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FLUE GAS WASTE ENERGY RECOVERY
OPTIMIZATION WITH SPRAY RECUPERATORS

by

DONG HO LEE, M. ENG.

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of
the requirements for the degree of
Master of Engineering

Department of Mechanical
and Aeronautical Engineering
Carleton University
Ottawa, Ontario

November, 1987

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ABSTRACT

The design methods of spray and packed columns have been investigated in this thesis for the condensation heat recovery of boiler flue gas using the volumetric heat transfer coefficient method.

For spray columns, the correlations developed by several authors, can not be applied directly to the design since each heat transfer device can have a different size and a different number of nozzles etc. In order to solve this problem, the device correction factor 'C' was introduced.

In packed column design, while new and more efficient packing materials are present in the market, there is no design information for those packing materials. Since the heat transfer of packed column mainly depends on the contact surface area of packing material, the data for Raschig rings or Berl saddles with similar contact surface area, which are available, can be adopted for design of a packed column.

It is concluded that the volumetric heat transfer coefficient method is suitable for the design of spray columns using a device correction factor, and for the design of packed columns selecting proper packing material data available.

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NOMENCLATURE

c_g	Specific heat of gas (Btu/lb $^{\circ}$ F)
$c_{g,A}$	Specific heat of gas, species A (Btu/lb $^{\circ}$ F)
c_s	Humid heat capacity of gas (Btu/lb $^{\circ}$ F)
C	Device correction factor for spray column
C_o	Factor defined by Equation 2.6
C_1-C_4	Constants
D	Heat transfer column inside diameter (ft)
Ex	Amount of excess air (%)
f_1-f_3	Factors defined by Equations 2.17 - 2.19
F_p	Packing factor
g	Conversion constant, 32.2 (lb-mass ft/lb-force sec 2)
G	Gas mass velocity (lb/hr ft 2)
G_F	Maximum allowable gas velocity for packing (lb/sec ft 2)
$h_{g,a}$	Gas phase vol. heat transfer coefficient (Btu/hr ft 3 $^{\circ}$ F)
$h_{l,a}$	Liquid phase vol. heat transfer coefficient (Btu/hr ft 3 $^{\circ}$ F)
H _g	Height of gas phase transfer unit (ft)
K	Factor of maximum CO $_2$ % in flue gas of fuel
L	Liquid mass velocity (lb/hr ft 2)
m	Exponent in Equation 2.16
n	Exponent in Equation 2.16
N_A	Rate of diffusion of species A (lb/hr)
Δp	Pressure drop (in-water)
Pr_g	Prandtl number of gas
Q_c	Heat transfer rate for condensation (Btu/hr)
Q_g	Sensible heat transfer rate for gas stream (Btu/hr)

Q_h	Heat transfer rate for humidification (Btu/hr)
Q_l	Sensible heat transfer rate for liquid stream (Btu/hr)
Q_L	Latent heat recovered (Btu/lb)
Q_S	Sensible heat recovered (Btu/lb)
Q_T	Total heat recovered (Btu/lb)
Q_T	Total heat transfer rate (Btu/hr)
Sc_g	Schmidt number of gas
t	Temperature ($^{\circ}\text{F}$)
t_i	Interface temperature ($^{\circ}\text{F}$)
t_g	Gas temperature ($^{\circ}\text{F}$)
t_l	Liquid temperature ($^{\circ}\text{F}$)
t_s	Adiabatic saturation temperature ($^{\circ}\text{F}$)
T_{c1}	Water temperature at place 1
T_{c2}	Water temperature at place 2
T_{h1}	Gas temperature at place 1
T_{h2}	Gas temperature at place 2
U_a	Overall vol. heat transfer coefficient (Btu/hr ft^3 $^{\circ}\text{F}$)
$(U_a)_{avg}$	Average overall vol. heat transfer coefficient (Btu/hr ft^3 $^{\circ}\text{F}$)
v_{design}	Design gas velocity (ft/sec)
v_{max}	Maximum allowable gas velocity (ft/sec)
V_T	Volume of contacting zone (ft^3)
W_c	Amount of water condensed per lb oil (lb water/ lb oil)
W_{ta}	Amount of theoretical air for combustion (lb/lb oil)
Z_T	Height of contacting zone (ft)

Greek letters

α	Correction factor for simultaneous heat and mass transfer (Equation 2.5)
Δt_m	Log mean temperature difference ($^{\circ}\text{F}$)
λ	Heat of vaporization (Btu/lb)
μ_l	Viscosity of liquid (lb/ft hr)
σ	Surface tension (dynes/cm)
ω	Humidity of gas at column inlet (lb water vapor/lb dry gas)
ω_s	Adiabatic saturation humidity (lb water vapor/lb dry gas)
ρ_g	Density of gas (lb/ft ³)
ρ_l	Density of liquid (lb/ft ³)
Ψ	Parameter in Equation 2.16

Subscripts

avg	Average
M	Mean

CHAPTER 1

INTRODUCTION

1.1 Introductory problem statement

Decreasing fuel supplies and increasing cost of those fuels make it vital to conserve our available sources of energy. For the same reasons it is important to investigate the possibilities of recovering some of the heat that is lost forever. These losses occur in the form of unburned but combustible waste materials (waste fuel), sensible heat discharged from drainpipes (drain water), and sensible and latent heat exhausted from stacks (flue gas).

In recent years there has been an increasing interest in the study of flue gas heat recovery using the direct contact spray condensation method. Condensation heat recovery is defined as the process of cooling exhaust gases below their dew point and recovering more than 50 percent of the latent heat content of the exhaust gas as well as a significant amount of sensible heat. This is performed in the absence of an intermediate wall, thereby allowing both fluids to be in direct contact. Usually this contact allows mass and heat transfer between fluids to take place until equilibrium is attained.

Direct contact heat exchangers can generally be classified into three categories :

(1) Spray column : A spray column heat exchanger

essentially consists of a cylindrical vessel. The hot flue gases enter the vessel at the bottom and are cooled by a spray of water fed from nozzles located at the top. The two streams are in direct contact inside the exchanger.

(2) Packed column : A packed column heat exchanger is an exchanger filled with packing material to promote high contact surface area. As used here, it also has a spray of water at the top. This water is heated by the flue gases as it flows through the packing.

(3) Baffle tray column : A baffle tray column heat exchanger is an exchanger which contains baffles to enhance heat transfer performance by increasing contact surface and residence time for the two streams.

The principle of direct contact heat recovery is based on wet scrubbing. But the method of wet scrubbing poses some difficulties when applied to the above three devices because too many experimental data, not available yet, are required.

In Europe these devices are quite popular, but only recently have they been marketed in North America. The design methods that they are using are not available yet [1]. But there are some experimental data for design of these direct contact heat exchangers. Fair[2] and Kiang[3] suggested the use of a design method based on volumetric heat transfer coefficient.

The volumetric heat transfer coefficient method is

based on equations defining the coefficient and some experimental correlations. This method simplifies the design of direct contact heat exchangers. This thesis deals with the design of spray and packed column heat exchangers using the volumetric heat transfer coefficient method.

1.2 Increase in efficiency

The efficiency improvement from the application of direct contact condensation heat recovery can be visualized by referring to the heat recovery curve shown in Figure 1.1 [4]. Cooling the flue gases from 300°F to the dew point of 140°F results in an efficiency increase of 3 percent. Further cooling, resulting in condensation of water vapor, increases the rate of heat recovery dramatically and 10 percent total efficiency improvement is obtained at a 100°F outlet temperature from the stack.

In drawing this figure, the outlet gas was assumed to be saturated and the humidity of outlet gas was calculated from equilibrium conditions. But the actual device reveals that surface effects in the condenser result in cold spots to less than saturated gas on average, and therefore greater latent heat recovery [5].

The heat recovery curve was drawn for boilers using natural gas, which have nearly twice the moisture content of No.2

furnace oil. That is, the heat recovery for No.2 oil fired boilers is less than that for natural gas fired boilers. This will be discussed further in Chapter 5.

