THE DESIGN OF AN AIR FILTRATION SYSTEM TO CLEAN HIGH TEMPERATURE/HIGH HUMIDITY RADIOACTIVE AIR STREAMS

Ty H. Proffitt, P.E. and James P. Burket, P.E.
ENSERCH Environmental Corporation

ABSTRACT

During normal operating processes or waste remediation efforts high efficiency (HEPA) filtration systems are used to remove particulate contamination from air streams. These HEPA filtration systems can accommodate a range of air humidities and temperatures and still retain their effectiveness. However, when the combination of high humidity and high temperature are present the effect of these highly saturated air streams can be detrimental to a HEPA filtration system. Couple this highly saturated air stream with the effect of radioactivity and a case for a "specialized" HEPA filter system can be made. However, using fundamental laws of heat transfer it is possible to design a HEPA filter system that can operate in a high temperature/high humidity radioactive environment.

INTRODUCTION

During many remediation or facility operations, a radioactive air stream high in humidity and temperature is created. This air stream must undergo filtration prior to being released to the atmosphere. HEPA filters and charcoal adsorbers are the standard system components which filter air borne radiation particles and gases. However, humidity reduces the efficiency of these components. Therefore, a high humidity air stream must undergo a process to reduce the humidity prior to filtration. If this operation is to occur upstream of the filtration, a radioactive waste stream will be generated. In order to circumvent the creation of a waste stream, a filtration system that can accommodate high temperature/high humidity operations is required. This paper will illustrate the design considerations required for such a system.

PURPOSE

The format used to illustrate the design of a filtration system for high temperature/high humidity radioactive environments will be as follows:

A section will be devoted to explaining the methodology behind the particular component in question and what its relevance is to the overall design. A subsection will detail any design considerations that should precede the determination of equipment size, materials of construction, performance characteristics, etc... A subsection will detail a typical calculation complete with equations and references that will allow the reader to perform a similar calculation by insertion of his air stream characteristics.

CRITERIA

The typical criteria used in the sample calculation is as follows:

- Process Air flow: 200 ACFM ± 0
- Air Stream Temperature: 120°F ± 0
- Air Stream Humidity: Saturated % ± 0 %
- Water Droplet Concentration: 5 gal/min (Max)
- Decay Heat Generation for Charcoal Adsorbers: 10 Watts (Max)
- Filter Housing Dimensions: Standard 1 x 1 Unit, 20 feet long

APPLICABLE CODES AND STANDARDS

- ASHRAE Handbook of Fundamentals - 1989 Edition
- ERDA 76-21 - Nuclear Air Cleaning Handbook - 1976 Edition
- ASME/ANSI N509 and N510 - Nuclear Power Plant Air-Cleaning Units and Components -1989 Edition

DEMISTER SIZING METHODOLOGY

Demisters are used in a filtration system to remove water droplets entrained in the air stream. Demisters contain drains to transfer this liquid to a collection tank.

The air stream demister sizing is based on the flow rate in scfm, water content, and maximum/minimum water droplet sizes in the air stream.

Design Considerations

For the purpose of simplicity, the mass flow rate of the air stream should be calculated at the inlet to the demister. This mass flow rate will be used throughout the calculation and will give conservative results.

The drain pipe from the demister must be equipped with a barometric seal if it operates in a vacuum environment. The addition of a barometric seal will prevent the removed droplets from being reintrained into the air stream. However, if the humidity of the air stream is low enough to evaporate the seal, a level instrument should be attached to the seal to indicate if the water level evaporates below a specified limit. This instrument will inform the operator to add water to the seal and prevent the fan from pulling the seal into the air stream.

System operating pressures must be determined at the demister. This information is required to calculate mass flow rates and air stream enthalpies.

Removal of all entrained water droplets (down to a prescribed size) will be accomplished before the water reaches the HEPA filters. Normally, a droplet size of 10 microns is acceptable. However, if the water droplets are larger, the heater size calculation should make some determination regarding the heat required to heat not only the air but also the water droplets.

