FlameGuard™
Fire Retardant FRP Duct/Pipe

Design Considerations
FlameGuard Fiberglass Reinforced Pipe & Fittings

CUSTOMER / PROJECT: ____

REQUIREMENTS

1.0 PROCESS DESIGN

Most reinforced plastic duct is used in corrosive fume exhaust and that is the area we will address. OSHA requirements and good business practice dictate using long established guidelines for designing fume pick up hoods and exhaust systems from process connections to exhaust safely away from personnel. Two such guidelines are the American National Standard Fundamentals Governing the Design and Operation of Local Exhaust Systems, Z9.2-1960, or Industrial Ventilation, a manual published by the American Conference of Governmental Industrial Hygienists 1970.

In designing a fume exhaust system, the objective is to keep the contaminants out of the general process area using the smallest amount of air movement (CFM) practicable. Additional air movement means higher operating costs, bigger environmental control units (scrubbers, incinerators, stacks), and generally higher first costs (duct size).

1.1 **COSTS:** The anticipated concentration limits of the process stream needs to be evaluated for chemical corrosion resistance at temperature. Specific recommendations should be made by the resin manufacturer whenever possible. Fiberglass pipe is not subject to many of the corrosion problems associated with metal pipes, such as galvanic, aerobic, intergranular corrosion or pitting.

1.2 **DESIGN PRINCIPALS:** Flow of air is quantifiable as the product of the cross sectional area of the system and the air velocity. Typically this is expressed as:

$$Q = AV$$

Velocity may be expressed as a function of velocity pressure, which is always exerted in the direction of air flow. For duct systems conveying (contaminated) air at room temperature and atmospheric pressure, this relationship is:

$$V = 4005*(VP)0.5$$

Total system pressure may then be expressed as the sum of the static pressure, which may be positive or negative, and the velocity pressure, which is always positive.

$$TP = SP + VP$$

Where: Q = air flow in CFM (cubic feet per minute)
A = cross sectional area of the (duct, hood, etc.) system in square feet
V = velocity in linear feet per minute
TP = total system pressure, in inches of water
SP = static pressure of the system, in inches of water
VP = velocity pressure, in inches of water

The pressure difference required to move air into an opening must be sufficient to: a) accelerate the air from rest to a velocity and, b) overcome the turbulence losses at the opening. For acceleration, the energy required is equal to the velocity pressure. Turbulence losses at the hood opening vary depending upon the opening geometry. The coefficient of entry (Ce), a factor usually ranging between 0.6 and 0.98, indicates the relationship between the actual hood turbulence losses, and that of a "perfect" hood with no turbulence losses (or the ratio of actual to theoretical flow). See hood design below for additional guidance on hood entry losses.

To use the above relationships, hood static pressure requirements are calculated. Using the velocity equation above, the velocity pressure is set equal to the hood static pressure and then degraded by the hood coefficient of entry. The formula then becomes:

$$V = 4005* Ce *(SPh)0.5$$

The hood static pressure is determined from the above equation. The difference between this number and the velocity pressure determined in equation 2 is the hood\ entry, he, loss due to turbulence. Expressed as a formula:

$$SPh = VP + he$$

Hood entry losses, above, are often expressed as a function of the velocity pressure.

$$he = Fh*VP$$

Bernoulli's Theorem for the conservation of energy is used to determine actual fan/ duct/ hood sizing requirements. To summarize the Theorem for duct systems, the static pressure plus velocity pressure at an upstream point in the system must equal the sum of the static and velocity pressures at a second point downstream in the system, plus any friction and dynamic (turbulence) losses.

$$Sp1 + VP1 = SP2 + VP2 + \text{losses}$$

Friction losses will vary directly with the length of the duct system, and inversely with the diameter or cross sectional area of the system. Turbulence losses will increase with the number and severity of any changes in direction caused by fittings such as elbows, tees, reducers, transitions and hoods. Once the overall system is designed, usually through an interative process, the fan or blower system may be selected to provide the required capacity (CFM) at the required static pressure.