1.3 Goals and overview of thesis

Azbel[6] indicated that there will be some significant errors for spray column design in using the volumetric heat transfer coefficient method. This is because neither physical properties of nozzles nor nozzle parameters are taken into account, and the correlations used in the method are based on the assumption that spray particle size and average spray velocity are constant for similar nozzles and the same nozzle pressure drop.

The goal of the thesis is the development of design methods for spray and packed columns.

In the work described herein, a device correction factor was introduced by the author to compensate for the errors which occur in the spray column design. This is described in Chapter 2 and Chapter 4.

Sufficient data on the packing material are not yet available for the method which uses the volumetric heat transfer coefficient method for packed column. However, it has been demonstrated in this thesis that the proper selection of other

existing packing material data gives good results.

In Chapter 2 the heat transfer mechanisms are presented for direct contact heat exchangers. Detailed gas cooling types are also included. The design methods of spray and packed columns are the most important part of this thesis. Chapter 3 presents test facilities and the experimental measurements.

Chapter 4 shows experimental results. The device correction factor for a spray column is determined experimentally. For the packed column, predicted and experimental results are compared. Increase of efficiency and decrease in the amount of carbon dioxide are discussed.

Finally in Chapter 5 a summary of achievements, conclusions, and recommendations for continued studies is made.

CHAPTER 2

THERMAL PERFORMANCE

2.1 Heat transfer mechanism

A direct contact heat exchanger is a vessel in which the process stream and cooling fluid are comingled directly. There is no heat flow resistance from a separating wall, no complexity due to arranging this wall for maximizing the heat transfer surface. The attractiveness stems from the potential economy and simplicity.

The four general classifications of direct contact gas-liquid heat transfer are :

- (1) Simple gas cooling
- (2) Gas cooling with vaporization of coolant
- (3) Gas cooling with partial condensation
- (4) Gas cooling with total condensation

In the application of flue gas condensation heat recovery, we are concerned with items (2) and (3) listed above.

Figure 2.1 shows the basic direct contact heat recovery system. The exhaust gas enters the bottom of the column and is cooled by direct contact with a cold water stream entering the top of the unit.

The heated water stream is passed through a secondary heat exchanger where the recovered energy is transferred to a clean water stream. The use of a secondary exchanger is generally recommended to isolate the potable water from the water in contact with the exhaust gas.

The exhaust gas outlet temperature is directly dependent on the inlet water temperature to the column, and as the water temperature increases, the outlet gas temperature increases, which results in decreased heat recovery rates.

The factors that limit the working fluid temperature are determined by the direct contact nature of the process. The total heat recovery process can be divided into four separate sections as shown in Figure 2.2

Section (1) and (4) in the figure are inlet and outlet sections required for exhaust gas inlet, water storage at the bottom of the tower to provide a liquid head for the working fluid pump, and outlet exhaust gas-water disengagement section.

In the humidification section (2), the exhaust gas contacts the water stream and the process of adiabatic humidification occurs. The exhaust gas is cooled to its adiabatic saturation temperature by vaporizing the working fluid until the gas is saturated.

The liquid is not heated during this process since all of the heat transferred from the gas is used to vaporize liquid. Therefore, the adiabatic saturation temperature of the flue gas represents the maximum limitation on the temperature of the heated working fluid.

The adiabatic saturation temperature of the exhaust gas is determined by its inlet temperature and humidity according to the equation [7, 8]:

$$\omega_s - \omega = c_s (t - t_s) / \lambda \quad (2.1)$$

where, ω_s = adiabatic saturation humidity (lb water/lb dry gas)
 ω = inlet gas humidity (lb water/lb dry gas)
 c_s = humid heat capacity of gas = $0.24 + 0.45 \omega$ (Btu/lb°F)
 t_s = adiabatic saturation temperature (°F)
 t = inlet gas temperature (°F)
 λ = latent heat of vaporization = 970.3 (Btu/lb)

Humid heat capacity of a mixture of air and water vapor, c_s , is the heat capacity of the mixture per lb dry air. The heat capacity of dry air in the temperature range from 30 to 200°F is approximately 0.24 Btu/lb°F, and of water vapor 0.45 Btu/lb°F.

But the convenient way to determine the adiabatic saturation temperature is to use the humidity chart shown in Figure 2.3.

In the dehumidification (or condensation) section (3) in Figure 2.2, the cold water provides the driving force to condense part of the water from the flue gas recovering both the latent heat added to the flue gas in section (2) and part of the additional latent heat of condensation associated with water vapor in the inlet flue gas. The flue gas exits section (3) with less water content than when it entered section (1), resulting in a net recovery of latent heat.

2.2 Heat transfer.

As mentioned earlier in this chapter there are four major regimes under which gas and liquid heat transfer occurs by direct contact : simple gas cooling, gas cooling with coolant vaporization, gas cooling with partial condensation, and gas cooling with total condensation.

(a) Simple gas cooling.

In the simple gas cooling process, only sensible heat transfer takes place, and also there is no mass transfer between gas and liquid, as it is purely a heat transfer process. A schematic diagram of the system in which this could occur is shown in Figure 2.4.

The required volume can be calculated from :

$$V_T = \frac{Q_T}{(Ua)_{avg} \Delta t_m} \quad (2.2)$$

where, V_T = heat transfer volume required (ft^3)

Q_T = total heat transfer rate (Btu/hr)

$(Ua)_{avg}$ = average volumetric overall heat transfer coefficient (Btu/hr ft^3 $^{\circ}\text{F}$)

Δt_m = log mean temperature difference ($^{\circ}\text{F}$)

The log mean temperature difference is based on terminal conditions. The above equation represents the definition of volumetric heat transfer coefficient, and is also used in calculation of the following regimes of gas cooling.

The volumetric overall heat transfer coefficient, Ua , must be determined empirically, and is a characteristic of the system used. The overall heat transfer coefficient can be segregated into two components as :

$$Ua = \frac{1}{\frac{1}{h_{ga}} + \frac{1}{h_{la}}} \quad (2.3)$$

where, h_{ga} = gas phase volumetric heat transfer coefficient (Btu/hr ft^3 $^{\circ}\text{F}$)

h_{la} = liquid phase volumetric heat transfer coefficient (Btu/hr ft^3 $^{\circ}\text{F}$)

This equation is similar to the usual overall heat transfer coefficient equation for series resistors. There is no

conduction resistance in the equation because of direct contact,

Both, h_{ga} and h_{la} are functions of the heat transfer device as well as the properties of the gas and liquid.

The typical operating line is represented by AB on humidity chart shown in Figure 2.5. Point A represents temperature and humidity at the initial condition. Point B is located between point A and the saturation line, and has the same humidity as at point A.

(b) Gas cooling with coolant vaporization

In a gas cooling with vaporization process, the gases vaporize a certain amount of coolant and carry it through the system.

The total heat transfer, Q_T , will include not only the sensible heat transfer but also the latent heat transfer. Thus, the overall heat transfer coefficient is : (Refer to Appendix A for the derivation of the equation)

$$U_a = \frac{1}{\frac{1}{a h_{ga}} + \left(\frac{1}{h_{la}} \right) \left(\frac{Q_L}{Q_T} \right)} \quad (2.4)$$

$$a = \frac{Co}{1 - e^{-Co}} \quad (2.5)$$

$$Co = \frac{N_A c_{g,A}}{h_g a V_T} \quad (2.6)$$

where, Q_l = sensible heat transfer rate for the liquid stream (Btu/hr)

Q_T = total heat transfer rate (Btu/hr)

a = correction factor for simultaneous heat and mass transfer.

N_A = rate of vaporization of the liquid (lb/hr)

$c_{g,A}$ = specific heat of the vaporized liquid (Btu/lb °F)

The multiplier a in Equation 2.5 is the Ackerman factor [9, 10], which accounts for the influence of mass transfer on heat transfer rate. When heat and mass are transferred in opposite directions as in humidification (gas cooling with coolant vaporization), N_A is negative and $a < 1.0$. In dehumidification (gas cooling with partial condensation), heat and mass will flow in the same direction, therefore N_A is positive and $a > 1.0$. Figure 2.6 [2] shows how a varies with Co .

