MECHANICAL DRAFT FANS FOR THE MODERN INCINERATOR

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ABSTRACT

The performance characteristics of forced and induced draft fans are presented with special reference to incinerator applications. The formulas and guidelines will be especially helpful in plant design and modification. Special blading and fan construction for use under corrosive conditions are discussed.

INTRODUCTION

During the last five years we have witnessed a tremendous change in the design concepts of refuse incinerators. The engineers responsible for units recently completed, presently under construction, or still now in the planning and design stage, have usually armed themselves with the latest data available on almost every facet of incinerator design. Some engineers, blessed with a surplus of time, experienced personnel, or perhaps just a healthy concern for their “professional necks”, or possibly all three, have gone to great lengths to execute a plant that is in their customer’s best interest — a plant that performs at the designed capacity or better, with a highly efficient cost of operation. Of this they can be justly proud.

Some engineers, whether through lack of any or all of these assets, have not been as fortunate. I have long felt that too few engineers have a proper appreciation for the critical role that mechanical draft fans play in the efficiency of incineration — a role so absolute in its power that the miscalculation of system losses, or the misapplication of fan performance to these system requirements, has led quite a few men to regret lighting the first match. This feeling, and the foregoing phrases, “usually armed” and “almost every facet”, have supplied the impetus and inspiration to write this paper.

PERFORMANCE CALCULATIONS

It is not the intention of this paragraph, nor is it within the authority of this paper, to develop criteria or instruct the engineer in the use of any, in the determination of air required for combustion or cooling, but only to assist him in making these requirements presentable to a fan engineer in a form which is familiar to him, and is consistent with prevailing industry standards.

Since the fan industry corrects all fan performance to 70°F and 29.92 in. Hg, it does not suffice for the engineer to ask for “X” CFM, at “Y” inches static pressure at “Z” degrees F. He must, in order to make the requirements valid, give the pounds of air or gas per hour, and/or the density in pounds per cubic foot, and the elevation in feet above or below sea level. Unless this information is given, a fan engineer can only proceed, if he doesn’t mind wasting valuable time, by assuming the density of standard dry air at sea level. In the case of forced draft fans handling essentially ambient air, the amount of error in the as-
sumption, however slight, can still cause an increase in fan size and power consumption. This error further aggravates the situation when the fan must be selected at a standard direct drive motor speed. If the fan is to be installed inside the plant in an area with an average ambient temperature of 80 to 90°F, the density at these temperatures should be given. Most plants, especially 24-hour plants, never see less than 80 or 90°F even in the winter. If a fan is mounted indoors, but pulls air from outside the plant, consideration still should be given to temperature increase in the ductwork, due to duct length through high ambient conditions in the plant before the fan inlet.

If the fan is selected on the basis of 70°F air at the inlet, but never actually sees less than 80 to 90°F at the inlet, the fan will not be able to develop the required static pressure at the specified volume, but will actually fall short by 2 percent to 3.8 percent. If the fan has the capability to develop the higher static pressure, it will move to the left on the fan pressure curve and deliver less than the specified volume. Either case compromises the job in the design stage!

It is therefore desirable to state the temperature and density at the fan inlet that is more apt to prevail, and also to keep ductwork on the inlet side to a minimum since a fan is more sensitive to changes on this side. Care should be taken to avoid sharp bends at the inlet and discharge. The fan rotation should complement duct direction on the discharge side. Ductwork at right angles to the inlet should be at least 1½ wheel diameters wide. If this is not possible, a fan engineer can recommend an inlet box or other solution.

The same statements made above on performance data also hold true for induced draft fans, but with a vengeance! Let us take this example:

150,000 CFM @ 5.0 in. S.P. @ 600°F

If we assume the density of standard dry air at 600°F of 0.0375 lbs per cubic foot, we would arrive at a corrected pressure ($P_c$) at 70°F, 0.0750 density of 10.0 in. static pressure.