Sample Calculation

No calculation is required for sizing a demister. Vendor catalog information will guide the designer to an acceptable choice. For our example criteria:

The worst case scenario for the air stream demister will occur when the flow rate is at 200 ACFM, and the air has the maximum amount of entrained water 5 gal/min. with a droplet maximum size of 10 microns. These parameters provide the information required by a vendor to supply a demister for this application.

DUCT HEATER SIZING METHODOLOGY

Any air stream can have the humidity level reduced by inducing heat into the process stream and increasing the temperature. This reduction in humidity levels allows the use of filtration and carbon adsorption.

The air stream in-line duct heater sizing is based on the process airflow and relative humidity. Heat transfer calculations are performed in accordance with ASHRAE Fundamentals 1989.

Design Considerations

In most circumstances, the duct heater and HEPA filter housing will be located upstream of the fan on the suction side. In order to account for the low absolute pressure and model the density correctly, a high altitude psychrometric chart can be used.

If the process air flow of the system varies, the maximum air flow must be used in the determination of a heater size.

In order to maintain a low relative humidity at the entrance of the charcoal adsorbers/filter bank and at the entrance to the exhaust fan, an acceptable temperature must be determined for each case.

Flow can be considered incompressible. This information is useful in the determination of fluid density.

If the air stream contains particulate matter or salts that may collect on heater coils, indirect heating methods may be used. Indirect heating methods are heating methods that do not contact the fluid being heated. For the purposes of this calculation, direct heating methods were used.

Duct Heater Calculation

The worst case possible for heating the air stream occurs at the maximum flow rate (200 ACFM).

The density of the air passing through the duct heater is given by this equation (utilizing the high altitude psychrometric chart):

The psychrometric chart for 120° *saturated air*

$$@\ 5000\,ft\ gives \rightarrow \rho = 0.0491\,\frac{lbm}{ft^3} \tag{1}$$

The mass flow rate of the entering air is given by the following equation:

$$Mass\ flow,\ \dot{m} = Q \times \rho \approx 10\,\frac{lbm}{\min} \tag{2}$$

where:

Q = the actual flow rate of air = 200 ft^3/min

ρ = the density of air at 5000 ft = 0.0491 lbm/ft^3

We will assume that the exit temperature of the air in the duct must be greater than 133°F in order to ensure a low relative humidity at the filter bank entrance. To minimize the size of insulation in the ducts between the heater and filter bank, a temperature of 195°F is assumed for the exit temperature of the heater. The heater exit temperature may be revisited based on the duct insulation thickness commensurate with an exit temperature of 195°F.

In order to raise the air temperature to 195°F, the inlet and exit enthalpies of the air mixture must be determined. From the 5000 ft psychrometric chart:

h_i = enthalpy of moist air at 120°F saturated = 140.67 Btu/lbm_{da}

h_o = enthalpy of moist air at 195°F, @ 120°F Dew Point = 162.2 Btu/lbm_{da}

The total heat added to the air stream can be determined from the following relation:

$$Heat\ Added = \dot{m} \times \frac{60\ \min}{hr}(h_o - h_i) = 12{,}918\,\frac{Btu}{hr} \tag{3}$$

The heat that must be added to the stream is 12,918 BTUH.

DUCT INSULATION THICKNESS METHODOLOGY

Duct insulation is required in the duct work between the duct heater and filter housing to reduce the heat loss from the process stream enroute to the filter housing.

The duct insulation thickness calculation is performed using fundamental methods of conductive heat transfer.

Design Considerations

If the process air flow of the system varies, the minimum air flow must be used in the determination of a duct insulation thickness.

Temperature drop across the metal duct wall can be considered negligible.

For simplicity, the average temperature for air entering and exiting the duct will be used as the air temperature to determine the heat loss across the duct wall. Iterative heat transfer calculations may be used if a more precise number is required.

Duct Insulation Thickness Calculation

The worst case scenario for temperature change through the duct will occur at the lowest air flow possible - 200 ACFM.

$$Mass\ Flow\ at\ 200\,ACFM = \dot{Q} \times \rho \times 60\,\frac{\min}{hr} =$$

$$200\,\frac{ft^3}{\min} \times 0.0491\,\frac{lb_m}{ft^3} \times 60\,\frac{\min}{hr} = 589\,\frac{lbm}{hr} \tag{4}$$

where:

Q = air flow at the inlet to the demister

ρ = Density of air at the inlet to the demister

To simplify this equation, it is assumed that the duct from the heater to the filter inlet is in a constant temperature environment and will only undergo conductive heat transfer. Similar to underground duct.