If the system is an adiabatic humidifier, the operating line is AC shown in Figure 2.5. C is located on the adiabatic cooling line AD.

(c) Gas cooling with partial condensation

In the gas cooling with partial condensation process,

the condensation of some vapours occurs during the heat transfer operation. The total heat transfer includes gaseous sensible heat transfer as well as the latent heat recovery and transfer. The overall volumetric heat transfer coefficient can be described by the equation :

$$Ua = \frac{1}{\frac{1}{h_1 a} + \left(\frac{1}{a h_g} \right) \left(-\frac{Q_g}{Q_T} \right)} \quad (2.7)$$

where a is as defined in Equations 2.5 and 2.6. Note that N_A is now positive because heat and mass flow are in the same direction. Refer to Appendix A for the derivation of the above equation.

The operating line is represented by ADE in Figure 2.5. The amount of condensation is larger than that of humidification. While Figure 2.5 shows the ideal operating lines, the actual operating lines in heat transfer columns are different. This is discussed in Section 5.3.

(d) Gas cooling with total condensation

In the gas cooling with total condensation process, there is total condensation of high temperature vapours. The total duty, Q_T , is the same as the condensation duty hence there is no sensible heat duty, Q_g . Thus Equation 2.7 becomes :

$$Ua = h_1 a \quad (2.8)$$

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This is the case of steam condenser. If the coolant liquid is the same as the condensate, there is relatively little resistance in the liquid phase, and extremely high U_a values result.

2.3 Spray column

2.3.1 Overview of spray column

The previous section dealt with heat transfer characteristics using volumetric heat transfer coefficients. Since all the volumetric heat transfer coefficients are functions of heat transfer devices used, they must be predetermined by experiment.

Fortunately there are some correlations for the gas phase volumetric heat transfer coefficient. Since the liquid phase volumetric heat transfer coefficient is much higher than the gas phase coefficient (around 60 times higher in usual values), the liquid side heat transfer resistance, $1/h_{l,a}$, can be neglected in the Equations 2.3, 2.4 and 2.7. Therefore, as long as the correlation for the gas phase heat transfer coefficient is suitable for the heat transfer device used, the design of the device is successful.

The heat transfer coefficients, $h_{g,a}$ and $h_{l,a}$, are mainly dependent on gas mass velocity, liquid mass velocity, column

height, and nozzle orifice diameter. Neglecting the liquid side resistance, $1/h_{ga}$, the gas side heat transfer coefficient, h_{ga} , can be determined by experiment.

Heat transfer data for commercial spray columns were correlated by Fair[2] by using the equation :

$$h_{ga} = \frac{0.043 G^{0.8} L^{0.4}}{Z_T^{0.5}} \quad (2.9)$$

where, G = gas mass velocity (lb/hr ft²)

L = liquid mass velocity (lb/hr ft²)

Z_T = height of contacting zone (ft)

The data were derived from hydrocarbon systems as well as from air and water systems. Solid cone spray nozzles were used in columns 2.5 to 6 ft in diameter.

Another correlation derived by Mehta and Sharma [11] is

$$h_{ga} = \frac{1.62 C_2 G^{0.82} L^{C_3}}{0.205 C_3 Z_T^{0.38}} \quad (2.10)$$

Values for the constants are presented in Table 2.1.

For a 15.2 in. diameter column with a typical solid cone nozzle used in commercial applications (Orifice diameter 0.33 in), Equation 2.10 reduces to :

$$h_{ga} = \frac{0.0146 G^{0.82} L^{0.47}}{Z_T^{0.38}} \quad (2.11)$$

A comparison of Equation 2.11 with Equation 2.9 indicates that they are quite similar in form but the numbers and powers differ somewhat.

Other design parameters for spray columns should be noted. For gas pressure drop, the following correlation has been suggested [2] :

$$\Delta p = 5.88 \times 10^{-5} L + 0.065 \quad (2.12)$$

where Δp is in inches of water and L is liquid mass velocity. For maximum allowable vapour capacity, the Souders-Brown coefficient can be applied [2].

$$v_{max} = C_4 \left(\frac{\rho_l - \rho_g}{\rho_g} \right)^{1/2} \quad (2.13)$$

where, ρ_l = density of liquid (lb/ft³)
 ρ_g = density of gas (lb/ft³).

The coefficient C_4 has not yet been determined by flood tests. Although values as high as 0.40 have been used successfully, a design range of 0.15 - 0.25 appears more appropriate. It is warned that very fine atomization by the nozzles can cause entrainment at relatively low gas rates.

2.3.2 Design of spray column

In the section on the overview of spray column, the correlations were made under different conditions, i.e. different column diameters, different nozzle orifice diameters, and different number of nozzles etc. Moreover, all of these correlations require well-arranged nozzles and well-distributed spray. All of these factors make it very difficult to make use of those correlations.

The author devised one method to solve the problems. By virtue of two definition equations (Equations 2.2 and 2.4 or 2.7), the exact value of h_{ga} for a system can be determined experimentally, since h_{ga} is a function of the system used. This is done by measuring Q_T , Q_g , Δt_m and V_T and substituting these values into the above two equations (Equations 2.2 and 2.4 or 2.7). As long as the same column diameter and the same nozzle configuration are used, the determined value of h_{ga} can be applied successfully.

Although Equations 2.9 and 2.11 are very similar, Equation 2.9 is more general [12]. Now a device correction factor, 'C', can be introduced as :

$$h_{ga} = \frac{C G^{0.8} L^{0.4}}{Z_T^{0.5}} \quad (2.14)$$

This is a very convenient way because the coefficient

'C' itself corrects all the errors so long as only Z_T , L and G vary.

The determination of 'C' is described in Chapter 4.

2.4 Packed column

2.4.1 Overview of packed column

There is no entirely satisfactory method for predicting the size of a packed column heat transfer unit required. In practice the value for a particular packing will depend not only on the physical properties and flow rates of the gas and liquid, but also on the uniformity of the liquid distribution throughout the column height and diameter. This makes it difficult to apply data obtained from the laboratory to an actual industrial size column.

Using the volumetric heat transfer coefficient approach, the heat transfer coefficient for the gas phase is [2] :

$$h_{g,a} = \left(\frac{Sc_g^{2/3} c_g G}{Pr_g} \right) \frac{1}{Hg} \quad (2.15)$$

where Sc_g = Schmidt number of gas

Pr_g = Prandtl number of gas

Hg = height of gas phase transfer unit (ft)

Hg is evaluated from mass transfer data for a particular type and size of packing material used.

Data for only Raschig rings and Berl saddles were summarized and generalized. This work led to methods for predicting heat transfer units [2,13] :

$$H_g = \frac{\psi S c_g^{0.5} D^n}{(L f_1 f_2 f_3)^m} \left(\frac{Z_T}{10} \right)^{1/3} \quad (2.16)$$

$$f_1 = \left(\mu_L / 2.42 \right)^{0.16} \quad (2.17)$$

$$f_2 = \left(62.4 / \rho_L \right)^{1.25} \quad (2.18)$$

$$f_3 = \left(72.8 / \sigma \right)^{0.8} \quad (2.19)$$

where, μ_L = viscosity of liquid (lb/ft hr)

σ = surface tension of liquid (dynes/cm).

Values for parameters are given in Table 2.2.

2.4.2 Design of packed column

The generalized pressure drop correlation has been developed as a practical aid to packed column design and incorporates flowrates, physical properties of the fluid, a wide range of packings and pressure drop on one chart shown as Figure 2.7 [14]. A line representing flooding, which is the maximum flow capacity condition for a packed column, would lie above the top curve and hence the correlation may be used safely in design procedures. The packing factor F_p which is employed in the

correlation is a modification of the specific surface of the packing. The pressure drop is also shown in the Figure 2.7.