$$P_c = 5.0 \frac{0.0750}{0.0375} = 10.0 \text{ in.}$$

Our air horsepower requirement would be:

$$AHP = \frac{150,000 \times 10.0}{6356} = 236.0 \text{ @ 70°F}$$

and 118.0 AHP at 0.0375 density. If we had selected a fan for 10.0 in. S.P. at 70°F, we would be trapped by the assumption of dry air. There is probably not an incinerator operating that does not have partially saturated gases going up the stack or through a fan, whether caused by cooling water sprays, or moisture in the refuse. If we assume a partially saturated gas at 600°F, having a density of 0.032, which is not unreasonable, our corrected pressure would now be

$$P_c = 5.0 \frac{0.0750}{0.320} = 11.72 \text{ in.}$$

The fan that we selected for 10.0 in static pressure at 70°F would fall 17.2 percent short of the pressure required at 150,000 CFM. Our air horsepower requirement would now be

$$AHP = \frac{150,000 \times 11.72}{6356} = 276.5$$

We have not been penalized by the higher corrected pressure because our air horsepower at 5.0 in. S.P. @ 600°F still is 118.0.

$$AHP = 276.5 \frac{0.032}{0.075} = 118.0$$

This difference in corrected pressure represents a difference of 5 in. in wheel diameter for a radial tip fan running at 720 RPM, a difference that makes the smaller fan selected for 10.0 in. static pressure at 70°F woefully inadequate. These examples show the importance of including accurate densities with fan requirements.

Another consideration in performance calculations should be in the addition of safety factors to the net requirements for volume and pressure. The engineer must decide whether 5 percent, 10 percent, 15 percent, etc. shall be added to his actual net requirement for volume. When he makes his choice, and remember his system has not changed, he must increase his net static pressure requirements by the square of this percentage increase, or 10 percent, 21 percent, 32 percent, etc. No safety factor should be added to the temperature since this will be kept fairly constant by air or water cooling during normal operation. Since the fan will be designed structurally for a continuous operating temperature in excess of the actual operating temperature, no useful purpose can be served by adding to it. These additions to the "net" requirements give us new values called "test block" requirements. These are the requirements that a fan must be guaranteed to deliver at its full rated speed with inlet dampers or inlet
vanes, or outlet damper in the wide-open position. The fan will operate at the net requirement by a reduction in speed, or by partially closing the inlet dampers or vanes, or the outlet damper.

When losses in ductwork are to be calculated, the engineer should select a maximum duct velocity, and then using the volume at the net condition, size his ducts and branches accordingly. Dampers are figured in their wide-open position with an allowance of one velocity pressure exit loss through the reduced area of the damper. When the total system loss is determined, this becomes the net static pressure. The test block percentages are then added. The amount of area lost in a damper due to blades is a function of the shaft span, number of blades, and the total pressure through the damper. Long shaft spans and high total pressures increase shaft diameters, which cut damper area.

APPLICATIONS OF FAN TYPES

Forced Draft Fans

Most fans for this type of service have either the backward inclined, flat blade wheel, or the backward inclined airfoil bladed wheel. Most have ten to twelve blades, and have a non-overloading characteristic in their horsepower curves.

The ratio of wheel diameter to outlet area is quite generous, and large volumes can be handled at fairly low outlet velocities with little penalty on fan size. Static efficiencies vary from 82 percent on the flat blade wheels to 89 percent on the airfoil wheels. The ideal forced draft fan set-up is either single or double inlet, depending on space available and volume requirements, direct driven at a standard motor speed of 1775 RPM, 1185 RPM, or 880 RPM. Variable inlet vanes should be used for volume control, and since very few plants can offer completely dust-free air even from outside the plant, vanes with all bearings and operating mechanisms outside the airstream should be used. This is especially essential if fans are pulling air from in-plant locations such as the ash pits or tunnels, storage pit, or even the operating floor. All vanes should be servo-operated, with automatic or manual control at the discretion of the engineer. Most systems have overfire air automatically controlled by temperature, and underfire air manually controlled by the operator.

Induced Draft Fans

The radial tip fan is by far the best choice for this service when the gas to be handled is not clean. Even if a mechanical collector is used, quantities of fly ash will still get to the fan.

Most fan manufacturers have different philosophies about wheel design, and the blade totals vary from 10 to 28, with the most efficient fans having 10 to 12. A great number of blades allows the wheel to be made smaller in diameter, and with a low ratio of diameter to housing growth, making it possible to develop a higher pressure for a given RPM than a wheel having fewer blades. The penalty here is the high velocities through the fan which accelerate the wear rates of fan parts and breeching.