Assuming an inlet temperature of 195°F and an exit temperature of 170°F, the following relation can be derived to determine the allowable heat loss from the duct.

$$Energy\ \varepsilon = Energy\ Out \rightarrow \dot{m} \times (h_i - h_o) =$$

$$q_{out} = 4236\,\frac{Btu}{hr} \tag{5}$$

where:

m = mass flow rate of air @ 200 ACFM = 589.0 lb_m/hr

h_i = inlet enthalpy at 195°F = 162.2 Btu/lb_m

h_o = outlet enthalpy at 170°F = 155.01 Btu/lb_m

q = heat flow from the duct = Btu/hr

Equating this heat loss to the conduction heat transfer equation, the overall heat transfer coefficient for the duct wall can be determined:

$$q_{out} = 4236\,\frac{Btu}{hr} = \frac{2 \times \pi \times L_{duct} \times (T_{air} - T_{environ})}{\frac{\ln \frac{R_{ins}}{R_{duct}}}{k_{ins}}}$$

$$\therefore R_{ins} \approx 3.29\ inches$$

where:

R_{ins} = radius of duct plus insulation required

R_{duct} = radius of duct = 3"

L_{duct} = length of duct = 20 ft

T_{air} = duct air stream average temperature = 183°F

$T_{environ}$ = outside environment temperature = 55°F

k_{ins} = thermal conductivity of typical duct insulation = 0.025 Btu/hr-ft-°F

3.29" - 3.0" = 0.29"

Therefore, use insulation with a thickness greater than 0.29"

CHARCOAL ADSORBER SIZING METHODOLOGY

Charcoal adsorbers are used in a filtration system to filter or adsorb radioactive gases and volatile organics using activated carbon.

Charcoal adsorbing filter design is based on an Iodine concentration, flow rate and residence time requested in the scope of work. The amount of charcoal needed is derived from a combination of the residence time requirement, the Iodine present in the air stream and the decay heat generated by captured Iodine during upset conditions.

Design Considerations

The ignition point and combustibility of charcoal should be a principal concern in the design of high temperature filter systems that contain charcoal adsorbing. Most charcoal burns at approximately 630°F. However, charcoal will release (desorb) previously *trapped material* at a much lower temperature. The ERDA 76-21 Nuclear Cleaning Handbook should be consulted for temperature and humidity concerns surrounding charcoal adsorbers.

The material present in the air stream should be assumed to evenly disperse on the face of the charcoal adsorber. However, this point should be discussed with the operating contractor to verify this assumption. Even dispersion on the face of the charcoal adsorber will prevent "hot spots" from occurring on the charcoal adsorber.

Decay heat in the charcoal could cause the charcoal to ignite. The air flow must be sufficient to prevent ignition of the charcoal in the filter system and provide a proper residence time.

Charcoal Adsorber Calculation

To calculate the required depth of the charcoal bed, a combination of the residence time and flow rate criteria must be used. Assuming a residence time of 0.25 sec and a maximum flow rate of 200 ACFM, the following equation will provide the required bed depth:

$$Bed\ Depth, t = \frac{Q \times 12 \times RT}{60 \times A} = 0.625\ inches \qquad (7)$$

where:

Q = actual flow rate of air = 200 ft^3/min

RT = residence time required = 0.25s

A = frontal area of adsorbing filter = 16 ft^2

To maintain adequate cooling in the charcoal adsorbers during normal operation or upset conditions, sufficient airflow must be present. This airflow can be calculated from an energy balance on the charcoal filter. The worst case for possible charcoal ignition will occur at the highest possible heat generation (upset conditions) and the lowest air flow (200 ACFM). We will use the exit air temperature from the duct between the heater and the filter bank as an inlet temperature to the charcoal adsorber. Assuming the air temperature cannot exceed the ignition temperature (630°F) of the charcoal, the mass flow rate of air required for cooling can be found from:

$$Energy\ In = Energy\ Out \rightarrow \dot{m} \times (h_o - h_i) =$$

$$q_{conc} \therefore \dot{m} = \frac{q_{conc}}{h_o - h_i} = 0.0042\,\frac{lb_m}{min} \qquad (8)$$

where:

h_i = enthalpy of moist air at 170°F = 155.01 Btu/lb_m

h_o = enthalpy of moist air at 630°F = 289.26 Btu/lb_m

q_{conc} = heat concentrated in the charcoal(10 w) = 0.567 Btu/min

The required flow rate can be found by dividing the mass flow rate by the density at 170°F.

