not be a problem.

Where substantial expansion must be accommodated the seal at the wall may take the form of slots or dished arrangements such as illustrated in Figure 28. In place of I beams it is also not

Figure 28. Flat and Dished Grid-to-Vessel Seals

uncommon to find grid supports in the form of struts or legs tied to the bottom head of the vessel or even to the foundation in the case of a low pressure brick lined plenum resting on ground level.

Support and seal arrangements are strongly dependent on temperature, refractory lining, materials of construction, vessel diameter and similar mechanical and structural considerations which vary as widely as the designers ingenuity. In many instances these details are considered proprietary, though unpatentable knowhow, not to be divulged to public agencies or non-licensees.

Jet Penetration

Mechanism of Bubble Formation

Fluidizing gas entering through a grid hole effectively faces a bed with a cross sectional area equal to that of the perforation. Since the hole velocity (via equation (2)) is enormous relative to the superficial incipient fluidization velocity at which the interstitial frictional resistance equals the weight of the bed, the solids will be lifted or pushed upwardly forming a dome as illustrated in Figure 29 (b). The surface of the dome is greater than the area of the grid hole and hence the velocity through this surface is lowered, but if still in excess of the incipient fluidization velocity it will continue to push the solids upwardly as in Figures 29 (c) and (d) until the area of the jet penetration is such that the velocity through the interface reaches the incipient fluidization velocity. The moment it attempts to exceed this surface

Figure 29. Bubble Formation from Bed-Penetrating Gas Jets

area, so that the interface velocity would drop below incipient fluidization, the vertical walls of the penetration cannot be held back and they collapse or slide down into the void, toward the hole, and cut off the gas as a trapped bubble (having a diameter of approximately $P/2$), after which the process begins anew. The ultimate shape of the jet, as illustrated in Figure 29(d) approaches that of a flame, which analogously represents a limited surface-to-surroundings velocity dissipation. The jet takes on the form of a prolate spheroid of nearly constant ratio of major to minor axis ratio and hence could lead to dimensional prediction based solely on this ratio, the volumetric flow through a grid hole and the solids' incipient fluidization velocity. However, in practice penetration depths are generally determined from empirical correlations.

Correlations of Jet Penetration

The earliest and simplest proposed correlation of jet pene-

tration is shown in Figure 30. Visual observations (in this instance of P) are always somewhat open to interpretation and hence there is reasonable scatter of the data about the "best fit" lines. There is industrial evidence that the line for upwardly directed jets correctly predicted the jetting out of a 6" i.d. nozzle and out of a 45" I.D. uncapped riser blowing into fluidized beds. Experiments in which particle-in-gas suspensions up to loadings of 10 lbs/cu.ft., were directed upwardly into powder beds also showed agreement with Figure 30 basing nozzle velocity on the volumetric gas flow and the density on that of the suspension of gas plus solids. The lesser penetration of downwardly directed jets is attributable to the difficulty the bed finds in yielding when driven in effect into a vessel wall or bottom head. The upper surface of a bed interfaces with a fluid plenum into which it can easily lift. By analogy a ball buried in a bed of powder can easily be pulled up and out with an attached string, but it can almost impossibly be pushed down via an attached rod.

Figure 30 is intended to apply to situations in which the jetting fluid is also, and solely, the bed fluidizing medium, and in which the penetration depth includes only the contiguous fluid and not an additional height including the upper surface of the generated bubble.

TABLE I.—Legend for Fig. 30.

Jet		Bed		
Medium	Diameter (in)	Material	Particle Size (in)	Symbol
Liquid	0.436	Sand	0.002-0.08	■
Gas	0.076	Air	—	□
Gas	0.269	Water	—	⊗
Gas	0.75	FCC catalyst	0.002	×
	0.324	Mill Scale	0.0068	○
	1-2	Wheat	0.1-0.2	✱
	11	Coal	0.01-0.4	●
	1.17	Wood	0.75-1.5	△
	5-6	Coke	0.75-1.5	▲
Gas	0.118-0.236	Catalyst-Fe ore	0.008-0.06	▨

Figure 30. Jet Penetration into Fluid - Particle Media

Erosion and Attrition

Erosion and attrition are concurrent phenomena; if a particle impinges on a surface with sufficient force to cause erosion, it is likely that the particle itself will also suffer wear, or structural damage. It may be argued that damage might be minor if the particles are elastic, or major if highly friable. However, neither of these conditions are likely to be frequently encountered and in any event at best only considered an incidental advantage to designs oriented to minimize if not eliminate attrition entirely. Whether a material is erosive or erodable is sometimes surprisingly contrary to intuitive expectations; the severe erosion of steel cyclone cones by a ground green vegetable product is at first encounter an incredible example.

Jet Impingement

Within a fluidized bed, erosion and/or attrition are principally the result of jet impingement. The erosion of internal surfaces or the abrasive wear of bed solids resulting from their mixing and churning is usually negligible. Reactors with beds of highly abrasive solids containing vertical tubular internals are known to have been in continuous operation for as long as 14 years without requiring tube replacement. Though the bubbling

bed as a whole appears abrasive, the highest local bulk solids velocity, in the downflowing shell around a rising bubble, only equals the bubble rise velocity. This amounts to a mere 4 feet/sec for a 12" diameter bubble, 5.7 ft/sec for a 24" bubble, and 7 ft/sec for a 36" bubble. Experimental investigations of particle attrition in fluidized beds have shown no detectable effect of bed depth; whatever attrition occurs in the bed must take place only at the grid. Japanese physicians have reported therapeutic benefits from the immersion of whole live human bodies (heads excluded) in fluidized beds of spent coffee grounds; presumably the effect is more in the nature of a massage than a shave with a well-used razor.

The action of the jet apparently involves plucking bed particles from the perimeter of the orifice from which it issues, rapidly accelerating them in the direction of the jet, and impinging them on their own specie at the tip of the jet or on any object placed close enough to the orifice to be contacted by the bed-penetrating gas jet. As indicated in Figure 31 in an upwardly directed jet the particles picked up at the orifice edge travel a longer distance into a decelerated zone against an aerated yielding jet tip, whereas in a downwardly directed sparger jet the particles travel a shorter distance to then impinge on a non-yielding unaerated bed. The latter results in a "harder" impingement and hence in an expectedly greater degree of attrition.

Figure 31. Attrition Mechanism at Grid Ports

Rate of Attrition

The only quantitatively interpretable attrition rate data published to date pertain to a conventional zeolite silica-alumina FCC catalyst and to average 125 micron diameter glass beads correlated empirically by the relationship:

$$\phi = C (V_o \sqrt{\rho_G})^{2.5} n \frac{\pi D_o^2}{4} \quad (3)$$

where

ϕ = grams of o to d_p produced/min.

V_o = hole velocity, ft/sec

ρ_G = density of gas issuing from hole, lbs/cu.ft.

n = number of holes

D_o = hole diameter, feet

For downwardly directed jets issuing into FCC catalyst the following values of C were determined experimentally:

<u>Size Range, microns</u>	<u>C</u>
0 to 2	0.00017
0 to 23	0.00138
0 to 50	0.00349

test hole velocities ranged from 108 to 910 ft/sec and hole diameters from 1/32" to 3/4". All tests were carried out with ambient air. Upwardly directed jets resulted in values of C lower by a factor 2, glass beads resulted in values of C lower by a factor of 12, and exploratory tests with some coals, cokes and lignites gave values of C lower by a factor of about 10.

Attritors

Far severer, though quantitatively uncorrelated, attrition will result from impingement of the particles on a stationary metal object as opposed to the porous or yielding bed. Fluidized bed processes in which the bed particles are intended to grow (e.g. black liquor oxidation, evaporation of radioactive uranyl nitrate solutions, incineration of chemical byproduct waste slurries, fluid coking, etc.) are generally provided with intentional "attritors" to control particle size in the event that the bed grows too coarse and then defluidizes or slumps because of insufficient natural generation of seed fines from the slurry sprays. These attritors consist of either simple high velocity steam nozzles inserted through the vessel shell as in Figure 32 (a), or

Figure 32. In - Bed Attritors

the same arrangement with a thick target plate provided for particle impingement, as in Figure 32 (b). The rate of attrition from the latter is substantially greater than from the nozzles alone. A further increase in attrition is generally attainable via opposed ejectors in which solids are drawn from the bed into the suction sides of the ejectors and the two high velocity streams forced to impact upon each other, as illustrated in Figure 32 (c). This in effect doubles the impact velocity and should therefore result in at least a 12 fold greater rate of attrition than accomplished by simply one open nozzle as in Figure 32 (a). Such opposing ejector-pulverizers are near-standard equipment offered by commercial vendors.

Erosion of Internals

Erosion is principally the result of either high velocity solids impingement or the "cutting" action of metal upon metal. As illustrated in Figures 33 (a) and (b) spargers must have suf-

Figure 33. Cause and Avoidance of Erosion of Vessel or Internals

ficient elevation that their jets never reach a vessel head or other internal bracing, supports, drawoff wells etc. As illustrated in Figures 33 (c), (d), and (e), tubular internals, cyclone diplegs, etc. must similarly be elevated out of reach of grid jets, or else the jets shortened by reducing grid hole diameter. Equally severe erosion of tubular internals can occur by the cutting action of metal to metal contact. The masses of solids thrown about on the bed surface by the bursting of bubbles, coupled with the fluctuating bed mass above any unit area of grid cross section causes substantial buffeting forces on typical tubular internals. These must therefore not only be braced, but have that bracing tightly welded to the tubes as in Figure 33 (f) to avoid not only gaps where gas can pass through, carrying an erosive stream of particles, but also to avoid any relative rubbing motion of a loose metal brace against a tube wall. Braces such as illustrated in Figure 33 (f) are generally staggered throughout a reactor's internals rather than all located at equal levels. The penetrated caps in Figure 16 (a) would require tight fit spacers in the annulus between cap and riser to avoid tube erosion by lateral impingement on the grid hole risers.

Sample Problem in Grid Design

Your company has purchased a refinery from an independent operator. This refinery has among its units an old catalytic cracker. Compared to the fluid cracking units operating in your

company's other refineries, this plant suffers from extremely high catalyst losses out of the regenerator stack and exceptionally high oxygen concentrations in the gas samples drawn from the plenum above the bed. The size of the regenerator is such that they would expect to be able to operate with a feed of 74,400 SCFM of air and a catalyst circulation of about 33.6 tons/minute whereas they dare not exceed 65,300 SCFM and 29.5 tons/minute, since at this rate they already lose on the order of 8 tons of catalyst per day. Their first thought is that the cyclones are either poorly designed, of insufficient staging, or perhaps damaged. The possibility is also suggested that cyclone diplegs might not be sealed in the bed. From the equipment drawings, installation dates of cyclone replacements, previous turnaround inspections, operating history, and catalyst inventory, there is no basis for ascribing the poor performance to cyclone design or configurations.

The high plenum O₂ concentration would suggest extremely large bubbles rising through the bed as a result of perhaps mal-distribution at the grid. Recycle catalyst and air enter the regenerator below a flat plate grid. A study of this inlet configuration also reveals no obvious reasons to suspect local maldistribution of either catalyst or air flows. However, it is noted that the grid holes are 2" and 3" in diameter. These seem a bit large and lead to the suspicion that perhaps the grid has insufficient pressure drop to achieve good distribution of air and catalyst into the bed. In order to check this point, a careful test run is made and the pressure drop across the grid measured at several points with manometers and gauges. The results indicate the grid to have a measured pressure drop equal to about 23% of the pressure drop through the bed above. This is sufficiently close to the general criterion of 30% that it is not sufficient basis to claim that distribution is so poor as to account for the enormous losses and lack of oxygen utilization.

Having been presented with the above information and conclusions with which you cannot disagree, and all seem reasonable, you are now asked "What is wrong?"

1) Explain to your management via calculations (all of the necessary figures are shown in Figure 34) what is actually happening to gas and solids within this vessel to cause the observed difficulties.

2) Show what you would propose as a physical change in order to remedy the situation and via calculations show how this would reduce the catalyst losses from 8 tons/day to something approaching 2 tons/day without additional cyclones or other recovery equipment.

Catalyst bulk density in loosest state (e.g. at $V_{mf} \approx 23$ lbs/cu. ft.

Catalyst apparent particle density = 77 lbs/cu. ft.

Figure 34.

3) Under your proposed change in (2) what would you predict to be the size of the bubbles bursting on the bed surface?

Solution

1) If all the holes in the grid are blowing equally (as we could assume they might be since the measured grid ΔP is close to the 30% of bed minimum criterion) then presumably we should be able to match the observed ΔP by applying the orifice equation with a coefficient of 0.8:

$$\text{ACFM at grid} = 65300 \frac{0.076}{0.0358} = 138,200$$

Solids in feed gas: 29.5 tons/minutes

$$29.5 \times 2000 / 138,200 = 0.426 \text{ lbs of cat./ft}^3 \text{ of gas}$$

$$\text{Density of air \& cat. stream} = 0.426 + 0.0358 = 0.46 \text{ lbs/ft}^3$$

$$\text{Existing grid hole area} = \frac{700 \pi (3)^2}{4 \times 144} + \frac{1540 \pi (2)^2}{4 \times 144} = 67.9 \text{ ft}^2$$

$$V_o = 0.8 \sqrt{2g \Delta P / \rho}$$

$$\frac{138,200}{60 \times 67.9} = 0.8 \sqrt{2 \times 32.2 (\Delta P) / 0.46}$$

$$\therefore \Delta P = 12.92 \text{ lbs/ft}^2 = 2.48" \text{ H}_2\text{O} \leftarrow \text{vs } 11" \text{ H}_2\text{O measured!}$$

$$\frac{\text{grid } \Delta P \times 100}{\text{bed } \Delta P} = \frac{12.92 \times 100}{12.5 \times 20} = 5.16\% \text{ of bed}$$

far below minimum criterion!

Now to convince operators this is true despite their 11" measurement:

Non-blowing holes must be draining catalyst via $W = 1.2 \rho_B D_o^{1/2}$ since the bed above these holes is un-aerated. The draining catalyst is picked up by the feed gases and carried up through the "blowing" holes.

$$\frac{\text{lbs/min} \times \text{ft}^2}{60} = 1.2 \times 23 \times D_o^{1/2}$$

Since D_p is 60 microns, $D_o = 1.5 D_p$ is essentially D_o
Let $D_o = 3"$, then rate of draining is given as:

$$\text{lbs/min} \times \text{ft}^2 = 2860$$

Procedure:

Assume some number of grid holes not blowing and calculate the solids efflux rate from these holes; add this rate to the reactor feed-gas-plus-solids and calculate ΔP through the remaining blowing grid holes; if this does not equal the measured 11" H₂O then repeat with a different assumption. This will yield the fraction of the grid area that is inactive.

Trial 1. Assume 10 ft² of grid hole area is inactive (e.g. is draining) and 57.9 ft² blowing gas + solids:

$$10 \times 2860 = 28,600 \text{ lbs/min draining and returning}$$

The stream through the blowing grid holes has therefore a density of

$$0.0358 + \frac{28600 + (29.5 \times 2000)}{138,200} = 0.6708 \text{ lbs/ft}^3$$

$$\frac{138,200}{60(67.9-10)} = 0.8 \sqrt{2g \frac{\Delta P \times 0.03613 \times 144}{0.6708}}$$

$\therefore \Delta P = 5" \text{ H}_2\text{O} \leftarrow$ Too low, less than the 11" obsv'd.

Trial 2. Assume 23 ft² of hole area draining:

$$\text{density} = 0.0358 + \frac{(23 \times 2860) + 59000}{138,200} = 0.9388 \text{ lbs/ft}^3$$

$$\frac{138,200}{60(67.9-23)} = 0.8 \sqrt{2g \frac{\Delta P \times 0.03613 \times 144}{0.9388}}; \Delta P = 11.6" \text{ H}_2\text{O}$$

close enough to 11" H₂O obsv'd.

$\therefore$ about 22 ft² of grid hole area is inactive

$$\text{or about } \frac{(67.9 - 22)}{67.9} \times 100 = 67.5\% \text{ of grid is actively blowing; the remaining } 32.5\% \text{ of the hole area is "draining"}$$

- 2) Solution is to reduce the hole area, but not the number of holes (e.g. weld "plates" over each hole and drill smaller holes in these "plates"; the question is what hole diameter?

For the conditions in Figure 34, how much hole area is required to achieve good distribution?

$$V_o = 0.8 \sqrt{2 g \frac{0.3 \times 12.5 \times 20}{0.46}} = 81.98 \text{ ft/sec}$$

$$\frac{138,200}{60 \times 81.98} = 28.1 \text{ ft}^2 \text{ of hole area}$$

$$\text{existing number of holes} = 1540 + 700 = 2240$$

$$\therefore 28.1 \times 144 / 2240 = 1.806 \text{ in}^2 \text{ area of each hole}$$

$$\frac{\pi d^2}{4} = 1.806 \quad d = 1.516" \text{ (say 1.5")} \leftarrow \text{answer}$$

This will then mean that the superficial velocity in the bed will be the anticipated:

$$\frac{138,200 \times 4}{\pi (41.167)^2 \times 60} = 1.73 \text{ ft/sec}$$

whereas before (with only 67.5% of the grid blowing) it was effectively

$$\frac{1.73}{0.675} \text{ or } 2.56 \text{ ft/sec}$$

$$\frac{V^2}{g D_p \rho_p^2} \text{ is therefore now} = \frac{(1.73)^2}{32.2 (60/254000 \times 12) (77)^2} = 0.0796$$

Then from Figure 10.8 of Zenz-Othmer $W/V \rho_G$ or $e / \rho_G \approx 3$

$$\begin{aligned} \text{before the change in hole diameters: } V^2 / g D_p \rho_p^2 &= \left(\frac{2.56}{1.73} \right)^2 \times 0.0796 \\ &= 0.1743 \end{aligned}$$

$$\therefore W/V \rho_G \text{ or } e / \rho_G \approx 10$$

$3/10 \times 8 \approx 2.4$ tons/day now a likely loss assuming equal cyclone collection efficiency, TDH, etc.

$$3) \quad v \sqrt{\rho_f} \approx 81.98 \sqrt{0.0358} = 15.5 \quad \therefore \quad P/D_o \approx 4$$

$$\therefore P \approx 4 \times 1.5 = 6" \quad \varphi \quad D_{Bi} \approx 3"$$

$$\therefore \quad \frac{D_B \text{ at bed surface}}{D_{Bi} \leftarrow 3"} = 0.15 \frac{L_B}{D_{Bi}} + 0.85 = 0.15 \frac{12.5 \times 12}{3} + 0.85$$

and hence D_B at bed surface $\approx 25"$ ~~←~~ this should in practice be checked to see if it exceeds the system's inevitable maximum stable bubble diameter, but in this example no data were given for the particle size distribution (from which one could obtain the D_p present in the greatest number).

Cyclone Collectors

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Introduction

Voluminous literature exists on the theory and design of cyclones; the book by Jackson (1)* for example not only represents a very comprehensive review but also points out the frustrations of apparently diametrically conflicting results in the vast accumulation of observations. As in nearly every field of science, it is frequently impossible to retrace the details of old experiments when all the related conditions were not recorded because their significance was not realized at the time, or when several variables or dimensional ratios were altered simultaneously. It is therefore not surprising to still find an apparently never ending flow of academic research being conducted on this relatively simple equipment. Over the past decade, spurred particularly by environmental regulatory agencies, various industrial organizations have quite thoroughly reinvestigated most, if not all, aspects of cyclone design and apparently resolved them to their satisfaction; but this has not as yet appeared in journals open to the public. These data are principally in the files of operating, and retained research, companies and perhaps surprisingly not in the files of such vendors as Buell, Emtrol, Ducon, Van-Tongerren, Fisher-Klosterman, etc. whose resources, test facilities and motivations never match those of the major users of this equipment in the petroleum, heavy chemicals, and mineral industries.

Considering the enormous amount and variety of materials recovered

*Numbers in parentheses refer to entries in the list of "References Cited".

in these industries, via cyclone separators requiring collection efficiencies of the order of 99.99 +%, any advances in the art would be eagerly sought and adopted. Yet the most highly efficient units still represent the fundamentally simplest configurations suggesting that the many apparently more sophisticated innovations with multiple ports, auxiliary aeration, rotating internals, vortex stabilizers, etc. afford no performance superior to simple properly proportioned cylinders with tangential inlets. Cyclone design consists of defining a means of optimizing these proportions to maximize the recovery of entering particulates and predicting the recovery as a function of such operating variables as velocity, solids loading and pressure drop.

As illustrated in Figure 1, a conventional cyclone consists of a cylinder with a tangential opening through which a dust-bearing gas stream enters at high velocity, thus imparting a centrifugal force to the particles, usually on the order of several hundred g's of acceleration. This force drives them to the cylinder wall. The particles slide down the wall to an exit port while the spiraling gas stream turns around and exits from the opposite end.

In petroleum refineries for example, cyclones are principally used in the reactors and regenerators of fluid catalytic cracking (FCC) units, in fluid coking units, and following fluid

Figure 1.

bed incinerators handling refinery waste streams and sludges. As normally designed and installed in these applications, their ability to collect particles smaller than 40 microns decreases exponentially with decrease in particle size so that essentially everything smaller than 5 microns is almost always lost. Where the cyclone effluent gas eventually discharges to atmosphere, as in FCC unit regenerators, it is common practice to employ two to

three stages of cyclones in series. Were it not for the constraints of pressure drop, erosion, particle attrition, and cost, cyclones in these applications could be designed according to experimentally corroborated theories to be significantly better in their collection of particles smaller than 5 microns. Subsequent examples based on other industrial data illustrate the collection of particles as small as 1 micron with reasonable efficiency consistently obeying the established blend of basic theory and associated empirical correlations discussed on the following pages.

Basic Theory

Fundamentally cyclones are simply particle settling chambers made more effective by increasing the particle's density through an applied centrifugal force. A particle with a density of 1 gram/cu. cm., as conventionally measured by water displacement and a laboratory balance, would be heavier by a factor of 56 if placed in an acceleration field 56 times that existing in the ambient atmosphere on the surface of the Earth. This could be exerted on the particle by introducing it at a velocity of 60 ft/sec tangentially into a 2 foot radius cylinder or essentially a 4 foot diameter cyclone ($v^2/2g = 60 \times 60/2 \times 32.2 = 56$). This viewpoint forms the basis of the theory for sizing and predicting the performance of cyclone collectors.

The "Cut-Point" Particle Diameter

Assuming that gas moves through a cyclone as a rigid spiral maintaining a constant velocity equal to the average velocity at the cyclone inlet, the distance travelled by the gas is the cyclone barrel perimeter, πD_B , multiplied by the number of spirals traversed, N_s . The travel time or residence time within the cyclone is then given by

$$\pi D_B N_s / v_i$$

where v_i is the gas velocity at the cyclone inlet. In order to reach the barrel wall, or be completely settled out within this period of time, a particle would have to travel a maximum net horizontal distance L_W across the width of the downwardly spiraling gas stream. It would encounter a resistance in this fall to the wall assumed to be definable by the drag coefficient of the centrifugally weighted particle. If it is also assumed that the particles are always falling in laminar or Stokes Law flow then this net particle velocity can be represented by

$$v_t = \left(\frac{v_i^2}{D_B/2} \right) \frac{D_p^2 (\rho_p - \rho_g)}{18 \mu_g} \quad (1)$$

or more generally over a broader range from $Re = 0.0001$ to $Re = 100,000$ by:

$$v_t = \frac{2.25 \left(\frac{v^2}{\mu} \right) D_p^2 (\rho_p - \rho_g)}{D^{1.425} \left[4 \left(\frac{v^2}{\mu} \right) \rho_g (\rho_p - \rho_g) \right]^{0.475} \mu_g^{0.05} + 36.38 \mu_g} \quad (2)$$

Equating residence time to free fall time as

$$\frac{L_w}{v_t} = \frac{\pi D_B N_s}{v_i} \quad (3)$$

and substituting equation (1) or (2) for v_t allows solving for the smallest particle diameter which could reach the barrel wall in the available time. Using equation (1) this results in the familiar expression:

$$D_{P_{th}} = \sqrt{9 \mu_G L_w / \pi N_s v_i (\rho_p - \rho_G)} \quad (4)$$

which defines the so-called cut point or the theoretically smallest completely collectable particle size (2).

Equation (4) could be regarded as an over simplification on at least the bases that:

1) It is most likely applicable only to very dilute particle-in-gas streams where each particle acts individually.

2) It is not clear how N_s is determined.

3) It is not immediately obvious that the centrifugally weighted particle always lies in the Stokes Law region and hence substitution of equation (2) in equation (3) might lead to a more

realistic relationship.

4) It is not likely that the spiralling velocity is constant at v_i along the entire spiralling path.

Dust Concentration

Certainly point 1) is well taken. Unfortunately there have not been substantial data reported under such conditions. In fact this begs the question as to how dilute a suspension needs to be in order that one can disregard any particle interactions. It is extremely difficult to obtain reliable and accurate cyclone efficiency data without sampling the overhead or loss stream, and if the feed suspension must be very dilute it becomes obvious that any single test would require a long and tedious steady state continuous iso-kinetic sample collection from the overhead or loss stream. The effect of inlet dust concentrations, determined experimentally and then empirically related to the cut-point diameter of equation (4), with simultaneous correlations of path length and fractional efficiency for dilute loadings, is presented in subsequent sections specifically devoted to these quantitative performance correlations, and their illustration via sample calculation procedures.

Length of Path

Since most performance tests are conducted in steel cyclones

it is not possible to observe or measure values of N_s . Since the number of spirals can be expected to vary with one or another operating condition it becomes a trial and error within a trial and error procedure to derive a consistent correlation between observed collection efficiencies, equation (4), and the prediction of N_s , assuming that the performance data entering the analysis all pertain to sufficiently dilute suspensions to avoid so-called loading effects. If such is not the case, then the loading effect must be entered into the correlation analysis as another integral trial and error component. There are few, if any, publications other than chapter 11 of API publication No. 931 (3) in which an author has presented the results of such a rigorous integrated trial and error approach.

Stokes Law Limits

Limiting the Stokes law region to $Re < 0.3$ and a corresponding drag coefficient of 80, it can be shown that equation (4) is valid as long as

$$\frac{v^2}{\mu} < \frac{5.4 \mu_G^2}{\rho_G D_p^3 (\rho_p - \rho_G)} \quad (5)$$

which is satisfied almost universally. Consider for example a situation in which $\rho_p = 80$ lbs/cu.ft., $\rho_G = 0.05$ lbs/cu.ft., $\mu_G = 0.03$ centipoises, v is taken constant at an inlet velocity of 60 ft/sec and the radius of the cyclone, r , is 2 feet. Equation (4) would predict a D_{pth} value of 16.7 microns and equation

(5), would suggest that equation (4) based on Stokes law is valid as long as it predicts D_{pth} less than 20.5 microns. Therefore it is not surprising to find that substitution of equation (2) in equation (3) also leads to D_{pth} of 16.7 microns, since in the Stokes law region equations (1) and (2) should give essentially identical results, any differences merely reflecting the inaccuracy which equation (2) suffers as a consequence of fitting a broader range of Reynolds numbers extending into the Newton's law region as well.

Velocity Decay

It is certainly reasonable, and in fact necessary, that the spiralling velocity decay as it progresses down the length of the barrel in the region occupied by the exiting vortex. This raises the question of the length of the exiting vortex and whether any quantitative deductions can be made as to the rate of gas diversion into this vortex.

From studies in small cyclones Alexander (4) deduced an empirical relationship for vortex length as:

$$\mathcal{L} = 2.3 D_O (D_B^2/A_I)^{1/3} \quad (6)$$

In his conventionally proportioned cyclones gas outlet and inlet areas were nearly equal, L_W equaled $D_B/4$, D_O equaled $D_B/2$ and h equaled $3 L_W$, so that by substitution in (6) $\mathcal{L}$ reduced to

$$\begin{aligned}
 \mathcal{L} &= 2.3 D_O (D_B^2 / [(D_B/4) (3 D_B/4)])^{1/3} \\
 &= 4 D_O \\
 &= 2 D_B
 \end{aligned}$$

In industrial practice $\mathcal{L}$ has been deduced on the basis of observed severe erosion at the apex of the cyclone cone when the cyclone is too short and the vortex touches the cone wall. These observations can be empirically related to the ratio of outlet to inlet area and as summarized below in Table 1, show reasonably good agreement with equation (6).

Table 1. Comparison of Vortex Length Determinations

<u>h/L_w</u>	<u>A_O/A_I</u>	<u>$\mathcal{L}$/D_B</u>	<u>Source</u>
2.5	1.3-1.5	1.75	A.P.I. (3)
3	1.0	2.0	Alexander (4)
2	0.8	2.1	A.P.I. (3)
2	0.6	2.6	A.P.I. (3)
2	0.4	3.2	A.P.I. (3)

If the distribution of the gas inflow along the length of the vortex is considered proportionately constant in all cases, then it becomes of interest to examine whether the average inflow velocity over the surface area of the vortex might also be constant.

The cyclone outlet tube area = $\pi D_o^2/4$ (7)

The cyclone inlet area = $(\pi D_o^2/4) / (A_o/A_I)$ (8)

But also the inlet area = $L_W \left(\frac{h}{L_W} \right) L_W$ (9)

So that equating (8) and (9) and solving for L_W :

$$L_W = \sqrt{\frac{\pi D_o^2}{4 (h/L_W) (A_o/A_I)}} \quad (10)$$

The ratio of barrel to outlet tube diameter =

$$\frac{D_o + 2 L_W}{D_o} \quad (11)$$

Substituting equation (10) for L_W :

$$\frac{D_B}{D_o} = 1 + \frac{2 \sqrt{\frac{\pi D_o^2}{4 (h/L_W) (A_o/A_I)}}}{D_o} \quad (12)$$

$$= 1 + \sqrt{\frac{\pi}{(h/L_W) (A_o/A_I)}} \quad (13)$$

Therefore:

$$\frac{\mathcal{L}}{D_o} = \left(\frac{\mathcal{L}}{D_B} \right) \left(1 + \sqrt{\frac{\pi}{(h/L_W) (A_o/A_I)}} \right) \quad (14)$$

If V_v represents the average gas velocity into the surface of the cylindrical vortex then

$$\frac{V_v}{V_o} = \frac{D_o^2/4}{D_o (\mathcal{L}/D_o) D_o} \quad (15)$$

$$= 0.25 / (\mathcal{L}/D_o)$$

$$= \frac{0.25}{\left(\frac{\mathcal{L}}{D_B}\right) \left(1 + \sqrt{\frac{\pi}{(h/L_W)(A_o/A_I)}}\right)} \quad (16)$$

Substituting values of A_o/A_I , h/L_W , and $\mathcal{L}/D_B$ from Table 1 results in the figures shown in Table 2.

Table 2. Calculated Average Vortex Inflow Velocities

A_o/A_I	h/L_W	$\mathcal{L}/D_B$	V_v/V_o	ft/sec		
				V_i	V_o	V_v
1.5	2.5	1.75	0.075	80	61	4.55
1.0	3	2	0.062	80	80	4.95
0.8	2	2.1	0.050	80	100	4.96
0.6	2	2.6	0.037	80	133	4.93
0.4	2	3.2	0.026	80	200	<u>5.24</u>

Avg: 4.93

Within the accuracy of the determinations of $\mathcal{L}/D_B$ it is difficult to draw indisputable conclusions, but it would appear that $\mathcal{L}$ could be determined for any cyclone simply on the basis

of a 5 ft/sec velocity into the surface of an exiting vortex having a diameter equal to that of the gas outlet tube.

The significance of the decline in spiralling gas velocity based on a constant velocity into the exiting vortex (in comparison with the assumption of a constant spiralling velocity as in equation (1)) can be gleaned from the example illustrated in Table 3. The terminal velocities in column 5 of Table 3 were calculated by assuming a cut point particle diameter, and its correct choice based on showing the last column to add up to the specified width, L_w , of one foot. In comparison to the D_{pth} of 19.6 microns from Table 3 the particle size predicted from the simple form of equation (4)

$$\begin{aligned}
 D_{pth} &= \sqrt{9 \mu_G L_w / \pi N_s V_i (\rho_p - \rho_G)} \\
 &= \sqrt{9(0.03)(0.000672)(1) / \pi 4 \times 60 (80 - 0.05)} \\
 &= 0.0000548 \text{ feet} \\
 &= 16.7 \text{ microns}
 \end{aligned}$$

illustrates that: a) the sophistication of accounting for velocity decay is not of enormous significance (especially in relation to the precision in measured performance data), and: b) the initial one or two turns have the greatest effect. This

latter observation points up the basis for the success of so-called "skimmers", "tangentials", "rough cut" cyclones, "short contact time" separators, etc., employed in various industrial sectors, all of which amount to essentially single-turn centrifugally-weighted particle settling chambers or cyclones. Though the relationships used in Table 3 could be integrated and solved analytically the results would not differ significantly and would in fact display even more prominently the rapid decline in the effectiveness of additional spirals.

Table 3 Predicted Theoretical Cut Point Particle Size
Based on A Spiralling Velocity Decay

Bases: Particle Density = 80 lbs/cu.ft. $L_W = 1$ foot
 Gas Density = 0.05 lbs/cu.ft.
 Gas Viscosity = 0.03 centipoises Inlet Veloc. =
 60 ft/sec
 $N_S = 4$, $D_B = 4$ ft.

Result: *corresponds to $D_{pth} = 19.6$ microns

Cumulative No. of Traverses of the Barrel Diameter	V_s ft/sec	$\frac{D_{BN_s}}{v_s}$	$\frac{v^2}{r} = \frac{v_s^2}{0.75(D_B/2)}$	V_t^* , From Eqn. (2), ft/sec	$\frac{D_{BN_s} V_t}{v_s}$
1	60	0.209	2400	2.162	0.452
2	41	0.306	1121	1.046	0.320
3	23	0.546	353	0.341	0.186
4	5	2.50	17	0.017	0.042

$$: 1.000 = L_W$$

Dimensional Considerations

Inlet Configuration

Since particles must reach the barrel wall in order to be collected, recovered, or separated from the inlet gas stream, it is desirable to maximize the number of particles already nearest the wall at the very beginning or inlet to the cyclone. A rectangular inlet as shown in Figure 2B is therefore preferable to the circular inlet of Figure 2A. The circular inlet is tangent to the barrel at only one point, whereas the rectangular inlet is tangent along approximately one third of its perimeter.