Evase outlets are required for quieter operation and to drop the velocity in order to regain pressure. Some fans depend on Evase outlets to reach even medium efficiencies of 70 percent to 75 percent. The wheels with few blades depend on greater diameter or a faster speed to develop pressure. The ratio of wheel diameter to housing growth is very high, and the velocities through the fan are low. This drastically lowers the wear rates of fan parts and breeching, lowers the air velocity noise to an acceptable level, and allows the fan to develop a static efficiency of 80 percent to 83 percent because of low internal losses. It also obviates the use of an Evase outlet.

Some people will argue that since the wheel with few blades must run at a higher tip speed, the wear rates will be higher. If we consider only impact velocity, this is true. But blades and wear plates can be made of materials such as "cor-ten" steel or "T-1" steel which have 150 to 340 Brinnel ratings, and can take high impact forces with relative impunity. The real problem is the escape velocity of the particle off the blade. This is what does the damage by cutting and grooving the blades and center plates. A fan with small inlet and outlet areas, having high velocities through the fan keeps these particles going at a great rate, but the fan with few blades and greater housing areas can quite often cut the air and particle velocity by one-half.

Depending on the volume and pressure required, an induced draft fan can be either single inlet or double inlet, with inlet boxes. A single inlet fan, although larger in wheel diameter, will oft-times be less expensive than a double width fan for the same service. It will be a lot less expensive to install and align, and will usually weigh less. This can be appreciated when upper story, platform, or roof locations are used. A double inlet fan on independent bearing pedestals probably would require an inertia-mass base under some of these conditions, but a single inlet fan might not.

If one were required, the block for a single inlet fan is less complicated and less costly. Volume control on a single inlet fan can be accomplished with a variable inlet
vane, designed for high temperature and "dirty" service with all bearings and operating mechanisms out of the airstream. Inlet dampers can be used on the boxes of double inlet fans. A more costly method is the use of variable speed drives. Few incinerators operate at the reduced loads long enough to justify the initial cost and the maintenance cost of these drives.

In some new plants, where electro-static precipitators are to be used, the obvious choice would be an airfoil bladed fan. The benefits are high efficiency, quiet operation, higher motor speeds. The use of forward curved squirrel-cage wheels with 48 to 60 blades, is losing favor because of their lower efficiency, higher velocities, and Eave requirements. Forward-curved wheels do not have a non-overloading horsepower characteristic as do airfoil wheels and a few radial tip wheels.

If a wet scrubber is to be used, we must consider a few things before we get to fan types. First, the gases leaving a scrubber are usually more injurious to fans than the fly ash that is being removed. Concentrations of dilute acids are always present. Eventually these acids will pit the wheel hub and backplate, and fatigue cracking will occur. Second, most scrubber systems require a higher pressure to operate. Although exit temperatures are usually in the 160 to 200 F range, corrected pressures can get fairly high. This requires a fan to run at a higher tip speed to develop the pressure. These are the problems, and here are some solutions:

After a determination has been made on the "test block" conditions, the scrubber manufacturer should be able to determine the quantities, and concentrations, of any acids that will not be cleaned out. This must often be an educated guess because of the variables in refuse as received at the plant. Now the fan manufacturer must check what materials or coatings he can use in his fan. Wheels coated with rubber or other materials such as neoprene or polyvinal chloride sheets tolerate a tip speed of only 13,000 to 14,000 F.P.M. If the pressure is high, we might have to forget this. Straight radial blade wheels might be able to develop the pressure within the tip speed limits, but the fans are usually much larger, have high air velocities, and are not too efficient. There are many types of stainless steels that will withstand most scrubber by-products and some that withstand just about anything, but their mill cost and fabrication costs usually make for very expensive fans. Some phenol-based, thermostetting or catalytic coatings similar to baked "heresite", or "plasite", when applied over mild steel after it has been sand-blasted, usually afford good service while the coating lasts. Radial tip, backward inclined flat blade, and airfoil wheels can be good choices, depending on the pressure required and the material costs. If a single fan is too costly, a less expensive option of two rubber-coated fans in series may do the trick.