$$Minimum\ Cooling\ Flow = \frac{\dot{m}}{\rho} = 0.094\,\frac{ft^3}{min} \qquad (9)$$

where:

ρ = density of moist air at 170°F = 0.0451 lb_m/ft^3

Therefore, use a 16 ft^2 filter with a bed depth of 0.625" and maintain a flow of 0.094 ft^3/min through the filter.

THE HEPA FILTER SIZING METHODOLOGY

HEPA filters are used to remove radioactive particulate from a process air stream.

The HEPA filter flow rate is based on the ventilation requirement given in the statement of work. HEPA filter housing dimensions are based on standard 24" x 24" x 11-1/2" HEPA filters.

Design Considerations

Standard HEPA filters can operate up to a temperature of approximately 220°F. If possible, this should be considered as a maximum air stream temperature to eliminate the stocking of special high temperature components. The ERDA 76-21 Nuclear Air Cleaning Manual gives further direction on the problems associated with high temperature HEPA filtration.

HEPA filters will experience a higher pressure drop when they become wet due to a dust and particulate which combine with the damp surface to create a "mud" and blind the filter. For this reason, pressure measurement across each filter or the use of an efficient pre-filter is suggested.

HEPA Filter Sizing Calculation

Standard HEPA filters are rated at 1000 ACFM with 1" wg pressure drop when tested in a clean condition. Our maximum system flow is 200 ACFM; therefore, a standard 24" x 24" x 11.5" HEPA filter will suffice.

FILTER AREA HEATING SYSTEM METHODOLOGY

Heating is required to maintain the filter area to a temperature which prevents condensation and is derived by fundamental heat transfer methods of free and forced convection.

Design Considerations

Condensation will occur if the temperature of any surface contacting the air stream is below 120°F; therefore, all surfaces that contact the air stream must be maintained above 120°F.

The area where the filters are to be located (room, building, etc..) must have a reliable heating system that can maintain a temperature range within ±5°F of the set point in most cases. For this reason, it is advisable to use a dedicated room or building to act as the filter area to monitor and control the elevated space temperature.

Filter Area Heating System Calculation

To prevent condensation on the inside surface of the filter bank, the filter housing wall temperature must remain above 125°F. The corresponding space temperature surrounding the filter housing can be determined by a heat flow analysis through the wall of the filter housing. An initial filter area space temperature of 100°F is chosen to calculate air properties. The temperature drop across the duct wall is considered negligible due to the high thermal conductivity of the duct material.

A heat flow circuit representing the temperatures at three locations is given by this relation:

$$T_{ai} \xrightarrow[H_i]{} T_s \xrightarrow[H_o]{} T_{ao} \tag{10}$$

where:

T_{ao} = temperature of the outside air - °F

T_s = temperature of the duct wall - °F

T_{ai} = temperature of the air inside the duct - °F

H_i = inside heat transfer coefficient - Btu/hr-ft^2-°F

H_o = outside heat transfer coefficient - Btu/hr-ft^2-°F

The free convective heat transfer coefficient must be calculated by assuming an average temperature between the duct wall and the space. The final space temperature corresponding to a duct wall temperature of 125°F can be solved by an iterative procedure. Assuming a filter area space temperature of 100°F, an average outside film temperature of 112°F is used for finding air properties. The Grashof number is used to calculate the free convective heat transfer coefficient:

Grashof Number (N_{Gr}) =

$$\frac{(L^3 \times \rho^2 \times \beta \times g \times (T_\infty - T_{os})}{\mu^2} = 38{,}962{,}041 \tag{11}$$

where:

L = characteristic surface dimension of the duct - 1 ft

T_∞ = room air temperature in the filter building - 100°F

T_{os} = duct surface temperatures - 125°F

μ = absolute viscosity of air at T_{avg} - 1.3044×10^{-5} lb$_m$/ft-sec

β = Inverse of T_{avg} - 0.0017/°R

ρ = Density of air at T_{avg} = 0.0696 lb$_m$/ft^3

g = gravitational constant - 32.2 ft/s^2

To determine which form of the Nusselt equation to use the Grashof number is multiplied by the Prandtl number:

$$N_{Gr} \times N_{Pr} = 28{,}052{,}669 \tag{12}$$

where:

N_{Pr} = Prandtl Number at T_{avg} - 0.72

Laminar flow natural convection relations apply.

Assuming the greatest free convection takes place on the top surface of the duct, the corresponding Nusselt Number and convective heat transfer coefficient can be found:

where:

ΔT = Difference between the duct wall temperature and the

$$H_o = 0.27 \times \left(\frac{\Delta T}{L}\right)^{0.25} = 0.6037 \frac{Btu}{hr - ft^2 - {}^\circ F} \tag{13}$$

air temperature - 25°F

L = Characteristic length of duct side - 1 ft

To begin the internal heat transfer iteration, the air flow is assumed to be 200 ACFM, and the inside average temperature is assumed to be 165°F. If the duct wall temperature is maintained at 125°F, the inside convective heat transfer coefficient must be determined to find the total heat transferred to the room. A film temperature of 135°F is used to determine property values. To find the inside convective heat transfer coefficient, the Reynolds number must be calculated:

$$\textit{Reynolds No.}\ (N_{Re}) = \frac{4 \times \dot{m}}{\pi \times D_h \times \mu} = 7906 \tag{14}$$

where:

$\dot{m}$ = mass flow rate of moist air = 0.16 lb$_m$/sec

D_h = hydraulic diameter of 1X1 HEPA Housing = 2 ft

μ = abs viscosity of air @ T_{Film} = 1.3424×10^{-5} lb$_m$/ft-sec

π = 3.142

From the calculation of the Reynolds number the flow is found to be transitional, the Reynolds number is then used to calculate the Nusselt number and ultimately the convective heat transfer coefficient.

$$\text{Nusselt No. } (N_{Nu}) = 0.023\,(N_{Re})^{0.8} \times (N_{Pr})^{0.3} = 27.37 \qquad (15)$$

where:

N_{Re} = Reynolds Number at T_{film}

N_{Pr} = Prandtl Number at $T_{film} = 0.72$

The Nusselt Number is used in the calculation of the inside heat transfer convective coefficient:

$$H_i = \frac{N_{Nu} \times k}{D_h} = 0.2203\,\frac{Btu}{hr - ft^2 - {}^\circ F}$$

where:

k = thermal conductivity of air @ T_{film} = 0.0161 Btu/hr-ft-°F

D_h = hydraulic diameter of the duct = 2 ft

Referring to the original heat transfer circuit equation (11), the heat flow through one segment is equal to the heat flow rate through the entire circuit. The heat flow rate from the inside air to the inside surface is equal to the heat flow rate through the system. This heat flow rate is given by the following equation:

$$q_{duct} = H_i \times (T_{ia} - T_s) = 8.81\,\frac{Btu}{hr - ft^2}$$

where:

H_i = inside convective coefficient - 0.2203 Btu/hr-ft^2-°F

T_{ia} = inside average temperature of the air - 165°F

T_s = surface temperature of the duct - 125°F

The heat transfer rate through any section of the heat flow circuit (outside air, duct wall, inside air) must equal the heat transfer rate through the total circuit. Therefore, the heat flow rate calculated in eqn (17) is compared to the outside air convective coefficient. The resulting new space temperature is then used in equations (11 - 13) to derive a new outside convective heat transfer coefficient. After several iterations, the final heat flow rate and surface temperatures are as follows:

Heat flow rate: $Q \approx 9$ Btu/hr-ft^2

Filter Area Space Temperature: $T_{oa} \approx 110$°F

HEAT LOSS THROUGH FILTER BANK METHODOLOGY

Heat loss occurring while the air is moving through the HEPA filter bank must not cause the air to exceed a relative humidity requirement. We assume 70% to be the maximum allowable relative humidity. An energy balance on the inlet and exit air temperatures will verify that the heat lost will not increase the relative humidity beyond a specified limit.