Figure 2A

Figure 2B

There is ample evidence among published and proprietary data that under comparable operating conditions tangential entries are preferable to volute or "wrap-around" inlets or to axial flow units such as illustrated in Figure 3. Fractional efficiency curves for all three types are presented in the section on performance correlations.

Figure 3 Common Cyclone Inlets

Gas Outlet Tube Penetration

Since the entering gas spiral must not only force particles to the barrel wall but must also give them an initial downward component, the outlet tube must protrude down into the barrel in order to prevent some portion of the inlet particle-bearing gas stream from bypassing toward the gas exit. As illustrated in Figure 4, the optimum length of the gas outlet tube has been found experimentally to be in the range of 1.0 to 1.2 times the height of the inlet duct. When the outlet tube is less than 80 percent of the height of the inlet duct, the loss in efficiency (or bypassing) becomes so great that it is frequently even visually distinctly detectable. Lengthening the gas outlet tube much beyond the height of the inlet duct not only serves no purpose but in fact can be detrimental to cyclone performance unless the barrel or cone are simultaneously lengthened. The distance from the bottom edge of the gas outlet tube to the point at which it's extension would touch the cyclone cone, or reach its apex, must at least equal the natural vortex length if par-

TOO LONG AN OUTLET
TUBE SHORTENS THE
EFFECTIVE LENGTH OF
THE CYCLONE

TOO SHORT A
GAS OUTLET
TUBE PERMITS
BYPASSING

THEREFORE THE OPTIMUM
LENGTH OF OUTLET TUBE IS
TO THE BOTTOM OF THE INLET
DUCT OR AT MOST 20% LONGER

Figure 4.

ticle reentrainment and/or erosion are to be avoided. There are instances in which the outlet tube is intentionally made shorter than the inlet duct height; this is frequently practiced in medium size internally lined units to permit easier access through the inlet duct when it is necessary for a man to crawl into the cyclone to replace or repair liner materials. The loss in performance attending too short an outlet tube penetration has been incorporated in design correlations; the loss in performance attending too deep a penetration has not been quantified, but is rather averted by maintaining the cyclone length at least equal to the natural vortex length.

Barrel Roof and Annulus

With the inlet gas flow directed to give a downwardly flowing spiral, there exists a relatively quiescent though spinning volume of gas between the top of the barrel and the

upper edge of the inlet duct. Particles accumulating in this region tend not only to be erosive but to slowly migrate across the barrel roof and down the outside of the gas outlet tube. At the edge of this tube they are drawn into the exiting gas stream.

The "shave-off" duct of the Van Tongeren (5) design was originally suggested to cope with syphoning off this particle accumulation and bypassing it externally to a point of reentry at the barrel periphery well below the gas outlet duct. Observations in plastic models as well as operational units indicate that a tangential inlet duct whose upper surface is flush with the cyclone roof, as illustrated in Figure 5, eliminates both the erosive accumulation and the migratory loss of particles.

As an alternative to the shave-off duct, a once popular concept involved a difficulty fabricated, downwardly sloping spiral roof. Although at first this seemed to be a logical solution, it was inevitably realized that so directing the gas stream reduced the number of spirals or path length, as well as the residence time within the cyclone. As a result, the potential collection efficiency is reduced and hence the helical roof concept has generally been abandoned.

The entering particle-bearing gas stream should have an unimpeded path to the barrel wall. Any interference that would cause particles to bounce and/or ricochet can lead to their es-

Figure 5.

cape in the exit gas. The gas outlet duct should therefore also be small enough, or the cyclone barrel large enough, to maintain an annular width around the gas outlet tube equal to at least the width of the inlet duct, as illustrated in Figure 6.

Figure 6.

Inlet Duct Dimensions

The contraction of the gas stream leaving the barrel through the gas outlet duct represents a pressure loss that should be minimized if possible. For a fixed barrel diameter, the annular width should be at least equal to the width of the inlet duct thereby establishing the outlet tube diameter, D_o , as $(D_B - 2 L_w)$ as illustrated in Figure 6. Since it is desirable to maximize the number of particles entering nearest the barrel wall and to

minimize the distance L_w that those particles farthest from the barrel wall must travel to reach it, the dimensions of the inlet duct are preferably a ratio of h/L_w greater than 1.0, as illustrated in Figure 7.

Figure 7.

As in free fall through an expanse of fluid, the entering dust particles encounter aerodynamic surface friction or drag as they migrate through the spiraling inlet gas stream toward the barrel wall. In order to promote their reaching the wall, L_w should be minimal. However, their residence time or the number of spirals within the barrel should not be sacrificed. Therefore, a longer cyclone would be necessary. Besides pressure drop, reentrainment, and interference from possibly overlapping spirals, practical considerations of body length dictate the existence of some reasonable limit of h/L_w . Experience has shown that the principal areas of pressure loss through cyclones exist at the inlet and in the contraction from the barrel to the gas outlet tube. Imposing a criterion of outlet pressure drop not exceeding the inlet pressure drop (essentially equal inlet A_I and outlet A_o areas) for any chosen L_w in conjunction with

the dimensional relationships in Figure 6 leads algebraically to combinations such as:

L_w :	$0.2 D_B$	$0.25 D_B$	$0.28 D_B$	$0.3 D_B$
h :	$7.0 L_w$	$3.0 L_w$	$2.0 L_w$	$1.4 L_w$
D_o :	$0.6 D_B$	$0.5 D_B$	$0.44 D_B$	$0.4 D_B$

It becomes obvious from the above that L_w should be at least $0.25 D_B$, as illustrated in Figure 8 leading to the dimensional ratios commonly acceptable for a first design trial.

Figure 8.

Cyclone Length and A_o/A_i Ratio

Extensive experimental investigations (6) have shown that under the acceleration field existing in all high efficiency cyclones, the entering annular, downwardly spiralling, gas can be

regarded as analogous to flow in a coiled duct or pipe. Therefore, it is subject to velocity gradients at the barrel wall that may or may not permit contained dust particles to make contact with the wall. The spiralling gas velocity must be low enough to permit such contact. This limiting or maximum allowable velocity is calculable as the analogous saltation velocity in pneumatic conveying. Without decreasing a cyclone's gas capacity or inlet velocity, the spiralling annular velocity can be reduced (if it is shown to exceed the corresponding saltation velocity) by enlarging the width of the annulus. It may more conveniently be reduced by simply reducing the diameter of the exiting gas spiral (for example, by reducing the effective diameter of the gas outlet tube).

Such necking down or "modification" of the gas outlet tube to an A_O to A_I ratio less than unity is accompanied by an increase in both the exit gas pressure drop and the depth of penetration of the exiting gas spiral. The effective length of a conventional cyclone of $A_O/A_I \geq 1.0$, as defined in Figure 9, has been shown experimentally to be 1.6 to 1.75 D_B . As A_O/A_I decreases, the cyclone must be lengthened to avoid dust reentrainment at the base of the exiting spiral.

Figure 9.

Cyclone Shape

As implied in Figure 10, the existence of a conical body portion in large cyclones, or the common hopper of a multicyclone cluster, is principally a convenience to gather the collected solids to a central discharge point. Aside from the cost of the barrel pipe alone, the area loss would, for example, prohibit extending a whole body diameter as a solids return pipe down into a fluidized reactor bed. In principle, the configuration in Figure 10A is equal in performance to that of the more common, practical hopped vessel in Figure 10B. However, the 10A configuration, or one approaching it, is sometimes a necessity. It is also the form more common to small diameter, multiclone arrangements where many barrels feed through a tube sheet into a common hopper. The cone angle of 60 degrees in Figure 10B is a conventional average suitable to very free-flowing solids. This can be a very important detail, depending upon the nature of the collected solids. Even under conditions using the dimensional ratios

Figure 10.

in Figure 9, theory supports the reasoning that the longer the barrel or body, the greater the possibility of maximizing the number of spirals traversed by the entering dusty stream, and hence the greater the anticipated cyclone efficiency. Referring to Figure 10, arrangement A is therefore theoretically the ideal. Since it is more practical to funnel solids to a central point, arrangement B would be considered superior to C, as it maintains a greater length for active spiralling. However, the choice between B and C also involves the properties of the collected solids.

If the collected solids tend to be sticky or simply have an angle of repose or angle of internal friction approaching 90 degrees, arrangement B would be unsuitable. By the time the collected solids reached the dipleg in arrangement B, their spiralling velocity might have slowed to such a rate that they would begin adhering to the cyclone wall and then quickly plug

the dipleg opening. In such a case, the slightly less efficient design C would provide continuous operation and would be the proper choice. It would be far better to operate continuously at for example 90 percent efficiency than to operate half of the time at for example 92 percent efficiency, particularly when one considers the cost and labor of periodic cleanouts. It is not uncommon to encounter this difficulty since the finer the material to be collected, the less its usual free-flow behavior. Therefore, theoretically, it appears that a greater number of spirals will achieve the highest collection efficiency. However, as pointed out, it is far more efficient in the long run to use a design having fewer spirals for such materials.

If the collected particles are not only such that would tend to adhere and plug arrangement B but are also quite erosive in nature, then arrangement C could fail by wear. It is not uncommon to find 1/2 inch deep spiral grooves cut through cone metal within a few weeks of operation. In such instances it can be effective to use the expedient of welding a flat bottom cylinder over the worn cone, creating essentially the arrangement shown in Figure 10D. In this configuration the collected solids form their own cone, and the spiralling particles roll against a cone of their own specie rather than against the cyclone body metal. If even a slight increase in particle attrition is intolerable, or if the gaseous media carrying the solids is corrosive (or potentially corrosive if allowed to lie stagnant and perhaps alter the solids' composition therein) then arrangement D can also

fail. The alternative in such an instance is to increase the cone angle, as in Figure 10B, to again reduce the solids' spiralling velocity, and then interrupt the erosive horizontal component of their velocity by dropping them into a vertical section as shown in arrangement E. There, the increased downward component along the vertical walls allows the particles to more readily pass through the transition into the dipleg.

The configuration in Figure 10E is commonly in evidence in modern FCC units. The location of a dust trap or "radish", such as shown in Figure 1, between the vertical section and the dipleg is a redundant carryover from earlier designs. At one time the dust trap served as an enlarged section to hold a "vortex breaker" or shield plate that protected the dipleg opening from the exiting spiral in cyclones of insufficient length. It still frequently appears only because of the conservative nature of change.

Dipleg Size

The dipleg can simply be regarded as a pipe stuck down into a fluidized bed or as a connection to a particle collection bin. Solids are poured into the top of this pipe at varying rates depending upon their feed rate to the cyclone. The minimum specified size of the pipe is generally based on the simplified (7) relationship for solids flow by gravity into an opening as given

by equation (17):

$$W_s = 1.2 \rho_B d_o^{1/2} \quad (17)$$

where W_s = solids flow rate, lbs/sec x sq.ft. of pipe area

ρ_B = solids bulk density, lbs/cu.ft.

d_o = dipleg, i.d., inches

This relationship, analogous to the Francis weir equation, is based on a vast amount of data and is sufficient for design purposes where the particle diameters are much smaller than the orifice diameters. The details of its derivation and its application to other fluids (gases and liquids) in a dimensionless form has been demonstrated experimentally (8).

Rearrangement of equation (17) solved for diameter yields:

$$d_o^{2.5} = 153 w / \rho_B \quad (18)$$

where w = solids collection rate, lbs/sec

d_o and ρ_B as in equation (17).

It has been argued that Equation (17) is too conservative since solids do not enter the top of a dipleg pipe in choked bulk flow. This is easily demonstrated (9) by an experiment such as illustrated in Figure 11. If solids are dumped instantaneously into a funnel (the funnel representing the cone of a cyclone), they will drain out of its bottom nozzle (representing

the dipleg pipe) at a rate calculable from Equations (17) or (18). If, however the solids are poured into the funnel at a rate such that the bottom opening is not choked with solids (such as is the action in an actual cyclone cone), they can pass through the funnel at a rate several times as great as that calculable from either equation.

Figure 11.

The question, therefore, arises as to why Equation (17) should be used for sizing dipleg diameters since it must be conservative, if one is to believe the easily demonstrated results of Figure 11. The answer presently lies simply in a compromise with unknowns.

When the dipleg is submerged in a fluidized bed it will contain a level of collected solids somewhat above that of the bed itself and hence it usually represents a parallel bed of

relatively high length to diameter. The degree to which this bed within the dipleg will operate in slug flow is usually unknown, particularly in view of the variable conditions at its bottom end. There have been instances where insufficient height was allowed within the dipleg (between its calculated solids level and its upper end, where it is attached to the cyclone cone or dust hopper) for bed slugs to shower down and be dissipated. Thus, the slugs were thrown back up into the cyclone cone. In such cases, these solids must flow back out under the rate given by Equation (17). The slug thrown back up into the cone is analogous to the instantaneous dumping of solids into the funnel of Figure 11. The effect of such a reverse slugging of solids into the cone is equivalent to partially or completely removing the affected cyclone from the system until such time as the cone can drain. As a rule of thumb, it has been suggested that a distance of 7 feet always be allowed between the cone bottom and the calculated solids dipleg level to allow slug dissipation before reaching the cone.

Though Equation (17) might be conservative in most cases, at the same time it does not lead to excessively large dipleg diameters. In fact, in many cases, it would call for third-stage diplegs of perhaps only an inch in diameter. In practice, it is not considered advisable to have diplegs smaller than about 4 inches inside diameter to minimize the possibility of plugging with defluidized solids. It has been demonstrated in small

scale Plexiglas units that, for example, a 10-foot long 1-inch inside diameter pipe fitted with a trickle valve or "dollar plate" at its lower end submerged 3 feet into a fluidized bed of FCC catalyst will barely, if at all, expel its contents. If the L/D (the ratio of filled dipleg length to the dipleg inside diameter) is large, the diameter too small to allow dipleg aeration without excessive slugging, and the solids residence time long enough to allow self-deaeration, the dipleg can effectively plug. Correlations of dipleg L/D versus capacity at incipient plugging have been proposed. However, the exploratory investigations have been meager and leave much to be desired in the way of a final, wholly logical correlation.

Presently, Equations (17) or (18) with a 10 percent safety factor can be confidently applied where the calculated diameter is greater than 4 inches. A clear dipleg height of at least 7 feet and a minimum inside diameter of 4 inches suffices in most instances.

Performance Correlations

Even prior to the establishment of such regulatory agencies as the EPA, the petroleum and heavy chemical industries represented those most involved and most contributive to the establishment of a comprehensive correlation between cyclone theory and practice, particularly in application to the control of carryover from flu-

idized bed operations. Laboratory experimentation in this field is extremely difficult and tedious, since it requires investigations with particles in the 0 to 40 micron size range; larger sizes are too easily collected and hence do not contribute to extrapolation into the region of practical interest. Simultaneously, an apparent substantial effect of loading requires long duration testing in order to determine quantitatively reliable figures. Past industrial studies have therefore relied principally upon matching cyclone operation and efficiency to measured plant losses and calculated entrainment loads. To establish any consistently comparable figures this requires in addition to knowledge of the fluid properties and flow rate:

- a) the particle carryover rate (cyclone loss rate)
- b) the carryover analysis based on aerodynamically equivalent spherical diameters
- c) the apparent particle density
- d) all cyclone dimensions
- e) the particle entrainment rate entering the first stage cyclones
- f) the particle size analysis of the entrainment, again based on aerodynamically equivalent spheres.

Since equilibrium catalyst particles are very nearly spherical, items a), b), c) and d) can be reasonably well established; however, e) and f) are as yet not accepted as being indisputably predictable. Even if they were, the determination of a fundamental fractional efficiency curve would require simultaneous trial and error procedures to establish a correlatable loading correction and a means of predicting the effective spiral path length, or N_s , within the cyclone. Such interrelated trial and error procedures, based on laboratory data, detailed testing of industrial installations, and in some instances on the best available entrainment correlation (10) (11) (12) (when inlet sampling was not feasible) were employed to arrive at the performance correlations presented here.

Particle Shape

All theories relating to any aspects of particle dynamics are always based on treating the particulates as perfect spheres. Particle sizes determined by techniques other than sedimentation represent some physical dimensions which are all too frequently incorrectly accepted as the diameters of spheres. It would obviously be incorrect to expect a cylindrical particle 1/4" long by 60 microns in diameter to behave aerodynamically like a 60 micron sphere simply because it could pass through a 60 micron opening in a screen. Unfortunately there is again a lack of universal agreement on means of determining shape factors and

for that matter whether they even apply to very small particles.

A recent paper (13) presented data intended to show that small irregular particles do not deviate significantly from spheres in their settling rates. Unfortunately this is not in accord with observations on cyclone collection efficiency, from which it would rather appear that the shapes of all sizes of a particular material, or grind, are the same; in other words that the shape factor determined for particles in at least an observable size range, if not for the very largest, is applicable down to the smallest.

In any case this raises the question of the most appropriate and/or definable shape factor. For the present purposes, and in accord with the subsequent industrial examples, the shape factor will be defined as the diameter of a sphere of equal density (calculated from an experimentally observed settling velocity and the standard drag coefficient versus Reynolds number correlation for spheres), divided by the characteristic physical dimension resulting from any of the conventional methods of particle size analysis. This approaches most closely the ratio of particle volume to volume of a sphere having the same surface area as the particle. The interrelationships between the variously defined shape factors and their estimations from solely physical dimensions are summarized in a number of publications (14).

N_s and Fractional Efficiency

Figure 12 presents a generalized correlation of collection efficiency for spherical particles of any diameter, or apparent density, suspended in any gaseous medium, entering a cyclone with proper dimensional ratios, at flowing densities no greater than one grain of solids/cubic foot of gas. This correlation is consistent with the cut point diameter as defined by equation (4), and an effective path length or residence time interpreted in terms of number of barrel perimeters (spirals), N_s , as given by Figure 13. Data from axial flow cyclones appear in agreement with the fractional efficiency curve for tangential inlets when D_{pth} is calculated for half the number of spirals given in Figure 13. This in effect suggests that axial flow cyclones have cut point diameters larger by a factor of $\sqrt{2}$ and therefore somewhat lower collection efficiencies under otherwise similar operating conditions.

The lower fractional collection efficiency of cyclones with scroll or volute type inlets is well documented in the literature (15), but has never been satisfactorily explained. There is evidence that collection efficiency is impaired under conditions of unsteady or pulsating inlet loading; it is conceivable that particle saltation on the ledge of the volute might lead to such flow disturbances though this would not be intuitively anticipated, and hardly to the degree demonstrated in Figure 12.

Figure 12.

*-inlet or outlet, whichever is the greater
 ** for axial flow cyclones divide ordinate
 by 2 to get true effective N_s

Figure 13.

Outlet Tube Modification and Cyclone Length

In conventional cyclones, fitting the dimensional ratios in Figures 6 and 8, the gas outlet tube area generally exceeds the inlet area and the length of the exiting vortex has been experimentally determined to be $1.6 D_B$.

In so-called "modified" cyclone designs in which the gas outlet tube area has intentionally been reduced, either to decrease barrel diameter or to drop the annular velocity below the

particles' saltation rates, the greater penetration of the exiting vortex results in some additional residence time correlated by use of the higher exit velocity in Figure 13. The greater spiralling velocity within the exiting vortex also exerts a drag on the annular spiral, further reflected in a lower cut point particle diameter and again accountable via substitution of outlet velocity for inlet velocity in equation (4).

Figure 14.

Figure 14 summarizes the "optimum" dimensional ratios and the recommended minimum necessary cyclone lengths to avoid re-entrainment of collected particles through contact with the bottom end of the exiting vortex. As discussed in a subsequent section dealing with saltation and reentrainment, it can be shown that gas outlet tube modifications to A_O/A_I values less than 0.4 become detrimental to cyclone performance.

Effect of Solids Loading

The magnitude of the effect of solids inlet loading (between 1 and 1000 grains per cubic foot) on overall collection efficiency has likely lead to the greatest confusion and apparently conflicting comparative design evaluations. The greater the loading, the greater the overall efficiency as empirically correlated in Figure 15 and consistent with Figures 11 and 12. The abscissa of Figure 15 represents the total inlet or feed loading to the cyclone of all the particles, and not the loading of each individual size or size increment. The parameters indicate the rate of increase in efficiency from that at zero or low load (for example, less than 1 grain/cu.ft.) calculable from Figure 12 for any feed particle size distribution. As illustrated in subsequent sample calculations, the zero load efficiency is taken as a point on the ordinate of Figure 15 at an abscissa of 1. A curve beginning at this zero load point and drawn by interpolation between adjacent existing curves yields

Figure 15.

the effect of loading on the total collection efficiency as applied to the bulk weight of total feed solids. It does not apply to individual or incremental feed components.

The empirical "fits" of performance data to the basic theory, as represented by Figures 12 through 15 bear as yet unpublished aspects to their application. At loadings of less than 1 grain/cu. ft. Figure 12 permits prediction of the extent to which any given particle size will be collected and hence from a summation for each feed component as illustrated in subsequent examples, the overall efficiency for a feed material of any given size distribution. From the feed composition and the calculated amount of the collected material) a simple material balance on each component then yields the composition or particle size distribution of the lost or uncollected material leaving in the exit gas stream.

At inlet loadings in excess of 1 grain/cu.ft. the improvement in cyclone performance, or in total collection efficiency, is correctly predictable from Figure 15. However, there are anomalies in the interpretation of this improved performance relative to individual component particle sizes. Component as well as overall material balances could obviously be satisfied by multiplying the collection efficiencies for each particle size at zero loading (e.g. at loadings less than 1 grain/cu.ft. as calculable from Figure 12) by the ratio of the overall collection efficiency at the higher loading (from Figure 15) divi-

ded by the overall collection efficiency at zero load. By this procedure the collection efficiency of each particle size is enhanced equally. This is neither intuitively correct nor in agreement with observations. Certainly it would be expected that the increased collection efficiency would be biased in favor of the coarser particles; if such were not the case then in passing through a series of cyclones the coarsest passing the first stage would never be completely removed unless the subsequent stages were designed to exhibit substantially smaller cut point capability. Even in instances where subsequent stages do have smaller cut point diameters, the losses do not contain particles as large as this calculation procedure to correct for loading would allow. Note that Figure 15 is perfectly satisfactory unless it is necessary to determine the particle size analyses of the collected or lost materials rather than simply their total weights. In calculating the performance of 2 or more stages of cyclones in series (where one or more operate at loadings in excess of 1 grain/cu.ft.) such component particle analyses are absolutely necessary since the loss from a preceding stage determines the analysis of the feed to a subsequent stage.

An alternate approach to the effect of loading on total as well as component collection efficiency, as practiced by some vendors, attempts to bias the distribution of increased collection efficiency toward the larger size particles by altering the cut point diameter. An empirical correlation of the effect of loading simply reduces the cut point diameter thus resulting in

an increased collection efficiency following the calculation procedure in the previously cited examples. As clever and intuitively satisfying as this might appear it also fails to satisfy all evidence to date. Simply adjusting the cut point diameter amounts to shifting the fractional efficiency curve of Figure 12 to the right, thus increasing the collection of all sizes, though in the same relative proportions defined by the curve. All too frequently this results in situations where the loss from the cyclone would be predicted to contain nothing larger than 1 or 2 microns when observations show a spectrum of sizes up to perhaps 40 microns. Within a specific industrial application such as for example fluid catalytic cracking (and in particular the catalyst regenerator where effluent is more precisely monitored) it is possible by judicious alteration of Figure 12 in conjunction with an empirical loading correction of the cut point diameter, to devise a procedure which reasonably matches the observed performance of a series of two or three stages of cyclone separators. The annoying aspect of this practice is in its heaping of empiricism atop empiricism and its known failure to agree with specific detailed experiences in other industrial applications.

From an accumulation of select data from properly proportioned industrial cyclones gathered over nearly 3 decades, and a methodical small scale experimental program, a family of curves such as in Figure 12 has been correlated (16) to predict the particle size distributions of catch and loss in conjunction with the overall efficiencies of Figure 15 at all loadings. This work is to

date still proprietary though available to individuals at a reasonable fixed price.

Effect of Outlet Tube Penetration

Though it is well recognized that the withdrawal of the gas outlet tube, to penetrations or immersions less than the height of the tangential inlet, leads to reduced collection efficiencies and is hence not recommended practice, there are instances in which partial, if not entire withdrawal is logistically desirable if not inevitable.

Short of internally-cooled double-wall construction, or relatively sophisticated innovations in refractory formulation and/or installation, there exist high temperature applications in which gas outlet tubes have either actually failed or been intentionally omitted because of anticipated problems in sustaining structural integrity.

Ter Linden's (17) data are often quoted as illustrative of the effect of shortened gas outlet tubes; a generalized correlation of his data and that of others is given in Figure 16.

Saltation and Reentrainment

According to equation (4), a cyclone's efficiency should continuously increase with an increase in inlet velocity. However,

Figure 16.

any simple experiments will show that the rate of increase drops off and eventually begins to decrease as inlet velocity is raised. The velocity at which this deviation occurs appears to be a function of many variables. Fortunately, it is frequently undetectable over the practical ranges of operation.

Stairmand's (15) suggestion that reentrainment might be related to attaining spiraling velocities (in the annulus between the gas outlet tube and the barrel wall) that exceed particle pick-up velocities appears reasonable. It not only appeals to the intuitive senses but also takes into account all the system variables, including cyclone size. It would appear to be partially substantiated by the vendor's practice of necking down the gas outlet tube when striving for exceptional efficiency. The means are now available for an exploratory analytical study of this effect, based on proposed correlations (18) of settling and pickup velocities.

It had been found (19) that a reduction in the outlet tube area increases cyclone collection efficiency. This could be regarded in conjunction with the simplified view of gas and particles spiraling down through the cyclone barrel as a wall-less coil having a rectangular cross section. The latter is represented by the height of the cyclone inlet and the width of the annulus between outlet tube and barrel shell. These mechanistic picturizations permit drawing an analogy to saltation phenomena in pipe flow and deriving therefrom a theoretical approach to predicting what is

commonly referred to as the A_O/A_I effect or "outlet tube modification." This A_O/A_I ratio represents the relative outlet tube area to the inlet duct area. As illustrated in Figure 17 when A_O/A_I is decreased, the annular width is consequently increased.

When entering a cyclone, gas and solids travel down the barrel in a spiral fashion. The largest and/or heaviest particles that first reach the wall form an easily discernible, dense narrow stream. The entering gases initially also form a spiral path within the barrel, but soon follow the aerodynamic pressure gradient and move inwardly to be exhausted through the upper outlet tube. This flow pattern suggests an analogy (6) to flow through a coiled pipe bearing a narrow slit along its inner length to allow gradual gas dissipation, as depicted in Figure 18.

The spiralling particle-gas stream is subject to a substantial centrifugal field. Relative to normal gravitational attraction, the number of g's exerted is given by $v_i^2 / 32.2 R$

so that for a conventional cyclone having an inlet port width, or inlet pipe diameter, one-quarter the diameter of the barrel, the number of g's exerted is given by $v_i^2 / 32.2 (3/8) D_B$.

Figure 18.

A conventional 3-foot-diameter cyclone at an inlet velocity of 60 feet per second imposes a field of 100 g's on entering particles. At 100 feet per second inlet velocity, a 1-foot-diameter unit exerts 830 g's, and at only 40 feet per second, even a 10-foot-diameter cyclone yields a 13-g field.

It becomes obvious from the magnitude of these figures that, in terms of the particle dynamics, the vertical orientation of the model coil is of almost minor consequence. The effect the centrifugal force has on the particles is equivalent to considering them to be heavier, or of a density equal to their normal weight per unit volume multiplied by the number of g's. The pipe coil of Figure 18 may thus be unrolled and considered as a length of horizontal pipe conveying effectively g-weighted particles as

pictured in Figure 19. Three fundamental mechanisms of particle separation and/or reentrainment may be described by this model.

Figure 19.

1. If a particle is to reach saltation (that is, be collected) it must pass through the streamlines of the spiralling gas as proposed by Rosin et al. (2).
2. By direct analogy to saltation (18) in horizontal conveying lines, it would appear that if the gas velocity in the pipe of Figure 18 or 19 were greater than the saltation velocity of the particle (the particle now exhibiting a g-weighted mass), then the particle would have little chance of reaching and remaining on the bottom of the pipe (see Figure 19) or on the outer diameter of the coil (see Figure 18). This leads to the conclusion that exceeding the particle saltation velocity should begin to decrease rather than increase the anticipated collection efficiency.
3. Having once reached saltation at the barrel wall, the particles may again be reentrained if their pickup velocity (18) is exceeded.

Cyclone efficiency would presumably be at a maximum when the annular velocity is so low that it is unable to carry any particles;

in other words, when it is below the minimum saltation velocity. This minimum velocity and hence minimum annular width, or consequent A_O/A_I ratio, can theoretically be calculated for any particle-gas cyclone system from existing correlations of saltation velocity data observed in conventional horizontal conveying. The necessary assumptions and procedure can be illustrated by the following typical example for collecting sand particles having a density of 161 pounds per cubic foot from an atmospheric air stream:

1. Assume the particles are so small that the solids inlet velocity essentially equals the gas inlet velocity.
2. The width of the gas inlet, at the cyclone mouth, is one-fourth of the cyclone diameter.
3. The width of the gas inlet is the effective pipe diameter for calculating saltation velocity.
4. Feed particles range down to less than 1 micron in diameter.
5. The solids loading is so low as to have a negligible effect on saltation velocity (for example $V_S = V_{SO}$).
6. The acceleration field, expressed as the number of g's (where one g equals 32.2 feet per second squared), is based on the solids inlet velocity and the cyclone radius

from its axis to the midpoint of the inlet width. In other words, the radius equals (diameter/2) minus (inlet width/2), or $(3/8)$ (cyclone diameter).

7. The effect of the pipe diameter on saltation velocity is proportional to the 0.4 power of pipe diameter (18).

For cyclone diameters of 12 inches to 36 inches, for example, and inlet velocities of 50 to 100 feet per second, the above assumptions permit calculating the maximum barrel spiralling velocities that can be tolerated and still permit all the feed particles to salt out. If these maximum allowable velocities are less than the inlet velocities, the outlet tube must be decreased in diameter in order to open up or increase the annular area in the barrel and bring the velocities down to the calculated maximum. If the required reduction in velocity is excessive, then the barrel diameter should also be increased. If the velocity needs to be halved, the outlet tube diameter would theoretically be squeezed to zero under cyclone dimensional ratios conforming with item 2 in the above-listed assumptions.

For particles with a density of 161 pounds per cubic foot in atmospheric air, the denominator of the ordinate of the saltation correlation (18) for 1 1/4-inch pipe is found to be 2.44 feet per second at one g. The ordinate goes through a minimum at a value of 2.1, therefore, in view of assumption 5 above, no re-

entrainment of any size particles would occur in a one-g field

when $(V_{SO}/2.44) = 2.1$. From assumptions 3 and 7:

$$\frac{V_{SO}}{(\text{No. of } g's)^{1/3}} = 2.1 \left[\frac{L_w}{1.25} \right]^{0.4} \quad (19)$$

and from assumptions 1 and 6:

$$\text{No. of } g's = \frac{12 v_i^2}{32.2 D_B} = \frac{12 v_i^2}{32.2 D_B^3} \approx \frac{v_i^2}{D_B} \quad (20)$$

where D_B is the cyclone diameter in inches. Therefore

$$V_S = \frac{4.7 (L_w)^{0.4} v_i^{2/3}}{D_B^{1/3}}$$

or

$$\frac{v_i}{V_S} = \frac{v_i^{1/3} D_B^{1/3}}{4.7 (L_w)^{0.4}} = \frac{v_i^{1/3}}{2.7 D_B^{0.067}} \quad (21)$$

and

$$\begin{aligned} \frac{A_O}{A_T} &= \frac{1}{4} \left[D_B - \frac{2v_i}{V_S} \frac{D_B}{4} \right]^2 \bigg/ \frac{\pi}{4} \left(\frac{D_B}{2} \right)^2 \\ &= \frac{4}{D_B^2} \left[D_B - \frac{v_i}{V_S} \frac{D_B}{2} \right]^2 \end{aligned} \quad (22)$$

so that equations (21) and (22) can be represented graphically as shown in Figure 20.

Several rather significant conclusions can be drawn from Figure 20:

1. The A_0/A_I effect becomes more necessary or beneficial with increasing inlet velocity. For the particle-gas conditions chosen in the sample calculations, it would appear that outlet tube modification would be beneficial above 32 to 40 feet per second inlet velocity. In other words, as inlet velocity is increased above these values, collection efficiency would become less than that normally expected, unless the outlet tube diameter were reduced.

2. The A_0/A_I effect does not strongly depend on cyclone

diameter, particularly as the diameter begins to exceed 3 feet.

3. The benefits of the A_O/A_I effect depend on particle diameter. Large particles or high density particles having a high saltation velocity would reflect little gain from outlet tube modification.

4. At velocities 2 to 2 1/2 times the saltation rates, particles can be picked up from a deposited layer. For the conditions chosen in Figure 20, it would have to be expected that at inlet velocities above 80 to 100 feet per second the 36-inch cyclone, for example, would begin to show a decline in overall collection efficiency.