Presently the data for the equations 2.16 to 2.19 are not available except for Raschig rings and Berl saddles. However, the data for Raschig rings and Berl saddles can be used to make conservative estimates for the other packings. In this case, the physical properties of packing material should be compared with those of Raschig rings or Berl saddle to adopt the proper one. The main physical properties are contact surface, free surface and packing factor etc. [15].

CHAPTER 3

TEST FACILITIES AND EXPERIMENTAL MEASUREMENTS

3.1 Test facilities

The direct contact heat exchange performance was evaluated experimentally using spray and packed column heat exchangers attached to the flue gas exhaust of a CARSWELL oil-fired furnace (Figure 3.1) which is located in the Thermodynamics Laboratory at Carleton University.

The furnace uses No.2 oil for domestic hot water and warm air generation. The specifications of this furnace are shown in Table 3.1.

The furnace was in good condition. Even though the exhaust gas temperature was a little higher than the temperature of inlet gas into the heat exchange column, it was found not to affect the performance test of the column.

The fuel flow rate was kept constant at 1 GPH for the packed column performance test since it was very difficult to change the nozzle. However, it was changed a little for the spray column performance test.

In order to check the correct amount of carbon dioxide before and after the column, two dampers were removed and the fan

with speed controller was used (Figure 3.2). The fan was located at the top of the heat exchange column. Due to the low pressure drop characteristics of the spray column without packing, the fan was removed for this column. For the packed column installed with GEMPAK, the fan was needed because of high pressure drop of packed column characteristics.

Originally the heat exchange column was designed and manufactured for the packed column heat exchanger, but without packing material inside the column, it can also be used as a spray column. Therefore, the same size of column was used for spray and packed column performance tests. The column is shown in Figures 3.3 and 3.4.

The column was made of stainless steel and has two parts. The bottom part includes water storage space, gas inlet hole, water outlet hole, and packing material holder. There are also ten thermocouple wire holes in which commercial Swageloks were installed to seal gaps. The upper part contains two water nozzles, a fan and two thermocouple holes. This makes it easy to install or remove the packing materials. The packing material, GEMPAK 4A, was supplied by Glitsch Canada LTD (see Figure 3.5). The specifications of GEMPAK 4A are listed in Table 3.2.

Unlike other packing materials, i.e. Raschig rings and Berl saddles, the GEMPAK is bundled material which can be easily installed in a heat exchange column. While Raschig rings, Berl

saddles, Pall rings, and Intalex saddles have been used in chemical engineering for quite some time now, the GEMPAK type of packing has only recently been introduced into the market.

Three bundles of GEMPAK were used to make 2.4 ft of column height. For effective use of these materials, the second GEMPAK should be placed at 90 degrees to the first circumferentially.

Two commercial water spray nozzles were installed. Each nozzle has an ability of spraying 19 GPH at water pressure of 40 psi. The required water flow rate was about 45 GPH. Considering 45-65 psi water supply in Carleton University, two nozzles are quite suitable. The details of the nozzle are shown in Table 3.3.

The heat exchange column was covered with 1 inch thick fiberglass. For the spray column, the device correction factor includes the effect of insulation. For the packed column, the heat loss through the insulating material was ignored for simplicity.

3.2 Experimental measurements

3.2.1 Temperature, fuel flow rate, and water flow rate

The temperatures of water and flue gases were measured

using copper-constantan thermocouples. For the spray column, eight more temperatures were measured between inlet and outlet temperature points. It was impossible to measure water and flue gas temperature inside the packing materials for the packed column.

Upright cups were used in order to measure the water stream temperature, while reversed cups were used for the gas temperature. Thermocouple junction points were installed in the cups. A mixing valve was convenient to make a proper inlet water temperature.

A water flow meter and a fuel flow meter were used for inlet cold water amount and fuel amount.

3.2.2 Humidity

The absolute humidity of flue gases can be calculated if the excess air amount is known. The excess air amount can also be calculated using the amount of carbon dioxide in the flue gases assuming complete combustion of the fuel. A Bacharach Test Kit was used to measure the amount of carbon dioxide.

Ganapathy[16,17] suggested a good estimation method of excess air by volume using the following equation :

$$Ex = 100 \times \left(\frac{K}{\% CO_2} - 1 \right) \quad (3.1)$$

where K is a factor depending on fuels. For oil fuels the factor K is 15.5. The derivation of this equation is based on the linearity between excess air and CO_2 amount by volume in flue gas. The linearity can be well assumed in the neighborhood of 10 to 60 % excess air. K represents maximum CO_2 % in flue gas for designated fuel. For the spray column in this thesis, larger amounts of air resulted in a device correction factor 'C' which deviated unusually from the average value of 'C', therefore two experiments were discarded, which is described in Section 4.1.1. The range of excess air used was between 7 and 60 % for spray column, and between 15 and 37 % for packed column.

In order to provide preliminary data, the inlet flue gas humidity is calculated in the next section. The calculations are based on the properties of No.2 oil in Table 3.4.

3.2.3 Calculation of inlet flue gas humidity

While fuel type and flue gas temperature are the most critical in improving efficiency, the effect of combustion air humidity is such that in a high humidity area the resulting efficiency increase can be 1 percent higher than the predicted with the assumption of zero humidity in air [18]. In this thesis the combustion air humidity is neglected in calculation

for simplicity.

Using the physical and chemical properties of No.2 oil in Table 3.4, the inlet flue gas humidity, is calculated as follows :

(a) Complete combustion with theoretical air

The theoretical air in lb per lb oil burned, W_{ta} , is calculated as follows: (Refer to Appendix B for details of theoretical air calculation)

$$\begin{aligned} W_{ta} &= 11.52 C + 4.32 (8 H - O + S) \\ &= 11.52 \times 0.871 + 4.32 (8 \times 0.1280 - 0.0500 + 0.050) \\ &= 14.026 \text{ lb air/lb oil} \quad (\text{Here } N = 0.00 \text{ assumed}) \end{aligned}$$

The total weight of the products when one lb of oil is burned is the weight of the oil plus that of the air = $1.000 + 14.026 = 15.026$ lb. The products are CO_2 , N, SO_2 , and H_2O . Therefore weights of products produced per lb of oil are:

$$CO_2 = 0.871 \times 44/12 = 3.194 \text{ lb/lb}$$

$$\begin{aligned} N &= (\text{from air}) + (\text{from oil}) \\ &= 14.026 \times 0.7685 = 10.779 \text{ lb/lb} \end{aligned}$$

$$SO_2 = 0.05 \times 64/32 = 0.100 \text{ lb/lb}$$

$$H_2O = 0.1280 \times 18/2 = 1.152 \text{ lb/lb}$$

$$\text{Total weight} = 3.194 + 10.779 + 0.100 + 1.152 = 15.225 \text{ lb/lb}$$

Therefore the inlet flue gas humidity is calculated as follows:

$$\begin{aligned}
 \text{Humidity} &= (\text{water vapor per lb oil}) / (\text{dry gas per lb oil}) \\
 &= (\text{H}_2\text{O}) / (\text{CO}_2 + \text{N} + \text{SO}_2) \\
 &= 1.152 / 14.073 \\
 &= 0.0819 \quad (\text{lb water vapor} / \text{lb dry gas})
 \end{aligned}$$

(b) Combustion with 30 % excess air

With the same method as above, the calculation is performed. Air contains 23.15 % oxygen and 76.85 % nitrogen by weight.

$$\text{CO}_2 = 3.194 \text{ lb/lb} \quad (\text{same})$$

$$\text{Excess O} = 4.208 \times 0.2315 = 0.974 \text{ lb/lb}$$

$$(\text{Excess air} = 14.026 \times 0.3 = 4.208 \text{ lb/lb})$$

$$\text{N} = (14.026 + 4.208) \times 0.7685 = 14.013 \text{ lb/lb}$$

$$\text{SO}_2 = 0.100 \text{ lb/lb} \quad (\text{same})$$

$$\text{H}_2\text{O} = 1.152 \text{ lb/lb} \quad (\text{same})$$

Therefore the humidity of the flue gas with 30 % excess air is calculated as follows:

$$\begin{aligned}
 \text{Humidity} &= (\text{water vapor per lb oil}) / (\text{dry gas per lb oil}) \\
 &= (\text{H}_2\text{O}) / (\text{CO}_2 + \text{O} + \text{N} + \text{SO}_2) \\
 &= 1.152 / 19.333 \\
 &= 0.0596 \quad (\text{lb water vapor} / \text{lb dry gas})
 \end{aligned}$$

CHAPTER 4

EXPERIMENTAL RESULTS

4.1 Spray column

4.1.1 Device correction factor 'C'

As mentioned earlier in Chapter 3, the column originally was designed for a packed column which effects heat transfer much better than a spray column of the same size which does not have packing. Therefore it is expected that the desired condensation of water vapor in gases will not occur in the spray column without packing because of insufficient height for condensation. Experimentally the dehumidification process, which occurs in the upper part of the column did not overcome the humidification of the gases which occurred in the lower part of the column.