Design Considerations

In order to use the correct natural convection relations, an assumption must be made regarding the surface that will experience the most heat loss. The most natural convection will be assumed to occur through the top of the filter housing.

Heat Loss Through the Filter Bank Calculation

In order to assure that the temperature of the air does not drop to the point where the relative humidity of the air exceeds 70%, an energy balance must be performed on the filter bank.

Energy Loss Through Filter Train =

$$\dot{m} \times (h_o - h_i) = 6336\,\frac{Btu}{hr} \qquad (18)$$

where:

$\dot{m}$ = mass flow rate of moist air at 200 ACFM = 600 lb$_m$/hr

h_o = enthalpy of air at the inlet to the filter bank (170°F and 120°F Dew Point) = 155.01 Btu/lb$_m$

h_i = enthalpy of air at 70% Relative Humidity and 120°F Dew Point) = 144.45 Btu/lb$_m$

If we assume the filter bank is 20 feet long, with a perimeter length of 8 feet, the heat rate in Btu/hr-ft^2 is found as follows.

$$\text{Heat Rate} \rightarrow \frac{Btu}{hr - ft^2} = \frac{6336\frac{Btu}{hr}}{20\,ft \times 8\,ft} = 39.6\,\frac{Btu}{hr - ft^2} \qquad (19)$$

This represents the maximum allowable heat transfer rate through the duct walls that will ensure a relative humidity less than 70% in the exit air stream. If we compare this heat rate with the actual heat rate calculated in section 4.3, we find that the heat rate calculated section 4.3 is less than the allowable rate and therefore the air stream will not exceed the 70% relative humidity requirement.

HEAT LOSS THROUGH THE DUCT FROM THE FILTER BUILDING TO THE EXHAUST FANS METHODOLOGY

The temperature of all surfaces contacting the air stream must be maintained above 120°F to prevent condensation from occurring. The combination of outlet temperature from the filter building and duct insulation thickness must be sufficient to prevent condensation from occurring in the duct between the filter building and the exhaust fans.

Design Considerations

The duct between the filter housing area and the fans is the most likely area for condensation to occur due to the heat loss experienced in the filter area. For this reason, it is advisable to attach a drain port to a low point in the duct that can be used to remove clean condensate if necessary.

Duct Heat Loss Calculation

The heat loss from the filter area to the exhaust fans is performed in the same manner as the heat loss in the duct upstream of the filter area. The combination of air exit temperature from the filter building and duct insulation thickness must be sufficient to prevent any condensation from occurring in the duct between the filter area and the exhaust fans. Using the filter building space temperature found in equation 17, the air stream temperatures exiting the filter building can be found.

From information given in equation 17 the total heat transfer rate can be found.

$$Q_{lost} = 9.0 \frac{Btu}{hr - ft^2} \times 20 \times 8 = 1440 \frac{Btu}{hr} \quad (20)$$

Performing an energy balance on the filter system, with the energy exiting the filter bank being equal to the heat lost in the bank, the exit temperature can be found from the following relation:

$$Q_{lost} = \dot{m} \times (h_i - h_o) \therefore h_o =$$

$$h_i - \frac{Q_{lost}}{\dot{m}} = 152.61 \frac{Btu}{lb_m} \quad (21)$$

where:

m = mass flow rate of moist air = 600 lb_m/hr

h_i = enthalpy of inlet air @ 170°F = 155.01 Btu/lb_m

h_o = enthalpy of outlet air = Btu/lb_m

From the psychrometric chart, for air at 120°F dew point, the corresponding temperature for an enthalpy of 152.61 Btu/lb_m is 159.53°F.

Using the same procedure as in the previous section with an inlet temperature of 160°F and an outlet temperature of 125°F, the insulation thickness for the duct from the filter area to the exhaust fans can be determined.

CONCLUSION

If careful consideration is given to temperature and insulation values, and adequate knowledge of process flow conditions exists, a system can be designed to operate and filter a high temperature/high humidity radioactive air stream.