5. It appears inadvisable to reduce A_O/A_I to less than 0.4.

Typical results (6) for a given powder fed to a given diameter of cyclone are shown schematically in Figure 21 which qualitatively conforms to what would be expected from a combination of equations (4), (21) and (22). At inlet velocities below V_s collection efficiency shows a steady increase with v_i . When v_i reaches V_s further increase is more gradual, until at sufficiently high velocities it actually begins to decrease. In other words, increasing the gas spiralling velocity increases the horizontal or saltation drag faster than it can be overcome by the simultaneously higher centrifugal force. Quantitatively, saltation theory predicts the inlet velocities at which efficiencies begin to deviate from those calculated from equation (4) in combination with trial-and-error derived fractional efficiency and N_s curves. Test data have indicated that the peak in efficiency occurs at inlet velocities of about twice V_s , which qualitatively agrees with past observations (18) of pickup versus saltation velocities. The change in efficiency with velocities exceeding V_s has been correlated

Figure 21.

empirically in various fashions. A simple relationship applicable to the standards of Figure 14 would show

$$\frac{100 - E_{v_i}}{100 - E_{v_s}} = \frac{1 + (v_i/v_s)}{2} \quad (23)$$

for v_i/v_s values less than 2.

Underflow

Drawing off a fraction of the feed gas, along with the collected solids, out of the bottom of a cyclone obviously lessens the particle losses in the overhead gas discharge tube since the underflow gas carries its proportionate quantity of fines which would otherwise have been carried out in the overhead stream. The direct effect is relatively small as illustrated by the relationship

$$\Delta E = \% \text{ UF } (100 - E) / 100 \quad (24)$$

where ΔE is the % increase in efficiency due to the underflow, UF (expressed as a % of the feed gas stream), and E represents the % collection efficiency without underflow. If, for example, a cyclone exhibited 90% collection efficiency without underflow then taking 5% of the feed gas out through the bottom (assuming the solids content would be removed by some other downstream means, as for example in U.S. patent 3,698,874) then the overall efficiency of this cyclone would increase by

$$\begin{aligned}\Delta E &= 5 (100 - 90) / 100 \\ &= 0.5\%\end{aligned}$$

or in other words increase from 90% to 90.5%.

Underflow is practiced primarily in low loading instances involving very fine particles where reasonable collection efficiencies are necessary with minimal allowable pressure drop. Such situations lead logically to multiple cyclones operating in parallel (which effectively substitutes small inlet width for high inlet velocity to achieve a reduced cut point particle size). Attaining near equal gas and particle flow to each cyclone of a group, operating in parallel, with minimal pressure drop, and assuring that no gas flows in reverse up into the cone of one or more when all discharge into a common hopper, is the principal motive in operating with a small underflow stream. The effect of underflow on the collection efficiency of an otherwise normally operating cyclone (as given by the foregoing equation) is not substantial;

however, in conjunction with maintaining any number in parallel operating normally (where one or more might otherwise malfunction) the effect of underflow can be significant.

Coupling Effects

All cyclone performance correlations capable of consistent substantiation are based on homogeneous well-dispersed particle-in-gas flows at the inlet. Cyclones hung high in the disengaging zone above fluidized beds represent such near-ideal feed conditions. Any alternate ducting of the inlet flow should obviously not intentionally counter the directional rotation within the cyclone barrel. Arrangements such as in Figure 22 (a), (b) and (c) are to be avoided unless the distances shown as x represent substantial multiples of the duct diameter sufficient to establish steady

Figure 22 Inlet Ducting Configurations

state velocity distributions and homogeneous particle suspension. Duct velocity should also be well above particle saltation velocity if consistent correlation of results is to be anticipated. Even in the case of Figure 22 (d), where the centrifugal force in the inlet elbow is directionally the same as in the barrel, the distance x should be substantial to avoid particle rebound altering particle flow paths in the vicinity of the cyclone mouth. The volute entry of Figure 22 (e) has been shown to exhibit lower collection efficiency than conventional tangential entries despite the directionally similar rotation; for lack of an as yet better explanation this has been attributed to the effect of whatever phenomena might accompany or be induced by the ledge forming the lower boundary of the volute.

Close-coupled bends such as illustrated in Figures 22 (a) through (d) are not uncommon sights associated with so-called "sheet metal" cyclones frequently seen on the roofs or alongside various light industrial plants. These cyclones generally collect rather large "dusts" such as wood shavings, nut hulls, metal dusts, feathers, threads, etc. where satisfactory previous installations suffices for selection purposes in comparison with more sophisticated design know-how, which cannot be economically justified.

The smaller the particles to be collected and the lower their inlet concentration the greater the necessary attention to such

details as inlet configuration and hence little wonder that this becomes particularly significant in the coupling of second or third stages. Just as the eye of a hurricane induces rotation of the surrounding atmosphere, or as a flat paddle stirrer in a tank of liquid induces circulatory motion of the entire contents, so the fluid throughout a cyclone takes on a rotary motion which carries on up into the gas outlet tube. If this tube then fed directly into a subsequent stage cyclone in a configuration such as in Figure 23 (a) or (b), it would be less conducive to directing particles toward the barrel wall of the second stage than if the discharge flow were maintained as a flat helix about a vertical axis as illustrated in the couplings in Figure 23 (c) and (d). Arrangement (c) is that in common practice; the theoretic-

Figure 23

cally preferable arrangement in (d) has been generally considered economically unjustifiable. It should be obvious in Figure 23 (c) that the gas outlet tube with its tangential drawoff, shown within the dotted circle, behaves much as a small cyclone driving particles, leaving the first or upstream cyclone, to the walls of the outlet tube (whose tangential drawoff then becomes the inlet duct of the second or downstream cyclone) thus acting as a pre-separator enhancing the performance of the second stage. This pre-separation must be taken into account in calculating the performance of a series of coupled cyclones. The particle size distribution of the dispersed solids entering the downstream stage will differ from the particle size distribution of the entire mass leaving the upstream stage; this difference lies in the separation performance of the tangential coupling.

Industrial Performance Data

The following examples, drawn from industrial plant and test operations wherein extraordinary efforts were made to be certain of the numerical figures (operating conditions, analyses etc.), cover a substantial range in cyclone size. Inlet widths vary from 5/16" to 24 inches and barrel diameters from a few inches up to 12 feet. The fact that the test observations are in reasonably good agreement with what would be predicted from Figures 12, 13 and 15 is to be expected, since some of these data were used to establish the correlating curves. Cyclone size enters into the calculation procedure simply in terms of the inlet

width in equation (4). These examples also cover a range of inlet velocities from 55 to 430 ft/sec. Unfortunately the sources must be held in confidence since some represent operations either at one time considered classified, irreconcilable and hence unaccepted by the equipment vendor as representative, or evidence gathered with the intent of litigation. Such considerations plague all existent industrial data such that neither the user nor the vendor dares release results which could lead to suits of defamation, fraud, non-compliance, etc. The examples chosen are intended to illustrate some extremes. It would appear rather unlikely that a simple theory and complementing correlations would yield such good agreement with practice, purely fortuitously.

Example 1. A Pharmaceutical Product

This operation explored the collection of essentially uniform size microbes about 1 micron in diameter. The loading was trivial and the cyclone of reasonably conventional design provided with an oversize barrel diameter to avoid saltation. The organic particulates were estimated to have a particle shape factor of 0.75 by microscopic observation relative to other particulates having similar contours and experimentally determined shape factors. The pertinent data were as follows:

cyclone gas inlet: 5/16" x 1"
cyclone gas outlet: 1 1/4" diam.

Gas density: 0.2 lbs/ft³
Gas viscosity: 0.0182 centipoises

cyclone inlet velocity: 430 ft/sec	Particle density: 62 lbs/ft ³
N _s , from Figure 13: 6	Particle shape factor: 0.75
Obsv'd. cyclone efficiency: 50%	Particle size: 1 micron (physical)

From Equation (4):

$$D_{p_{th}} = 12 \times 25400 \sqrt{\frac{9(0.0182)(0.000672)(5/16)}{12 \pi 6 \times 430 (62 - 0.2)}}$$

$$= 0.729 \text{ microns (spherical)}$$

$$\frac{D_p}{D_{p_{th}}} = (1) (0.75) / 0.729$$

$$= 1.0288$$

∴ E_o from Figure 12 = 51% vs. obsvd. 50%.

This represents a very simplistic case; however, other than at most a disputable shape factor, it leaves little room for error. It represents an extreme in terms of inlet width and inlet velocity, and yet appears in good agreement with Equation (4) and Figure 12. This cyclone operated at a pressure drop of nearly 22 psi which was also found in excellent agreement with the calculation methods given in a subsequent section.

Example 2. A Bark Boiler

This operation involved the collection of char plus sand carryover from a so-called bark boiler. It represents a far more

common application of somewhat larger cyclones and also an instance where knowledge of particle shape factors was of pronounced importance in predicting observed behavior. The pertinent data were as follows:

192 axial flow cyclones handling 156,237 ACFM (wet)
cyclone gas inlet: 0.2454 ft² gas density: 0.0411 lbs/cu.ft.
cyclone gas outlet: 6" diameter gas viscosity: 0.027 centipoises
cyclone inlet velocity: 55.27 ft/sec particle density: sand/155 char/30
Annular width: 1.5 inches particle shape factor: 0.497* 0.0883*
solids loading: 1.27 grains/dry SCF

Particle Size Analysis** of feed solids:

Char		Sand	
<u>wgt. fraction</u>	<u>Microns (physical)</u>	<u>wgt. fraction</u>	<u>Microns (physical)</u>
0.02	12	0.02	7.5
0.03	22	0.03	12
0.05	35	0.05	17
0.10	64	0.10	26
0.10	112	0.10	47
0.10	185	0.05	72
0.10	380	0.05	112
0.088	1,200	0.012	170
0.588		0.412	

*- from sedimentation and free fall tests in air

**--with U.S. Std. Sieves and a Coulter Counter

Observed cyclone collection efficiency = 58%.

In calculating the anticipated cyclone performance either the values of D_{pth} from eqn. (4) should be divided by the shape factor or else the physical "diameters" from the sieve and Coulter Counter analyses should be multiplied by the shape factor to determine the ordinates of Figure 12.

	<u>Char</u>	<u>Sand</u>
From eqn. (4) D_{pth} :	13.66 (spherical)	6.01 (spherical)
D_{pth} /shape factor:	155 (physical)	12.09 (physical)

Collection of Char:

<u>wgt. fraction</u>	<u>Microns D_p, (phys.)</u>	<u>$D_p/155$</u>	<u>E_o</u>	<u>Collected (wgt. fract.) $\times E_o$</u>
0.02	12	0.0774	---	---
0.03	22	0.1489	0.010	0.0003
0.05	35	0.2258	0.045	0.0023
0.10	64	0.4129	0.17	0.0170
0.10	112	0.7226	0.365	0.0365
0.10	185	1.194	0.56	0.0560
0.10	380	2.452	0.77	0.0770
0.088	1,200	7.742	0.965	0.0849
				Σ : 0.2740

Collection of Sand:

		<u>$D_p/12.09$</u>		
0.02	7.5	0.6203	0.31	0.0062
0.03	12	0.993	0.49	0.0147
0.05	17	1.406	0.62	0.0310
0.10	26	2.151	0.74	0.0740
0.10	47	3.89	0.858	0.0858
0.05	72	5.96	0.925	0.0463
0.05	112	9.26	0.99	0.0495
0.012	170	14.06	1.00	0.0120
				Σ : 0.3195

Total Collected = (0.2740 + 0.3195) 100

= 59.35% vs. obsvd. 58%

Example 3. A Coal Combustor

The background data regarding particle density and size analysis were in this instance reported by a third party as representing equivalent spherical sizes. Accepting this at face value the re-

maining pertinent facts were as follows:

cyclone gas inlet: 4" x 10"	gas density: 0.1219 lbs/cu.ft.
cyclone gas outlet: 4.25" diam.	gas viscosity: 0.045 centipoises
cyclone inlet velocity: 80 ft/sec	particle density: .62 lbs/cu.ft.
cyclone outlet velocity: 229 ft/sec	inlet loading: 2.58 grains/ACF
Annular width: 4"	

Observed cyclone collection efficiency = 50%

Cyclone Feed		Avg. $\frac{D_p}{5.59^*}$	E_o	Collected lbs/hr.
D_p (spher.)	lbs/hr.			
16 - 20	0.944	3.22	0.82	0.7741
12.7 - 16	1.454	2.54	0.78	0.1341
10.8 - 12.7	1.938	2.04	0.73	1.4147
8 - 10.1	2.576	1.61	0.67	1.7259
6.3 - 8	3.111	1.28	0.58	1.8044
5 - 6.3	3.315	1.01	0.50	1.6575
4 - 5	3.366	0.805	0.40	1.3464
3.2 - 4	3.009	0.644	0.31	0.9328
2.5 - 3.2	2.346	0.51	0.22	0.5161
2 - 2.5	1.964	0.403	0.15	0.2946
1.6 - 2	1.326	0.322	0.095	0.1260
1.3 - 1.6	0.969	0.26	0.06	0.0581
0 - 1.3	0.172	0.116	nil	---
	26.5			11.7847

$$*D_{pth} = 12 \times 25400 \sqrt{\frac{9(0.045)(0.000672)4/12}{\pi 229 \times 6 \times \sqrt{(62.4 - 0.1219)}}$$

$$= 5.59 \text{ microns}$$

$$E_o = 11.7847/26.5 = 44.5\%; \text{ from Fig. 15, } E_L = 51\% \text{ vs. vs. obsvd. 50\%}$$

Example 4. A Flyash Test

This represents one of a number of tests carried out on a two stage system in which unfortunately particle size measurements were not made on the catch or loss from the first stage. The run reported here is considered representative. The pertinent data were as follows:

cyclone gas inlet: $5 \frac{5}{16}$ " x 13"	gas density: 0.0765 lbs/ft ³
cyclone gas outlet: 12" i.d.	gas viscosity: 0.0182 centipoises
cyclone inlet velocity: 70 ft/sec	particle density: 124.8 lbs/cu.ft.
N _s from Figure 13: 4.4	particle loading: 600 grains/cu.ft.
	Obsv'd. cyclone efficiency: 98%

from Eqn. (4) $D_{pth} = 11.48$ microns (spherical)

<u>cyclone inlet</u>					
<u>D_p (spher.)</u>	<u>wgt.</u>	<u>D_p</u>	<u>E_o</u>	<u>Collected</u>	
<u>microns</u>	<u>fract.</u>	<u>11.48</u>		<u>(wgt. fract.) × E_o</u>	
0 - 5	0.06	0.2178	0.08	0.0024	
5 - 10	0.19	0.6530	0.32	0.0608	
10 - 20	0.24	1.307	0.59	0.1416	
20 - 40	0.18	2.613	0.782	0.1408	
40 - 80	0.24	5.23	0.905	0.2172	
80+	0.09	7.85	0.97	0.0873	
				$\Sigma: 0.6501 = E_o$	

from Figure 15, $E_L = 98.4\%$ vs. obsv'd. 98.0%

Example 5. A Fertilizer Plant

This operation involved the collection of Gypsum particles in a 12 foot diameter cyclone. The data were taken during plant operation via pitot traverses and isokinetic sampling in 37" diam-

eter ducts. The data are therefore subject to the inaccuracies to be anticipated under such non-ideal circumstances as well as to the effects of fluctuations in operating conditions which could have occurred in the course of the test, despite requests to the operators to maintain conditions as constant as reasonably possible. The pertinent data were as follows:

	<u>inlet loading, gr/ft³</u>	<u>E_L, observed</u>
Cyclone gas inlet: 2' x 3'	119	95.3
Cyclone gas outlet: 4' diam.	73.3	92.5
Cyclone inlet velocity: 93.3 ft/sec	55.4	91.6
N _s from Figure 13: 4.9		
Gas density: 0.0585 lbs/cu.ft.		
Gas viscosity: 0.021 centipoises		
Particle density: 145 lbs/cu. ft		

From eqn. (4), $D_{Pth} = 10.65$ microns (spherical)

<u>D_p, microns</u>	<u>wgt. fract.</u>	<u>D_p, microns</u>	<u>wgt. fract.</u>
1.5	0.02	18.2	0.10
2.0	0.03	27	0.10
3.0	0.05	42	0.10
5.0	0.10	80	0.10
7.0	0.10	170	0.05
9.8	0.10	360	0.03
13.2	0.10	800	0.02

From Figure 12: $E_o = 59.11\%$

<u>Ldg. gr/cu. ft.</u>	<u>E_L from Figure 15</u>	<u>E_L observed</u>
119	95.8	95.3
73.3	94.8	92.5
55.4	92.5	91.6

Example 6. A Scaled Test of a Third Stage FCC Catalyst Collector

This test was one of a number conducted on a scale model of a cluster of third stage cyclone collectors proposed for a refinery installation. The collected material was carefully size analyzed and was in perfect agreement with the calculated catch distribution.

Cyclone gas inlet: 1 3/16" x 2 5/8"	Gas density: 0.0765 lbs/ft ³
Cyclone gas outlet: 1 3/16" I.D.	Gas viscosity: 0.0182 centipoises
Air feed: 127.3 ACFM/cyclone	Particle density: 92 lbs/cu.ft.
N _s from Figure 13: 6	Particle loading: 0.168 grains/cu.ft.
Obsv'd. cyclone efficiency: 89.0%	Underflow: 2.43%
	D _{pth} = 1.45 microns (from Eqn. 4)

Cyclone Inlet				
D _p microns	wgt. fraction	D _p /1.45	E _O %	% Collected
1.0	0.02	0.69	35	0.70
1.31	0.03	0.90	45	1.35
1.78	0.05	1.23	57	2.85
2.75	0.10	1.90	71	7.10
4.6	0.10	3.17	82	8.20
7.2	0.10	4.97	89.6	8.96
10.3	0.10	7.10	95.5	9.55
14.1	0.09	9.72	99.6	8.96
>16.1	0.41	>10.0	100.0	41.00
				<u>88.67</u>

$$E_O = (0.8867 (1 - 0.0243) + 0.0243) 100$$

$$= 88.94\% \text{ calculated efficiency}$$

Pressure Drop

Gas Only

Methods of calculating cyclone pressure drop have been in dispute for nearly three decades. Measurements taken during operation frequently differed significantly from correlations developed in laboratory investigations which used gas-only results as a base. In recent years several investigators (20) (21) have demonstrated the reduction in pressure drop which accompanies the addition of a light loading of dust to the inlet gas. The effect appears analogous to the reduction of wall drag by addition of liquids to long gas pipelines, or the natural "greasing" of conduits in the pneumatic conveyance of live fowl; small particles apparently disturb the laminar wall film and thereby reduce the drag. The method of calculating pressure drop outlined below applies to inlet loadings in excess of 4 lbs of solids per second per square foot of inlet area. The pressure drop for gas only, at zero dust loading, as predicted from such relationships as proposed by Stairmand (21) are of the order of 2.5 times as great. This should be borne in mind if cyclones are installed in a system which might flow continuously but handle solid only intermittently.

Under Collection Conditions

The total pressure drop through a collecting cyclone can be

considered as composed of the summation of 5 component losses.

Figure 24.

1. An abrupt contraction of the gas stream from the superficial velocity, V_{Gs} , in the vessel (assuming the cyclone to be located in the freeboard or disengaging height above a fluidized bed, otherwise this term is neglected) to V_{Gi} , the inlet velocity to the cyclone. Referring to Figure 24 and its attendant nomenclature:

$$\Delta P_{(s-i)G} = 0.00298 \rho_G (V_{Gi}^2 - V_{Gs}^2 + K V_{Gi}^2) \quad (25)$$

where K is a function of the ratio of the inlet area at i to the superficial vessel cross sectional area, as in the following table:

<u>Area Ratio</u>	<u>K</u>
0	0.50
0.1	0.47
0.2	0.43
0.3	0.395
0.4	0.35

2. A solids acceleration loss from v_{ps} to v_{pi} taken as:

$$\Delta P_{(s-i)p} = L v_{Gi} (v_{pi} - v_{ps}) / 167 \quad (26)$$

For small particles having a low terminal velocity, v_{ps} is usually taken as v_{Gs} and v_{pi} taken as v_{Gi} .

3. A barrel loss is treated as straight pipe pressure drop for N_s number of spirals through the annulus between the barrel and the gas discharge tube, calculated at a velocity equal to v_{Gi} in a pipe of hydraulic diameter corresponding to the inlet area expressed as:

$$d_{hi} = 4 \text{ (inlet area) } / \text{ (inlet perimeter)}$$

so that:

$$\Delta P_B = f D_B \rho_G N_s v_{Gi}^2 / 26.2 d_{hi} \quad (27)$$

where f is the friction factor obtained from a conventional Fanning friction chart giving f as a function of Reynolds number, Re .

$$f = \frac{g_{c,hi} \Delta P}{2 \rho_G v_{Gi}^2 N_s \pi D_B}$$

$$R_e = d_{hi} v_{Gi} \rho_G / \mu_G$$

In equation (27) D_B and d_{hi} are to be expressed in inches; f is generally of the order of 0.004 to 0.008.

4. A reversal loss equivalent to one velocity head due to reversal in gas flow direction going from the barrel toward the gas discharge tube calculated at the inlet or spiralling velocity:

$$\Delta P_r = \rho_G v_{Gi}^2 / 335 \quad (28)$$

5. An exit contraction loss, going from a velocity based on the superficial barrel cross section to that in the exit pipe, expressed as:

$$\Delta P_e = 0.00298 \rho_G (v_e^2 - v_B^2 + K v_e^2) \quad (29)$$

where K is obtained from the table in step 1. corresponding to the area ratio given as $(D_e/D_B)^2$.

The foregoing relationships, properly applied, have shown excellent agreement with operating data ranging from 5 inches

of water to 20 psi. Each component of the total loss, other than the solids acceleration, contains a velocity head term, so that the total may be approximated by an empirical relationship between the inlet velocity and the number of velocity heads for particular cyclone proportions. This is the general practice among vendors. Increasing a relatively well-loaded cyclone's efficiency by higher outlet tube velocity as opposed to higher inlet velocity entails less total pressure drop, since the exiting gas bears far fewer solids and hence a lesser and probably insignificant particle acceleration term. The stability or equality of gas and solids distribution in any system of cyclones operating in parallel also depends on sufficient pressure drop to ensure such distribution, as is the case in any instance of division of streams in parallel fluid flow.

Pressure Balance

The treatment of a fluid bed and a cyclone dipleg as the two legs of a manometer is illustrated in Figure 25. The analogy is ideally simple, but in practice it must be borne in mind that levels are frequently difficult to define with great precision. They are average figures. Bed level may fluctuate 4 feet or more as bubbles rise and burst. The dipleg inventory might slug to heights of 7 feet or more above the calculated levels. The greater the attention given to the details of good grid and cyclone system design, the less the fluctuation in these levels, but inevitably some will exist. The calculated dipleg levels

or pressure balances will depend upon cyclone pressure drops, entrainment rates, and particle size (and hence deaeration rates and bulk densities) of the material collected in each stage.

Figure 25

Figure 25 illustrates the influence of cyclone pressure drop on required vessel height above a bed of fluidized solids, or conversely the limited allowable cyclone pressure drop if the vessel already exists or has been specified. In a series of stages (e.g. two shown in Figure 25) the back-up level ($L_D - s$) in the last stage is the greatest, since the pressure drops are cumulative. In addition each subsequent stage collects

finer particles which generally exhibit lower bulk densities than the bed; this is further aggravated by deeper submergences. From process considerations the collected fines should be returned to the bottom of the bed, as near the grid as possible, to avoid segregation and maintain a well mixed homogeneous bed. If this is considered paramount and if in addition high collection efficiency (probably requiring high velocities and hence high cyclone pressure drop) is of equal importance, then cyclone elevation can become so great that internal location might no longer be practical; in which case the cyclones would better be located external to the bed vessel.

A proper pressure balance, to determine solids backup levels, requires a means of predicting the bulk densities in the diplegs. If these are simply open ended immersed pipes, as implied in Figure 25, then their contents can be regarded as high aspect ratio fluidized beds operating in parallel with the bed into which they are immersed. Their densities are then calculable from the voidage corresponding to a bed of their mean particle size fluidized at the prevailing superficial rate. The mean particle diameter of the material in any dipleg is calculable from the particle size distribution of its collected solids. If the diplegs are provided with any effective aeration-inhibiting valves, or deflectors, then their bulk densities may be estimated from typical curves of mean particle diameter versus void fraction in their loosest unaerated bulk state as illustrated in the subsequent design example. In such an in-

stance it is not inconceivable to find the bulk density in a first stage dipleg to be greater than the density of the surrounding fluidized bed.

Manifolding

Figure 26 illustrates typical manifold arrangements as exist in large fluidized bed reactors (Fig. 26a); in external multicyclone, low loading, high efficiency collectors (Fig. 26b); and in conventional duct manifolds. Arrangement (a) has the virtue of simplicity and requires only a single vessel penetration for the outlet of the combined cyclone gas discharges. Arrangement (b) represents a radial feed distributor intended to assuredly achieve equal flow of gas plus solids to each of the cyclones. Since all feed a common collection hopper there could occur a reverse flow of gas up the dipleg of one or more if pressure drops in all parallel paths were not substantially equal. If oriented in more than one elevation as in Figure 26 (b), it is necessary to consider particle trajectories as well, unless they are so small that they assuredly follow the gas streamlines, otherwise the lower cyclones will receive a greater solids load than those on the upper level.

The improvement in distribution afforded by cyclone pressure drop is illustrated, in an example in the subsequent section dealing with sample calculations, by comparing the relative flows

Figure 26.

from a straight duct manifold (as in Figure 26c) before and after the installation of cyclones bearing a given predictable pressure drop.

Orientation in Fluidized Bed Reactors

Elevation Above Bed Surface

Since bubbles passing up through a fluidized bed and bursting at the surface thrust particles into the disengaging zone at rates which the far lower superficial velocity cannot carry, there will be an immediate fallback region resulting in a particle concentration or density gradient over several feet above the bed surface. As illustrated in Figure 27, the Transport Disengaging Height (TDH) represents the height above the surface of a dense-phase bubbling fluidized bed at which the particle entrainment, or dilute-phase density, shows only a negligible decrease with further increase in height (23). At this level

Figure 27.

the gas velocity profile has stabilized, the coarser nonconveyable particles have all fallen back, and the gas stream is as saturated as it can be with the entrainable solids. Further fallback occurs only partially, along the vessel's walls where the velocity profile inevitably shows a gradient. There is no advantage in providing additional vessel height in excess of TDH unless pressure balance calculations (shown in Figure 25) require higher elevation of internal cyclones. Providing a height less than TDH above the bed results in higher entrainment or cyclone loading.

Prior to economic optimization studies, TDH is recommended to establish an initial overall vessel height or cyclone elevation. Where vessel cost or other height limitations exist, it can sometimes be desirable to locate cyclones below TDH and offset the higher entrainment loading with higher efficiency (higher pressure drop) or multistage units.

Internal vs External Location

The installation of cyclone recovery equipment as auxiliary to process vessels frequently leaves open the choice of their installation within the vessel or external via ducts leading gas and solids to the cyclones and diplegs returning collected material through nozzles back into the process vessel. The choice generally involves the economics of the vessel and structure assembly and the characteristics of the solids being handled. The latter is a process aspect which in most instances is resolved by operating experience so that it becomes either an overriding limitation or, as in most instances, a condition able to be circumvented, so that the economics of overall construction becomes the inevitable criterion.

The economics are usually dictated by the following considerations:

- 1) If the process vessel operates at a substantial internal pressure then if the cyclones were located externally they, as well as the ducting, would also require wall thickness to be specified to code. Therefore internal location would represent a considerable cost reduction.
- 2) If the process vessel operates at a substantial temperature level, external location of the cyclones would require additional insulation. Since ducting to and from the cyclones would be anchored to the vessel the external arrangement would also require expansion joints at appropriate locations.
- 3) If the cyclone system operates at a substantial pressure drop (for example several stages to achieve high collection efficiency) it may require such long diplegs that if located internally the process vessel itself would have to be heightened to accomodate the cyclones' level. If the process vessel is large in diameter, and costly, economics could dictate external cyclone location some distance above the vessel head with ducting and dipleg penetrations either through the head or return through the vessel shell.
- 4) If the cyclone system occupies considerable volume, requiring the process vessel to be enlarged to maintain reasonably low velocity up to the cyclone inlets, external location could be far more economical.

The process or operating limitations are usually dictated by the following considerations:

- 5) If the recovered particles have tendencies to be non-"free-flowing" at any operating condition it can be desirable to locate cyclones external to the vessel to permit dipleg cleanout or the provision of other means to maintain flow such as aeration, supplementary heat tracing or cooling etc. External cyclone location frequently requires the return dipleg to leave the cyclone at an angle thus presenting a less favorable flow condition. Normally if this angle exceeds the collected solids angle of internal friction (usually of the order or 72° from the horizontal) there is no flow hindrance. However, this steep a return angle is frequently difficult

to accomodate without undue vessel height. A lesser angle represents a compromise, with other provisions for inducing flow, unless experience with the particular material to be handled indicates its angle of internal friction to be some figure less than the intended return line inclination. In this regard it should be borne in mind that solids can, and have plugged, even vertical return lines. For this very reason it is frequently desired to install external cyclones in newer processes where less experience is available with the specific solids. In such instances it is felt prudent to risk a less favorable external arrangement which is more easily accessible than a possibly more favorable internal arrangement which is inaccessible and requires plant shutdown for correction with the correction likely to be unknown and therefore perhaps only temporary.

- 6) The above considerations also favor external arrangements in a new process involving unknowns which could lead to difficulties within the bodies of the cyclones. External equipment has the decided advantage of accessibility for replacement, repair, alteration or even blanking in instances where excessive turn down or similar flexibility may be warranted.
- 7) External multiple cyclone arrangements also afford the opportunity to modify the components with restrictions in cases where there might occur insufficient pressure drop to achieve uniform distribution to all units. Such modifications could for example be carried out in the outlet tubes to increase efficiency as well.

Dipleg Operation

The effective operation of any cyclone system within a fluidized bed reactor can be completely destroyed by improper dipleg sizing and/or improper dipleg operation. A gross error is easily traceable, but a minor error can be extremely elusive and can cause diverse conclusions possibly leading to useless cyclone revisions. If, for example, no solids (or relatively few, as in a third-stage cyclone) were entering the dipleg pipe, then the pipe would simply contain a portion of the bed since it

would obviously fill up through its bottom end to a level at least that of the surrounding bed. In such a case, the pipe represents a small-diameter, high L/D (ratio of filled length to inside diameter) fluidized bed operating in parallel with the larger diameter and lower L/D bed in which it is immersed. Gas passing up through the reactor vessel would now have two paths in leaving the reactor vessel. One path would be up through the main bed surrounding the dipleg pipe, passing through the disengaging height, entering the first-stage cyclone, passing through the cyclone train and leaving through the outlet tube of the last stage. The other path would be up through the dipleg pipe, through the cyclone cone and barrel, and out through the cyclone's outlet tube. The quantity of gas that would use the latter path would be a fraction of the total gas passing through the reactor bed. It would equal the ratio of the dipleg pipe cross-sectional area to that of the reactor vessel. The superficial velocity up the dipleg would simply be equal to the superficial bed velocity.

The gas passing up through the bed surrounding the dipleg pipe must pass through the cyclone inlet and barrel before reaching the same cyclone outlet tube used by the gas passing up the dipleg pipe. In fact, the bed gas might have to traverse through one or two cyclones in series before reaching this outlet tube. The pressure in the cyclone body above the dipleg pipe is, therefore, lower than the pressure in the plenum space

above the surface of the main reactor bed by an amount equal to the pressure drop encountered by the bed gas in passing through the cyclone system. This lower pressure atop the dipleg pipe induces more bed gas to escape up the dipleg pipe. The entering gas draws in more bed solids and builds up a bed level within the dipleg pipe that is above the level in the main bed, by a height equivalent to the cyclone system pressure differential. The reactor bed and the dipleg act like two legs of a manometer in which the fluid level (in this case fluidized solids) differences reflect the pressure difference existing above the fluid surfaces in the two legs (see section on "Pressure Balance").

This differential causes the bed in the dipleg pipe to have an even higher L/D, or aspect ratio, than that corresponding to its bed immersion depth. In any event, its L/D is usually enough to lead to a slugging-bed form of fluidization, particularly if its diameter is small relative to the size of gas bubbles rising through the main bed. This action, which has been observed in Plexiglas scale models, is illustrated schematically in Figure 28. When a void (or bubble) engulfs the lower end of the dipleg pipe, its contents have no bed to rest against. They rush out the bottom, thereby instantaneously losing inventory in this "manometer" leg, resulting in an immediate inrush of gas and solids from the bed leg of the manometer. This sawing back and forth accentuates the dipleg slugging phenomenon and can affect its normal solids-handling capacity.

Figure 28

Dipleg Seal

Since the open bottom end of a dipleg provides a path for gas to escape from the reactor by circumventing pressure drop through the cyclone inlet, it is obvious that catalyst loss can be significant during the time required to fill the reactor vessel with bed solids. It is not often practical or desirable to charge solids to a reactor without a simultaneous flow of gas through the bed support grid. Frequently, it is even most feasible to fill the vessel by blowing the charge in with the feed gases. In either case, as illustrated in Figure 29, the open-ended dipleg can cause a rapid loss of charge while filling.

To circumvent such loss, which usually consists primarily of the finer size fractions of the charge, various valving arrangements have been devised. The most practical of these valve forms is the trickle valve, which consists simply of a loosely hung

Figure 29.

plate suspended at the open mouth of the dipleg, which has been bent via a mitered elbow, so that its opening is in a vertical plane. Though the plate is not usually machined to form a perfect seal against the dipleg mouth, it is sufficiently flush with the opening so that only a minimal amount of gas, and hence of solids, can squeeze through the clearance. The cyclone atop the trickle-valve-sealed dipleg, therefore, becomes operative, and collected solids can accumulate to some extent within the valve, as illustrated in Figure 29B. The function of the trickle valve is primarily to prevent excessive solids loss during reactor charging.

It has been claimed that a trickle valve also retards dipleg slugging and hence stabilizes the density and inventory within the dipleg during reactor operation. This occurs not

because of its valve-like characteristics, but simply because the dipleg opening is in a vertical plane. Exploratory experiments in Plexiglas equipment have shown that the valved dipleg expels solids only when the valve plate finds itself engulfed in a gaseous (bubble) void. It then quickly sucks gas and solids back up, either before slamming shut or afterwards through the plate-to-face clearance. In a high temperature reactor, it is seldom assurable that relative expansions will not distort dimensions and geometrics so that closures or clearances will deviate significantly from their cold counterparts.