In Appendix C.1, a sample calculation was performed to obtain the device correction factor 'C'. Also refer to a computer program in Appendix D.1. The input and the output values of the computer program are as follows:

Input : Water flow rate

Fuel flow rate

CO₂ content in inlet flue gas

Inlet flue gas temperature

Outlet flue gas temperature

Inlet water temperature

Outlet water temperature

Output : Device 'correction factor 'C'

All the input values are dependent on experimental condition. Detailed computer program results are shown in Table 4.1.

The Table 4.1 shows that generally the absolute humidity of the outlet gases was higher than the inlet gas humidity. This means that the vaporization of coolant water occurred in the lower part of the column, but due to the insufficient height of the upper column the condensation was not enough to overcome the humidification. Therefore the calculations were correct consistent with the assumption of cooling by coolant vaporization.

Since the device correction factor, 'C', is independent of type of heat transfer, this correction factor can be used in the partial condensation application, which is applied in the next section.

The experiments were performed through 13 measurements with various values of gas mass velocity, G , and liquid mass velocity, L . The excess air amount was also changed. Discarding the highest and lowest values of 'C' (Table 4.1), the distributions are between 0.091 and 0.103. This is shown in Figure 4.1.

Other results derived by Fair[2] and Mehta[11] show 'C' is constant in the wide range of $G^{0.8} L^{0.4}$. Averaging 'C' values gives 0.099, and now Equation 2.14 becomes :

$$h_{ga} = \frac{0.099 G^{0.8} L^{0.4}}{Z_T^{0.5}} \quad (4.1)$$

Comparing this equation with the Equation 2.9 developed by Fair[2], the new value of 'C' (0.099) is greater than twice the original value of 'C' (0.043). Both of the equations were developed with various G and L and constant Z_T . This thesis used 2.4 ft of column height and Fair used 4.3 ft of column height. This means that by adjusting the exponent of Z_T , we may get a better equation.

The Equation 2.11 derived by Mehta and Sharma [11] was based on 8.0 ft of column height. Table 4.2 shows the main data used to derive the gas phase heat transfer coefficient equations for all the above three cases. In order to compare the device correction factor 'C', Equation 2.11 is changed to an equation with the same exponents of G, L and Z_T as others.

Figure 4.2 shows the plot of device correction factors against $G^{0.8} L^{0.4}$ for the above three cases. There is a relationship between 'C' values and column height Z_T . Smaller height column has bigger 'C' value. Therefore if we adjust the height correction exponent of Z_T , the three equations can be reduced to a single equation. The exponent of Z_T will be greater than 0.5,

which means the column height effect is more significant.

Adjusting the 'C' value and the exponent of Z_T , a new equation is obtained as follows:

$$h_{ga} = \frac{0.276 G^{0.8} L^{0.4}}{Z_T^{1.7}} \quad (4.2)$$

Using this equation, prediction of values of h_{ga} can be made. Table 4.3 and Figure 4.3 show comparisons of h_{ga} between prediction and experiment.

The absolute humidity of outlet gas shown in Table 4.1 was calculated from humidification duty divided by heat of vaporization (see Appendix C.1 for sample calculations). Therefore the humidity may include not only vapor but also mist content. This is why the relative humidities in the Table 4.1 present values larger than 100 %. Actually this occurred at the case of lower inlet gas temperature and lower inlet cold water temperature.

In the next section, using the new value of 'C', 0.276, and height correction exponent, 1.7, a spray column design, which satisfies the condensation requirement, is discussed.

4.1.2 Proper size of spray column

Now using Equation 4.2 for the gas phase heat transfer coefficient, the design can be made for a spray column.

The requirements for condensation heat recovery devices are cooling exhaust gases below their dew point and recovering more than 50% of latent heat.

In Appendix C.2, a sample calculation was performed for a spray column design in agreement with the above requirements. The input values and the output results are as follows : (Refer to computer program results for details in Appendix D.2)

Input :

Fuel flow rate : 1.15 GPH

Flue gas inlet, outlet temp. : 410, 135°F

Water inlet temp. : 95°F

Amount of excess air : 41%

Desired condensation : 70%

Adiabatic saturation temp. of inlet gas : 134°F

Column diameter : 0.7 ft

Output :

Sensible cooling duty : 12249 Btu/hr

Condensation duty : 6415 Btu/hr

Total duty : 18665 Btu/hr

Column height : 3.6 ft

Now comparing this column with the original one used in experiment, its height is one and half times the height of the column used. The additional height (1.2 ft) would provide condensation of water from the flue gases, which were nearly saturated at some lower part of the column, to 70% of inlet gas absolute humidity.

Fuel flow rate

Water flow rate

Flue gas inlet and outlet temperatures

Water inlet and outlet temperatures

Excess air amount

Column size

And output result is a condensation rate at column outlet.

(Computer program in Appendix D.4)

With above experimental results, predictions can be made in reverse. The condensation rate obtained above is now used as an input condition. The input conditions are:

Fuel flow rate

Flue gas inlet and outlet temperatures

Water inlet temperature

Excess air amount

Condensation rate

Column size

And output results are outlet water temperature and water flow rate. (Computer program in Appendix D.5)

Table 4.4 shows the comparison between the experimental and predicted results. While the prediction shows higher water flow rate and lower outlet water temperature, the experimental result shows lower water flow rate (average 2.4 GPH lower) and higher outlet water temperature (average 1.8 °F higher). Since the total duties in both cases were nearly same, the actual

performance of the packed column was more effective than the predicted performance in obtaining high temperature of outlet water.

The actual outlet water temperature is 1 to 3°F below the adiabatic saturation temperature of flue gases for each of 8 experiments.

4.2.2 Ultimate design of packed column

There will be some cases that the compactness of packed column is important, or the cost of packing material is critical.

The packed column can also be designed to operate at the highest economic pressure drop. Recommended design values of pressure drop for absorbers and strippers are between 0.18 and 0.6 in of water per foot of column height [15]. This is equivalent to 40 % to 60 % flooding. Therefore using generalized pressure drop correlation (Figure 2.7), the packed column diameter can be determined.

If the packed column installed with GEMPAK is designed using 40 % flooding value, the equivalent size will have 0.42 ft diameter and 0.8 ft height to produce the same performance as one having 0.7 ft diameter and 2.4 ft height discussed in the previous section. The computer program for the ultimate design

of packed column is in Appendix D.6.

4.3 Relative humidity

The cooling of a gas is complicated by the fact that in some zones of the heat exchange column the gas may be cooled below its saturation temperature and the moisture precipitated as a mist. To determine whether a mist will be produced it is necessary to carry out a stepwise calculation to evaluate the gas temperature and humidity at several intermediate points in the column.

Typically the humidity of vapor is plotted against temperature, as shown in Figure 4.4. The curve AB is the saturation curve, point C represents the temperature and humidity of the gas at the inlet and point D the state of the gas leaving the column [19].