**CONSTRUCTION DETAILS**

**Forced Draft Fans**

These fans, whether single or double inlet, should be minimum 3/16 in all welded construction with split housings for wheel removal, access doors and drains in fan scroll, flanged inlets and outlets. If a fan has an unducted inlet, it should have a protective screen even if it has an inlet vane. Double inlet fans should be arrangement #3 for either direct or belt drive. Independent bearing pedestals with sole plates should be used. Single inlet fans should be either arrangement #1, #3, #7, #8 or #9 for belt or direct drive. Belt drive single inlet fans should be arrangement #9 where possible, because this saves floor space and mounting costs. All belt driven fans should have heavy gauge belt guards; direct driven fans should have coupling guards. Bearings should be self-aligning ball or spherical roller types with oil or grease lubrication. Sight glasses for lubricant inspection should be used. If the fans are subject to atmospheric corrosion, it is advisable to make them of "cor-ten" steel or equal.

**Induced Draft Fans**

These fans have usually been double inlet fans with inlet boxes, and independent bearing pedestals. They should be of all-welded construction with a minimum plate gauge of 3/16 in. Fans down-stream of wet sprays or mechanical collectors should be made of "cor-ten," or similar steels which have a high resistance to atmospheric corrosion. Scroll liners should be used in the housings when no wet sprays or collectors are used, and should also be seriously considered even when they are. If used, they should be 3/16 in. minimum gauge "cor-ten" or equal, bolted in place with counter-sunk heads. Wheels should be of "cor-ten" or equal, with the center plate notched between the blades to reduce the WR² of the wheel, and to omit an area sensitive to wear. Blade wear-plates of "cor-ten" or "T-1" steel, extending across the notched center plate, should always be used, even if a collector is used. This does not apply to electro-static precipitator systems. Wear plates should be bolted on with counter-
sunk head bolts, and should be welded to the blade along their leading edge to keep out any fly ash.

Inlet boxes should be tapered to cut down bearing spans. There should be an access door in the fan housing, and one in each inlet box. The housing will almost always have many splits to facilitate shipping and assembly, so wheel removal should be no problem. All splits should have asbestos gaskets. Valved or piped drains should be provided in the housing and each inlet box. Bearings can be air or water cooled, self-aligning, ring oiled, sleeve type; or they can be ball or spherical roller bearings that depend on shaft coolers and heat shields to keep cool.

If fans are to be installed outdoors, and water cooled bearings are preferred or must be used, provisions for running with an ethylene glycol anti-freeze system should be investigated, or bearings will have to be drained every day if the plant is not in continuous operation. If the fans are to be insulated, and most are, a simple and inexpensive installation is aided by 6 in. lengths of 9 gauge wire, tack welded to housing and boxes on 12 to 18 in. centers. Parallel blade inlet dampers with 3/16 in. bent channel frames, and 10 gauge minimum blades are most often used for volume control. They are interconnected with a common shaft and are servo-operated from this shaft. The bearings used on the damper shafts should be high temperature graphite type good for 700 F.

If a single inlet fan is used, it should be arrangement #3 or #8 for direct drive. Arrangement #3 requires independent pedestals on both sides of the fan, and dictates the use of an inlet box to keep the bearing out of the hot inlet. If arrangement #8 is used, usually no box will be required, and excellent control can be had by using a high temperature inlet vane.

**INSTALLATION AND MAINTENANCE**

Whenever a fan manufacturer receives a complaint that a fan is vibrating, the odds are about 99 percent in his favor that the fan was not set properly. Fans should be set straight and level, and all anchor bolts should be tightened evenly. A good nonshrink grout should be used between the fan base and foundation slab.

Usually, the only maintenance a fan requires is at the bearings, or grease fittings on dampers and vanes. Great care should be taken to follow the manufacturers recommendations on what lubricants are acceptable. An induced draft fan will require inspections at shorter intervals than forced draft fans due to the nature of their service.

**CONCLUSION**

While I have tried to cover the questions that most often are voiced, I am sure there are areas where further discussion could go on and on.

It is my great hope that this paper will afford some answers to the engineers who have had trouble specifying fans, and the engineers who have lost a little of their sanity trying to make fans and systems work. I hope that this paper can be of special help to the unfortunate engineers who have to stand by and watch fans and breechings disintegrate because of inadequate materials and designs.

Please remember that fans are not magic wands! If a fans and system are mismatched, the best fan in the world will fail! Also remember that you get exactly what your specification calls for. Be explicit! Leave no loose ends! You and your customer deserve the best!