The effectiveness of a vertical orientation of the dipleg opening lies simply in its limiting gas admission from only a portion of the area surrounding the periphery of the dipleg. As illustrated in the plan view sketches in Figure 30, a bubble approaching the bottom of the straight dipleg in view A can enter the pipe all around its periphery. In view B, a bubble rising slightly off center of the dipleg would find no open peripheral edge and possibly rise up completely diverted and thus unable to enter the dipleg. The bubble in Figure 30C could only partially enter the dipleg. The relative ineffectiveness of a "coolie hat" or flat "dollar plate" hung below the dipleg end is illustrated in Figure 30D. The gas passing around these baffle plates still faces 360 degrees of peripheral dipleg opening. The effects illustrated in Figure 30 have been qualitatively demonstrated in Plexiglas models.

Figure 30.

The experimental observations illustrated in Figure 30 suggest that in any event the angled dipleg end (sloped preferably to the solid's angle of internal friction) is desirable, with or without a plate hung in front of its vertical opening. In fact they suggest that other than in startup or filling operations (e.g. as in Figure 29) it would also likely be preferable not to even have the plate present. Patents have consequently been issued for Trickle valves with consumable (wooden) flapper plates or with consumable support rings, all intended to be burned off when the bed is brought up to temperature, after first filling to submerge the diplegs.

The counterweighted type of dipleg valve illustrated in Figure 29C is generally limited to above-bed installation since its operation depends heavily on maintaining free movement of the hinge pin at its pivot point. Since the clearances must be

close to obtain a reasonable seal, there is too ample an opportunity for particles to become lodged in these clearances and freeze the closure plate in an open or shut position if it were submerged in a bed. There are no tight clearances between the flapper plate and hanger rings of the Trickle valve (24) making it therefore a more versatile and operationally acceptable design.

It is neither desirable nor common practice to terminate diplegs in the disengaging height above a fluidized bed. The recovered elutriated particles should be returned to the bottom of the bed to maintain homogeneity and ample reactive surface. When returned to the top of the bed they are immediately available for reentrainment and result in an unnecessarily high recirculation through the cyclone system. Trickle valve sealed diplegs are intentionally terminated in the disengaging space only under rare instances where the density of the bed and the density of recovered fines differ so widely that a pressure balance indicates an intolerable backup level forcing the cyclones to an unjustifiable elevation. Unsubmerged sealed diplegs are more common in "transfer line" or "riser" reactors which utilize fluidized beds minimally, if at all.

In order to seal a Trickle-valve-ended dipleg in a dilute phase it is necessary to slope the valve mouth, or pitch the entire assembly, in order that some reasonable torque (presumably due to accumulated weight of collected solids) will be necessary

to "lift" the flapper plate off the valve mouth. Theoretically a perfectly perpendicular hung flapper plate would require negligible force to push it an incremental distance off the valve face, break the seal, and allow back flow of gas to destroy collection efficiency. In such instances the valves, or their seats, are usually sloped about 3° from the vertical when installed. Unless braced substantially, this slope could be altered when the vessel is brought to any appreciably elevated operating temperature. If the rate of collected solids inflow to the dipleg is low and/or the solids deaeration time rather short, it is not inconceivable to find that by the time the dipleg has filled to a height sufficient to have an inventory weighing enough to push the flapper open, this inventory will have deaerated and its weight therefore be borne on the walls of the high L/D dipleg and hence never brought to bear on the flapper, as originally conceived by the designer.

Height Above Grid

Figure 31 illustrates the blown dipleg situation which is not uncommon with poorly distributed grid designs. An even more common difficulty occurs when diplegs extend into poorly, if not totally, unaerated bed regions, as illustrated in Figure 32. Either condition is not as obviously detectable as might be implied in the illustrations. In a large reactor there might exist several sets of cyclones operating in parallel (12 sets,

Figure 31.

Figure 32.

for example, is not uncommon for example in a 40 foot I.D. vessel); and if only one or two diplegs were inoperative, it would not necessarily be discernable from, for example, a basically poor cyclone design. There is all too often a tendency to blame excessive loss rates on poor cyclone design when the fault might lie in dipleg location or size, grid design, and the like.

The blown dipleg in Figure 31 represents a situation in which solids find difficulty in passing down counter to the up-flowing gas. In diplegs located more judiciously, there exists a net counterflow of gas to varying degrees depending upon the

cyclone's collection rate, the size of the dipleg, and the aeration rate. It is good practice in situations where dipleg aeration taps are necessary (because of potential deaeration plugging) to locate the taps at intervals no greater than 10 feet. The aeration gas should then be introduced at rates such that the total quantity introduced results in a superficial gas velocity, leaving the solids level within the dipleg, not exceeding the incipient bubbling velocity. Excessive aeration can reduce cyclone efficiency, particularly in second-or third-stage units where the collected solids consist of fines having a low terminal velocity and the reverse upflow in the dipleg can lead to excessive reentrainment. In more heavily loaded diplegs, excessive net aeration upflow can limit dipleg solids downflow capacity. The countercurrent gas-solids capacity of a vertical pipe can be estimated from countercurrent gas-liquid correlations (25). The solids capacity calculated from Equation (18) plots on such a correlation superimposed on the liquid capacity at zero gas flow.

As illustrated in Figure 33 where cyclones must be located externally (to cool, to increase disengaging height, to allow for ease of servicing, and the like) and where the dipleg may be difficult to insert down into the bed (because of staging plates, multiple grids, baffles, or cooling or heating coils contained in the bed), its reentry through the vessel wall conveniently and effectively results in a vertically oriented dipleg

Figure 33

opening. The slope at the point of entry, shown in Figure 33, should preferably be equal to the solids angle of internal friction, α , whenever possible.

Riser Cyclones

The trend toward so-called "riser cracking" in FCC processes or "transfer line" reactors in chemical processing applications frequently results in arrangements having cyclones directly coupled to the line, as illustrated by the two examples in Figure 34. In such instances, whether the diplegs are submerged or not, the pressure, P_2 , inside the cyclone is greater than the pressure P_3 in the vessel surrounding the cyclone. As opposed to the conventional arrangement in Figure 25, where the pressure inside the cyclone is lower than that inside the vessel, and where bed solids tend to rise up into the dipleg, the arrangement shown in Figure 34a would cause gas to tend to blow out of

Figure 34.

both the overhead gas outlet tube and the dipleg. If submerged, the solids level in the dipleg could be below the level of the bed. An oversized dipleg in the first-stage cyclone of Figure 34a could therefore never seal without an intentional torque on the seal plate of the Trickle valve. Sealing is usually carried out by tilting the valve and increasing the plate weight, thereby affecting a counterweighted seal, as discussed previously and illustrated in Figure 34a. In such an arrangement, one can at least be certain of the inlet loading to the first stage cyclone.

Particle Attrition in Cyclones

Attrition Effects

The correlation of cyclone efficiency data requires

proper incorporation of the effects of particle attrition occurring within the cyclone system. In operating plants it is usually only practical (and then with great analytical difficulty) to sample the particle stream entering a cyclone and the stream leaving the cyclone (the losses) in the outlet gas duct. The difference between these two determines the collection efficiency, assuming that the fines in the exit stream are those that entered in the feed gas. The analysis of what is collected is thereby calculable to satisfy the material balance shown as loop 1 in Figure 35. In laboratory tests with smooth-walled ideal cyclones,

Figure 35.

collection efficiencies are most often obtained from a knowledge of the solids feed and the weight and analysis of the collected material. In this instance, the loss is calculated to satisfy a material balance. If particle attrition did not occur, these procedures would be satisfactory. That attrition does occur in some (if not all) commercial installations is obvious from an overall material balance as shown in loop 2 of Figure 35. No cyclone system exhibits 100 percent collection efficiency. The losses leaving the system consist of the finest bed particle sizes entrained into the cyclones. To maintain bed inventory, makeup solids must be added to the bed. If the makeup catalyst has the same size distribution as the bed, or as the original bed, it is obviously coarser than the material lost from the reactor via the cyclones. Therefore, if no attrition occurred within the reactor-cyclone system, the bed particle size analysis should become progressively coarser until theoretically the losses would be reduced to zero. However, this does not occur in practice. Beds do reach an equilibrium analysis, and this can consist of as much as 30 weight percent less than 40 microns, approaching, if not exceeding, the fines content of the makeup stream.

Correlating Attrition

Though an easily defined phenomenon, the problem of quantitatively correlating attrition rates is not only enmeshed in a multitude of variables and possible mechanisms but also suffers to date from a lack of distinct data, on any scale of operations.

From observations of the characteristics of many reactors accumulated over a period of years, coupled with some equally questionable small-scale tests, a reasonable correlation (26) (27) of attrition rates has evolved based on the following simplifying assumptions and definitions:

1. Attrition occurs mainly from a breakdown of the coarser particles. Fines do not get finer at any significant rates.

2. Fines are defined as all particles smaller than 44 microns. This is not only a conveniently measurable demarcation but also approaches an apparent optimum natural demarcation between easily and more difficultly attrited sizes, or, simply, sizes more or less able to follow changing gas streamlines.

3. Fines production by attrition occurs only in the first cyclone stage. Any subsequent stages handle not only a small fraction of the solids entering the first stage but also principally fines, and hence their influence on fines production is negligible.

The resulting correlating plot is shown in Figure 36 with the width of the drawn curve covering the scatter of the limited data on which it is based. The use of Figure 36 in calculating equilibrium bed fines content, entrainment, and cyclone system

Figure 36.

loss rates is illustrated by the examples in the following section. It has been experimentally verified that FCC catalyst attrition occurs principally in the first-stage cyclones. There is also commercial evidence that excessive gas outlet tube velocities give rise to severe attrition. In the normal range of inlet velocities of 60 to 100 feet per second, it is therefore recommended that outlet tube modification not be carried to values of A_O/A_I less than 0.4, or, in other words, to outlet tube velocities in excess of 250 feet/second.

Equilibrium Bed Particle Size Distribution

A correlation such as that shown in Figure 36 would be most useful in predicting the equilibrium bed particle size distribution attainable for a given overall cyclone system efficiency and a given fresh catalyst particle size distribution. Conversely, since the cyclone system efficiency is a function of the bed particle size distribution, it affords a means of prescribing the efficiency necessary to maintain the equilibrium bed and its attendant loss rate. As presented in Figure 36 the calculation of the equilibrium fines content would involve the following steps:

1. Assuming a bed particle size distribution.
2. Designing an apparently satisfactory cyclone system.

3. Calculating the anticipated fines content after passage of one bed inventory through the cyclone system.

4. Redefining the bed fines content or the cyclone system, if the results obtained in step 3 did not check with that assumed in step 1.

The above involves an extremely tedious series of iterations with a large number of interrelated variables. However, a solution using Figure 36 does not require trial and error or even iteration if the following simplifying assumptions, probably well within the accuracy of the data supplied in Figure 36, can be accepted:

1. In order to maintain bed inventory, fresh catalyst is added after every bed turnover at a weight rate equal to the loss rate. Admittedly, in practice the makeup inventory would seldom be added quite so frequently.

2. The superficial velocity in the reactor exceeds the terminal or free-fall velocity of the largest particle size in the bed. All of the bed particles are entrainable. This is quite common in FCC units and other fluidized beds operated to maximize production.

When equilibrium has been reached, the bed size distribution both before and after any turnover of one bed inventory must remain unchanged. The equilibrium bed fines concentration is therefore given by

$$\chi_E = \frac{(\text{total fines}) - (\text{fines lost}) + (\text{fines added})}{(\text{inventory})} \quad (30)$$

$$\text{total fines} = \chi_t (\text{pounds of inventory}) \quad (31)$$

$$\text{Fines lost} = y(1 - E_{OA}) (\text{pounds of inventory}) \quad (32)$$

$$\text{Fines added} = \chi_f (\text{pounds of fresh catalyst added}) \quad (33)$$

$$\begin{aligned} (\text{Pounds of fresh catalyst added}) &= \quad (34) \\ &(\text{pounds of catalyst lost}) \\ &= (1 - E_{OA}) (\text{pounds of inventory}) \end{aligned}$$

Where:

χ_t is obtained from Figure 36 and represents the amount of fines that would be present if nothing had been lost from the last cyclone stage.

E_{OA} is the combined overall collection efficiency of the cyclone system.

y is the weight fraction of fines in the material lost from the last cyclone stage.

x_f is the weight fraction of fines in the fresh makeup catalyst.

Combining Equations (30) through (34) gives:

$$x_e = x_t - (1 - E_{OA}) (y - x_f) \quad (35)$$

At equilibrium, x_e must equal x_c so that Equation (35) may be represented as:

$$x_t - x_c = (1 - E_{OA}) (y - x_f) \quad (36)$$

which multiplied by $e \cdot v_i$ represents the ordinate of Figure 36. At equilibrium, Figure 36 is therefore a plot of

$$(1 - E_{OA}) (y - x_f) e v_i$$

versus x_e , under the simplifying assumptions made in deriving equations (30) through (36), as labeled schematically in Figure 37. The simplest use of Figure 37 can be illustrated by reference to the four examples in Table 4.

Figure 37

Table 4. Calculation of Fines Content

	<u>Case 1</u>	<u>Case 2</u>	<u>Case 3</u>	<u>Case 4</u>
x_F	0.22	0.39	0.22	0.25
y_F	1.0	1.0	1.0	1.0
e	0.5	0.29	0.32	1.05
v_i	66.0	62.0	75.0	72.0
E_{OA}	0.999961	0.999982	0.999786	0.9999705
Ordinate from Figure 37				
$[K v_i (1 - E_{OA})$				
$(y - x_F)]$	0.001	0.0002	0.004	0.0067
	0.16	0.34	0.03	0.13

In all of the cases listed in Table 4. the composition of the makeup catalyst (and hence x_F) is known. When a cyclone system is offered by a vendor, the first-stage inlet velocity v_i is also established. If the vendor claims the system should not lose any particles larger than 40 microns ($y = 1.0$), this point can be checked if the vendor's assumed bed composition differs greatly from the calculated equilibrium composition. The entrainment rate

or first-stage loading e has been established from conventional correlations and will not change significantly over reasonable ranges of bed size distribution, since the operating conditions are such that even the maximum particle size in the bed is entrainable. (This can again be checked once the bed equilibrium has been approximated.) The preceding information coupled with Figure 36 permits estimating the overall cyclone efficiency necessary to achieve a desired equilibrium fines content or, conversely, what fines content might be anticipated if the vendor quotes a claimed overall system collection efficiency. Since at fixed operating conditions e is dependent on X_f , E_{OA} is dependent on X_f , v_i , and the number of stages, and y is dependent on E_{OA} , it is obvious that changing any single variable requires a review of the effect on all the others. The examples in Table 4 illustrate typical values encountered in commercial fluidized bed units using conventional silica-alumina base (for example, where $K = 1$) carriers.

In the broadest sense, the abscissa of Figure 37 could be labeled $(X_o)_c$, representing the weight fraction of fines in the stream entering the cyclone system. The ordinate could be labeled

$$K \sum v_i (X_t)_c - (X_o)_c$$

where $(X_t)_c$ represents the weight fraction fines in the combined streams leaving the diplegs and gas outlet duct of the

cyclone system. If the entrainable weight fraction of the bed is given by W_e , then

$$(x_t)_c = x_o / (W_e)_o$$

and equation (35) would be simply

$$\frac{x_e}{(W_e)_E} = (x_t)_c - (1 - E_{OA}) (y - y_f) \quad (37)$$

The design of the cyclone system for any fluidized reactor begins with a statement of the initial or desired equilibrium, that is, the bed particle size distribution. Whether this can be attained, or what the equilibrium distribution would converge to for any given design and operating conditions, is then the result of a trial-and-error procedure as outlined in Figure 38.

Figures 36 and/or 37 represent at best the current approach to attrition in cyclones. There are essentially no data available on the effect of particle properties, cyclone surface characteristics, attrition to sizes other than simply $\pm$ 325 mesh, and g forces within the cyclone. Whether or not the conceivable sophistication in design procedures that would be possible from a correlation of all such variables will ever evolve, it is obvious that attrition could be responsible in many instances for apparent discrepancies in measured cyclone collection efficiencies. Attrition occurring in the cyclone raises the question of whether the

collected fines are "fed fines" (those fed into the cyclone) or "generated fines" (generated or attrited in the cyclone) and hence on what size distribution one should base fractional collection efficiency.

Equilibration Time

Figure 36 is intended here simply as an exploratory or suggestive correlation for further study. It requires careful judgement of the accuracy of the information used in its applications and, as expected, is quite sensitive to cyclone efficiency. When used iteratively, Figure 36 should enable calculation of the number of inventory turnovers and hence the time interval necessary to reach an equilibrium bed particle size distribution. As illustrated by the sample calculations in reference (27) this equilibration time can vary from hours to years depending on bed solids inventory, entrainment rate, and cyclone efficiency.

Design Innovations

Acceptance of the basic theory allows a reasonably realistic evaluation of the probable virtues of the myriad design innovations suggested and explored over the years. These include addition of vortex breakers, vortex stabilizers, underflow, secondary rotation-inducing streams, spinning outlet tubes, nested cyclones, reduced outlet tube diameters etc. It would be futile to attempt

to list all the cyclone patents ever issued and make an evaluation of each. Were any of significant value, there is little doubt that their vendors would have come to the foreground surpassing all others in acceptance of the innovation. Differences among such current offerings represent principally scientifically unverifiable claims intended to permit an entree to competitive bidding. The purchaser's final analysis resolves itself to eliminating the bids bearing irreconcilable dimensions, and then selecting from the remaining on the basis of price, delivery promises, mechanical guarantees, prior credibility, and luncheon menus, in that order. Some design modifications, proposed and practiced, are worthy of review relative to the fundamentals of cyclone theory.

Vortex Breakers

The intent of vortex breakers was to limit the length of the exiting gas vortex so that it not touch the cone of the cyclone and reentrain already-separated solids. This has obvious virtues if the cyclone were designed shorter than its natural vortex length, $\mathcal{L}$. Among cognizant cyclone designers it is now the practice to simply supply cyclones whose length exceeds $\mathcal{L}$ as specified in Figure 14. It might be argued that though the vortex breaker overcomes the reentrainment in a cyclone that is too short, it must simultaneously augment a more rapid decay of the spiralling velocity and thereby effectively decrease N_s . From the illustration in Table 3 it is obvious that this would con-

tribute almost negligibly to any deterioration in cyclone performance, in contrast to the loss in efficiency attending re-entrainment from the cone surface, as borne out in past practice. In many applications major difficulty with vortex breakers lies in their eventual erosion, failure, and possible plugging of dip-legs.

Vortex Stabilizers

It has been suggested that the exiting gas vortex tends to wander randomly back and forth and hence "lap" at the cone walls, thereby leading to particle reentrainment even in a cyclone designed to exceed $\mathcal{L}$. The latest innovation (28) intended to stabilize the position of the vortex suggests the insertion of a rod along the centerline or the axis of the cyclone reaching from the gas outlet tube down to the level of a vortex breaker, or to the terminus of $\mathcal{L}$. Aside from attempting a demonstrable experiment in a cyclone of proper length, this is difficult to justify even intuitively in light of an experimental program carried out in 1956 by one of the major engineering contractors which showed that the collection efficiency of a well-proportioned cyclone was independent of its orientation. The cyclone was as efficient in a horizontal position as in either a tilted or vertical position. If the vortex truly wandered in a vertically oriented cyclone, it is difficult to imagine that it would not dip into contact with the cone to a greater extent in a horizon-

tally oriented cyclone. Neither theory nor practice (relative to well-designed alternates) have lent support to the concept of substantial vortex instability or to its stabilization via axial rods.

Underflow

Drawing off some 2 or 3% of the feed gas along with the collected solids, out of the bottom of the cone or dipleg, is widely practiced in the petroleum industry in the third stage collectors of Fluid Catalytic Cracking Unit regenerators. This obviously lessens the particle losses in the cyclone overhead since the underflow gas leaving through the cone carries its proportionate quantity of fines which it would otherwise have carried out through the overhead gas discharge line (see equation (24) under the section on performance correlations).

Underflow is generally practiced in multicyclone arrays to assure no draft or flow, in reverse direction, up through the cone of any single unit in the array. Underflow may take preference over the more conventional practice of simply providing sufficient pressure drop to assure equal flow distribution, since in later stages the collected fines frequently exhibit poor flow characteristics, which would benefit from a cocurrent fluid stream. The Underflow principle is in any event not at variance with the premises on which cyclone theory and two-phase gas-solids flow are founded.

Secondary Streams

In accord with cyclone theory it would appear beneficial to thwart the spiralling velocity decay by injecting secondary rotation-inducing gas streams. Experimental evidence appears to support such a procedure when practiced in small scale (e.g. 6" diameter) cyclones. However, in 36" to 72" industrial scale units there is no evidence of enhanced performance. It is unclear whether this is due to impractical secondary gas requirements and/or the greater difficulties in achieving desirable patterns of injected flow distribution across wider annuli.

Mechanical Rotation

An alternative to secondary gas injection would appear to lie in mechanically spinning the gas discharge tube. Though again not at variance with theory this presents practical difficulties. The rotational speed must be substantial in order to have a significant effect and the rotating bearing must be sealed and reliably gas purged at all times. As a means of thwarting decay of the spiralling velocity it is also obvious from Table 3 that its effect would have to be felt lower down in the cyclone, after an initial one or two turns. This could only be accomplished by extending the gas discharge tube, thus requiring an additional internal sealable bearing and an equally lengthened cyclone, both of which contribute significantly to cost, assuming that a sealed

bearing were feasible in such environments. There are no known industrial applications of rotating gas discharge tubes in cyclones. (as distinct from centrifuges or what have been referred to as centrifugal separators).

Nested Cyclones

The concept of nested cyclones differs from simple series staging by requiring that only the outer or first stage be designed as a pressure vessel and/or offering substantial saving of space in applications where the cyclones are installed within a process vessel in the disengaging space above a fluidized bed. Since subsequent stages must capture progressively finer particles they must be progressively smaller in the dimensions controlling path width and inlet velocity to conform with equation (4). If subsequent stages are thus made smaller, then they could obviously be nestable. Some years ago a patent was granted on the claim that overall performance is enhanced by this reduction in the size of the second and third stages; if ever challenged or attempted to be enforced, it would surely have been found invalid in the obvious face of equation (4) and years of prior practice. The complexities in fabrication and the difficulties in maintenance have ruled out industrial acceptance of the nesting concept.

Widened Annuli

C.E. Lapple (19) noted that "Reducing the gas outlet duct diameter will increase collection efficiency." As this was gradually applied in industrial practice it was discovered that the natural vortex simultaneously lengthened and the pressure drop increased dramatically as the diameter of the outlet tube approached

zero. A mechanism leading to the increase in efficiency was based on particle saltation velocity (6). It simply showed that if the spiralling velocity exceeded the saltation velocity (the limit above which a particle could never come fully to rest in the bottom of a horizontal pipe) then the particle could not remain on the cyclone wall even if its terminal velocity were such as would allow it to reach the wall. Reducing the diameter of the outlet tube has the effect of increasing the annular width and thereby reducing the spiralling velocity, if it otherwise exceeded the saltation velocity of the particles intended to be collected. The counter-effective lower acceleration field leads to an optimization in design which is covered at length in Chapter 11 of API publication 931 (3) concluding that the optimum size is that which exhibits an annular velocity of 1.35 times the saltation velocity. This corresponds to an outlet to inlet area ratio of 0.4 for a cyclone with a rectangular inlet and a cylindrical gas outlet tube and essentially matches empirically optimized industrial practice. Further increase in the ratio of annular to saltation velocity shows that when the velocity reaches twice saltation the efficiency begins to decrease as already collected particles are picked up and reentrained; experimental data show pickup velocities from "salted" layers to be approximately twice the saltation velocities. The saltation velocity concept is incorporated in a programmed calculation procedure suggested by Licht et.al. (29).

Shave-Offs, Skimmers and Tangentials

The concept of removing collected solids by allowing them to fall into slots in the cyclone wall thus skimming or shaving off

the centrifuged solids before they travel any additional distance where they might encounter obstructions or other reentrainment inducing means, has not been accepted as general industrial practice. The original intent of the Van Tongeren (30) shave-off was better accomplished by an inlet duct flush with the cyclone roof. From an efficiency point of view it is the finest particles, last to reach the cyclone wall, which would best be skimmed. However, in view of the velocity decay in the later spirals it becomes more effective to reenter a higher velocity subsequent stage. Skimming through slots in the gas discharge tube is similarly less effective than a subsequent added stage. Its effectiveness is predictable from equation (4) if the outlet tube is designed to yield a calculable spin velocity.

Document FE-2357-44 relating to the CFCC Development Program at the Schenectady Laboratories of the General Electric Co. under Contract No. EX-76-C-01-2357 stated that "outlet skimming and the substantially increased engagement length, which extends the inlet annular swirl region provided significant improvements in separative efficiency over a modified Stairmand cyclone." Increasing N_s by lengthening the outlet tube insertion while simultaneously increasing cyclone length, as well as bleed off skimming, are all effects in conformance with equation (4); it would obviously be of interest to determine whether the degree of improvement in efficiency is precisely calculable and whether the calculations would also be in good agreement with the results observed on the "modified Stairmand cyclone." In all such claims of improved performance it is most important to know whether the comparison is being made with a well-designed basic cyclone and whether the

comparison is predictable and made under comparable operating conditions. This requires first a sound correlation for the basic cyclone with best dimensional ratios.

Commercially available (31) "Skimmers", sometimes referred to as "Tangentials" are essentially cyclones without cones, a barrel height equal to the inlet duct height, and an operating residence time less than one spiral. Their performance is predictable from equation (4) and the correlations in Figures 12 through 16. Table 3 suggests the rationale behind the surprisingly good performance of such short separators.

The so-called "Wedge Separator " (32) is also essentially a less than one turn centrifugal separator or skimmer. On a small scale, with a narrow inlet width, it exhibits predictably high efficiencies.

Bed Recycle

The increase in anticipated removal of fines in a conventional cyclone may be attributed to the impingement of coarse and fine particles whereby the coarse overtake and sweep fines along with them. It has been found in practice that this mechanism can be supplemented by introducing a secondary stream of coarse material into the cyclone (33). Though shown experimentally to increase collection efficiency, the added complexity of the necessary piping makes it industrially unattractive. A suggested arrange-

ment for recycling bed catalyst as the coarser stream to increase the efficiency of collection of finer particles entrained from the bed surface in a fluidized solids reactor is illustrated in Figure 39.

Figure 39.

Mechanical Design Aspects

Refractory Lining

All cyclones subjected to high solids loadings, including cones and ducting, are usually lined with 3/4- to 1-inch thick

refractory concrete reinforced with steel grating panels. These panels are either directly welded to the cyclone or occasionally supported by 1/4-inch thick spacers. The lining material is a high temperature abrasion-resistant refractory preferably applied pneumatically. The supporting steel grating is generally 1 3/4 by 1 3/4 inches and is of No. 14 gage USS Type 410.

When supporting spacer bars are used, they are generally installed circumferentially and located longitudinally on 9-inch centers. The bars are attached to the cyclones with 1-inch long staggered fillet welds on approximately 9-inch centers. The panels or shapes are usually oriented so that the grating bars are perpendicular to the spacer bars and attached thereto with 1/4-inch fillet welds spaced on approximately 9-inch centers. All first and last grating bars are tack-welded to the spacer bars. Figure 40 illustrates installation details of these steel grating panels and bars. The panels or shapes are cut from standard size panels and rolled or bent the "easy way". A 1/8-inch wide clearance is maintained between all panels or shapes, and this clearance is also filled with refractory cement. The refractory is leveled flush with the inside or exposed surface of the grating panels and then carefully allowed to dry or cure. The freshly poured or gunited refractory must be prevented from becoming dry any time within 24 hours of its application. In the initial stages of moisturizing with curing water sprays, care

must be exercised not to flood the surfaces. The curing water must not be applied until it is no longer possible to wipe damp

Figure 40.

cement from the surface. Any artificial drying should not begin until 24 hours after the final application of the refractory. The drying procedure then consists of maintaining a surface temperature of 125°F to 200°F for 18 hours, and 200°F for the last six hours. The temperature is then increased 50°F per hour until a surface temperature of 500°F is reached. This temperature is maintained for at least 12 hours. Soundness is tested by rapping with a ball-peen hammer. Any void areas indicating incomplete adherence are chipped out, refilled, and re-cured.

The top portions of cyclones are frequently stiffened with knee braces attached with full-thickness (3/8-inch to 1/2-inch) welds to the gas outlet tube and the flat top head of the cyclone. Strips are attached to the knee braces to form cross-sectional tees or ells.

In order to prevent bending caused by expansional movement of the cyclones, each cyclone is supported by a hanger that usually consists of two flat straps. Necessary adjustments to the correct position of the cyclones are usually carried in the field. If hanger rods with clevises are to be used, stiffeners should be added between the clevises. This is also done in the field, after the cyclones have been installed.

In vessels bearing exothermic combustion reactions subject to so-called above-bed after-burning it is common practice to

provide one or more steam connections to the shell leading to internal alloy steel pipes. The ends of these pipes are located in the center and in front of each first-stage cyclone inlet.

The plan view clearances between adjacent cyclones and between cyclones and vessel walls are normally maintained at a minimum of 3 to 6 inches respectively.

Manholes, inspection openings, dipleg connections, and all joints are of full-welded design. No bolted connections are acceptable since experience has shown them to inevitably leak, if not fail.

Leak Testing

After installation, cyclone systems should be leak or pressure tested using any suitable arrangement. A typical one is illustrated in Figure 41. Vents, inlets, and Trickle valves can be sealed with 3/4-inch foam rubber gaskets, clamps, and 3/4-inch plywood plates where necessary. Plant air is admitted through any dipleg aeration tap, or other nozzle specifically provided therefore, until bubbling is observed through a fixed head of fluid (usually 1 1/2 pounds per square inch is sufficient, depending on the design pressure drop). All joints are then tested for leaks by applying a conventional soap solution.

Figure 41.

Internal Cyclone Support and Bracing

Cyclone systems installed internally are normally supported from the vessel head, hung either individually with rods or straps, or as an assembly from pipe or I-beam cross members. An almost infinite variety of details exists, bounded solely by the ingenuity of the mechanical designer and the metallurgical characteristics of the construction materials. Basically, the objectives for support devices are simplicity, flexibility with regard to thermal expansions, maintenance of weld integrities with minimum stress (particularly of plenum welds), ease of replacement and servicing, and sufficient rigidity to withstand repeated random impacts from the flow of several hundred pounds per second of bed solids from any and all directions. Such impacts on diplegs submerged within the fluid bed are restrained via networks of interdipleg bracing which, in the case of long diplegs, may be

Figure 42.

located at more than one level and sometimes tied to the vessel wall.

The variety in detail among different installations is augmented by the vessel designer's choice of plenum arrangements. Figure 42 for example, illustrates details of a typical strap-type support system in a small FCC regenerator provided with six sets of second-stage cyclones and an annular plenum. Figure 43 illustrates schematically a beam-type support system for a larger vessel with a centrally located plenum. The arrangement in Figure 43 relies solely on sliding, as opposed to pivotal, connections to permit thermal expansion. This minimizes distortive stresses on the dished plenum membrane and considerably simplifies the mechanics of installation. Hanger straps may be made to ride within a slotted pipe, hang over a pipe, or hang from the flanges of an I-beam.

Inter-dipleg bracing can be fabricated from angle or channel iron, but most frequently consists simply of approximately 3-inch Schedule 80 pipe, typically ASTM A312, Type 304. As illustrated in Figure 44, the dipleg clip might be a complete annular disc or a partial disc with a reinforcing saddle backup.

Erosion

Figures 45 a and b illustrate the severe erosion and surface

Figure 43.

Figure 44.

A

C

B

Figure 45.

attack that can occur at the top of the barrel in the vicinity of the inlet duct as a result of excess stress due to after-burning. Figure 45C illustrates erosion at the apex of the cone, just above the dust hopper, as a result of insufficient cyclone length following installation of an outlet tube modification.

Vendor Responsibilities

Specifications and Guarantees

Unless the purchaser of a cyclone system for any fluidized bed process carries out the design of the cyclone recovery system, even if only on a relatively superficial basis, his specifications might impose restraints on a vendor that could make the desired performance level unattainable. In any event, a vendor's guarantees, other than those of mechanical integrity, usually relate either to a quoted cyclone efficiency or a maximum particle loss rate. In either case such guarantees are essentially unenforceable simply because no agreement has been reached as to the calculation of the particle entrainment rate from the bed entering the first-stage cyclones. If the loss rate is greater than desired, the vendor can always claim the entrainment to be greater than what was specified to him. Unfortunately few, if any, vendors are equipped with the capability and/or ever provided with the process data necessary to calculate the anticipated entrainment. As has been pointed out, even the design of

the grid can become an important influence on entrainment. It is therefore obvious that cyclone design should be the function of the process design group, whether it be the engineering contractor or the operating purchaser. The vendor's position can only be that of a fabricator with at best the responsibility for structural integrity and possibly installation, lest he become in effect a limited engineering company charged with all the commensurate responsibilities, process design data, and operational guarantees.

Assuming a conventional vessel arrangement, with a well-designed grid distributing feed gases uniformly over the bed cross section, the proper design of the cyclone system can require the following information:

1. The density, viscosity, temperature, pressure, and actual volumetric flow rate of the gas leaving the bed surface.
2. The inside diameter of the vessel at all elevations from the bed surface to the tangent line at the head, as well as the head shape and the dimensions of any preconceived plenum and/or exit gas nozzle.
3. The bed depth, dimensions of any internals, depth of gas jet penetrations into the bed at the grid, and points of solids feed and/or drawoff from the bed, as well as the nozzle sizes and

the solids flow rates through these nozzles.

4. The apparent particle density, the operating bed density, the loosest bulk density of the bed, any data on the rate of deaeration or settling time of the bed, the drained angle of repose and/or angle of internal friction of the bed solids, and any other information on particle properties relating to their stickiness, or attrition resistance at operating conditions.