But our experiments do not have sufficient data to enable us to draw this Figure. However we can plot some data points, i.e. temperatures and humidities of gases at inlet and outlet of the column. Figure 4.5 shows the experimental gas cooling lines for the spray and the packed columns. Refer to Appendix E for the plotting of operating lines.

Lines A and B in the Figure 4.5 (a) were typical of the spray column experiments. While the heat transfer represented by

line A requires more coolant water and less fuel, that represented by line B requires considerably less water and more fuel. A mist is produced at the outlet column in the case of line A.

Line C in the Figure 4.5 (b) represents a typical type of gas cooling in the packed column experiments. The relative humidities are shown in Table 4.1 for spray column and in Table 4.4 for packed column experiments.

4.4 Increase in Efficiency

The heat recovery curve for natural gas fired boiler shown in Figure 1.1 was drawn on the assumption of outlet gas saturation and no excess air. With the same assumption, the heat recovery curve can be drawn for No.2 oil fired boiler. The curve is shown in Figure 4.6. Because of lower moisture content, the efficiency increase of No.2 oil is less than that of natural gas. For the curve plotting data of the No.2 oil fired boiler, refer to Appendix F.

As mentioned in Section 1.2, the actual heat transfer column revealed that surface effects in the condenser resulted in cold spots leading to less than saturated gas on average, and therefore greater latent heat recovery. The relative humidities shown in Table 4.4 have values between 57 and 70 %. This means that additional latent heat was recovered.

The increase of efficiency is defined as the ratio of the recovered heat to the higher heating value of fuel. Table 4.4 shows the increase of efficiency for each of 8 experiments. This was calculated from total heat recovered, Q_T , divided by higher heating value of No.2 oil. The average efficiency increase of the packed column is 10.0 %. This is also shown in Figure 4.7.

4.5 Carbon dioxide

While the amount of carbon dioxide was decreased by an average of 0.45 % for the packed column, the spray column did not show any decrease in the amount of carbon dioxide. This is because the contact surface of the spray column was much less than that of the packed column. The amounts of the carbon dioxide in the inlet and the outlet flue gas are shown in Tables 4.1 and 4.4.

CHAPTER 5

CONCLUSIONS AND RECOMENDATIONS

5.1 Summary

The design methods of spray and packed columns have been investigated in this thesis for the condensation heat recovery of boiler flue gas using the volumetric heat transfer coefficient method.

For spray column, the correlations developed by several authors cannot be applied directly to the design since each heat transfer device can have a different size and a different number of nozzles etc. In order to solve this problem, the device correction factor 'C' was introduced.

In packed column design, while new and more efficient packing materials are present in the market, there is no design information for those packing materials. Since the heat transfer of packed column mainly depends on the contact surface area of packing material, the data for Raschig rings or Berl saddles with similar contact surface area, which are available, can be adopted for design of packed column.

Six computer programs were written. Two programs were developed for the spray column and four programs for the packed column.

The experimental results have been compared with predicted values.

5.2 Conclusions

The following conclusions can be made as a result of this study :

1. The volumetric heat transfer coefficient method is suitable for the design of spray and packed columns, using device correction factor 'C' and selecting proper packing material data available.
2. A device correction factor 'C' for a spray column can be determined experimentally.
3. Packing material data can be selected from presently known data of other packing material.
4. The increase in efficiency for the packed column was in the range of 9.2 - 10.3 % corresponding to the inlet flue gas temperature between 339 and 376 °F.
5. While the amount of carbon dioxide was considered to decrease in the packed column because of larger contact surface area, the spray column, which has the same height, did not show variation of carbon dioxide amount.

5.3 Recommendations

In pursuing further investigations of the direct contact heat exchangers, the following recommendations are made:

1. The gas cooling lines were plotted on humidity chart with only inlet and outlet points of temperature and humidity (Figure 4.5). In order to plot exact cooling lines, both of temperature and humidity should be measured at many places inside the heat transfer column.

2. For the economic evaluation of the heat exchanger, more extensive experimentation is recommended with a secondary heat exchanger (Figure 2.1).

Table 2.1 Constants for Equation 2.10 [2, 11]

Column Dia. (in)	Orifice Dia. (in)	C_2	C_3
8.0	0.17	5.15×10^{-5}	0.93
15.2	0.17	4.90×10^{-4}	0.70
15.2	0.22	8.97×10^{-4}	0.62
15.2	0.33	4.28×10^{-3}	0.47

Table 2.2 Parameters for Packed Column Heat/Mass Transfer [2, 19, 20]

	Raschig Rings			Berl Saddles	
	0.5 in	1 in	2 in	1 in	2 in
Exponent m	0.6	0.6	0.6	0.5	0.5
Exponent n	1.24	1.24	1.24	1.11	1.11
Parameter ψ					
40% flood	105	110	210	60	95
60% flood	115	105	210	60	95
80% flood	95	80	210	60	95
Contact surface (ft ² /ft ³)	127	63	31	76	32
Free space (%)	85	92	92	69	72
Packing factor	300	115	57	110	45

Table 3.1 Specifications of CARSWELL Oil Furnace

Model	Bonnet capacity (Btu/hr)	Maximum fuel input (GPH)	Nozzle / solid spray	Aero burner model No.
300C	114,000	1.0	80 °AR	FAFC-2-300

Table 3.2 Specifications of GEMPAK 4A

Diameter (in)	Height (in)	Density (lb/ft ³)	Contact surface (ft ² /ft ³)	Free space (%)	Packing factor	Maximum liquid (GPM/ft ²)
8.4	9.5	14.5	137.1	97.2	32	8

Table 3.3 Specifications of Water Spray Nozzle

Nozzle type	Orifice dia. (in)	Spray type	Spray angle (degree)	Capacity at 40 psi (GPH)
Monarch H-261	0.06	solid cone	80	19

Table 3.4 Physical and Chemical Properties of No.2 Oil [17]

Physical properties			Chemical composition (% by wt.)			
API	lb/gal	HHV (Btu/lb)	C	H	O+N	S
33.00	7.185	19360	83.30	13.28	2.42	1.00

Table 4.1 Performance Test Results of Spray Column with No Packing

Water Flow Rate (GPH)	Fuel Flow Rate (GPH)	CO ₂ (%)		Excess Air (%)	Temperature (°F)				HI (lb/lb)	HO (lb/lb)	C	QT (Btu/hr)	RH (%)
		Before	After		IC	OG	IW	OW					
29	0.87	11.5	-	35	301	112	93	113	0.600	0.071	0.103	6041	(108)
29	0.925	11.5	-	35	365	128	97	125	0.060	0.072	0.094	8098	71
23	1.04	11.3	-	37	364	128	83	112	0.059	0.086	0.091	9168	84
35	1.10	14.5	14.7	7	355	114	91	118	0.076	0.076	0.098	7844	(115)
30	1.10	12.5	13.0	24	382	121	92	113	0.066	0.082	0.100	9754	95
25	1.15	11.0	10.5	41	410	135	95	127	0.058	0.093	0.102	12116	74
20	1.15	9.5	9.3	63	425	149	100	132	0.050	0.097	0.107	13970	49
15	1.15	9.0	9.0	72	427	194	104	134	0.047	0.092	0.082	12417	7
35	1.04	12.8	11.7	21	368	117	93	119	0.067	0.077	0.098	8674	(103)
30	0.98	12.0	12.0	29	390	124	94	123	0.063	0.079	0.099	9202	85
25	1.04	11.4	9.7	36	410	135	97	127	0.060	0.092	0.101	10594	73
20	1.04	9.9	9.7	60	415	150	101	132	0.052	0.093	0.102	11665	<5
15	1.04	10.5	9.5	48	397	156	95	130	0.055	0.093	0.098	10033	<5

* TG: Inlet flue gas

OG: Outlet flue gas

IW: Inlet coolant water

OW: Outlet hot water

HI: Inlet gas humidity

HO: Outlet gas humidity

C: Device correction factor

QT: Total heat recovered

RH: Relative humidity of outlet gas

* Water flow rate and fuel flow rate were measured within the error limits of ± 0.5 GPH and ± 0.005 GPH respectively.