1.3 PROCESS DESIGN OPTIONS:

A. Local enclosure of fumes to the maximum extent possible
B. Consideration of push/ pull air flow systems
C. Optimal configuration of duct runs
D. Optimal selection of down stream control equipment

A. **PROCESS ENCLOSURES & HOODS:** The quantity of air required to capture and convey the air contaminants depends upon the size and shape of the hood, its position relative to the points of emission, and the nature and quantity of the contaminants. Good hood design will create air flow past the source of contamination sufficient to remove the contaminated air around the source and to draw that air into the hood.

A complete enclosure will be the most efficient from an exhaust standpoint, but may be unrealistic from an operations point of view. If enclosure is not practicable, the hood should be located as close as possible to the source and shaped to control the area of contamination.

While the specific gravity of the contaminant has an effect on the dispersion fo the contaminant (rising or falling relative to clean air), the overall effect on a properly designed exhaust system is negligible. This is due to the relatively low concentrations of the contaminant in the contaminated air mixture to be exhausted. Of much greater importance is locating the fume pick up point as close as possible to the point where the fumes are generated.

In operation, air will move to the openings in the hood. Critical to effective design is achieving an air velocity necessary to overcome any opposing air currents. This velocity is known as the "capture velocity" - the air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents and to capture the contaminated air causing it to flow into the hood.

The velocity of the air stream moving toward the hood is (approximately) inversely proportional to the square of the distance from the slot or hood opening. This emphasizes how important hood location is. The equation for calculating flow for round or essentially square free hanging hoods is:

$$V = Q/(10*X2 + A)$$

Where V = centerline velocity at distance X from the hood, fpm (feet per minute)
X = distance along the axis in feet (and where X is less than or equal to 1-1/2*D)
Q = air flow in CFM
A = area of the hood opening in square feet
D = diameter of round hoods or the side of essentially square hoods
Air volume and velocity ratios for some rectangular shapes may be represented as follows:

Hood Type	Aspect Ratio, W/L	Air Volume
Regular slot, plain end	<0.2	Q = 3.7 L*V*X
Regular slot, flanged end	<0.2	Q = 2.8 L*V*X
Regular slot, plain end	<0.2	Q = V*(10*X2 + A)
Regular slot, flanged end	<0.2	Q = 0.75*V*(10*X2 + A)
Open Booth	As Required	Q = V*A = V*L*W
Canopy Hoods	As Required	Q = 1.4*P*D*V
Where W = width of slots or rectangular opening, L = length of slots or rectangular opening, X = distance from opening, P = perimeter of the hood, D = distance of the canopy above the work, All units in feet		

The Industrial Ventilation Manual gives ranges of capture velocities for various processes as shown in the table below:

Contaminant Dispersion Method	Examples	Capture Velocity, fpm
Released without velocity into quiet air	Evaporation from tanks, degreasing, etc.	50 - 100
Released at low velocity into moderately still air	Spray booths, intermittent container filling, plating, pickling, low speed conveyor transfers, welding	100 - 200
Active generation into a zone of rapid air motion	Spray painting, barrel filling, conveyor loading, crushers	200 - 500
Released at high initial velocity into a zone of very rapid air motion	Grinding, abrasive blasting, tumbling	500 - 2000

Within each of the above categories and ranges, the following should be considered: Minimal room air currents favor the lower end of the range. A large hood, which generates a large air mass, favors the low end of the range. Contaminants of high toxicity and/or a high production, heavy use process favors the upper end of the range.

Once the air reaches the hood opeinng, it must enter the hood, which generates a pressure drop known as the hood entry loss. Hood shapes can very widely, but must have hood openings of between 70% and 100% of the corresponding duct area. Some typical entry loss coefficients are as follows:

Hood Type	Coefficient of Entry, C_0	Entry Loss
Duct - round, square or rectangular, plain end	0.72	0.93 VP
Duct - round, square or rectangular, flanged end	0.82	0.49 VP
Bell Mouth Inlet (flared reducer)	0.98	0.04 VP
Tapered Cone, 30° included angle	0.96	0.08 VP
Tapered Cone, 90° included angle	0.93	0.15 VP
Tapered Rect, 30° included angle	0.93	0.16 VP
Tapered Rect, 90° included angle	0.89	0.25 VP
Orifice plus flanged duct (duct velocity = slot velocity)	0.55	2.3 VP
Tapered hoods (face area > twice the duct area)	0.82 - 0.98, dependent on angle chosen, divide hood into simple shapes, sum factors for each shape	