5. The particle size distribution desired in the equilibrium bed and the size distribution of makeup solids and of re-entering streams in the case of circulating solids flows.

6. Limitations on allowable pressure drop through the cyclone system and the allowable particle loss rate.

If the cyclone designer is not familiar with a reliable means of predicting the particle entrainment rate and size distribution from the above information, then such data must be supplied as a function of the bed size distribution and the location of the cyclones. In addition, if he is unable to estimate dipleg densities from the size distributions of the collected solids, then such information must also be supplied.

Malfunctions

All too often when losses exceed anticipated levels the immediate reaction is to ascribe the malfunction to poor cyclone design. It is not likely that conformance to the criteria described in this report would lead to gross design errors. In many instances excessive loss rates are discernibly attributable, by careful calculation procedures, to erroneous or poorly estimated design specifications or to poor bed or vessel design or both. Some questions arising from typical areas that should be reviewed are listed below.

1. Does bed analysis correspond to that on which entrainment was based?
2. Do bed, dipleg, and particle densities correspond to those specified or used in the original design?
3. Has the plant been in operation long enough to have reached an equilibrium bed size analysis?
4. Was the entrainment rate properly calculated?
5. Is gas distribution uniform at the grid, and, if not, was the entrainment rate then based on the correct superficial velocity, and/or are diplegs possibly inoperative?

6. Under the operating conditions, is the cyclone long enough to avoid reentrainment or erosion at the cone?

7. Have the particle saltation or pickup velocities been exceeded in the annulus?

8. Are the diplegs large enough to handle the downflow of collected solids in all stages?

9. Are the solids unexpectedly sticky and the cyclones therefore too long?

10. Do cyclone pressure drop measurements indicate plugging or maldistribution at the inlet?

11. Is the inlet piping arrangement such as would lead to unequal solids and/or gas rates to cyclones operating in parallel?

12. Could poor hanger design have caused a weld break in plenum or other locations, or could a poorly placed refractory have fallen into a dipleg?

Sample Design and Performance Calculations

Collection Efficiency

As is the case with any piece of process equipment there is no one precisely definable optimum design when one considers space allocations, performance, efficiency, pressure drop, cost, etc. as a whole. This is particularly the case with vendors' selections from standardized dimensions. However, in terms of performance (collection efficiency) and pressure drop, calculational procedures can converge toward the best suited dimensions and certainly permit an accurate evaluation of vendors' proposed or existing designs. Vendor's proposed designs are basically derived through a trial and error process matching their standard dimensions to the process specifications. Frequently a rapid estimate of cyclone size or number, in non-critical applications, is based on assuming an inlet gas velocity of 60 ft/sec, an inlet area of 2 to 3 times L_w^2 (whatever is necessary to avoid a fractional number of cyclones), and a barrel diameter of 4 L_w ; if subsequently calculated collection efficiencies or pressure drop do not meet specifications then necessary adjustments are appropriately made in either inlet areas, outlet areas, number of parallel units, or combinations thereof.

To illustrate the procedure from a more fundamental approach, consider the design of cyclones specified to recover at least 60%

of the particles having a diameter of 10 microns in a 1,470,000 ACFH gas stream bearing 200 lbs/hr of particles with a size distribution as subsequently given below. The particles have an apparent density of 110 lbs/cu.ft.; the gas viscosity and density are 0.02 centipoises and 0.1 lbs/cu.ft. respectively. The pressure drop through the cyclone collectors is not to exceed 6 inches of water.

From Figure 12 for a tangential cyclone $D_p/D_{pth} = 1.31$ at 60% fractional collection efficiency. Since the loading is less than 1 grain/cu.ft. ($200 \times 7000 / 1,470,000 = 0.95$) this will also be the operating efficiency. Therefore to meet specifications the cyclone or cyclones will need to achieve a cut point diameter: given by:

$$\begin{aligned} D_{pth} &= 10/1.31 \\ &= 7.63 \text{ microns} \end{aligned}$$

Substitution in equation (4) yields:

$$\frac{7.63}{25400 \times 12} = \sqrt{\frac{9 \times 0.02 \times 0.000672 L_w}{12 \pi N_s v_1 (110 - 0.1)}}$$

So that $L_w/N_s v_1 = 0.0215$

Where L_w is in inches and v_1 in ft/sec

Maintaining dimensions within 1/8" increments the following number and dimensions of cyclones would satisfy the specified performance:

No. of cyclones in parallel	U_i	U_s	L_w "	h "	D_B "	A_O/A_I
1	120	5.4	14	35	56	1.26
2	100	5.0	10.75	27.375	43	1.23
3	90	4.8	9.25	23.5	37	1.24
4	79.8	4.6	7.875	23.375	31.5	1.06
14	60	4.0	5.125	13.625	20.5	1.18

In this instance the gas stream represents a typical feed to a third stage collector in a series so that the pressure drop calculations will not involve a significant, if any, contraction loss. At a loading of 0.95 grains/cu.ft. the solids acceleration term, equation (26), is also negligible. The remaining barrel, reversal, and exit losses from equations (27), (28) and (29) yield the following:

No. of cyclones:		1	2	3	4	14
d_{hi} "	=	20	15.4	13.3	11.8	7.5
$Re/10^6$	=	1.49	0.957	0.74	0.58	0.28
f	=	0.0033	0.0035	0.0037	0.004	0.0045
ΔP_B	=	2.7	1.84	1.50	1.18	0.67
ΔP_{re}	=	4.3	3.0	2.4	1.9	1.07
U_B	=	23.9	20.3	18.2	18.8	12.7
U_e	=	95.2	81.3	72.6	75.3	50.8
ΔP_e	=	3.64	2.65	2.12	2.28	1.04
ΔP_{total}	=	10.64	7.49	6.02	5.36	2.78

Four cyclones 31.5 inches in diameter with an inlet velocity of about 80 ft/sec would apparently satisfy the specifications. Equal performance at lower pressure drop requires a greater number of smaller cyclones; this illustrates why high collection efficiency of fines coupled with low pressure drop forces a multiclone type of design.

If the feed particle size distribution were typically as given in the first two columns below, the overall collection efficiency at the given loading (less than 1 grain/cu.ft.) would be 45% which is simply the summation of the collected fractions of each of the feed components.

Feed wgt. fract.	Avg. D_p microns	$D_p/7.63$	fig. 12 $E_o\%$	Col. 1 times Col. 4
0.005	1.5	0.197	3	0.015
0.005	1.8	0.236	5	0.025
0.01	2.1	0.275	7.5	0.075
0.03	2.5	0.328	11	0.33
0.05	3.1	0.406	16	0.80
0.10	3.9	0.511	24	2.40
0.10	4.8	0.629	31	3.10
0.10	5.6	0.734	38	3.80
0.10	6.5	0.852	44	4.40
0.10	7.4	0.970	49	4.90
0.10	8.5	1.11	54	5.40
0.10	10.0	1.31	60	6.00
0.10	12.0	1.57	66	6.60
0.05	14.4	1.89	71	3.55
0.03	16.4	2.15	74	2.22
0.01	18.1	2.37	76	0.76
0.01	21.0	2.75	79	0.79
				Σ : 45.165%

Had the specifications been given in terms of an overall required collection efficiency, or in terms of all particles greater

than a given size (instead of specific to one diameter) then a more complex trial and error procedure would have been necessary to match a summation of the last column above to that specified. In this instance 68% of the particles greater than 60 microns and 38% of the particles smaller than 60 microns would be collected. Since the inlet loading was less than 1 grain/cu.ft., the figures in the last column above, divided by their summation, represent the weight fractions of each of the corresponding particle sizes in the second column, found in the total collected material.

If the inlet loading had been specified as 10 grains/cu.ft., then the overall collection efficiency, from Figure 15, would be approximately 65% with a greater relative efficiency of collection of the coarser sizes; this would again have necessitated a trial and error procedure even if the specifications had been specific to only one given size.

Similar examples illustrating the application of Figures 12 through 15 in evaluating the performance of existing cyclone installations are presented in the section on Industrial Performance Data.

Pressure Balance and Dipleg Densities

Following a procedure such as in the example, or as outlined in reference (3), the geometric weight mean particle diameters of a fluidized bed, and the materials collected in each of 3 stages of overhead cyclones, have been determined as listed in the first column of the table below. For this particular catalyst or bed material, having an apparent particle density of 75 lbs/cu. ft., the void fractions in the most loosely packed beds, before the onset of bubbling, have been determined as shown by the dashed curve in Figure 46. If the fluidized density of the bed is 30 lbs/cu.ft., all cyclone diplegs are submerged 10 feet below the bed level and the pressure drops across each stage are as given in the table. The calculated level of the solids in the third stage dipleg, above the level of the bed surface, is calculable from Figures 25 and 46 as follows:

<u>Location</u>	<u>Mean D_p microns</u>	<u>ΔP "H₂O</u>	<u>From Fig. 46 ε</u>	<u>ρ</u>
Bed (unfluidized)	300		0.50	37.5
" (fluidized)				30.0
1st Stage Cyclone		10		
" " Dipleg	250		0.51	36.8
2nd Stage Cyclone		20		
" " Dipleg	50		0.60	30.0
3rd Stage Cyclone		30		
" " Dipleg	12		0.65	26.3

$$L_{D_3} \rho_{D_3} = S \rho_B + (P_1 - P_4)$$

$$L_{D_3} (26.3) = 10 (30.0) + (10 + 20 + 30) (5.2)$$

$$L_{D_3} = 23.27 \text{ ft.}$$

Therefore level above bed surface = $23.27 - 10$
 = 13.27 feet.

If the dipleg was submerged only 5 feet the 13.27 foot figure would drop to 12.6 feet.

The level in the first stage dipleg is 0.4 feet below the bed for a 10 foot submergence

$$L_{D1} \rho_{D1} = S \rho_B + (P_1 - P_2)$$

$$L_{D1} 36.8 = 10 (30.0) + (10) (5.2)$$

$$L_{D1} = 9.6$$

Level below bed surface = 0.4 feet

Figure 46.

Manifolding

To illustrate the influence of cyclone pressure drop on improving flow distribution, consider the estimation of air distribution to a 12-ported manifold, as illustrated in Figure 47, before and after the attachment of 12 identical cyclones which have an additional pressure drop of 14" of water.

Figure 47.

The relative flows out of the first and last ports can be estimated from the correlation (34) shown in Figure 48 which represents a composite of the data of the principal investigators (35) (36) of blowing manifolds.

The manifold in Figure 47 can be regarded as two 24" x 24" cross section manifolds, side-by-side, each with 6 ports. For either one, based on the nomenclature of Figure 48 and the hy-

Figure 48.

draulic diameters of the rectangular cross sections:

$$d_h = 4(9 \times 24) / 2(9 + 24)$$

$$= 13.091 \text{ inches}$$

$$D_h = 4(24 \times 24) / 2(24 + 24)$$

$$= 24 \text{ inches}$$

$$n d_h^2 / D_h^2 = 6(13.091/24)^2$$

$$= 1.785$$

$$\alpha = \frac{n d^2}{D^2} \times \frac{D}{4L}$$

$$= 1.785 \times \frac{2}{4 \times 20}$$

$$= 0.0446$$

$$F = 0.003/1.85 \times 0.0446$$

$$= 0.036$$

From Figure 48 at intersection of $n d^2 / D^2 = 1.785$ and $F = 0.036$ the ordinate is approximately 0.8 and the abscissa about 20,000.

There is therefore an 80% greater flow out of the last port, at the far end of the manifold, than out of the first port nearest the inlet end of the manifold.

Without the cyclones attached the pressure drop across the manifold

$$P_I - P_D = \frac{0.08}{20000} \left(\frac{86400}{60 \times 2 \times 4} \right)^2$$

$$= 0.1296 \text{ psi}$$

With the cyclones attached the pressure drop is increased by 14" of water or 0.506 psi making the abscissa of Figure 48:

$$\frac{\rho V^2}{(P_I - P_D)} = \frac{20,000 \times 0.1296}{0.1296 + 0.506}$$

$$= 4078$$

so that at $F = 0.036$ the ordinate now becomes approximately 0.2 indicating at most 20% greater flow into the cyclones at the far end than into those nearest the inlet. Judicious alterations in manifold or cyclone dimensions, within allowable pressure drop limitations can further improve the distribution.

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Fine Particle Recovery
(Via dry bed filtration)

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The suggestion that extraneous solids could be removed from gaseous streams by passage through a bed of particles is to this day regarded by many as novel, if not improbable, despite the fact that the average engineer is well aware of the technology of treating liquid streams by passage through beds of particles for purposes of removing undesirable elements via ion exchange and more simply for removing any suspended solids. The suggestion becomes more acceptable and credible by considering the analogy between the well accepted bag filter, which represents a tortuous path through the interstices in a layer, or bed, of interlaced cylinders (fibers), as opposed to the granular bed which represents a tortuous path through the interstices in a layer of interlaced spherical or angular shapes.

The novelty of the concept, in principle, vanishes rapidly when a search of the patent and technical literature reveals that shallow bed, granular, filtration devices in various forms have been proposed, explored, tested and even marketed over the period from at least 1896 to the present day.

Purposes and Applications

The principal interest in development of Granular Bed Filters today rests in removal of particulates from hot pressurized process gases which could then be utilized to drive turbines and other equipment for efficient recovery of otherwise lost energy. A secondary

interest lies in insuring that exhaust gases meet EPA emission standards for particulate matter, and a third in the simple incentive for an efficient filter not subject to high maintenance costs.

Although no universally accepted standards have been specified or agreed upon for the allowable particulate loadings of gases driving turbines, it is generally agreed that the lifetime and performance efficiency of such turbines can justifiably be increased by reducing the particulate content of the feed gas stream via a reasonable cost filter. As an example, some typical particulate loading specifications are summarized in Table 1. Electrostatic precipitators, scrubbers and fabric filters have been successfully used for removing particulates from exhaust gases at moderate levels of temperature and pressure, but not under high temperature (e.g. 1000° to 2000° F) and pressure conditions. Lowering the temperature and pressure of a turbine inlet gas stream for the purpose of facilitating particle removal by using proven techniques would result in large losses of energy. The increased prospective utilization of combined cycle gas/steam turbine/electrical generating systems, coal gasification, fluidized bed combustion, and synthetic gaseous fuels has intensified the interest in granular bed filters. Unfortunately the effect on turbine blades appears to be a function of the physical properties of the impinging particles. In catalytic cracking of petroleum fractions, carry-over silica-alumina catalyst fines remaining in the exhaust gases after passing through three stages

Table 1. Typical Turbine Specifications
(NTIS- PB 266 231 Feb. 77)

<u>Source</u>	<u>Particulate Loading (Maximum Allowable)</u>
United Aircraft	0.8 lbs/million SCF low BTU fuel gas (or approximately 12 ppm)
Westinghouse	0.03 percent (in fuel oil) (or 300 ppm by weight) 0.002 grains/SCF all smaller than 6 microns (for Fluidized Bed Coal Combustors)
General Electric (for aircraft-type turbines)	30 ppm by weight in fuel gas (10 micron maximum)
Brown Boveri	1-2 ppm by weight (in gases entering turbine)
D.O.E. (estimates)	0.75 grains/SCF (or approx. 2.6 ppm by weight) in 0 to 2 micron range 0.001 grains/SCF (or approx. 35 ppm by weight) in 2 to 6 micron range.
Exxon (estimates)	45 to 1 mg/m ³ (0.02 to 0.0004 grains/SCF)

of cyclones have shown power recovery turbine blade erosion to a degree necessitating re-blading only at 5 year intervals (1)*.

The severity with coal combustion or gasification fines is reportedly far greater.

Other than the incentives of power recovery the increasingly stringent EPA emissions standards create definitive advantages for dry collection to avoid the handling of slurries from wet scrubbers. With cyclones not likely to meet the standards, with electrostatic

* Numbers in parentheses refer to entries in the list of "References Cited".

precipitators representing high installation and maintenance costs, and with bag houses at 2 to 10 CFM/sq.ft. representing large installations requiring considerable space, the potential of a 40 to 100 CFM/sq.ft. high efficiency, low maintenance, granular bed filter is an inevitable attraction even in conventional electrical utility applications.

Other than power recovery and meeting emission standards there exist a host of industrial applications in which a "sand" medium would offer considerable process advantage over a fabric. Condensibles in a gas stream can cause severe bag failures particularly in systems carrying cement kiln and similar calcined effluents which can solidify on the bags. Sand media can be effectively dried and even washed in situ, and light accretions removed by the grinding action of the media. Where collected particulates can represent a fire hazard as in the collection of carbon black, cellulosic solids, and similar materials, a non-combustible filter medium would have immeasurable advantage.

In the popular literature are found such terms as "gravel bed filters", "panel bed filters", "expandable bed filters", "moving bed filters", "sand filters", "loose-surface filters", "porous bed filters", "MB filters" and a host of others all of which pertain to versions of what are generically referred to as granular bed filters.

All potential commercializations of the concept rest on four

basic interrelated factors:

- 1) Collection efficiency
- 2) Cleaning or regeneration capability
- 3) Capacity
- 4) Competitive cost

Their practicality lies simply in the operational details of their technology which might best be illustrated by a relatively chronological review of the principal attempts at commercial development over the past 30 years and the status of such work today.

Porous Sintered Granule Beds

Particle filtration via porous membranes rests primarily on the formation of a filter cake, removable by a reverse flow of fluid when the resistance of the cake (or the pressure drop) exceeds any desired level. The porous membrane may take the form of a bed of particles or for that matter a mat of fibres such as is the case with cloth bag filters. The thinner the membrane the lower the overall resistance with or without a filter cake and hence by analogy to the near mono layer of fibres in a cloth collector one could construct a thin bed or sheet of granules by sintering a shallow layer of metallic granules in a high temperature furnace. Such sheets of sintered metal granules in various forms are sold commercially for filtration purposes. Their principal application

is in liquid systems, such as the maintenance of dirt-free fuel lines in aircraft engines. They have had limited application in the recovery of carryover from fluid bed reactors and similar fine-particle processing, but are not broadly acceptable. Their commercialization stemmed from the work of Dr. David Pall whose interest in the early 1940's lay in the development of a gaseous diffusion barrier for isotope separation in connection with the Manhattan District project during World War II.

In the late 1940's the Pall Corporation (now Pall Trinity Corp.) supplied such porous metal in tubular form with the suggestion that their application would be analogous to a cloth bag filter in which the cloth is replaced by a rigid thin bed of sintered metal particles (2). As illustrated in Figure 1, in normal operation valve 1 would be open, allowing dusty inlet gas to flow through the elements, depositing a filter cake on their outer upstream surfaces and leaving through the exhaust plenum. When the cake has grown to a thickness exhibiting high pressure drop, valve 1 is closed, valve 2 is opened, and a short, high-pressure pulse of air is admitted in reverse flow through the porous elements to dislodge the filter cake which falls to the bottom of the containing vessel, to be eventually withdrawn through valve 3. In continuous use, the valves operate on a timed cycle and the containing vessel is provided with a multiplicity of porous elements and separate plenums for localized reverse cleaning.

There are a number of inherent disadvantages to this form of

Figure 1. Porous Sintered Granule Filter

filter. In order to obtain structural strength, the granules making up the porous element must be small in size to present sufficient bonding surfaces. This results in high-pressure drop or low gas capacity despite the only approximately 1/8 inch wall thickness of the elements. Simultaneously, the thin-wall tubes are very subject to cracking as a result of repeated thermal shocks between normal high-temperature operation and relatively colder reverse-flow cleaning blasts. In addition, over extended operation, the sub-micron fines penetrate the interstices of the elements. These fines become trapped and are unable to be blown out with the reverse cleaning blast. Thus these elements build up a residual pressure drop which further limits their capacity. They are not recommended where only very fine particles are to be collected; they operate most satisfactorily in handling streams containing sufficient coarse particles, in effect, to build up their own pre-coat. Such instances are rare, since where they occur the particle loading is usually so high that the cost of the elements is prohibitive even if they were never to fail by thermal shock.

Continuous Moving Bed Filters

As early as 1924 Cramp (3), (4), proposed a so-called "dust curtain" system for filtration of blast furnace effluent gases. The dust curtain was a vertical layer of the collected dust itself, held in place by an arrangement of grids and louvers packed under its own weight. Dusty gas passed through this curtain hori-

zontally adding more dust to it. The excess dust gradually overflowed the louvers and fell to the bottom. The curtain was renewed periodically by dropping dust out of the bottom of the grids and adding more at the top. It is not clear whether this concept was ever put to industrial use.

In the early 1950's the late Morton Dorfan (Mechanical Industries, Inc.) introduced the Dorfan Impingo Filter (5), (6), which consisted basically of a downflowing bed of gravel retained between louvered walls. Dust-laden gas was filtered by blowing through the bed normal to the direction of gravel flow and filtered dust hopefully carried downward with the gravel, trapped within its interstices, as illustrated in Figure 2.

Four units, each of 17,000 CFM air capacity and consisting of two cells each, were installed in a plant to collect asbestos rock dust from a stream of flue gas coming from a direct-fired dryer in which the rock was dried prior to milling. The dust was 100% finer than 100 mesh and 60% finer than 10 microns. The concentration entering the collector was approximately 6 grains/cu. ft. and that leaving about 0.2 grains/cu. ft.

Dorfan's filter circumvented the thermal shock failure of Pall's sintered beds by avoiding reverse-flow cleaning and instead circulating the bed granules, plus collected dust, through a vibrating screen. However, this consequently required continual replenishment of the bed and hence an enormous and costly gravel

Figure 2. Continuous Moving Bed Filter

circulation system. In order to provide reasonable gas throughput capacity, the bed granules had to be relatively large so as not to be blown off the retaining louvers; in practice "gravel" of plus 1/4 to minus 1 1/2 inch was recommended. The commensurately large size of the bed interstices resulted in lower collection efficiency, and required staging beds in series, as shown in Figure 2, to achieve reasonable overall collection efficiencies. Such staging only added to the enormity of the required gravel circulation. Basically, micron-size particle collection efficiencies of the order of 99 +% could never be attained with such a moving bed system since intergranule movement in a downflowing mass physically dislodges collected particles and makes them subject to reentrainment.

Simultaneous with Dorfan's development a similar device was reported in the Russian literature by Zhitkevich (7) for the removal of peat dust from air by passing it through a vertical moving bed of pieces of peat 5mm in diameter. The rate of downward movement of the bed was controlled by the speed of a screw conveyor removing the dusty peat at the base of the unit in much the same fashion as rotary valves (not shown in Figure 2) at the bottom of each louvered column in Dorfan's arrangement. It is not known to what degree Zhitkevich' device found industrial application.

In 1954 Egleson et al (8) published information on pilot

tests of a downwardly flowing coke bed proposed as a possible means of filtering coal dust from a stream of synthesis gas produced in the gasification of pulverized coal. An 8½ foot deep bed of 0.125" to 0.40" diameter coke particles moved vertically downward by gravity flow through a 12" i.d. column counter to an upwardly flowing stream of dusty air. At inlet loadings of 2 to 8 grains/cu.ft. of 200 mesh dust, collection efficiencies were reported as high as 99.9% with superficial air velocities of 33 CFM/sq.ft. giving pressure drops of only 0.3 inches of water per foot of bed depth. The dust laden coke leaving the bottom of the column was washed with water and continuously recirculated to the top of the column. Though achieving, by means of counterflow, a higher efficiency than the Dorfan or similar cross flow devices, it became obvious that the enormity of the granular media handling equipment would again make industrial application impractical.

Intermittent Moving Bed Filters

In the late 1950's it occurred to A.M. Squires (presently associated with the Chemical Engineering Department at the Virginia Polytechnic Institute) that the Dorfan filter could be made far more efficient by using a finer medium and arresting the bed motion during a filtration period, after which an accumulated cake of filtered fines would be removed by moving just the dusty sand. The desirable arrangement illustrated in Figure 3 would circumvent the fouling action of unremovable penetrated fines (as experienced with Pall

Figure 3. Intermittent Moving Bed Filter

type filters) and hopefully lessen the burden of circulating the immense quantities of sand associated with continuously moving beds (as exemplified by Dorfan type units).

In the arrangement pictured in Figure 3 the dusty gas passes through a thin bed of stationary sand held between a panel of louvers and a fine mesh screen. The filtered fines build up a cake on the exposed bed surfaces and to some extent penetrate the interstices. When the resistance of the cake has reached an undesirable level, the clean gas outlet valve 1 is closed and a short pulse of compressed air is blasted in reverse flow through the sand bed by opening valve 2. In continuous use, the valves operate on a timed cycle and the containing vessel is provided with a multiplicity of panels exhausting to a partitioned plenum, permitting localized reverse flow cleaning of individual panels.

The backwash pulse is sufficient to physically lift the sand beds as a mass, with minimum interparticle movement, so that a surface layer of sand between each pair of louvers is physically ejected from the panel and falls to the bottom of the filter vessel along with the collected filter cake. The expelled sand is immediately replaced by downward movement of fresh sand from the overhead hoppers. This only intermittent downward movement, coupled with a fine size of "gravel" permits high efficiency collection and avoids the buildup of resistance due to penetrating submicron fines since these are expelled along with the surface layer of

their surrounding sand. Incomplete removal of the sand containing penetrated fines would lead to their eventual accumulation near the bottom of the panel, by gravity flow, after repeated cleaning cycles, causing a gas flow maldistribution and excessive pressure drop. In order to minimize penetration and hence minimize sand circulation, the gas throughput was limited to approximately 10 to 12 ACFM/sq.ft. of exposed bed surface.

Referred to as the "Loose-Surface" or LS filter, several models were built and tested in the period of 1959 to 1961 at the laboratories of F.A. Zenz then in Roslyn Harbor, L.I. Figure 4 illustrates the largest of these. Typical experimental results using hand-sieved bank sand as the filter medium, and bagged flyash from a Consolidated Edison power plant as the redispersed dust, are given in Table 2.

Table 2 Typical Loose Surface, Panel Bed Filter, Test Results

Sand medium particle size	<u>16 to 30 mesh</u>	<u>30 to 100 mesh</u>
Flyash loading in feed gas, grains/cu.ft.	2 to 6	7 to 9
Face velocity, ACFM/sq.ft.	8 to 12	6
Initial (clean) bed pressure drop, "H ₂ O	0.1	0.38
Maximum pressure drop (prior to blowback), "H ₂ O	0.6	0.6
Flyash laydown between cleanings, oz's.	14	2 to 3
Collection efficiency	99.7%	99.8%

FIGURE 4
FLOW DIAGRAM
OF TEST UNIT
LS NUMBER 2

The simplicity of the laboratory unit of Figure 4, appeared sufficiently attractive to warrant its purchase in 1960 by the Fuller Company (of Catasauqua, Pennsylvania, a subsidiary of General American Transportation). After filing their patent (9) in Squires' name and spending 3 years in an attempt to build equally successful test models in their own laboratories Fuller abandoned the commercial development of Squires' filter and currently maintain solely a non-exclusive interest in the original patent. Fuller's decision to halt their development was based on 1) an inability to restrict the amount of sand lost on each cleaning cycle to levels commensurate with economical recirculation costs, 2) the realization that the cost of steel and support structures was far greater than Squires' original estimates, and 3) that at 10 CFM/sq.ft. the capacity was not competitive with bag filters and hence a limited market would possibly exist only in a few very high temperature applications. At the time of its purchase by Fuller the LS filter was considered a potential competitor of electrostatic precipitators for power plant flyash collection.

In view of subsequent developments it is interesting to note that during the early period of laboratory development of the LS filter it was suggested that the circulation of sand could be entirely avoided by installing a retaining screen on the clean side of the panel thus trapping the sand in a number of superimposed beds which could be cleaned by a reverse fluidizing flow of gas to elutriate out the collected fines. Squires reasoned

that this would not permit high filtration efficiency because the mixing action accompanying fluidization would result in backmixing of collected fines so that eventually the sand bed would be dusty throughout and therefore some amount of collected dust would find its way into the clean side of the filter. Exploration of this suggestion was therefore not carried further.

Upon joining the faculty of City College in 1966 Squires obtained substantial financial support from EPA and similar government agencies to essentially repeat the LS development with, in some instances, minor modifications in bed depth or louver configuration (10), (11), (12).

Inspired by Squires' modification the principle of the Dorfan filter was investigated anew at the U.S. Bureau of Mines (13), (14). In this instance the test apparatus, designed by Combustion Power Corporation (CPC) utilized downward moving beds of sand grains. The CPC design comprises essentially the panel form of the Dorfan filter rotated into a cylindrical arrangement and operated with intermittent media downflow. With the expectation that the CPC filter would live up to the claims of its developers on economically competitive grounds, the management of Weyerhaeuser Industries purchased all the outstanding stock of Combustion Power Corporation and installed such filters on a number of their own bark boilers. Unfortunately such complete installations were found to require the following component steps illustrated in Figure 5:

- 1) sieving of the collected fines from the bed media
- 2) conveyance of the media to a superimposed vessel
- 3) elutriation of residual fines from the media by fluidization
- 4) recovery of the elutriated fines in a "small" bag filter
- 5) redistribution of the granular media to the filter annulus

Figure 5.

C.P.C. Annular Bed Filter

A complete installation therefore takes on many undesirable and costly complexities quite apart from the otherwise simple principle. In addition to an inadequate overall collection efficiency (including the periods during media movement) a complete installation reaches levels of cost which encounter a severe sales resistance. Weyerhaeuser has reportedly been disenchanted with their own installations and further development only minimally supported with U.S. Department of Energy (D.O.E.) funds.

Fluidized Bed Filters

It should be obvious from the foregoing experiences with cross flow moving beds that high efficiency collection is not possible unless the downwardly moving "wall" is very thick (e.g. representing a deep bed). The rubbing action of inter-granule movement and the continuously changing interstitial configurations allow entrapped dust particles to be blown by the gas stream to deeper and deeper penetrations of the filter bed. Therefore very deep beds are required to achieve only reasonable collection efficiencies. It would, therefore, be expected that gas fluidized beds would similarly yield relatively low collection efficiencies. This conclusion is generally borne out by several such reported investigations (15), (16), (17).

Under U.S. Army Chemical Corps Contract DA-18-064-CML-2578 a mechanically (vibrated) fluidized bed of sand was investigated for its possible filtration potential. The point of this investigation was to determine whether the poorer efficiency of gas

fluidized beds was solely attributable to intergranule movement (as in moving bed filters) or significantly affected by the passage of unfiltered gas in the bubbles rising through the bed. The vibratory fluidized bed was operated with gas downflow thus avoiding bubble formation and the accompanying bypassing of dusty gas. The results showed insignificant improvement over a bubbling fluidized bed thus leading again to the conclusion that high filtration efficiency is compatible only with stationary media.

It is, however, of interest to note that a shallow moving fluidized bed of raw bauxite feed has been used to filter the carryover of particulates from the off-gases of electrolytic cells used in the production of aluminum metal. Such a unit has been operated by Aluminum Co. of America following the description in their patent (18). The filtered particulates are primarily pitch or asphalt employed as a binder in the cell anodes. Though they appear to be dry they are sufficiently tacky to adhere to most surfaces on contact. Kaiser Aluminum installed an electrostatic precipitator at their Chalmette, Louisiana plant which operated for only a short time until it became completely coated with a layer resembling the undercoating on an automobile. The sticky nature of the collected particulates might be the principal reason for the practical application of a fluid bed filter in this instance; particularly since the filter medium is itself the raw material continuously fed to the cells thus requiring no recirculation equipment and efficiently returning the asphalt fines to the cells as further fuel. The filtered off gases leaving the fluid bed are in any case subsequently passed through a bag house for cleanup

of non-sticky fines to achieve the level of overall efficiency necessary to pass EPA standards. A filter of the Dorfan type was also proposed for such aluminum plant operations (19).

Granular Bed Filters Mechanically Cleaned

As opposed to the LS filter's attempt to compromise between the principles illustrated in Figures 1 and 2 (e.g. fixed bed filtration but minimal sand circulation) Max and Wolfgang Berz (20) in 1957 proposed an arrangement requiring no sand circulation. The cleaning would be carried out by flowing a gas in reverse direction through the bed while subjecting it to mechanical vibration of sufficient magnitude to cause the intergranule movement necessary for removal of entrapped fines. The principle of their proposed filter is illustrated in Figure 6.

Figure 6. Mechanically Cleaned Granular Bed Filter

In normal operation dusty feed gas would flow upward through a stationary fixed bed of sand held on a horizontal screen of a mesh size smaller than the bed granules. A filter cake might develop and adhere to the lower surface of the screen and some amount of fines penetration would occur within the interstices of the bed. Since they employed a granule size many times larger than the approximately 30 mesh sand investigated by Fuller, they experienced a greater degree of fines penetration (as opposed to cake buildup), required deeper beds, and operated at considerably higher through-put capacities per unit of bed surface area.

When resistance to flow reached a level requiring bed cleaning, the dusty feed gas was diverted to another unit and a reverse flow of gas at low velocity passed down through the bed, while it was simultaneously vibrated in a manner such as to impart an orbital rotation to the particle mass. Thus the mechanically induced inter-particle motion dislodged the penetrated fines so that they could be swept out of the bed by the downflowing breeze. In continuous use, the reverse flow and bed vibration were automated on a timed cycle and the containing vessel again provided with multiple units cleaned individually in sequence.

The Berz' MB filter was sold commercially by Lurgi Apparatebau G.M.B.H. of Frankfurt, Germany, until about the end of 1969 when it was abruptly withdrawn from the market. Its shortcoming lies in the strain imposed on the necessary flexing membranes associated

with the vibrating technique. It is conceivable that at low temperatures where rubber membranes and spring-supported bed mounts are feasible, such a filter could operate with a reasonable life. However, at low temperatures it could never economically compete with the simplicity of bag houses. At high temperatures only metal bellows would suffice as the flexing membrane, and their life expectancy in the hot and dusty environment is impractically short.