Table 4.2 Spray Column Data

	Gas Mass Velocity G lb/ft ² hr	Liq. Mass Velocity L lb/ft ² hr	Mass Vely. Correction G ^{0.8} L ^{0.4}	Gas Phase H.T. Coeff. h _{ga} Btu/hrft ² °F	Device Cor. Factor C
Thesis (Column Height Z _T =2.4)	325	621	1339	89	0.103
	348	620	1413	86	0.094
	395	485	1418	83	0.091
	331	753	1468	93	0.098
	381	634	1533	99	0.102
	449	524	1620	107	0.102
	352	751	1540	98	0.098
	353	641	1449	92	0.099
	392	527	1460	95	0.101
	449	413	1476	96	0.102
	424	308	1251	79	0.098
Fair (Column Height Z _T =4.3)	215	793	1060	22	0.043
	290	727	1302	27	0.043
	430	679	1736	36	0.043
	560	910	2411	50	0.043
	600	679	2266	47	0.043
	720	798	2797	58	0.043
Mehta and Sharma (Column Height Z _T =8.0)	850	2600	5123	50	0.028
	1500	2600	8070	70	0.025
	1450	2600	7854	85	0.031
	2200	2600	10960	80	0.021
	3000	2600	14050	100	0.020
	3000	2600	14050	140	0.028
	900	4200	6496	55	0.024
	1500	4200	9776	80	0.023
	1500	4200	9776	110	0.029
	2200	4200	13281	80	0.017
	2200	4200	13281	125	0.027
	3000	4200	17021	110	0.018
	3000	4200	17021	170	0.028

Table 4.3 Comparison of h_a between Prediction and Experiment of Spray Column

	Column Height Z_T (ft)	Mass Vely. Correction $G^{0.8} L^{0.4}$	Gas Phase H.T. Coeff. h_a (Btu/hr ft ² °F)	
			Experiment	Prediction
Thesis	2.4	1339	89	83
	"	1413	86	88
	"	1418	83	89
	"	1468	93	92
	"	1533	99	96
	"	1620	107	101
	"	1540	98	96
	"	1449	92	90
	"	1460	95	91
	"	1473	96	93
	"	1251	79	78
Fair	4.3	1060	22	25
	"	1302	27	30
	"	1736	36	40
	"	2411	50	56
	"	2266	47	52
Mehta and Sharma	"	2797	58	65
	8.0	5123	50	41
	"	8070	70	65
	"	7854	85	63
	"	10960	80	88
	"	14050	100	113
	"	14050	140	113
	"	6496	55	52
	"	9776	80	79
	"	9776	110	79
	"	13281	80	107
	"	13281	125	107
	"	17021	110	137
	"	17021	170	137

Table 4.4 Comparison of Results Between Prediction and Experiment
of Packed Column with GEMPAK 4A

Water Flow Rate (GPH)	Fuel Flow Rate (GPH)	CO ₂ (%)		Ex Air (%)	Temperature (°F)				HI (%)	HO (%)	Cond (%)	QT (Btu/hr)	RH (%)	IE (%)
		Before	After		IG	OG	IW	OW						
- =	44.6 (47.0)	12.2	11.8	(27) "	370 "	102 "	94 "	133 (131)	(0.064) (")	(0.024) (")	(62.0) "	(14375) (14369)	(60)	(10.3)
- =	44.6 (47.1)	11.5	11.7	(35) "	376 "	103 "	95 "	134 (132)	(0.061) (")	(0.029) (")	(52.9) "	(14375) (14387)	(62)	(10.3)
- =	44.6 (46.4)	12.4	12.0	(25) "	364 "	101 "	93 "	131 (130)	(0.065) (")	(0.025) (")	(61.4) "	(14006) (14009)	(63)	(10.1)
- =	44.6 (46.5)	12.3	11.1	(26) "	372 "	103 "	95 "	132 (131)	(0.065) (")	(0.030) (")	(53.2) "	(13637) (13633)	(67)	(9.8)
- =	44.6 (48.4)	13.0	12.6	(19) "	359 "	100 "	93 "	131 (128)	(0.068) (")	(0.022) (")	(68.0) "	(14006) (13992)	(57)	(10.2)
- =	44.6 (46.4)	12.2	11.9	(27) "	365 "	101 "	93 "	131 (130)	(0.064) (")	(0.026) (")	(59.2) "	(14006) (14005)	(63)	(10.1)
- =	44.6 (47.4)	13.5	13.2	(15) "	339 "	99 "	92 "	127 (125)	(0.071) (")	(0.025) (")	(65.5) "	(12900) (12915)	(68)	(9.2)
- =	44.6 (47.0)	11.3	10.5	(37) "	375 "	103 "	95 "	133 (131)	(0.060) (")	(0.032) (")	(46.7) "	(14006) (13997)	(70)	(10.1)

* - : Experimental result

= : Prediction

* HI : Inlet gas humidity
HO : Outlet gas humidity
Cond : Condensation
IG : Inlet flue gas
OG : Outlet flue gas

QT : Total cooling duty
RH : Relative humidity
IE : Efficiency increase
IW : Inlet coolant water
OW : Outlet hot water

() : Computer outputs or calculated values

Figure 1.1 Heat Recovery Curve (Natural Gas Fired Boiler) [4]

Figure 2.1 Direct Contact Heat Recovery System [7]

Figure 2.2 Column Conditions [7]

Figure 2.3 Humidity Chart [8]

Figure 2.4 Countercurrent Direct Contact Heat Exchanger

Figure 2.5 Operating Lines of Heat Exchangers

Figure 2.6 Variation of Correction Factor a with Co [2]

Figure 2.7 Generalized Correlation for Flooding Rates in Packed Column [14]

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This thesis contains coloured
photographs & explanatory tables
which when microfilmed lose their
significance. The hardcopy of the
thesis is available upon request
from Carleton University Library.

Figure 3.1 Heat Exchanger
Equipped To Boiler

Figure 3.2 Direct Contact
Heat Exchanger

Figure 3.5 Packing Material (GEMPAK 4A)

Figure 3.2 Schematic Illustration of the System

Figure 3.4 Schematic Diagram of Heat Exchange Column

Figure 4.1 Determination of Device Correction Factor, C

Figure 4.2 Device Correction Factor vs Mass velocity Correction

Figure 4.3 - Measured vs Calculated Heat Transfer Rate, Spray Column

Figure 4.4 Operating Line for the Cooling of a Gas by Direct Contact [19]

Figure 4.5 Experimental Gas Cooling Lines
(a) Spray Column without Packing
(b) Packed Column with GEMPAK 4A

Figure 4.6 Heat Recovery Curve (No.2 Oil and Natural Gas Fired Boilers)

Figure 4.7 Efficiency Variation with Inlet Flue Gas Temperature

APPENDIX A Overall volumetric heat transfer coefficient in terms of liquid and gas phase volumetric heat transfer coefficients [10]

(a) Gas cooling with coolant vaporization

Frequently, some of coolant liquid vaporizes while gas is being cooled in direct contact gas coolers. Considering the vaporization of part of the coolant at outlet of direct contact heat exchanger, the mass transfer occurs from liquid to gas.