Slot hoods are most commonly used to provide uniform exhaust air over a discrete length of contaminant generation, such as an open top tank, or over the face of a large hood. The purpose of the slot is to distribute the air flow. Slot velocity does not contribute to capture velocity - only to slot pressure drop. Capture velocity is related to exhaust volume and slot length. Slot hood designs can vary widely. Although variable slot widths offer flexibility, they are subject to tampering. Fixed slots and unobstructed (no internal baffling) plenums are generally the most reliable designs. Plenum velocities are often designed at 1/2 the slot velocity.

Some additional quidelines for the use of slot hoods are as follows: If the width of a tank is 20" or less, a slot on one side is acceptable. If the width is 20" - 36", slots on both sides are desirable. If the width is 36" - 48", slots on both sides are necessary, unless all other conditions are optimal. If the width is over 48", local exhaust is usually not practicable, and enclosure or push/pull systems should be considered. If the length of the plenum is over 6 feet, multiple take-offs are recommended, and if over 10 feet considered necessary.

Based upon the above criteria, tables have been constructed to provide guidelines for tank ventilation with lateral (side slot) exhaust:

Required Min. Control Velocity (fpm)	CFM per Square Foot Tank Area to Maintain the Required Minimum Velocities at the Following Tank Width to Length Ratios (W/L)				
	0.00 - 0.09	0.10 - 0.24	0.25 - 0.49	0.50 - 0.99	1.00 - 2.00
Hood along one side or two parallel sides of a tank when one hood is against a wall or baffle. Also for a manifold along the tank centerline					
50	50	60	50	50	100
75	75	90	110	130	150
100	100	125	150	175	200
150	150	190	225	250	250
Hood along one side or two parallel sides of a free standing tank not against a wall or baffle. Also for a manifold along the tank centerline					
50	75	90	100	110	125
75	110	130	150	170	190
100	150	175	200	220	250
150	225	250	250	250	250

Note. Use W/2 as the tank width when the manifold is along the centerline or two parallel sides of the tank.

B. **PUSH / PULL SYSTEMS:** A push/ pull ventilation system uses a nozzle to push a stream of air across a tank surface into an exhaust hood. This style is particularly useful when exhausting across large tanks (i.e., width greater than 48"), but also may be of benefit for smaller systems, due to their ability to reduce overall exhaust volume requirements. This reduced requirement can be made possible by the push air's ability to maintain its profile over a relatively long path, compared with exhaust air, which pulls equally in all directions.

Push pull systems should be designed so they can be easily modified or adjusted to obtain the desired results. The push air, or Pressure Slot, needs to supply the required amount of air to the exhaust hood as calculated by the methods above. This exhaust requirement is modified by the following formula:

$$QP = QE/(D*E),$$

QP = Quantity of air pushed
QE = Quantity of air exhausted
D = length of the throw, or distance from the push nozzle to the exhaust hood
E = the entrainment factor, = 2.0 for D<8, 1.4 for D<16, 1.0 for D<24, and 0.7 for D>24

In addition to the above, the hood height should be sufficient to capture the pushed air within a 10° dispersion. Using the tangent of 10° = 0.18, the height of the hood is determined to be 0.18*D.

C. **OPTIMAL CONFIGURATION OF DUCT RUNS:** Sizing and layout for duct systems follows the same logic and many of the same calculations as that for pipe layout. Larger diameters with few fittings cost more to install and less to operate than systems with smaller diameters and many fittings. The key is determining the economical balance between first and operating costs.