Undaunted by their mechanical failure Berz devised an alternate arrangement now marketed through Gesellschaft fur Entstaubungsanlagen G.m.b.H. of Munich, Germany. In this version, illustrated in Figure 7, the gravel medium is agitated during the cleaning cycle by the stirring action of a rotating rake whose fingers are imbedded in the gravel medium. Though the long term mechanical integrity of the raking system as well as its possible introduction of flow paths of lower efficiency might be objectively questioned, it appeared that this filter might receive reasonable acceptance in at least the cement industry. Users, however, were not as completely satisfied as might be implied in Berz' optimistic presentation to the 1972 IEEE cement industry technical conference held in May, 1972. The cleaning period requires several minutes and the module undergoing cleaning is isolated from the dusty gas by a valve system during the cleaning period. Carrier Corp. (Rexnord Division) installed such filters at over a dozen cement plants (21), however rake failures began, as anticipated, within about a year. Present

Filtering Cycle

Cleaning Cycle

Figure 7. Alternate Form of Mechanically Cleaned Granular Bed Filter

efforts are concentrating on possible incorporation of a means of periodically removing and replacing the bed medium.

Granular Bed Filters Pneumatically Cleaned

In 1970 the Ducon Co. introduced the "Expandable Bed" filter (22), (23), (24), in which a number of superimposed beds of sand, trapped between retaining screens, would be cleaned by a reverse flow of fluidizing gas. Its principles are illustrated in Figure 8.

Figure 8. Ducon Expandable Bed Filter

In normal operation dusty gas passes through vertical arrays of parallel shallow granular beds held within metal-walled compartments sealed at top and bottom by perforated meshes finer in aper-

ture than the size of the bed granules. The dusty gas enters the beds at relatively high velocities, in the range of 40 to 100 CFM/sq.ft. of bed surface area. When flow resistance reaches a level requiring the bed to be cleaned, a sufficient momentary reverse flow of compressed gas is admitted to fluidize the bed granules, or "expand" the bed, and thus by entrainment from these agitated beds expel the particles collected and agglomerated in the bed interstices. No loss of sand can occur since any granules which might reach the end of a compartment cannot escape through the perforated retaining mesh.

The fluidized expansion of the granule bed and the accompanying intergranule movement, which allows efficient cleaning, are directly analogous to the action in bag filters. Upon cyclically reversing the flow of gas, particulates trapped between the fibers are expelled as the cloth flexes or expands. This movement of the fibers in the expansion of a bag allows such complete cleaning (depending upon the number of flexings during each cycle) that bags can always be returned to near their original pressure drop characteristics. By sufficient and proper duration of backwash flows, the trapped fines in the beds of Figure 8 can also be nearly completely expelled by the analogous intergranular bed flexing, bed expansion, or bed fluidization. Thus fluidization affords an economical non-mechanical cleaning mechanism compatible with high-temperature filter operation, requiring no costly recycle conveying of sand, no complex redistribution hoppers, no subsequent need to separate

collected fines from filter element sand, and hence no attrition of bed granules which normally accompanies repeated solids handling operations.

In practice it was discovered that the intermingling of collected fines within the granular bed, which inevitably occurs as a result of the fluidized agitation during cleaning, resulted in an eventual buildup to an equilibrium fines content which then increased the collection efficiency, presumably simply because it presented a more tortuous interstitial path for the dusty gas flow. It was also found that the economic optimum operation lay in high gas capacity with minor sacrifice in efficiency. In this connection it is interesting to note that, in a paper dealing with the panel bed filter, presented at the Washington, D.C. A.I.Ch.E. meeting December 3, 1974, R. Pfeffer (12) reported also observing higher efficiencies when using dust as opposed to fresh sand. However, it is also obvious that following a reverse flow cleaning there could exist a small amount of the collected dust adhering to "sand" media lying directly atop the bed supporting grid. This dust might well be blown off into the clean side of the filter when it is again placed in filtering operation and so contribute to a less than 100% filtration efficiency depending on cleaning frequency.

Technological Status of Systems
under Development
and under Commercialization

The technological success and eventual commercialization, of any one or more of the many granular bed filter concepts, lies in

- a) satisfactory dust collection efficiency
- b) reproducible regeneration or filter media cleaning
- c) competitive total installation cost.

Unfortunately these three aspects are rather intimately interrelated and in all likelihood only eventual full-scale testing will narrow the field; this becomes partially evident in even a cursory review of the major commercial contenders.

The Fuller-Squires Panel Bed concept though abandoned by Fuller has undergone a series of studies related principally to improvements in louver design. There is no doubt that this concept affords excellent collection efficiency and that following each blowback the dusty gas faces a reproducibly cleaned filter bed. The questions which remain as yet incompletely answered are whether sieving the dust laden sand drawn from the filter will produce sufficient cleaning. Will some residual dust particles cling to the media and escape into the cleaned gas stream upon return to the louvers? Would an additional fluidized bed elutriation column provide sufficient cleaning; how would this fluidizing gas stream then be cleaned and what would such addi-

tional equipment add to the total cost? How many louvers can be uniformly cleansed upon each blast of blowback gas? What is the investment increment for the media circulation and distribution system and what degree of media attrition might be anticipated? No tests of the Fuller-Squires panels have ever been conducted with continuous media recirculation or in an industrial environment.

The Combustion Power Corporation Annular Bed as illustrated in Figure 5 has undergone full-scale operation on bark boiler effluent giving results as shown in Table 3. It is unknown whether the reported media efficiencies include periods during which the media were in downward movement or only periods when the media were stationary. The media pressure drops may also refer ¹² to clean conditions prior to dust buildup. In any event the media efficiencies are substantially below the 99+% levels normally sought in granular bed filtration. Efficiencies in the 90% range could more effectively be attained with multiclone centrifugal separators. Reliable cost figures for a complete installation including all the elements in Figure 5 are unavailable.

C.P.C.'s principal development experience centered around a pilot unit installed at the Snoqualmie, Washington plant of Weyerhaeuser Lumber Co. to clean up effluent from a hog mill waste combustor. The pebble bed filter was in this instance a single downflowing sand annulus 8½ feet O.D. and about 6 feet I.D. with an effective filtering height of 16 feet. The face velocity

Table 3 CPC Annular Bed Filter Test Data
(Power Boiler effluent- Waste Wood
and Fuel Oil Fired)

Weyerhaeuser New Bern N.C. Pulp
Mill, January 1975

<u>Media Size, Inches</u>	<u>Media Face velocity ft/min.</u>	<u>Media Pressure Drop "H₂O</u>	<u>Cyclone Pressure Drop "H₂O</u>	<u>Loading, Gr/dry SCF</u>			<u>Collection Efficiency, %</u>		
				<u>Cyclone Inlet</u>	<u>Media Inlet</u>	<u>Media Outlet</u>	<u>Cycl.</u>	<u>Media</u>	<u>Total</u>
0.125-0.250	125	6	1.2	2.768	0.875	0.075	68.4	91.4	97.3
0.125-0.250	150	9.3	2.0	1.486	0.609	0.080	59	86.9	94.6
0.065-0.130	150	11.8	1.4	2.542	0.80	0.07	68.5	91.3	97.3
0.065-0.130	125	9.7	1.0	4.719	0.618	0.026	86.9	95.7	99.4

and effective inlet area are quoted by CPC on the basis of the mean diameter of the annulus and in this instance correspond to 100 to 163 ft/min. (or ACFM/sq.ft.) and to 365 ft² respectively. The pebbles were angular in shape and ranged from 1/8" to 1/4" in average diameter. They move downward through the annulus in gravity flow at a bulk velocity of 2 to 4 feet/hour. The pebble inventory is of the order of 40 tons though only about 20 tons are in the region exposed to gas flow. At 35,000 ACFM gas flow (~100 ft/min) with inlet loadings of 0.17 to 0.3 grains/dry cu.ft. and outlet loadings of 0.053 grains/dry cu.ft. the pressure drop across the downflowing pebble bed was of the order of 12" of water. The unit was operated at 300° to 350° F. Collection efficiencies in all tests ranged from 75% to 97%. The feed particulates showed at best about 27% smaller than 0.7 microns and 17% larger than 20 microns. The loss analysis showed 57% smaller than 0.7 microns and 11% larger than 20 microns.

Relative to public utility flue gas cleanup requirements the above performance figures fall far short of the necessary goals. Nevertheless CPC appears interested in promoting their filter for such application based on the premise that the Weyerhaeuser unit was not intended for high efficiency and that finer size of pebble combined with removal of the coarser feed solids by tangential feed gas entry into the containing vessel shell might yield the necessary overall efficiency for utility purposes.

CPC's conceptual design for a 1000 MW electric utility burning 12,000 tons of coal/day with 3,300,000 ACFM of 400°F flue gas, would consist of 3 banks of scrubbers each having 8 cylindrical filter vessels and each vessel containing 4 annular elements. They estimated a pressure drop of 8" to 10" of water across these filters and in 1974 a turnkey installed cost including fans and motors of \$2.00 to \$2.50 per ACFM. This would involve the sieving and circulation of 225 to 450 tons of pebbles per hour or as much as 10 times the ash rate assuming the flyash loading to the filters is of the order of 0.2 grain/cu.ft. Higher inlet loadings would require proportionately higher pebble circulation rates. The conceptual design is based on recirculating the flyash-freed pebbles pneumatically to the top of the filters where the pebbles are collected via cyclones and the conveying gas vented through bag filters. The pebbles are again freed of the collected flyash via screens and elutriation; the latter would involve more bag house area but less screen wear. It was not clear whether the \$2 to \$2.50/ACFM cost estimate included the solids handling and/or separation equipment.

In terms of a test unit for installation in a public utility's flue gas system operating on a slip stream, CPC felt that a 100,000 ACFM unit would be the minimum size for obtaining reliable and scalable data.

The Canmet-Prasco Hitec System (25), (26) represents a development very similar to the CPC design as illustrated in Figure 9.

Figure 9. The CANMET - Prasco Ltd. Hitec System

Initially, the packed-bed region is filled with appropriately sized granular material, as shown by the shaded portion of the drawing. The hot "dirty" gases enter at the location marked '1' and are directed by pipes to pass horizontally through the packed bed, which is held in place by louvres. As the dust is removed, the pores of the filter material begin to plug up, causing an increase of pressure. When this happens, some filter material drops from a chute onto a conveyor belt at '5', and fresh, properly sized material moves down the column '3' from above the louvred section to replace it.

The dirty filter material goes to the top of the unit in a bucket elevator and is dumped onto an inclined screen '6' where most of the dust '7' is sifted out for disposal. The remainder of the filter material is then fed back into the unit near the top of the vertical column into what is called an elutriation column. The gases to be cleaned enter at the bottom of this elutriation column and come in contact with the falling filter material. By controlling the speed of these gases, the finer particles of dust still in the filter material that were not removed by the screen are carried up with the gas and follow the path shown by the arrows marked '2', leading into the packed bed. After cleaning, the gases are vented to the atmosphere at '4'. The granular filter material is too heavy to be carried up by the gas stream and thus falls down into the storage area above the louvres for reuse. In this manner the problem of the plugging of the filter material has been overcome.

Tests carried out in CANMET laboratories have shown the system to be capable of capturing dust particles down to 1 micron (0.00004 inch) in diameter at efficiencies of over 99.9 per cent. Under appropriate operating conditions, the results obtained on the experimental unit showed the exhaust gases to contain less than 0.05 grain per cubic foot. A commercial-size unit capable of treating 7,500 cubic feet of "dirty" gas per minute has been constructed and installed in Winnipeg on a furnace used to melt cast iron. This unit is undergoing testing to determine its operational capabilities and whether it will meet the Manitoba Government regulations for pollution abatement devices.

The concurrent sizing of the filtration bed material by the gas being treated plus the baffle above the bed which results in a desired size gradation across the packed bed, are the novelties claimed to be essential to the achievement of the performance goals. Typical laboratory scale tests with ambient air are summarized in Table 4. Subsequent tests on an Ancast Industries Ltd. cupola in Winnipeg, Manitoba showed average collection efficiencies of 99.4% for particles larger than 25 microns, 97.3% for particles larger than 1 micron, and 50.3% for particles smaller than 1 micron. The resulting overall efficiency of 87.9% was insufficient to bring the effluent loading to below the Manitoba Department of the Environment's limit of 0.25 grains per standard cubic foot. Failure to reach the efficiencies anticipated from laboratory tests was attributed to poor cupola management which led to an unusual amount of sub-micron material in its effluent. License for fabrication and sale of the Hitec system has been granted to Prasco Ltd. of Winnipeg, Manitoba.

The Kawasaki louvred bed development is again patterned on the Dorfan moving bed principle. Little is known of this work (27) other than tests carried out at 85 to 295 CFM passing through a 10" wide x 40" high louvred panel holding 8" thick beds of down-flowing silica sand either 1250 to 2500 microns or 2500 to 5000 microns in diameter. Figure 10 summarizes results obtained with an oil fired utility flyash as the test dust.

Table 4

LABORATORY SCALE FILTRATION DATA FOR CANMET-HITEC SYSTEM

Media			Dust				*** Dusty Air Flow Rate, SCFM
Mean Diam., Microns	Rate lb/min	ΔP "H ₂ O	** Type	Inlet Conc. gr/SCF	Outlet Conc. gr/SCF	Coll. Eff. %	
1100	0.97	2.5-7.5	MCD	135	0.050	99.96	10
1100	0.62	5.0	"	134	0.036	99.97	10
1100	0.44	7.0	"	134	0.054	99.96	10
1100	0.96	5.0	"	45.5	0.175	99.62	15
1100	0.98	5.0	"	87.8	0.196	99.78	15
1100	0.98	3.5-6.5	"	81.6	0.156	99.81	15
600	0.62	5.0	MCD	135	0.011	>99.99	10
600	0.36	5.0	"	132	0.023	99.98	10
600	0.96	5.0	"	82.4	0.056	99.93	15
600	1.21	5.0	"	85.1	0.196	99.77	15
600	0	1.4	GB	56.6	0	100	10
600	0	1.6	"	65.8	0.14	99.79	10
600	0	1.9	"	137	1.28	99.07	10
600	0	2.0	"	150	3.02	97.98	10
600	0.50	2.2	"	148	4.22	97.15	10
600	1.22	2.2	"	118	1.18	99.00	10
600	1.74	2.2	"	153	0.880	99.42	10
600	2.20	2.2	"	130	0.420	99.67	10
250	1.44	5.0*	MCD	30.5	1.91	93.74	10
250	1.44	5.0*	"	31.4	1.59	94.94	10
250	1.68	5.0*	"	35.7	0.637	98.22	10
800	0.19	3.5	MCD	141	0.061	99.96	10
800	0.38	3.5	"	144	0.082	99.94	10
800	0.76	3.5	"	153	0.109	99.93	10
900	0.28	3.5	MCD	144	0.010	>99.99	10
900	0.53	3.5	"	142	0.016	99.99	10
900	0.89	5.0	"	82.6	0.091	99.89	15
1500	0.04	3.5	MCD	38.0	0.023	99.94	10
1500	0.23	3.0	"	81.2	0.051	99.94	10
1500	0.44	3.0	"	118	0.068	99.94	10

*- Pressure varied widely, figure reported is average pressure drop.

** - MCD- Mixed Cupola Dust (8% < 10 μ m, 10% < 30 μ m, 50% < 150 μ m, 100% < 1000 μ m)

GB- Glass Beads (1 to 25 μ m)

***- Filter Face Area: 6" x 6"

Filter Bed Depth: 3"

Figure 10

*Reported Filtration Efficiencies
of 8" deep beds of 1250 - 5000
micron diameter sands*

The PEMMCO Restricted Circulation Filter illustrated in Figure 11 is one (30) of two concepts under development designed to minimize the filter media losses experienced in the Fuller-Squires type of unit or to permit cleaning (31) without any reverse flowing gas. Only the shallow layers facing the dusty gas in Figure 11 are blown off by the reverse flow cleaning gas; the gaps at the bases of the louvres prevent immediate media replacement and therefore excessive

Figure 11. Design Principle of Restricted Circulation
Granular Bed Filter - Adsorber

media blow off. A basic advantage of such granular bed filters, in which the medium is removed and replenished, lies in their adaptability to simultaneously act as gaseous adsorbers or chemical reactors as in the RESOX process for flue gas desulfurization and conversion to elemental sulfur (32) (33).

The Melcher Electrofluidized Bed Filter consisting typically of millimeter sized particles is stressed by an imposed electric field that effectively polarizes the particles. They then act as

collection sites for previously charged fine particulates entrained in the dusty fluidizing feed gases. The advantages claimed for the electro-fluidized bed derive from the greatly reduced residence time for effective cleaning of the gas, realized by virtue of the bed's large collection surface area per unit volume, and the ease of handling the bed particles for removal of the collected fines (35) (36).

Collection efficiencies in excess of 98% have been reported (37) for submicron asphaltic particles in 10 cm deep beds of 2 mm sand. Superficial velocities ranged from 1.5 to 2 meters/second with bed pressure drops typically about 12 cm. of water. Electrical energization required less than 80 Watts/100 CFM. Though these conditions are a vast improvement over even multi-staged (38) non-electrified fluidized beds they still fall short of such typical fixed bed results as illustrated in Figure 11 despite their substantially greater face velocity.

Superimposing an electrical field on the bed particles was considered in the early days (1967) of the development of the Ducon filter illustrated in Figure 8. At that time it was felt to be impractical, economically unattractive, and above all unnecessary since the media alone showed essentially 100% collection efficiencies. There have been no reports of electrofluidized bed filters in successful commercial use; a pilot installation at an industrial site in Illinois was recently abandoned.

The application of an electrical potential gradient to a bed of ceramic beads for filtration of petroleum fluid catalytic cracking fractionator liquid bottoms has been reported (39) as achieving exit solids contents as low as 0.01% by weight. Offered by Gulf Science & Technology Co. it again suggests the possibility of application to fixed bed gas filters though this concept had been considered beyond the limits of economic feasibility.

Representatives from the Peoples Republic of China described their version of a "gravel bed filter", which is cleaned by a reverse flow of bed-fluidizing gas (40), at a recent symposium on iron and steel pollution abatement technology. Their presentation dealt primarily with the design principles which are covered in even greater depth in the ensuing discussion of the development of the advanced Ducon GBF concept. The authors reported that four units of their design were in industrial operation in China cleaning sintering machine fumes, clay drier effluent and other steel furnace operations. Capacities ranged from 8800 to 71,000 ACFM, pressure drops from 2.8 to 3.4 inches of water and collection efficiencies from 95 to 98%. These low pressure drops and low efficiencies (relative to figures quoted in the U.S. by Ducon, Westinghouse, Squires, etc.) are a reflection of the relatively large, 1300 micron (0.05") diameter of their filter media particles. What is most significant in this instance is the fact that units based on reverse flow fluidized cleaning are in operation in four industrial applications.

The Ducon Granular Bed Filter

Of all the GBF filter arrangements proffered to date, none has likely undergone as continuing a development effort as the Ducon version. This stems from the highly attractive virtue of its freedom from media circulation and historically from the naive approach to the aerodynamics of its operation. The early arrangement illustrated in Figure 8 was accepted as an infallible fait-accompli, by analogy to bag filters, and tested in several pilot plant applications without substantial success. Though it filtered perfectly, it eventually plugged or attained unacceptable pressure loss simply because the impulse blowback never fluidized the media but rather lifted the beds en masse and pinned them against the dusty side inlet screens. In 1975 (following 9 unrewarding years) with a change in management personnel, Ducon finally constructed a Plexiglas model which permitted visualization of the cleaning process and the ensuing development of an arrangement which if given proper attention still promises to achieve the intended goals. The details of these developments are common to all such devices, and to the design of fluid-particle systems in general.

Ideal Single-Particle Dynamics

The theoretical bases on which granular bed filtration was founded have been topics of interest and research for decades. Yet in spite of modern numerical techniques and perhaps improved correlations for particle-granule interactions, the same overall

approach for estimating capture efficiency is still employed within the academic community. This approach views individual bed granules as being placed in the particulate-containing gas stream and the particulate contaminants being captured when they come into contact with the bed granules. Typically, the capture mechanisms thought to be of importance are: inertial impaction, interception, Brownian motion, diffusion, and sometimes sedimentation. Neglecting sedimentation, the overall efficiency is given by an expression of the following form:

$$\frac{E\%}{100} = 1 - \exp(-KL) \left[\underbrace{\left(\frac{1.29 \rho_p r^2 U_0}{\mu R} \right)}_{\text{Inertial Impaction}} + \underbrace{2.34 \left(\frac{r}{R} \right)^2}_{\text{Interception}} + \underbrace{2.25 \left(\frac{RT}{6\pi \mu R U_0} \right)}_{\text{Diffusion}} \right] \quad (1)$$

Inertial impaction efficiency falls off rapidly with decreasing particle size, will increase with velocity, will decrease as gas viscosity goes up with temperature, and will increase with smaller collection media.

Interception is a collection mechanism that operates when the particle (of radius r) approaches within a distance r from the granule surface, so that even particles of low inertia but finite size will be collected to some degree by the mechanism of interception. Particles are collected by interception from a layer of gas which extends a distance r from the granule surface. As

observed, interception is purely a function of ratio of particle-to-granule size (r/R); it is only important when the bed medium is on the same order of size as the particulates being captured. For fine particle capture, this mechanism is of importance only in the cake filtration mode, when the collectors are indeed the same size as the particulates.

Very small particles are often collected more effectively than would be anticipated on the basis of the inertial impaction mechanism. This has been attributed to the effect of Brownian movement. Bombardment of small particles by gas molecules sets them in random motion similar to molecular diffusion. They thus tend to migrate toward regions of low particle concentration (that is, the surface of the collecting element). It is assumed that particles will be deposited on granules by diffusion if they move along streamlines which are at a distance from the surface equal to the mean Brownian displacement. The collection due to diffusion increases with higher temperatures and smaller particle sizes, and decreases with higher velocities.

For fixed gas properties and granule size, it can be shown that the overall efficiency will have a minimum with respect to particle size in the region where the inertial impaction mechanism has fallen off and the diffusion mechanism has not yet become very large. Figure 12 shows in a qualitative way how the different mechanisms combine.

Figure 12 Single Particle Collection Efficiencies

This simplified analysis indicates that judicious choice of the available design parameters of granule size, velocity, and bed depth will permit a design for very high efficiencies in granular bed filters. It should be noted that all the considerations to this point have been based on a clean bed surface wherein each bed particle acts individually; this ignores the "bed" aspect and is justified academically on the basis that the incoming dust particles are so small relative to the dimensions of the interstices between the bed particles that the bed particles can be considered as free standing targets.

Multiparticle Effects Within the Bed

Some aspects of the foregoing rather idealistic approach are open to partial question, or at least embellishment. The term R , which represents the radius of a bed granule, as it appears in equation (1) is fundamentally correct within the envisioned flow mechanics. However, since the granule is in contact as a fixed

bed, with neighboring particles, one is inclined to conjecture that a better measure of the dimension governing the interception effectiveness would be one related to the interstitial space. R might therefore be replaced with the radius of the largest sphere which can exist or pass through the opening formed by 3 touching spherical bed granules.

Though it is obvious, within the mechanisms considered in equation (1), that the larger the target, R, the greater the collection efficiency, it is also obvious that in a fixed bed grid the larger the bed particles the lower the collection efficiency; thus in a grid the target size of individual particles is not the controlling issue. In a grid, large particles present fewer targets and larger interstitial distances; smaller particles present more numerous targets and smaller interstitial distances.

As illustrated in Figure 13 this would replace R in equation

Figure 13- Particle Screening Effectiveness of Spheres on equilateral triangular pitch

(1) by (0.154 R) and thereby increase the first and third terms within the brackets by a factor of about 6.5, and the second term by a factor of about 45. The moment any material is collected by

either impaction, interception, or diffusion the second term builds up rapidly as the overriding collection mechanism.

There is also evidence among the experimental data that beyond some velocity threshold, collection efficiency at a fixed bed depth begins to decrease with further increase in face velocity. To a first approximation this appears to relate directly to particle "pick-up" velocity. When the gas velocity through the bed interstices reaches the pick-up velocity of the collected dust particles then these are re-entrained and carried through the bed. This aspect of filter operation has not been given much attention since practical pressure drop limitations are usually attained before velocities reach pick-up levels.

Optimization of Bed Characteristics

Since the granular bed filter is cleaned via a bed- fluidizing reverse flow of gas which elutriates the agglomerated collected fines from the surfaces and from the interstices of the beds, it is obvious that bed characteristics must be optimized from both filtering and cleaning considerations. Large granules lead to low pressure drop (desirable) and/or high gas capacity (desirable), but can lead to low collection efficiency (undesirable) and high blowback cleaning gas rates (undesirable). Smaller granules lead to higher pressure drop (undesirable) and/or low gas capacity, but high collection efficiency (desirable), and low blowback

cleaning gas rates (desirable). The deeper the granule beds the higher the collection efficiency (desirable), but also the higher the operating pressure drop (undesirable), the higher the cleaning flow pressure drop and the greater the necessary restriction in the bed support grid in order to achieve distribution of the blow-back gas. The choice of bed granule size must therefore consider both the filtration and cleaning processes.

Within the constraints of a competitive market position and the strong dependence of the foregoing variables on granule size, an optimum of 250 to 600 microns is approached rather rapidly. This size affords excellent collection efficiencies in the shallowest practical bed depth of $1\frac{1}{2}$ ". Shallower depths would impose impractical "out of level" tolerances on fabrication and erection, though at compatible filtration gas rates, dust penetration has never been significant beyond a depth of about $\frac{1}{2}$ " to $\frac{3}{4}$ " (eg. 30 to 45 granule layers) below the bed surface. A bed of 250 to 600 microns bubbles vigorously, well above its incipient fluidization velocity, at a superficial blow back face velocity of 2 ft/sec. At this velocity the elutriation rate of collected fines is so high that a duration of only 6 seconds is sufficient to clean the beds to a minimal equilibrium dust concentration and yet this velocity will not splash bed granules higher than about 12" above the bed surface, nor entrain any granules out of the bed compartments. Coarser granules would require far greater blow-back face velocities to achieve an equally vigorous fluidization

or degree of bed cleaning. Finer granules would be blown out of the compartments and would also require a still finer-mesh bed-support screen. The bed support screen represents a critical item both in terms of cost and mechanical integrity.

Support Grid Design

The GBF medium support-grid must not only have sufficiently small openings to retain the bed particles but must also present sufficient resistance (pressure drop) to the reverse blowback cleaning flow to assure that this flow is distributed uniformly over the entire area or cross section of the bed. This should theoretically be accomplished by providing a grid whose pressure drop under the 120 cfm/sq.ft. blowback rate will be at least 30% of that through the 1½" deep bed of sand medium.

$$\text{e.g.: grid } \Delta P = \rho_g v^2 / (0.8)^2 2g \quad (2)$$

for blowback with ambient air:

$$\frac{0.3 \times 1.5 \times 90}{12} = \frac{0.076 \times (120)^2}{0.64 (60F) 64.4} \quad (3)$$

Where $F = 0.0467$, the maximum fractional open area of the support grid.

This 4.67% open area grid is achieved by means of sandwiching a fine mesh stainless steel screen between two perforated plates (with perforations aligned). The perforations consist of ¼" holes on ½" triangular pitch centers; this yields 22.67% open

area so that the screen may have an open area of $0.0467/0.2267$ or 20.6%. This can be provided via a 24 x 110 mesh dutch weave wire cloth as illustrated in Figure 14 resulting in an overall F in

$$\% \text{ open area} = \frac{(0.0276 \times 0.0054) 100}{(0.0276 + 0.015)(0.0054 + 0.01)} = 22.7\%$$

Figure 14 Wire Cloth Dimensions

$$\begin{aligned} \text{equation (3) of } F_{\text{actual}} &= 0.227 \times 0.2267 \\ &= 0.05 \end{aligned}$$

which is considered close enough to the theoretical 0.0467 to achieve satisfactory gas distribution (pressure drop is 26.2% of bed pressure drop as opposed to the theoretical 30%). Blow-back gas densities greater than 0.076 lbs/cu.ft. would result in higher grid pressure drops and greater assurance of uniform gas distribution. The 110 mesh opening (137 microns) is smaller than the 250 micron smallest granule size.

The grid gas distribution criterion of 30% of the pressure drop through the bed is based on experimental investigations in beds 12" to over 100" in depth. It is, therefore, reasonable to

expect that this criterion could be relaxed in the case of much shallower beds (e.g. the $1\frac{1}{2}$ " depth of a GBF bed). In all likelihood the truer criterion might be 30% of the bed depth lying above the level of the penetrating gas jet issuing from the grid ports. In deep commercial beds this would be a negligible correction; however, for beds only of $1\frac{1}{2}$ " in total depth the correction might well be significant. Since it is also desirable to minimize the dead (unperforated) area on a grid where uncleaned media might concentrate, a pattern consisting of $\frac{1}{4}$ " holes on $\frac{3}{8}$ " triangular pitch centers was investigated in a Plexiglas model. Though a change from $\frac{1}{2}$ " to $\frac{3}{8}$ " centers might appear relatively trivial it makes a substantial difference in open area for gas flow and in metal surface for bed stagnation. This can be visually appreciated by reference to Figure 15. Visual observations of the surfaces of $1\frac{1}{2}$ " deep beds of 30-60 mesh sand fluidized at 2 ft/sec through screened grids employing pattern (a) of Figure 15 lead to the conclusion that distribution was satisfactory over the entire bed surface. Pattern (a) was therefore chosen for grid fabrication. The overall grid fractional open area is therefore $0.4 \times 0.227 = 0.09$ which from equations (2) or (3) would imply an effective bed depth of $\frac{1}{2}$ " which appears in order with $1\frac{1}{2}$ " overall minus a 1" bubble penetration.

1/4" diameter. Staggered. 3/8" centers. 8 holes per square inch
Open area 40%

1/4" diameter. Staggered. 1/2" centers. 5 holes per square inch
Open area 23%

Figure 15. Grid Hole Patterns

Blowback Cleaning

When the pressure drop across a GBF reaches the maximum allowable, or desirable, the bed is "cleaned" via a reverse fluidizing clean gas flow of 120 acfm/ft². This flow may be activated via a set pressure differential or a cycle timer. For bed dust loadings which would not exceed reasonable pressure drop limits, it has been determined that for the optimized 30-60 mesh filter medium the beds are cleaned to a minimal residual dust concentration at a continuous 120 acfm/ft² flow for a period of

six seconds. The rate of removal of collected dust by entrainment, is calculable from conventional fluidized bed entrainment correlations integrated over the blowback duration as bed dust concentration is depleted. This blowback gas is generally compressed air provided by means of a separate compressor or drawn from an existing plant compressed air line. To minimize the quantity of compressed air or the size of the required compressor the air volume may be slowly accumulated in a surge tank or augmented with secondary induced gas by means of an ejector.

Blowback Ejector Sizing

Figure 16 qualitatively illustrates a 2 element GBF module with one element filtering and the other being backwashed. The compressed air line discharging into the cleaned gas exit pipe acts as an ejector inducing a recycle of cleaned gas, which added to the expanding compressed or motive air, must achieve the required reverse gas flow of 120 acfm/sq.ft. of bed surface in the element being cleaned.

In the sizing and subsequent operation of such ejectors it must be assured that the combined blowback streams are not excessive (far greater than 120 acfm/sq.ft. lest they blow sand out of the compartments) and also that they are not too small (far less than 120 acfm/sq.ft., lest they fail to clean the beds). Assuming that the induced gas stream W_1 undergoes an abrupt contraction

Figure 16. Typical Ejector Arrangement

pressure loss $P_O - P_S$ in passing from the clean gas plenum into the ejector duct; assuming a nozzle, D_n , coefficient of 1.0 for the critical flow of the compressed blowback air; and fixing the combined flow of induced plus motive gas to yield a blowback face velocity of 2 ft/sec the flows and pressures can be expressed by the relationships shown in Figures 17 and 18.

These relationships have been computer programmed to define the required motive gas pressure and ejector throat diameter which

D_T, D_N in inches

P_M, P_S, P_D in PSIA

T_D, T_0, T_M in $^{\circ}F$

$P_0, \Delta P_F$ in "H₂O gage

ΔP_F = Filter Pressure Drop

A_{FE} = ft² of filter area blown back

F = fractional open area of bed grid

ρ_D = mixture density, lbs/ft³

$$\rho_D = 0.03613 (P_0 + \Delta P_F) + 14.778 + 0.00067 P_D / T^2$$

$$C_p / C_v = 1.4$$

$$P_M = (\text{Motive Gas Pressure, psig}) + 14.7$$

$$\text{Blowback Velocity} = 2 \text{ ft/sec}$$

$$W_I, \text{lbs/sec} = \frac{5.377 \rho_D A_{FE}}{(460 + T_D)} - W_M$$

$$W_M, \text{lbs/sec} = 0.418 P_M D_N^2 / (T_M + 460)^{1/2}; \text{ at } P_D / P_M < 0.528$$

$$W_M, \text{lbs/sec} = \frac{1.607 D_N^2 P_M}{(460 + T_M)} \left(\frac{P_D}{P_M} \right)^{0.7143} \left[(460 + T_M) \left(1 - \left[\frac{P_D}{P_M} \right]^{0.2587} \right) \right]^{1/2}; \text{ at } P_D / P_M > 0.528$$

$$P_S = 14.7 + 0.03613 P_0 - \frac{2.01946 (460 + T_0)}{(0.03613 P_0 + 14.7)} \left[\frac{W_I}{D_T^2} \right]^2$$

Figure 17. Ejector Flow Relationships

Figure 18 Optimum Ejector Design

must be provided to clean the beds, but which if exceeded could cause loss of sand inventory. Typical input which must be supplied consists of:

1. The filter area of an element, A_{fe} - for 12 beds each 6" x 36" this amounts to 18 sq. ft.
2. The fractional open area of grid plus screen, F - which for the screen plus perforated plate discussed earlier equals 0.05.
3. The range of operating pressure drops, ΔP_f - inches of water between the dirty side and the clean gas plenum.
4. The diameter of the pipe leading to the ejector, D_o - in inches.
5. The diameter of the motive gas orifice, D_n - in inches.
6. The temperatures T_d , T_o , and T_m , in degrees Fahrenheit
7. The gauge pressure in the clean gas plenum, P_o , in inches of water.

Typical output consists of the required motive pressure and the ejector throat diameter commensurate with performance at peak efficiency for a specified filter pressure drop. The curve in Figure 18 applies to conditions of peak performance; it represents the locus of the maximum values of the ratio of horsepower output (the compression of the induced flow) to horsepower input (the compression of the motive gas from the ejector discharge pressure to that of the motive gas upstream of the nozzle D_n). Unfortunately the entire family of curves for ejectors of varying throat-to-nozzle diameter ratios has never been satisfactorily determined. A program of such experimentation has been initiated and at this stage indicates that Figure 18 is likely to be represented more completely as shown in Figure 19. This suggests that once an ejector is sized for a given set of operating conditions it might be quite sensitive to changes in back pressure or filter pressure drop, and require commensurate alterations in motive pressure to maintain the necessary sum of motive and induced flows for proper bed cleaning.

Compressed Air Surge Tank

In instances where compressed air is not a readily available in-plant commodity, a small compressor can be utilized in connection with a surge tank or reservoir filled by the compressor between blow-backs. This surge tank would then be vented as the motive gas to an ejector. The size of the surge tank is based on containing the amount of compressed air, W_m , calculated from the relationship

Figure 19. Qualitative Indication of Ejector Performance Curves

given in Figures 17 and 18 over the time period of a blowback. The size of the motive air compressor is based on filling this tank to the required motive pressure within whatever time interval is available between blowbacks. Figure 20 represents a rapid graphical solution for surge tank size based on an element having 18 sq.ft. of filter bed surface blown back for a period of 8 seconds. Surge tank volume is directly proportional to these two conditions.

Element Dimensions

Figure 21 illustrates the standardized dimensions of the rectangular form of the GBF element; these are based solely on minimal resistance to gas flow. The ultimate restrictions lie across the beds and through the clean gas exit pipe, D_t , which might also be the throat of a blowback gas ejector. The resistance to clean gas flow through D_t can be conservatively taken as an abrupt contraction loss:

$$\Delta P, \text{ "H}_2\text{O} = 0.04174 \rho_g \left[\frac{(A_{fe} \text{ ft}^2) (ACFM/\text{ft}^2)}{D_t^2} \right]^2 \quad (4)$$

At a design face velocity of 35 acfm/sq.ft. through a 12-bed element 15" x 36" x 78" having a filter face area of 18 sq. ft. and a D_t of 6", equation (4) would yield a pressure drop of about 1" H_2O if the gas density were 0.0765 lbs/cu.ft.

The diameter D_t should be examined critically for every pro-

Figure 20
Design Chart for Sizing
Compressed Motive Air Tank

P_M = Motive Air Tank Pressure, PSIA
 P_0 = Absolute Pressure in clean gas duct, PSIA
 $\Delta P = P_I - P_0$ = Pressure drop across GB element, psi, at moment before motive air valve V-1 is opened

Bases: 18 ft² of bed surface per element
 8 seconds blowback duration
 2" maximum bed depths
 Motive air compressor capable of recharging tank in time interval between blowbacks

Example:

How large a blowback tank is required if a GB Filter is blown back when it reaches 20" H₂O pressure drop, if motive air is made available at 80psig and if the clean gas plenum duct operates at 6" H₂O gage?

Solution:

$$P_M = 80 + 14.7 = 94.7 \text{ psia}$$

$$P_0 = (6 \times 0.03613) + 14.7 = 14.9 \text{ psia}$$

$$\Delta P = 20 \times 0.03613 = 0.72 \text{ psi}$$

$$\frac{P_M - P_0}{\Delta P + 0.1} = \frac{94.7 - 14.9}{0.72 + 0.1}$$

$$= 97.3 \text{ abscissa}$$

$$\frac{P_M}{P_0 + \Delta P} = \frac{94.7}{14.7 + 0.72}$$

$$= 6.14 \text{ parameter}$$

ordinate from chart:

$$Q_M = \underline{\underline{9 \text{ cubic feet}}}$$

Figure 21. Typical Ducon GBF Dimensions

posed arrangement since its greatest back pressure effect will occur during backwash cleaning of an adjacent element. If the foregoing element were for example one of 6 operating in parallel then during blowback of one of the 6 the face velocity to the remaining 5 would be

$$\frac{(1) (120) (18) + 5 (35) (18)}{5 (18)} = 59 \text{ ACFM/ft}^2$$

and the pressure drop through D_t would rise to 2.78" H_2O . This should be taken into account in the ΔP_f term of the first equation in Figure 17 to obtain the discharge pressure P_D against which the ejector must operate.

The modular design presently offered (28) is illustrated in Figure 22; its operating principle is still the same as portrayed in Figure 8, but its design details are now regarded with extreme care. For example:

a) bed depth has been optimized at 1 1/2" since shallower beds impose unrealistic fabrication and erection tolerances and deeper beds add unnecessary and undesirable pressure drop;

b) filter media sand particle size has been optimized at 250 to 600 microns since coarser materials require excessive fluidizing velocity (blowback gas consumption) and entail reduced collection efficiency whereas finer particles would entrain too readily

Figure 22. Modular arrangement of the Ducon filter

upon fluidization and would require too fragile a bed support;

c) reverse flow fluidization at 120 CFM/sq.ft. for at most 6 to 8 seconds appears an optimum for cleaning to maintain an equilibrium low concentration fines inventory without excessive pressure drop; the degree of cleaning is calculable from fluidized bed entrainment correlations;

d) the bed supports are designed as screened perforated plates with a multiplicity of tiny holes to distribute the fluidizing gas bubbles over the entire bed in close proximity and with sufficient pressure drop to distribute equally to each bed in the stacked module;

e) the disengaging height above each bed is set at no less than 10½" in order to avoid filter media loss (without an inlet screen) compatible with the sand size, density and reverse flow velocity.

Collection Efficiency

No tests have as yet been carried out in an industrial environment with a unit bearing all of the above design considerations and limited to face velocities of about 40 CFM/sq.ft. with a maximum 1 psi buildup of pressure drop before reverse flow cleaning. As in the case of all granular bed filters, laboratory scale tests show

excellent filtration efficiencies as illustrated for example in Figure 23 from the results published by Westinghouse researchers (29). The ultimate technical feasibility of this concept lies in the ability to achieve cleaning of the beds in a practical manner. It is currently conjectured that past pilot plant failures may have been linked to a failure to appreciate ejector output as a function of operating pressures as illustrated in Figure 19. Laboratory tests conducted by Ducon, even over extended periods of time, but under steady and reproducible conditions, indicated essentially immeasurable loss rates within the accuracy of day to day weighings of isokinetic probe thimbles.

Media Pressure Drop

Accurately predicting the pressure drop through any fixed bed of particles is questionable even for clean beds of uniform size spheres. With angular particles having a range of sizes, ill definable shape factors and packing densities, the prediction becomes even more tenuous. When a collected powder of equally unknown characteristics forms a superimposed layer plus an unknown degree of interstitial penetration, the prediction methods become at best only qualitatively directional. The approximations employed to date have been based on matching experimental data to conventional fixed bed pressure drop correlations via appropriate, or effective, assigned particle diameters. As illustrated in example 1) of Figure 24 the effective mean spherical diameter of 30 to 60 mesh

Gas Velocity 50 fpm $\pm$ 10%

Bed Material 16-20 mesh Ottawa sand

Bed Depth 4"

Particle Size	Inlet Dust, mg/m ³	Outlet Dust, mg/m ³	Collection Efficiency, %
0-0.3	32	0.39	98.8
0.3-0.45	20	0.08	99.6
0.45-0.75	33	0.12	99.6
0.75-1.5	53	0.08	99.8
1.5-2.3	47	0.04	99.9
2.3-3.3	32	0.04	99.9
3.3-5.0	53	0.04	99.9
5.0-8.0	46	--	100
8.0	108	--	100
	513	--	100

Run No.	Inlet Loading g/m ³	Overall Efficiency %	
6-24	5.8	99.96	(99.92)
6-25	0.94	99.92	(99.8)
6-26	1.17	99.90	(99.7)

Figure 3.2 Typical GBF Filtration Efficiencies
(source: Westinghouse, Ref. 29.)

Figure 24

Design Chart for Calculating Gas Flow Pressure Drop Through Packed Beds of Filter Sands and/or Layers of Collected Particulates

Example 1. What is the pressure drop for the flow of essentially ambient air at a rate of 40 CFM/ft² through a 1.75" deep bed of 30-60 mesh sand?

$$\frac{jD_p \rho_g (\text{CFM}/\text{ft}^2)}{0.484 \mu_g} = \frac{0.01 \times 0.076 \times 40}{0.484 \times 0.0182} = 3.45 \leftarrow \text{Abcissa}$$

$$\text{ordinate at } \epsilon = 0.55: 50 = \frac{301550 jD_p \Delta P}{\rho_g L (\text{CFM}/\text{ft}^2)^2}$$

$$\therefore \Delta P = \frac{50 \times 0.076 \times 1.75 \times 40^2}{301550 \times 0.01} = 3.5 \text{ "H}_2\text{O}$$

Example 2. Suppose the 40 CFM/ft² flow in example 1 bore 4 grains of flyash/ft³ of gas. If the flyash had a particle density of 130 lb/ft³, a bulk density of 65 lb/ft³ and an effective mean particle diameter of 17 microns what would be the anticipated pressure drop through a filter bed after 5 minutes of filtering?

The pressure drop through the underlying sand bed would be the same as in example 1 (3.5" H₂O)

The layer of collected flyash will introduce an additional pressure drop. For this layer:

$$jD_p = 17 \text{ microns} / 25400 = 0.00067$$

$$\text{Abcissa} = \frac{jD_p \rho_g (\text{CFM}/\text{ft}^2)}{0.484 \mu_g} = \frac{0.00067 \times 0.076 \times 40}{0.484 \times 0.0182} = 0.224$$

$$\epsilon = 1 - (\rho_g / \rho_p) = 1 - (65 / 130) = 0.5$$

$$\text{Ordinate} = 900$$

$$\text{Layer Depth} = 40 \times 5 \times 4 \times \frac{1}{7000} \times \frac{1}{65} \times \frac{1}{16} \times \frac{1}{16} = 0.0211 \text{ ft}$$

$$\Delta P_{\text{thru layer}} = \frac{900 (40)^2 (0.0211) (0.076)}{301550 \times 0.00067} = 11.8 \text{ "H}_2\text{O}$$

$$\text{Total } \Delta P = 3.5 + 11.8 = 15.3 \text{ "H}_2\text{O}$$

angular sand has been found to be approximately 0.01 inches and its bed voidage in the loosest state, following a blowback, about 55%. Predicting the pressure drop through a collected cake of fine particulates is far more difficult since it is extremely sensitive to the effective agglomerated diameter of the cake particles. Over essentially all operating ranges of high efficiency granular bed filters the bed pressure drops are proportional to the reciprocal of the square of particle diameter; halving the diameter quadruples the pressure drop.

Parallel Module Overload

Figure 25 represents a filter composed of four modules each containing two filter elements. In the normal continuous mode of operation valves D-1, D-2, etc. would always be open so that the dusty gas would be distributed from the feed manifold equally into all elements of all modules continuously. To maintain any desired pressure difference between the feed and product manifolds, with minimal fluctuations with time as the filter elements accumulated dust, the elements would be blown back individually on a timed cycle by sequentially opening and closing valve B-1, then B-2, then B-3 etc. Such sequential blowback cleaning of individual elements follows conventional bag house practice. Consider for the moment the effect of such flow reversal on the other elements within the filter. During the time that valve B-1 is open no dusty gas can enter element E-1; therefore, elements E-2 through

Figure 25 Illustrative Arrangement of GBF Modules

E-8 are subject to a dusty gas feed rate increased by a factor of 8/7 or 14%. However, in addition to this relatively minor increase, these seven elements must also absorb the blowback gas from element E-1. Though dependent upon such factors as bed particle size, frequency of blowback, size and density of collected fines, etc., it is current practice to specify a blowback rate which creates a superficial fluidizing velocity of 2 ft/sec or 120 ACFM/sq.ft. of filter bed surface. If the elements were operating at a design face velocity of 50 ACFM/sq.ft. then obviously blowing back E-1 would increase the flow rate to the remaining seven elements from 50 ACFM/sq.ft. to:

$$50(8/7) + (120/7) = 74 \text{ ACFM/sq.ft.}$$

during the blowback period. This can be expressed analytically in the form:

$$\% \text{ OL} = 100 \left[\frac{T_E + (12/5)}{T_E - 1} \right] - 1$$

where % OL = % increase or overload in dusty gas face velocity through elements not being cleaned

T_E = total number of elements in filter and only one element backwashed at any moment.

Represented graphically in Figure 26 it becomes obvious that any

such installation whether on a pilot or industrial scale should preferably be provided with no less than 10 to 20 elements no matter what their size. Lower gas capacity for pilot tests should be provided with elements containing a smaller number of stacked beds rather than a smaller total number of elements.

Note in relation to Figure 25 it is assumed that reverse blowback from E-1 would be equally distributed to all seven remaining elements; however, this is probably unrealistic since it requires an absolutely zero frictional resistance in valves B-1, B-2 etc. as well as in the entire dusty gas manifold, and in addition, ignores the inertia transients. In practice, element E-2 would be considerably more overloaded. Some reasonable limits must be placed on this overload since it can result in dust breakthrough by the force of the high velocity interstitial gas circumventing particle impact by heightened particle pickup. For example, 100 ACFM/sq.ft. through a bed of 40% voids amounts to an interstitial velocity a little over 4 ft/sec which will exceed the saltation velocity of a variety of particulates depending on their properties. If the bed interstices were filled to some equilibrium content with collected fines, this effective interstitial velocity could easily reach 8 ft/sec and cause re-entrainment. The permissible overload cannot be specified without detailed knowledge of the system characteristics and properties.

Past and Current Trials

It is unfortunate that the early field trials of the Ducon concept were conducted with an incompletely developed design. Operating on slip streams from FCC regenerators (41), asbestos recovery units, etc. the results could only be called promising, but not overwhelmingly attractive from a commercial or competitive standpoint. The recent Exxon Mini-Plant test experience (42) involved a modification of the original Ducon design, but failed to approach their current arrangement in several critical areas. Figure 27 and Table 5 illustrate the significant comparisons between the Exxon unit tested and the present Ducon design.

Figure 27 — Comparison of GBF designs.

Table 5 GBF Comparisons and Recommendations

	<u>Exxon Test</u>	<u>Ducon Design</u>
Disengaging Height, inches	7"	10½"
Support Grid, % hole area	1.7%	15% ^[1]
Support Grid, hole diam., inches	0.1406"	0.25" ^[1]
Face Velocity, cfm/sq.ft.	52 - 100	40 ^[2]
Superficial blow-back velocity, ft/sec	1 to 2.7	2.0
Blow-back duration, seconds	2 to 30	6 to 8 ^[3]
Interval between blow-backs, minutes	3 to 90	
ΔP buildup before blow-back, psi	3.2 to 9	<1.0 ^[4]
Number of elements in parallel	2	10 ^[5]
Screen + Support grid ΔP at 2 ft/sec blow-back velocity		>0.75" H ₂ O
Number of beds per element (or per stack)	5	6
Ft. of wall perimeter/cfm x ft ² of bed surface	0.581	0.0778
Sand size, microns	250 - 600 425 - 2000 850 - 1400	250 - 600

Notes:

1. 1st models; current unit employs high resistance "screen" type grid
2. recommended for design purposes
3. with slow (e.g. not instantaneous) shut off
4. when operating under ambient conditions
5. preferred; to avoid excess loading when one element in group is blowing back.

Five significant differences are apparent:

1. The Ducon design has 50% greater disengaging height to avoid loss of bed material on blowback. This height was determined experimentally by measuring the "throw" of 30-60 mesh sand media upon instantaneous initiation of the 2 ft/sec or 120 cfm/sq. ft. blow-back velocity.
2. The Exxon unit was generally operated at face velocities averaging nearly 80 cfm/sq. ft., which is twice the Ducon standard. This velocity leads to more rapid filter saturation and higher pressure drop, and requires a more frequent blow-back cycle. If the element undergoing cleaning is not taken off-stream, then one additional parallel element would have its face velocity increased from 80 to 280 cfm/sq.ft. during cleaning; two additional parallel elements from 80 to 180 cfm/sq.ft., three additional parallel elements from 80 to 147 cfm/sq.ft., etc. Ducon recommends no less than 10 units in parallel, giving 40 to at most 57.8 acfm/sq.ft. This recommendation is based on observations of breakthrough when the dusty side gas rate is abruptly increased.
3. Ducon experience indicated that the bed support grids must have sufficient pressure drop to distribute the blow-back gas uniformly over the entire filter bed cross-sectional area. Preferably, this drop should be produced by as many holes as possible to avoid dead spaces on the grid between holes where dust can slowly accumulate and never be expelled. The hole pattern of the Exxon design

represents perhaps an excessively blank grid; only 1.7% open area versus 15% in the early Ducon design.

4. Ducon experience suggests that buildup in dust inventory should be limited to an amount reflected in at most a 1 psi increase in filter pressure drop, whereas the Exxon tests allowed a 3 to 9 psi buildup. It is not clear to what degree this is attributable to the higher gas density and higher gas viscosity in Exxon's operation. It is also not clear whether a 1 psi buildup should be an absolute criterion independent of the gas properties. In operating the filter, it is important that the amount of dust removed during the back-flush fluidization period should equal the amount collected. Otherwise the dust in the bed may increase, become the "continuous" phase, and pass through the filter.

5. In addition, fine particles may short circuit the granular bed by travelling along smooth walls or other surfaces (as frequently cited against the rakes of the Rexnord filter). It is interesting to note in Table 5 that, per unit of gas throughput, the Exxon version has nearly 7.5 times the wall surface per unit of bed area exhibited by the Ducon design.

Items 1, 2, 3 and 4 could, singly or in combination, contribute to so severe a buildup in dust inventory that the dust would become the continuous (more abundant particulate) phase, and hence would control bed fluidity upon blow-back. This could lead to either

rat-holing through the beds or agglomerative "balling" of the dust when bubbling fluidization of the sand medium became too subdued to provide any grinding or dust dispersing action. Attention to all of the above items will lead therefore to decreased penetration of the GBF by fine particles and increased removal efficiency.

In view of these differences Westinghouse has contracted with the Department of Energy to conduct tests with Ducon's current design at their Waltz Mill facility with simulated PFBC effluent particulates. Westinghouse's interest stemmed from their own investigations (43) of the concept in which a limestone dust ground to less than 10 microns (with an average size of about 1.5 microns) was blown down through a shallow bed of sand. When the pressure drop across the bed reached a predetermined level, flow through the bed was reversed; this fluidized the bed and blew the accumulated dust out through a vent at the top. Inlet and outlet samples were taken simultaneously, using cascade impactors, to determine overall efficiencies and grade efficiencies as a function of filter velocity, bed depth, loading, and granule size. Overall efficiencies in the range of 97.5 to 99+% were typically observed; high efficiencies were maintained through the difficult to capture size ranges of 0.3 to 1 micron. Pressure drops through the bed ranged from 10 to 20 inches of water for face velocities of 35 ft/min. and bed depths of 3 to 4 inches. Encouraged by this series of tests, Westinghouse designed and fabricated a pilot scale 7 sq.ft. granular bed filter, in which high temperature, ambient pressure, tests could be carried out at flow

rates of up to 500 acfm. This test facility was used to conduct preliminary tests at temperatures up to 1100°F on the same 10 micron limestone dust as was used in the bench scale work. Table 6 presents typical data.

Table 6 Granular Bed Filter Results*

<u>Particle Size</u> <u>microns</u>	<u>Inlet</u> <u>%</u>	<u>Outlet</u> <u>%</u>	<u>Collection</u> <u>Efficiency</u> <u>%</u>
0.0 - 0.5	11	13	96
0.5 - 1.1	5	15	88
1.1 - 2.2	11	25	91
2.2 - 3.7	16	26	93
3.7 - 8.2	12	8	97
8.2 +	43	12	99

Overall Efficiency: 96%

Comparitive efficiency for low-temperature laboratory unit under similar operating conditions: 99.5%

*Laboratory Unit Conditions: Gas Velocity: 30 fpm,
 Temperature: 1100°F
 Bed depth: 4 inches
 Bed Material, 14-28 mesh
 alumina

Although some apparent loss in efficiency from ambient conditions was observed, the efficiencies remained very high throughout the spectrum of particle sizes. On the basis of these in-house experimental data coupled with supporting theoretical background, Westinghouse

believes that very high filtering efficiencies in PFBC systems can be achieved with granular bed filters. It is expected that the tests at Waltz Mill will be more exhaustive and more conclusive.

A Granular Bed Filter of the Ducon design, but with a wider channel between facing stacks of beds to minimize dust recycling has been installed on fluidized bed combustor designed and operated by an east coast development company supported entirely by private funds. Thirty days after startup the operators reported that the filter has been operating satisfactorily. Their further experience, in conjunction with the Westinghouse-DOE tests should provide more substantiation than can be gleaned from the work of Yan and Wang (40). Simultaneously the PEMMCO filter (30) is also undergoing further development, supported by a major petroleum company, for a specific application.

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Eductors

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Eductors

The State-of-the-Art with regard to eductors, or ejectors, is quite reminiscent of that at one time concerning cyclones. Both are exceedingly simple pieces of equipment with no moving parts, no extraneous internals and coincidentally each bearing 3 ports; in the case of the cyclone one inlet and 2 outlets; in the case of the eductor one outlet and 2 inlets.

The uses to which eductors have been applied in the past have generally required sustained operation at a fixed condition. For example, steadily maintaining a given vacuum in a distillation unit, purging gas from a vessel or maintaining ventilation at a given rate, etc.; all instances wherein the designer strives for the optimum eductor dimensions or maximum operating efficiency. The design "correlations" have therefore been tailored to produce the optimum unit, since logically one would neither desire, nor expect, less. Particularly in the case of eductors, operating at conditions other than specifically intended in the design, can result in such drastically poorer performance that little attention has been paid to these "off peak" regimes. As a result, this neglect has left the full comprehension of an eductor's "operating diagram" somewhat clouded. Such a diagram may be qualitatively defined from simple logic guided by knowledge of the results of a few broad experimental investigations

such as were carried out by Process Equipment Modelling and Manufacturing Company of Nelsonville, New York, at the request of the Ducon Co. of Mineola, Long Island, in connection with their development of a granular bed filter. Similar tests had been carried out by the Foster Wheeler Corp. for the Central Electric Generating Board in England and very recently by Westinghouse Electric Corp. at their Waltz Mill laboratories under U.S. Department of Energy sponsorship.

Operating Principles

An eductor is basically a gas compressor without mechanically moving parts. A small amount of high pressure motive gas is allowed to expand to a lower pressure and in so doing to drag along a second induced gas stream and compress this stream from some lower pressure to the higher pressure to which the motive gas is allowed to expand. The work, or power, input lies in the motive gas which had to be compressed from the eductor's discharge pressure to the level of the motive pressure. The work output lies in the compression of the induced gas stream from some lower pressure to that at the eductor discharge. The efficiency of the unit is related to the quantities of the two streams and the relative pressure differentials expended and achieved.

Nozzle Dimensions and Orientations

Conventionally the motive pressure is sufficiently greater than the discharge pressure that it flows into the eductor through a nozzle at a sonic throat velocity; or viewed conversely, the motive to discharge pressure differential is maintained via a restriction nozzle or orifice. In order to be effective in inducing and compressing a secondary flow, the stream discharging from this nozzle must be guided into a properly sized throat as depicted in Figure 1(a). If the motive gas nozzle is improperly, or not optimally, oriented relative to the throat as in Figure 1(b), or if the throat is too large in size as in 1(c), there can be little if any inducement or compression of the induced flow. Obviously there must also be negligible resistance to the inflow of the induced stream through duct D_I , if it is intended that it reach its maximum possible rate. The overall operating diagram can then be qualitatively deduced by considering the composite of a properly proportioned and oriented nozzle, throat, and induced flow port attached to a closed vessel provided with a valved opening, as illustrated in Figure 1(d).

Correlation of Operating Variables

For such fixed eductor dimensions there exist five operating variables:

Motive gas pressure ----- P_M

Figure 1. Dimensional Considerations in Eductor Design

Discharge gas pressure -----	P_D
Suction gas pressure -----	P_S
Motive gas flow-----	W_M
Induced gas flow -----	W_I

These variables are not entirely independent. For example, with dimensions fixed, the motive gas rate, W_M , through nozzle D_N is a function of the ratio of P_M/P_D , whether the flow is critical or sub critical; the induced flow and/or its source pressure, P_S , are in turn a function of the motive gas flow and its pressure dissipation. It has been found empirically that all five variables can be represented graphically in terms of flow ratios versus pressure differential ratios, in much the same fashion as the classical heat and mass transfer numbers plotted versus Reynolds number, which also represent simply driving force ratios versus flow ratios.

Operating Characteristics

Consider the operation of the eductor in Figure 1(d), with fixed D_T and D_I , relative to the coordinates of Figure 2 under conditions at which the unit discharges to atmosphere and the motive pressure is held constant at some substantially higher pressure. This corresponds to holding constant the numerator of the abscissa and the denominator of the ordinate. With valve V , in Figure 1(d), closed the eductor will draw gas flow through opening D_I until the pressure in the closed volume upstream of D_I reaches some minimum value where further vacuum would result

in back-leakage of motive gas. At this maximum achievable vacuum, or minimum achievable pressure P_s , the induced flow W_I is zero, as represented by point A in Figure 2. If valve V is then partially opened, the vacuum is broken due to the in-leakage of gas, and the eductor sucks out the inleakage as the induced gas stream through D_I ; however, it cannot maintain the same degree of vacuum as existed prior to the inleakage. As a result $(P_D - P_s)$ decreases and W_I/W_M increases, moving the operating point in Figure 2 from A to B. As valve V is progressively opened, the eductor attempts to remove the increasing inleakage and maintain the suction pressure P_s at as low a value

as possible (e.g. remove the inleakage as rapidly as possible). Thus W_I progressively increases, and the vacuum, or $(P_D - P_S)$, simultaneously decreases moving the operating point along the curve of Figure 2 from A to B to C to D to E etc.

Optimum Performance

At any point along the curve ABCDE the "work" accomplished by the eductor is represented by the amount of the induced flow and the attained pressure differential $(P_D - P_S)$, which is a measure of the amount of compression added to the induced flow. At constant W_M and $(P_M - P_D)$, the value of the ordinate divided by the abscissa, at any point along the curve in Figure 2, represents a measure of this work and obviously yields a peak or maximum at the point of tangency with a line of 45° slope, since the coordinates are logarithmic. The curve beyond the point of maximum efficiency, represented by D in Figure 2, is not as easily definable, since it comes under the influence of extraneous resistances imposed for example by dimensions D_I and/or D_T coupled with the flow of W_I and of $W_M + W_I$. If dimension D_I or valve V in Figure 1(d) were infinite then P_S could be regarded in the practical sense as equal to P_D so that the abscissa of Figure 2 would reach infinity with the eductor drawing a maximum induced flow. On the other hand, any restriction in

D_I or particularly in D_T could result in P_S , upstream of D_I , approaching P_D at some lesser induced flow. The operating diagram beyond point D can therefore take on varied extrapolations as illustrated in Figure 2, from point D through F.

The curve of Figure 2 is typical of any of an infinite number each relating to a fixed unit with a given design ratio of D_T/D_N . This is illustrated by the family of dashed lines in Figure 3 which represents the conventional eductor design correlation; the solid curve in Figure 3 is the locus of points of maximum performance for which the corresponding ratios of D_T/D_N are generally represented in the scale along the left hand ordinate.

Design Data

The coordinates of Figure 3 have been found to correlate a wide range of experimental and industrial data whether the motive gas passes through nozzle D_N in critical flow (which corresponds to conventional practice) or in sub-critical flow. In general, vendors' quotations appear conservative relative to carefully conducted "academic" research results; the vendors' locus of optimum performance (e.g. the "design" curve) appears to be reduced by a factor of nearly 1.5 on the W_I/W_M scale. The material which follows summarizes reasonably carefully conducted laboratory investigations with neither optimistic nor conservative interpretations.

Vacuum Inducers

The shape of the performance curve for an eductor of fixed dimensions is typified in the data in Figure 4(a), based on experiments carried out in a simple arrangement such as illustrated in Figure 4(b). In these experiments, no attempt was made to optimize the distance between the nozzle D_N and the entrance to the throat, D_T , as either a function of the motive pressure and/or the abscissa of Figure 4(a). Such optimization, in varying degrees, has given rise to the apparent "discrepancies" between results reported by different investigators. Rather than strive for the

Figure 4. Typical Performance Curve for a Simplistic Eductor Configuration

ultimate in designing an item generally contributing negligibly to the overall cost of its associated equipment, satisfactory eductor dimensions and capacities can readily be determined from the conservative design curve of Figure 5. A typical gas eductor design procedure utilizing Figure 5, is illustrated in a subsequent section devoted to "Sample Calculations."

Compressors to 850 psi Discharge

Though conventionally regarded as a vacuum producing device an eductor can conversely be regarded as a compressor raising an induced gas stream from a suction pressure, P_s , to the discharge pressure, P_D . In effect such operation corresponds to imposing a back-pressure and determining whether all the pressures will rise simultaneously and whether the correlants of Figure 5 still apply. At least two substantial, albeit limited range, investigations of this nature have been conducted. The most recent was reported by Westinghouse Electric Corp. in the "2nd" Quarterly Project Report covering the period of 1/1/81 through 3/31/81 under DOE contract DE-AC21-80 ET 17093. In these tests motive pressures ranged to 300 psig and discharge pressures to 105 psig. The experimental results, from the apparatus shown in Figure 6, are summarized in Figure 7.

Figure 7 also includes data on an ejector tested by Foster

For "design" purposes this Figure, which closely matches vendor's past quotations, is conservatively based on approximately 60% of the maximum capacities achievable under field-adjusted optimum conditions

Figure 5. Optimum Eductor "Design" Correlation

Figure 6. Arrangement of Apparatus for Small Scale High Pressure Eductor Experiments

Figure 7. Performance Data for Experimental Eductors Operating Against High Discharge Pressures

Wheeler Ltd. at a CEGB power station in England in the late 1950's. This ejector operated with 1800 psia motive gas and a 570 psia suction pressure. It appears from the foregoing that barring extraneous flow restrictions, an in-line critical flow nozzle optimally oriented relative to the eductor throat can be expected to yield performance curves such as depicted in Figure 7, with the curve of Figure 5 representing a reasonably conservative design basis.

Particle Flow Induction

In Chapter 3 of C.E. Lapple's book on "Fluid and Particle Mechanics" (Universtiy of Delaware, March 1951), G.E. Alves suggests that "solids suspended in the secondary (induced) stream may be taken into account" by using the mixture density. No supporting data are presented though the author states that special ejectors were being manufactured commercially with large throats and provisions for admitting the solids along the center line, and for admitting the primary fluid through an annulus formed by the solids inlet tube and the ejector wall.

In the July 1961 issue of Chemical Engineering, E.J. Gibbons proposes design criteria for a "venturi feeder" for solids induction as shown in Figure 8. Though not an eductor in the full sense, his design is in any event not helpful in the sense of

suggesting how any significant solids flow would affect performance. His design criteria call for a superficial air velocity of at least 50 ft/sec and a loading of no more than 0.02 lbs of solids per cubic foot of air at location 3 (discharge) of the feeder in Figure 8. The presence of the solids is in no way taken into account in his calculation of pressures along the flow path from points 1 to 3 and no air is assumed to enter with the solids; thus the solids are merely "dribbled" into the contracted air flow at point 2 at so low a rate that they essentially "go along for the ride."

Under a DOE contract related to the Reevesville Fluidized Bed Combustor, Pope Evans and Robbins were supposed to conduct an experimental investigation of eductor performance in feeding coal into a fluidized bed. Roth, Divilio and Reed reported on

this in a paper entitled "Eductor Performance in the Coal Feed System of a Fluidized-Bed Combustor." However, in their experimental work they kept the coal feed port of the eductor blanked off, and in effect merely investigated various eductor designs determining the maximum vacuum attainable at various back pressures up to 30 inches of water gage. They were apparently unaware of the earlier data and never reported any results under actual solids feed conditions. At best their consumption of taxpayers funds showed that a single central motive gas nozzle required less motive pressure and mass flow to achieve a desired suction pressure, P_s , than a peripheral arrangement of multiple nozzles. This may be a matter of the direction of the jet exiting from the peripheral nozzles and does not eliminate a true annular in-line motive gas flow eductor from further consideration. There is a significant difference between an arrangement such as in Figure 9 (a) and the Pope, Evans & Robbins arrangement in Figure 9 (b).

A basic experimental investigation of the effect of induced solids has currently been initiated in the Department of Chemical Engineering at Manhattan College supported by grants from several industrial sponsors. The experiments will initially be carried out in an apparatus such as illustrated in Figure 10. In this arrangement with an eductor of fixed D_T and D_N , the solids enter through V-1, which constitutes an orifice of variable diameter. The back pressure, or discharge pressure, P_D , will be controlled via valve V-2 (with valve V-5 shut). Suction pressure, P_S , and solids induction rate, W_S , will be controlled via valves V-1 and V-3. The object of any of the proposed series of tests will be to determine to what degree W_S is increased, over and above its gravity flow rate through the orifice, V-1, as a function of the following:

- a) back pressure, P_D
- b) motive pressure, P_M
- c) particle size and bulk density
- d) suction pressure, P_S

In the dimensioning of the initial test apparatus it will be assumed that the solids efflux rate through the orifice at V-1, will simply correspond to that across any orifice of equal size bearing the pressure differential, $P_A - P_S$, in Figure 10. The maximum that would be drawn is predictable from the abscissa of Figures 3, 5 or 7 when the ordinate is equal to zero. However, this predictable solids flow is quite substantial and could

Figure 10. Proposed Arrangement of Ejector for Solids Induction Tests at Manhattan College

then conceivably alter the pressure drop through the throat of the eductor, thereby also substantially altering eductor performance. The object of the test program will be to determine how severe this effect would be and how it might be taken into account in eductor design. Variations in eductor dimensions, as well as orientation, (horizontal, sloped and vertical) will also be investigated.