Duct and fitting losses for some common shapes are given below as a fraction of the VP. These are often converted to equivalent lengths of straight run duct for summing losses:

Fitting Style	Loss as Fraction of VP
Standard Long Radius 45° Elbow (1.5:1)	0.20
Standard Long Radius 90° Elbow (1.5:1)	0.39
Branch Entry Loss @ 45° Entry, as Fraction VP in Branch	0.28
Branch Entry Loss @ 90° Entry, as Fraction VP in Branch	1.00
Reducer @ 10° Taper	0.06
Reducer @ 20° Taper	0.10

The friction losses for common fittings as a function of linear feet of straight duct are as shown below:

Friction Loss in Fittings as Equiv. Ft of Duct			
Duct Diameter (inches)	Long Radius Sweep 45° Elbow (1:5:1)	Long Radius Sweep 90° Elbow (1:5:1)	Branch Entry Loss @ 45° Entry
2	1	3	2
3	1	4	3
4	3	5	5
6	4	7	7
8	5	10	11
10	7	13	14
12	9	17	18
14	10	21	21
16	12	24	25
18	13	28	28
20	16	32	32
22	18	37	36
24	20	40	40
26	22	44	44
28	24	49	47
30	26	51	51
36	32	63	
48	44	89	

The losses for straight duct may be interpolated from the chart above. This chart is conservatively based upon friction losses in galvanized duct with joints on 48" centers. Fiberglass duct is typically smoother than galvanized, with a Williams and Hazen friction coefficient of 150, and joints on 120" or 240" centers.

D. **OPTIMAL SELECTION OF DOWN STREAM CONTROL EQIPMENT:**
Downstream equipment must be chosen for its ability to perform the required function (scrubbing or incineration efficiencies, filtration efficiencies, stack height, etc.). However, within each category choices usually exist. Scrubbers, for instance, can vary widely in removal efficiencies and pressure drop requirements, and the relationships are not necessarily linear. Scrubbers, for instance, can be spray scrubbers, wet dynamic scrubbers, cyclonic spray scrubbers, impactor scrubbers, venturi scrubbers and augmented scrubbers. Within each of these major categories exist many further divisions. Pressure drops can vary widely depending upon the final control equipment choice, from as little as 2" w.c. for a spray column, to over 40" w.c. for a high energy venturi scrubber. It is not the purpose of this section to go into these alternatives in detail, only to point out that they exist, and that equipment choice can have a major impact on overall exhaust design.

2.0 MECHANICAL DESIGN

Once the process has determined the required duct sizes, pressure, vacuum, and temperature handling requirements, design of the duct itself can begin.

2.1 **INDUSTRY STANDARDS:** Various industry guidelines have been in circulation since as early as the 1960's. A common reference remains the Bureau of Commerce Voluntary Guidelines PS-15-69, issued in 1969.

Other standards have been issued since, including the SPI Quality Assurance Manual in the 1970's, ASTM D3299, ASTM D4097, ASME RTP-1 in the 1980's (with recent revisions), and the SMACNA Thermoset FRP Duct Construction Manual, issued for the first time in 1997. Some of these standards were specifically issued to address fiberglass tank construction (such as the ASME RTP-1), but are often referenced for design, manufacturing, or quality assurance criteria on duct systems as well.

As the most recent quidelines to be published, the SMACNA Manual is a good starting point for discussions of fiberglass duct mechanical design. As with the other codes, SMACNA expands upon more than changes, the criteria first put forward in PS-15-69. One change worth noting, however, is the reduction in safety factors for vacuum operation from 5:1 in PS-15-69, to 4:1 in SMACNA. This is a reasonable adjustment, based upon the additional 20 years experience since the issuance of PS-15-69. The factor of safety for internal pressure is 10:1 for both standards.

2.2 **CORROSION LINER:** In setting the mechanical design, one of the first issues to be addressed is the nature of the gas stream to be conveyed. The corrosion liner is the portion of the duct in contact with the fluid being conveyed. The liner must resist corrosive attack, as well as provide any required abrasion resistance or electrical conductivity for dissipation of static electricity (as required in many pharmaceutical plants). The corrosion liner is a resin rich layer, with either a (C-glass) fiberglass, (Nexus) polyester, (Halar) ECTFE, graphite or other material to impart the properties desired. The liner reinforcement should be selected based upon the corrosive / electrical properties desired.

In addition to the liner reinforcement, consideration should also be given to the overall corrosion liner thickness. Non-continuous chop strand mat generally provides a greater degree of corrosion protection compared with continuous glass filaments from woven roving or filament winding. A balance needs to be made between these corrosion benefits of additional chop strand mat and the strength benefits of continuous glass. Composites USA can assist with this evaluation.