As an example of the anticipated flows to be accomodated, consider the sizing of an initial test unit such as illustrated in Figure 10. The "heart" of the apparatus lies in the size of the motive gas nozzle, the throat diameter, the depth of the bed of solids above V-1 (which is controlled by the depth of immersion of the retractable tube below V-3) and the size of the "orifice" at V-1. If dimensions such as in Figure 11 were chosen for the initial tests, then with V-1 wide open and no solids upstream of V-1, Figure 5 would suggest that at $D_T/D_N = 0.875/0.125$, or 7.0, $(P_M - P_D) / (P_D - P_S) \approx 80$ and W_I/W_M should reach at least 3:1. Under the conditions of Figure 11, P_S should reach about 29.5 inches of water vacuum. If V-1 were opened to an area equal to a 7/8" diameter hole (equal to the eductor's throat diameter) then a 29.5 (0.03613) pressure differential of 1.066 psi across 24" of bed depth above this hole would predictably (I. & E.C. Proc. Des. & Dev., 3, 348-355, 1964) suggest about 4 times the gravity flow rate of about 0.4 lbs/sec at zero pressure differential

Figure 11. Principal Dimensions of Initial Test Eductor

(I. & E.C. Fundamentals, 18, 345-348, 1979). The point of interest lies in whether 4×0.4 or 1.6 lbs/sec of solids can be passed through the 7/8" throat along with the 22.5 SCFM of air leaving the 1/8" nozzle, still drawing the same vacuum. Volumetrically this quantity of solids is less than the volume of air that would be induced if there were no solids in the 2" tube upstream of V-1. However, the kinetic energy loss for the suspension passing through the throat would be nearly eight times that of the combined gas streams. It is anticipated that the eductor research program at Manhattan College during 1982 and 1983 will resolve these and similar questions concerning the fluid mechanics of two-phase inductions on eductor performance.

Sample Calculations

Critical Flow

In the diagram below, pressure P_3 is held constant at 200 psig by a pressure switch hooked to an air compressor in the usual fashion. The pressure at P_2 is held constant at 35 psig by the pressure regulating valve V-1. The orifice is $1/2"$ in diameter and all the piping shown is 8" i.d. The back pressure P_1 is controlled by valve V-2. Downstream of V-2 the pipe exhausts to essentially ambient at $P_0 = 15$ psia.

Assume: the orifice to have a constant flow coefficient of unity; no frictional resistance in any of the piping or valves; the ratio of specific heats, K , is constant and equal to 1.4; the entire system operates isothermally at 60° F ; and the ideal gas laws apply.

1) If V-2 is wide open show whether flow through the orifice is critical or subcritical.

2) When V-2 is wide open what is the velocity of flow of air out of the 8" i.d. pipe at P_0 in ft/sec?

3) When V-2 is wide open what is the velocity of the air through the orifice based on conditions in the throat of the orifice?

4) If V-2 is partially closed to the point where P_1 is increased to 20 psig, what will be the flow of air out of the 8" i.d. pipe in lbs/second?

5) If the partially closed valve V-2 exhibited a flow coefficient of 0.6, what is its degree of opening expressed as a percentage of the area of the 8" i.d. pipe?

Solutions:

1) Critical flow occurs if $P_1/P_2 \leq [2/(K + 1)]^{K/(K-1)}$

$$\left(\frac{2}{K+1}\right)^{\frac{K}{K-1}} = \left(\frac{2}{1.4+1}\right)^{\frac{1.4}{0.4}} = 0.528$$

$P_1/P_2 = 15/(35 + 15) = 0.3$ when V-2 is wide open
since 0.3 is smaller than 0.528, flow is critical.

2) Since flow will be critical (see Appendix equation 15)

$$Q_{crit. at P_2, T_2} = A_o \sqrt{g K R T_2 \left[\frac{1+K}{2} \right]^{1+K/1-K}}$$

$$= \frac{\pi (0.5)^2}{4 \times 144} \sqrt{\frac{32.2 \times 1.4 \times 1545 \times 520}{29} \left[\frac{2.4}{2} \right]^{2.4/-0.4}}$$

$$= 0.88 \text{ ACFS at upstream conditions}$$

$$\text{Velocity out of 8" pipe} = 0.88 \times \frac{15 + 35}{15} \times \frac{4 \times 144}{\pi (8)^2}$$

$$= 8.4 \text{ ft/sec}$$

3) From equations (17) or (18) in the Appendix

$$\frac{Q_{crit. at throat cond's.}}{A_o} = \sqrt{g K R T \left(\frac{K+1}{2} \right)}$$

$$= \sqrt{\frac{32.2 \times 1.4 \times 1545 \times 520}{29} \left(\frac{2.4}{2} \right)}$$

$$= 1224 \text{ ft/sec}$$

$$\text{or: } \frac{(0.88 \text{ from 2) above} \times 4 \times 144}{\pi (0.5)^2} \times \frac{50}{50 \times 0.528} = 1224 \text{ ft/sec}$$

4. Flow will be subcritical through orifice since $P_1/P_2 =$

$$\frac{20 + 15}{35 + 15} = 0.7$$

Flow will be critical through valve since $P_0/P_1 =$

$$\frac{15}{20 + 15} = 0.429$$

(0.7 exceeds the critical pressure ratio of 0.528 as calculated previously whereas 0.429 is less than 0.528)

From equation (14) of the Appendix

$$\begin{aligned} Q_{\text{at cond's. upstream of the orifice}} &= \left(\frac{P_1}{P_2}\right)^{1/K} A_o \left(\frac{2gKRT}{K-1} \left[1 - \left(\frac{P_1}{P_2}\right)^{(K-1)/K} \right] \right)^{1/2} \\ &= \left(\frac{35}{50}\right)^{1/4} \frac{\pi (0.5)^2}{4 \times 144} \left[\frac{2 \times 32.2 \times 1.4 \times 1545 \times 520}{29 (1.4 - 1)} \left(1 - \left(\frac{35}{50}\right)^{0.4/1.4} \right) \right]^{0.5} \\ &= 0.822 \text{ ACFS at upstream conditions} \end{aligned}$$

$$\text{or: } 0.822 \times 0.076 \times 50/15 = 0.208 \text{ lbs/sec.}$$

5. Flow through partially open valve is critical; from equation (15) of the Appendix:

$$Q_{\text{crit. at } P_1, T_1} = C_o A_2 \sqrt{g K R T \left[\frac{1+K}{2} \right]^{1+K/1-K}}$$

$$\frac{0.208}{0.076} \frac{15}{15+20} = 0.6 A_2 \sqrt{\frac{32.2 \times 1.4 \times 1545 \times 520}{29} \left[\frac{2.4}{2} \right]^{2.4/-0.4}}$$

$$\begin{aligned} \therefore A_2 &= 0.003033 \text{ ft}^2 \\ &= 0.4367 \text{ in}^2 \end{aligned}$$

$$\frac{0.4367}{\pi (8)^2/4} \times 100 = 0.869\% \text{ open}$$

Subcritical Flow

In the diagram below, the upstream 100 psig pressure is held constant via a pressure switch hooked to a compressor. Assume: both orifices have a flow coefficient of unity; the ratio of specific heats, K , is constant and equal to 1.4; there is no frictional resistance in the piping; the entire system is isothermal at 60° F; and the ideal gas laws apply.

- 1) What is the pressure in the section of pipe between the two orifices?
- 2) What is the minimum suction volume of the compressor, in SCFM, necessary to maintain the conditions shown in the diagram?
- 3) What is the ideal adiabatic horsepower of the compressor?

Solutions:

- 1) There are 4 possibilities:
 - a) flow through "A" is subcritical and flow through "B" is subcritical.
 - b) flow through "A" is critical and flow through "B" is critical.
 - c) flow through "A" is subcritical and flow through "B" is critical.
 - d) flow through "A" is critical and flow through "B" is subcritical.

Though the correct possibility may be intuitively obvious consider its determination by the logical process of elimination: consider possibility a):

$$\text{The critical pressure ratio} = \left(\frac{2}{K + 1} \right)^{K/K-1} = 0.528$$

If flow through "A" were subcritical then $P > 0.528 \times 114.7$,
or > 60 psia

Then through "B", $14.7/P$ must be $14.7/>60$ which is < 0.528
and hence flow through "B" would have to be critical which
eliminates a) as a possibility.

Consider possibility b):

This requires both flows to be critical, but obviously also
equal in quantity, as must be the case in all instances.

From equation (15) of the Appendix:

$$Q_{\text{crit. at upstream Press.}} = A_o \sqrt{g K R T \left[\frac{1+K}{2} \right]^{(1+K)/(1-K)}} = A_o \sqrt{\text{Constant}}$$

and since SCFS through "A" = SCFS through "B"

$$\frac{114.7}{14.7} (\text{area of "A"}) \sqrt{\text{constant}} = P/14.7 (\text{area of "B"}) \sqrt{\text{constant}}$$

If flow through both orifices critical, then $28 \text{ psia} < P < 60$
psia, since the critical pressure ratio is 0.528.

(e.g. $P \geq 14.7 / 0.528 = 28$; and $P \leq 114.7 \times 0.528 = 60$ psia)

but areas of "A" and "B" differ by a factor > 5

$$\frac{114.7}{14.7} (0.75)^2 (\text{constant}) = \frac{28 \text{ to } 60}{14.7} (1.7)^2 (\text{constant})$$

$$4.39 (c) = 5.5 \text{ to } 11.8 (c)$$

which indicates that flow through "B" would exceed flow
through "A" which is impossible and therefore possibility
b) is eliminated.

Consider possibility c):

If flow through "A" is subcritical then $P > 60$ psia. Since
the area of "B" is 5 times as great as that of "A" the cri-
tical flow through "B" would have to be much greater than
that through "A".

$$\frac{P > 60}{14.7} (1.7)^2 >>>> \frac{114.7}{14.7} (0.75)^2$$

$$> 11.8 >>>> 4.39$$

which again eliminates possibility c) on the basis of the
material balance.

Consider possibility d):

From equation (15) of the Appendix the volumetric flow through "A" under critical flow conditions calculated at 114.7 psia is represented by:

$$Q_{\text{at 114.7 psia}} = \frac{\pi (0.75)^2}{4 \times 144} \sqrt{g K R T \left[\frac{1+K}{2} \right]^{1+K/1-K}} \quad (\text{A})$$

From equation (14) of the Appendix the volumetric flow through "B" under subcritical flow conditions calculated at 114.7 psia is represented by:

$$Q_{\text{at 114.7 psia}} = \left(\frac{P}{114.7} \right) \left(\frac{14.7}{P} \right)^{1/K} \frac{\pi (1.7)^2}{4 \times 144} \sqrt{\frac{2g K R T}{(K-1)} \left[1 - \left(\frac{14.7}{P} \right)^{K-1/K} \right]} \quad (\text{B})$$

Equating (A) to (B) and solving for P results in P = 23 psia

- 2) Since flow through "A" is critical solving (A) above:

$$Q_{\text{at 114.7}} = \frac{\pi (0.75)^2}{4 \times 144} \sqrt{\frac{32.2 \times 1.4 \times 1545 \times 5.20}{29} [1.2]^{2.4/-0.4}}$$

$$= 1.984 \text{ CFS at 114.7 psia}$$

$$\text{SCFM} = 1.984 \times \frac{114.7}{14.7} \times 60$$

$$= 928.8$$

$$3) \text{ IAHP} = \frac{144}{33000} \frac{K}{K-1} V_s P_s \left[\left(\frac{P_2}{P_1} \right)^{K-1/K} - 1 \right]$$

$$= \frac{144}{33000} \frac{1.4}{0.4} 928.8 \times 14.7 \left[\left(\frac{114.7}{14.7} \right)^{0.4/1.4} - 1 \right]$$

$$= 166.8$$

Gas Eductor Design

Using 100 psig motive gas it is desired to exhaust to atmosphere 20,000 lbs/hr of air at a suction pressure of 15" of water vacuum. How much motive air is necessary and what are the design dimensions of the throat and nozzle of the eductor assuming isothermal conditions at 60°F and ideal gas laws apply?

Solution:

$$\frac{P_M - P_D}{P_D - P_S} = \frac{114.7 - 14.7}{14.7 - [14.7 - 15(0.03613)]}$$

$$= 184.5$$

from Figure 5 at optimum operating conditions, $W_I/W_M \approx 5$ and

$$D_T/D_N \approx 12$$

Therefore required motive air, $W_M = 20,000/5 = 4000$ lbs/hr.
 Since flow through D_N will be critical (e.g. $14.7/114.7 < (2/K+1)^{K/(K-1)}$)
 D_N is calculable from equation (15) of the Appendix where now:

$$\frac{(4000)}{3600 \times 0.0765 \times 114.7} \times 14.7 = \frac{\pi (D_N)^2}{4 \times 144} \sqrt{\frac{32.2 \times 1.4 \times 1545 \times 520}{29} \left[\frac{2.4}{2} \right]^{2.4/-0.4}}$$

$$\begin{aligned}\text{so that } D_N &= 0.0115 (4000)^{1/2} \\ &= 0.727 \text{ inches}\end{aligned}$$

and therefore $D_T = 8.72$ inches

AppendixDerivation of Critical and Subcritical Gas Flow Relationships

With respect to the nozzle shown below

an energy balance yields:

at P ₁		at P ₂
kinetic energy	+	kinetic energy
internal energy	+	internal energy
net heat transfer	=	net work or fric- tion
$\frac{V_1^2}{2gJ}$	+	$\frac{V_2^2}{2gJ}$
+ h_1	+	+ h_2
+ Q	=	+ W

(1)

where J is the mechanical equivalent of heat (1 BTU = 778 ft-lbs)

Since flow is adiabatic, $Q = 0$

No work is done in the nozzle so $W = 0$

Since no fluid is lost

$$\frac{A_1 V_1}{v_1} = \frac{A_2 V_2}{v_2} \quad (2)$$

Solving equation (2) for V_1 and substituting in equation (1) yields

$$V_2 = \left[\frac{2 g J (h_1 - h_2)}{1 - [A_2 v_1 / A_1 v_2]^2} \right]^{1/2} \quad (3)$$

if A_2 is small relative to A_1 then

$$V_2 = [2 g J (h_1 - h_2)]^{1/2} = 223.7 (h_1 - h_2)^{1/2} \quad (4)$$

otherwise $[1 / 1 - (A_2 v_1 / A_1 v_2)^2]^{1/2}$ is a correction factor.

For a perfect gas $h_1 - h_2 = C_p (T_1 - T_2)$

therefore equation (4) may be written as

$$V_2 = 223.7 \left[c_p T_1 \left(1 - \frac{T_2}{T_1} \right) \right]^{1/2} \quad (5)$$

If the change in volume is isentropic then

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{(K-1/K)}$$

and equation (5) becomes

$$V_2 = 223.7 \left[c_p T_1 \left(1 - (P_2/P_1)^{K-1/K} \right) \right]^{1/2} \quad (6)$$

The weight flow through the nozzle is given by

$$W = AV/\nu \quad (7)$$

For an ideal gas under isentropic conditions

$$\nu_2 = \nu_1 (P_1/P_2)^{1/K} = (P_1/P_2)^{1/K} (R T_1/P_1) \quad (8)$$

Substituting equations (6) and (8) in (7)

$$W = 223.7 \frac{A_2 P_1}{R} \left[\frac{c_p}{T_1} \left(\left[\frac{P_2}{P_1} \right]^{2/K} - \left[\frac{P_2}{P_1} \right]^{K+1/K} \right) \right]^{1/2} \quad (9)$$

The maximum velocity through the "orifice" A_2 occurs when

$$\left(\frac{P_2}{P_1} \right)^{2/K} - \left(\frac{P_2}{P_1} \right)^{K+1/K}$$

is a maximum; the pressure, P_2 , which makes this term a maximum can be found by equating its differential, with respect to P_2 , to zero. Thus:

$$\frac{d \left[\left(\frac{P_2}{P_1} \right)^{2/K} - \left(\frac{P_2}{P_1} \right)^{K+1/K} \right]}{d P_2} = 0$$

$$\left(\frac{1}{P_1^{2/K}} \right) \left(\frac{2}{K} \right) \left(P_2^{(2-K)/K} \right) - \left(\frac{1}{P_1^{(K+1)/K}} \right) \left(\frac{K+1}{K} \right) \left(P_2^{1/K} \right)$$

$$P_2 = P_1 \left(2/(K+1) \right)^{K/(K-1)} \quad (10)$$

where P_2 is the critical pressure. The critical pressure ratio is therefore

$$\frac{P_2}{P_1} = \left(\frac{2}{K+1} \right)^{K/(K-1)}$$

Substituting the right-hand side of equation (10) for P in equation (9), referring to A_2 as A_t (throat or orifice area), and simplifying:

$$W = A_t P_1 \left(\frac{2}{K+1} \right)^{1/(K-1)} \left[\frac{2gK}{RT_1(K+1)} \right]^{1/2} \quad (11)$$

This results because $(2gJ)^{1/2} = 223.7$, and because it may be shown that $c_p = KR/J (K-1)$ on the basis that in a constant pressure process:

$$Q = \Delta U + w \quad (12)$$

where $Q = W C_p (T_2 - T_1)$

$U = W C_v (T_2 - T_1)$

$$\text{and } w = \int P dv = \int_{v_1}^{v_2} dv = P (V_2 - V_1), \text{ ft. lbs} \\ = \frac{WR (T_2 - T_1)}{J}, \text{ BTU}$$

Substituting in (12) yields:

$$W C_p (T_2 - T_1) = W C_v (T_2 - T_1) + WR (T_2 - T_1) / J$$

but $C_p = C_v + R/J$, and letting $K = C_p/C_v$ yields

$$K C_v - C_v = R / J$$

so that $C_v = R / J (K - 1)$

$$\text{and } C_p = KR / J (K-1)$$

For the case where A_t/A_1 is very small, $K = 1.4$ (e.g. air), and the orifice coefficient is essentially unity, equation (11) reduces to:

$$W / A_t = 3.88 P_1 / (R T_1)^{1/2}$$

Going back to equation (6), note this is difficult to solve if P_2 is close to P_1 . This can be circumvented by expanding equation (6) by the binomial theorem to give:

$$V_2 = 223.7 \left\{ C_p T_1 \left(\frac{K-1}{K} \right) \left[\frac{P_1 - P_2}{P_1} + \frac{1}{2K} \left(\frac{P_1 - P_2}{P_1} \right)^2 \right] \right\} \quad (13)$$

If $P_1 - P_2$ is small relative to P_1 then the second term within the brackets becomes negligible and equation (13) becomes:

$$V_2 = \sqrt{2 g R T_1 (P_1 - P_2) / P_1}$$

but since $R T_1 / P_1 = v_1 = 1 / \rho_1$

$$V_2 = \sqrt{2 g \Delta P / \rho_1}$$

which is the simple form of the orifice equation with a coefficient = 1.0.

Going back to equation (9) and substituting $(2 g J)^{1/2}$ for 223.7; $K R / J (K-1)$ for C_p ; and simplifying, yields:

$$W = \left(\frac{P_2}{P_1} \right)^{1/K} A_2 \frac{P_1}{R} \left[\frac{K R}{J(K-1)} \frac{2 g J}{T_1} \left(\frac{T_1}{T_1} \right) \left(1 - \left(\frac{P_2}{P_1} \right)^{(K-1)/K} \right) \right]^{1/2}$$

$$\text{or } Q_{\frac{at}{P_1, T_1}} = \left(\frac{P_2}{P_1} \right)^{1/K} A_2 \left[\frac{2 g K R T_1}{K-1} \left(1 - \left(\frac{P_2}{P_1} \right)^{(K-1)/K} \right) \right]^{1/2} \quad (14)$$

Substituting the critical pressure for P_2 as given by the right hand side of equation (10) yields

$$Q_{\frac{crit.}{at P_c, T_1}} = A_2 \sqrt{g K R T_1 \left[\frac{1+K}{2} \right]^{1+K/1-K}} \quad (15)$$

$$Q_{crit.} \text{ at } P_2, T_1 = A_2 \frac{P_1}{P_2} \sqrt{g K R T_1 \left[\frac{1+K}{2} \right]^{1+K/1-K}} \quad (16)$$

$$= A_2 \sqrt{\frac{g K R T_1 P_1^2}{P_2^2} \left[\frac{1+K}{2} \right]^{1+K/1-K}}$$

$$= A_2 \sqrt{\frac{g K R T_1 P_1^2 \left[\frac{1+K}{2} \right]^{1+K/1-K}}{P_1^2 \left[\frac{2}{K+1} \right]^{2K/K-1}}}$$

$$= A_2 \sqrt{g K R T_1 \left(\frac{K+1}{2} \right)^{-2K/1-K} \left(\frac{K+1}{2} \right)^{1+K/1-K}}$$

$$= A_2 \sqrt{g K R T_1 \left(\frac{K+1}{2} \right)} \quad (17)$$

So that for adiabatic conditions the critical velocity in the throat or orifice becomes:

$$V_{crit.} = \sqrt{g K R T \left(\frac{K+1}{2} \right)} \quad (18)$$

Nomenclature

V	=	velocity, ft/sec
g	=	gravitational acceleration, 32.2 ft/sec x sec
J	=	778 ft lbs per BTU
v	=	specific volume, ft ³ /lb
A	=	cross sectional area, ft ²
h	=	internal energy, BTU/lb
T	=	temperature °R
C_p	=	specific heat at constant pressure BTU/lb x °F
P	=	pressure, psf abs.
K	=	ratio of specific heats C_p/C_v
C_v	=	specific heat at constant volume
W	=	weight rate of flow, lbs/sec
R	=	the ideal gas constant, 1545.5 / MW, ft lbs/°F x mol
Q	=	volumetric flow rate, ft ³ /sec (measured at conditions noted)

Internals in Fluidized Beds

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Internals in Fluidized Beds

Internals other than cyclone diplegs, intentionally immersed within the dense bubbling beds of fluidized solids reactors generally have either of the following purposes:

- 1) the removal of exothermic heats of reactions; included hereunder would be such applications as drying, heat treating for crystal reorientation, thermal shock treatments, and similar heat oriented functions.
- 2) the retardation of bubble rise velocities which in effect increases gas residence time and is frequently interpreted as a bubble size control function.
- 3) the breakage of "clumps" of bed solids such as result in the neighborhood of liquid feed points where the liquid feed rate exceeds the bed's ability to locally flash this feed instantaneously.
- 4) the breakage of agglomerates or of individual particles which grow in size in the course of the reaction and would cause poor flow characteristics and bed slumping if not controlled by some internal particle attrition device.

Geometrical Arrays of Tubular Internals

The uniformity of bed temperature, resulting from the enormous surface area and high rate of solids mixing makes fluidized beds particularly attractive for carrying out exothermic reactions and withdrawing the heat through immersed coils which in many instances can very effectively become internal steam generating boilers. It is therefore little wonder that considerable attention has been given in the past to the orientation and design of such tubular internals.

Whether arrived at intuitively or on the basis of mechanical stress, support, surface wear, or thermal expansion considerations, tubular internals in such industrial processes as acrylonitrile, Fischer-Tropsch synthesis, oxychlorination, silicon, ore treaters, etc., have quite consistently been oriented vertically. The mechanical details vary, and where deemed advisable may take on a modular form in the fabrication. Figure 1. illustrates several common types such as serpentine coils, "harps", bundles, or "bayonets". Serpentine coils and harps vary in individual size, or number of tubes depending on the chord length across which they traverse the bed cross section; harps require upper and lower internal headers. Bayonets, because of their own inherent number and cost, the equal number of fluid feed lines and support hangers, etc., are not common in large diameter beds. The "bundles" illustrated in Figure 1. represent one modular form in which the upper steam

Figure 1. Typical Arrangements of Vertical Tubular Internals

drum and bottom boiler are steel spheres connected by equally spaced tubes on an equilateral triangular pitch emanating from the spheres in such a fashion that any cross section of the bundle from just below the upper sphere to just above the lower sphere would exhibit a hexagonal array as illustrated in Figure 2. Such array then permits stacking the bundles next to each

Figure 2.
Cross-Section of
Nested Hexagonal
Bundles (Plan
view of
Figure 1d)

other into the bed so that over the major portion of the bed height any vessel cross section would show a uniform pattern of tubes on equilateral triangular pitch centers over its entire area (barring smaller "fillers" or even "dead men" at appropriate points along the vessel periphery). Typically such modules require individual water feed lines to each boiler (in this instance entering through the steam drum and passing down through the center tube of the hexagonal bundle) and an individual steam drawoff line from each of the upper steam drums.

Uniformity of tube spacing over the vessel cross section is a characteristic design criterion regardless of the type of internal tubular arrangement (whether serpentine coils, harps, bundles, etc.). This is based on the premise that rising bubbles would preferentially seek any path, or channel, which was freer of such obstruction and therefore bed uniformity would suffer.

Effect on Bubble Hydrodynamics

In fluidized beds bubbles rise by displacement with inflowing bed solids. If surrounded in effect by a "cage" of such vertical tubes retarding the solids flow, the bubble will rise more slowly and its residence time within the bed will consequently be proportionately increased.

Within a uniform array of vertical tubes, or rods, the effective reduction in bubble rise velocity can be correlated in terms of the size of the bubble of equivalent hydraulic diameter defined as:

$$D_H = \frac{6(\text{free volume within the bubble})}{(\text{surface area within the bubble})} \quad (1)$$

The free volume within the bubble is taken as its spherical volume, $\pi D_B^3/6$, minus the volume occupied by the engulfed or penetrating

tubes. The surface area within the bubble is taken as that of the spherical surface, πD_B^2 , plus the total surface area of all tubes exposed within the bubble volume. Ignoring the shaded bubble surfaces lost at the points where tubular surfaces penetrate, as illustrated in Figure 3, equation (1) can be expressed as:

$$D_H = \frac{6 \left(\frac{\pi D_B^3}{6} - \frac{L_T \pi d^2}{4} \right)}{\pi D_B^2 + L_T \pi d} \quad (2)$$

$$= \frac{D_B^3 - 1.5 L_T d^2}{D_B^2 + L_T d} \quad (3)$$

where L_T and d are defined in Figure 3.

The larger the diameter and the closer the spacing of the vertical tubes, the smaller a bubbles' equivalent or hydraulic diameter. Equation (3) can be conveniently represented graphically by dividing both sides D_B and expressing all dimensions as ratios to the center-to-center spacing of the tubes so that:

$$\frac{D_H}{D_B} = \frac{(D_B/cc)^3 - 1.5 (L_T/cc)(d/cc)^2}{(D_B/cc)^3 + (L_T/cc)(d/cc)(D_B/cc)} \quad (4)$$

The smaller the bubble diameter relative to the tube spacing, the

Figure 3. Bubble Rising through a Square Array of Vertical Tubes

greater the possible variations in its hydraulic diameter, as illustrated in Figure 4. The conceivable extremes are reflected in the approximate minimum and maximum possible values of L_T given in Table 1.

Table 1. Extremes of Total Tube Lengths Exposed Within Rising Bubbles

<u>D_B/cc</u>	<u>L_T/cc</u>	
	<u>Minimum</u>	<u>Maximum</u>
1	0	1
2	2	5.7
3	10.6	15.9
4	29.2	34.6
5	60.8	69.7
6	107.7	117.7
7	169.0	180.4
8	260.4	275.1

Based on the figures in Table 1., equation (4) is represented graphically in Figures 5 (a) and 5 (b) for tube arrays on a square pitch. As an example, an 18" diameter bubble, D_B , rising through a square pitched array of 2" diameter tubes on 6" centers could actually be expected to rise through the fluidized bed at a velocity equal to that of a bubble, D_H , about 12" in diameter, or conversely the 18" bubble would in such case have a residence time about $\sqrt{18/12}$ or 1.22 times as long as it would have in a freely bubbling bed. It is this longer residence time which has erroneously given support to the theory that such internals limit bubble size, whereas in fact the overall physical dimensions of

Figure 4. Extremes in Tube Penetration Density
Within a Square Array

Figure 5. Hydraulic Diameters of Bubbles Rising Through Square Arrays of Vertical Tubes

the bubbles remain as in a bed without internals. Similarly the correlation of reactor performance with and without such internals has in the past been approached empirically in terms of the hydraulic diameter of the vessel rather than more fundamentally in terms of the hydraulic diameter of the bubbles.

Miscellaneous Arrangements of Internals

Limits such as illustrated in Table 1. and Figures 5(a) and 5 (b) can be derived for any regular geometric pattern of vertical tubular internals. Such regularity permits proper consideration of the effect on bubble hydrodynamics and therefore on the reaction kinetics. This is obviously of major importance in chemical reactions requiring a high degree of conversion per pass in order to be economically feasible. In other instances involving cruder operations such as combustion, drying, incineration, catalyst regeneration, etc. a regular geometric pattern is of minor importance; heat transfer coefficients from bed to immersed surfaces are inevitably substantial over their solids-swept areas and a difference of 20% to even 100% or more for optimally oriented internals is not necessarily critical. It is not uncommon in such instances to find tubular internals in the form of cylindrical or helical coils, or even horizontal or sloped tube banks as has been (perhaps ill-advisedly) practiced in fluidized bed coal combustors on the argument that horizontal tubes lend themselves to simpler support mechanisms, notwithstanding the experiences reported in

Exxon's miniplant operations or proposed alternate grid designs which would permit even deeper in-bed tube submergences and support.

Other than tube coils and banks of various configuration, so-called "plate-coils" have also been advantageously employed in panel and spiral arrangements, as illustrated in Figure 6, to equalize and extend solids path lengths or residence times and approach plug flow from the point of solids feed to the point of solids withdrawal.

In instances where the feed solids may be overly moist and tend to "cake", as in the drying of certain food products; or in instances where one or more reactor feed streams is of necessity in the liquid state not with the intent of simply being partially or completely evaporated (as in some pharmaceutical or polymerization type reactions), it is not uncommon to provide motor driven stirrers or agitators to break up or circulate any agglomerated lumps. These devices are usually custom designed to fit within the vessel shell or within matched contoured grids. They most frequently take the form of either "paddle wheels" or helical "ribbons" driven from horizontal or vertical shafts respectively.

In-Bed Attritors

As opposed to mechanically driven devices intended to break

Figure 6. Panel or Platecoil Internals

large and relatively loosely bonded unfluidizable agglomerates, it is equally necessary in some instances to break or attrite individual particles which have grown too large to remain unsegregated in the bed. Such particle growth occurs in numerous processes where the solids are partly or wholly the end product; fluid coking, black liquor oxidation, sludge incineration, etc. are typical. Generation of new nuclei via the feed mechanism is not always feasible nor necessarily sufficient to avoid defluidization and hence the individual bed particles must be shattered, chipped, or at least eroded.

In the order of increasing effectiveness free jets, jets impinging on a target plate, and opposed jets impinging upon each other as illustrated in Figure 32 of Task 1 have been employed effectively in-bed to control particle size and have been tested externally to establish their relative merits. The attrition at free jets is calculable from the relationship given by equation (3) in Task 1 dealing with "Distributors and Plenums in Fluidized Beds." Experimental data on the severest attrition resulting from opposing jets has not been reported in the literature in absolute quantitative form; however, a design procedure for dimensioning such equipment has been developed based on analogy to so-called "jet mills."

In 1960 Rumpf (Chem. Ing. Tech., 32, No. 3, 129-135) derived a relationship for the minimum distance, P_M , required to accelerate

a particle to velocity V_i , starting at an initial velocity of zero, which took the form:

$$P_M = \frac{4 D_p \rho_p}{3 \rho C_D} \left[\left(\frac{V_p}{V_p - V_i} - 1 \right) - \left(\ln \frac{V_p}{V_p - V_i} \right) \right] \quad (5)$$

The average velocity of the gas leaving the mouth of a nozzle at a velocity V_n , decays with distance P from the nozzle to lesser velocities, V_p , approximately as given by:

$$V_p = \frac{V_n}{1 + 0.1926 (P/D_T)} \quad (6)$$

Assuming that the gas picked up a particle at the mouth of the nozzle it is possible via equations (5) and (6) to calculate the distance P_M from the point of pickup at which the particle velocity reaches the maximum impact velocity V_i . This maximum particle velocity exists because as particle velocity increases, due to its acceleration by the gas, the gas velocity simultaneously decreases due to spreading of the jet. To affect the greatest degree of attrition or maximum impact, opposing nozzles should presumably be neither more nor less distant from each other than $2 P_M$.

Consider for example the design of an opposing jet in-bed attritor with nozzle i.d. equal to 1" and a nozzle velocity of

300 ft/sec intended to impact particles about 200 microns in diameter. If the particles' density is 80 lbs/cu.ft. and the nozzle gas density is 0.1 lbs/cu.ft. then from equation (6):

$$V_p = 300 / (1 + 0.1926 P) \quad (7)$$

and from equation (5) with $C_D = 0.4$:

$$P = 21 \left[\frac{V_p}{V_p - V_i} - 1 - \ln \frac{V_p}{V_p - V_i} \right] \quad (8)$$

Solving (7) and (8) simultaneously results in the figures shown in Table 2. which suggest that the nozzles be spaced only 6"

Table 2. Solutions of Equations (7) and (8)

<u>P, inches</u>	<u>ft/sec</u>	
	<u>V_p</u>	<u>V_i</u>
1	251.6	63.98
2	216.6	72.2
3	190.1	73.7
4	169.5	72.65

apart for maximum particle impact. If the nozzle i.d. were increased to 2" the spacing would increase from 6" to 10". The rate of attrition reverts to that from a simple nozzle or grid hole if the spacing between the opposing nozzles were equal to or greater than the sum of their jet penetration depths as discussed in Task 1. on design of "Distributors and Plenums in

Fluidized Beds." The nozzle diameter, D_T , is based on the optimum induced to motive flow ratio of about 3:1 as discussed in Task 4 dealing with the design of "Eductors."