The overall heat transfer is :

$$Q_T = Q_l + Q_h \quad (A.1)$$

$$= Q_l + N_A \lambda_A \quad (A.2)$$

where, Q_T = total heat transfer rate (Btu/hr)

Q_h = rate due to vaporization of liquid (Btu/hr)

Q_l = sensible heating duty of liquid (Btu/hr)

N_A = mass transfer rate of liquid A (lb/hr)

λ_A = heat of vaporization of liquid A (Btu/lb)

Also,

$$Q_T = U_a V_T (t_g - t_l)_M \quad (A.3)$$

$$= a h_{ga} V_T (t_g - t_l)_M \quad (A.4)$$

$$= h_{la} V_T (t_l - t_l)_M + N_A \lambda_A \quad (A.5)$$

Where, U_a = total vol. heat transfer coefficient (Btu/hr ft³ °F)

V_T = total heat transfer volume (ft³)

h_{ga} = gas phase vol. heat transfer coeff. (Btu/hr ft³ °F)

h_{la} = liquid phase vol. heat transfer coeff. (Btu/hr ft³ °F)

t_g = gas temperature (°F)

t_l = liquid temperature ($^{\circ}\text{F}$)

t_i = interface temperature ($^{\circ}\text{F}$)

subscript M : mean

Rearranging Equations A.3 to A.5 gives :

$$\frac{1}{U_a} = \frac{1}{a h_{g,a}} + \frac{1}{h_{l,a}} - \frac{N_A \lambda_A}{Q_T h_{l,a}} \quad (\text{A.6})$$

$$= \frac{1}{a h_{g,a}} + \frac{1}{h_{l,a}} - \frac{Q_l}{Q_T} \quad (\text{A.7})$$

or

$$U_a = \frac{1}{\frac{1}{a h_{g,a}} + \left(\frac{1}{h_{l,a}} \right) \left(\frac{Q_l}{Q_T} \right)} \quad (\text{A.8})$$

In Equation A.4 the multiplier a is the "Ackerman correction" factor which is defined as follows :

$$a = \frac{Co}{1 - e^{-Co}} \quad (\text{A.9})$$

$$\text{where, } Co = \frac{N_A c_{g,A}}{V_T h_{g,a}} \quad (\text{A.10})$$

For humidification, N_A is negative and multiplier a is less than unity, since mass and heat transfer are in opposite directions.

(b) Gas cooling with partial condensation

When partial condensation of the gas stream accompanies cooling, the mass transfer occurs from gas to liquid.

The overall heat transfer is :

$$Q_T = Q_g + Q_c \quad (A.11)$$

$$= Q_g + N_A \lambda_A \quad (A.12)$$

where, Q_c = rate due to condensation of gas (Btu/hr)

N_A = mass transfer rate of condensibles in gas (lb/hr)

Also,

$$Q_T = U_a V_T (t_g - t_l)_M \quad (A.13)$$

$$= a h_g a V_T (t_g - t_i)_M + N_A \lambda_A \quad (A.14)$$

$$= h_l a V_T (t_i - t_l)_M \quad (A.15)$$

Rearranging Equations A.13 to A.15 gives :

$$\frac{1}{U_a} = \frac{1}{a h_g a} + \frac{1}{h_l a} - \frac{N_A \lambda_A}{Q_T a h_g a} \quad (A.16)$$

$$= \frac{1}{h_l a} + \frac{1}{a h_g a} \cdot \frac{Q_g}{Q_T} \quad (A.17)$$

or

$$U_a = \frac{1}{\frac{1}{h_l a} + \left(\frac{1}{a h_g a} \right) \left(\frac{Q_g}{Q_T} \right)} \quad (A.18)$$

For condensation, N_A is positive and a is greater than unity.

APPENDIX B Amount of air required for combustion [21]

The oxygen required for combustion is obtained from air. If just enough air is supplied to effect complete combustion, this weight is called theoretical air (W_{ta}). Generally, however, more than the theoretical amount of air must be supplied in order to insure good combustion. The difference between the air actually supplied and the amount theoretically required is called excess air; percent of excess air is the amount of the excess divided by the amount theoretically required.

The combustion equations for carbon, hydrogen and sulphur are as follows :

Equation	Weights (lb) per lb fuel
$C + O_2 \longrightarrow CO_2$	$1 + 2.67 = 3.67$
$H + \frac{1}{2}O \longrightarrow H_2O$	$1 + 8 = 9$
$S + O_2 \longrightarrow SO_2$	$1 + 1 = 2$

Since air contains 23.15 percent oxygen and 76.85 percent nitrogen by weight, it is evident that in order to obtain 1 lb of oxygen, $1/0.2315 = 4.32$ lb of air must be supplied. By inspection of the above equations, it is seen that 1 lb of carbon requires $8/3$ lb of oxygen or $8/3 \times 4.32 = 11.52$ lb of air; also, 1 lb of hydrogen requires 8.0 lb of oxygen or 34.56 lb of air, and 1 lb of sulphur requires 1 lb of oxygen or 4.32 lb of air. Thus :

$$W_{ta} = 11.52C + 34.56 (H - O/8) + 4.32S \quad (B.1)$$

$$= 11.52C + 4.32 (8H - O + S) \quad (B.2)$$

The quantity $O/8$ represents that portion of hydrogen in the fuel which is assumed to be already in combination with oxygen in the form of water and therefore not available for combustion, the denominator being used since eight parts by weight of oxygen are required to burn one part of hydrogen.

APPENDIX C.1 Sample calculations for design

C.1 Device correction factor 'C' for spray column

In order to determine the device correction factor 'C', 13 tests were made with a fixed size spray column. The spray column has a contact volume which consists of 0.7 ft inside diameter and 2.4 ft height.

There are three main Equations, i.e. Equations 2.2, 2.4 or 2.7, and 2.14 for spray column design. Since the column was originally designed for a packed column, the gas cooling with coolant vaporization may be assumed to be used as a spray column. This can be checked later. Therefore Equation 2.4 is selected to determine the device correction factor 'C'.

$$V_T = \frac{Q_T}{(Ua)_{avg} \Delta t_m} \quad (2.2)$$

$$Ua = \frac{1}{\frac{1}{ah_g a} + \left(\frac{1}{h_l a}\right) \left(\frac{Q_g}{Q_T}\right)} \quad (2.4)$$

$$a = \frac{C_o}{1 - e^{-C_o}} \quad (2.5)$$

$$C_o = \frac{N_A C_{g,A}}{h_g a V_T} \quad (2.6)$$

$$h_{ga} = \frac{C G^{0.8} L^{0.4}}{Z_T^{0.5}} \quad (2.14)$$

Where, a is the Ackerman factor as explained in Section 2.2. Since the gas cooling with coolant vaporization was assumed, the rate of diffusion, N_A , now has a negative value.

The liquid side heat transfer resistance, $1/h_{la}$, is much greater than the gas/phase heat transfer resistance, $1/h_{ga}$. Therefore $1/h_{la} = 0$ is assumed. Now, there are 5 unknowns, i.e. U_a , h_{ga} , a , Co , and C with 5 equations. Other quantities are evaluated from the experimental results.

The data obtained from an experiment are as follows:

Data #1 : (taken from Table 4.1)

Water flow rate : 29 GPH

Fuel flow rate : 0.87 GPH

CO₂ content in flue gas : 11.5%

Inlet gas temperature : 301 °F

Outlet gas temperature : 112 °F

Inlet water temperature : 93 °F

Outlet water temperature : 113 °F

Calculations are first performed to get quantities other than the above 5 unknowns, and finally the above 5 equations are solved. (Refer to Appendix D.1 for computer program)

Calculations :

- * Water flow rate : (GPH $\rightarrow$ lb/hr)

$$29 \text{ GPH} = 29 \text{ Gal/hr} \times 3.785 \text{ l/Gal} \times 0.992 \text{ Kg/l} \times 1\text{lb}/0.453 \text{ Kg} \\ = 240.37 \text{ lb/hr}$$

- * Fuel flow rate : (GPH $\rightarrow$ lb/hr)

$$0.87 \text{ GPH} = 0.87 \text{ Gal/hr} \times 7.130 \text{ lb/Gal} = 6.203 \text{ lb/hr}$$

- * Excess air : EX (from Equation 3.1)

$$\text{Ex} = 100 \times (15.5/\% \text{CO}_2 - 1) \\ = 100 \times (15.5/11.5 - 1) = 34.8 \%$$

- * Inlet gas humidity : (refer to Section 3.2.3)

Combustion with 34.8% excess air :

$$\text{CO}_2 = 3.194 \text{ lb/lb oil}$$

$$\text{Excess O} = (14.026 \times 0.348) \times 0.2315 = 1.130 \text{ lb/lb oil}$$

$$\text{N} = (14.026 + 14.026 \times 0.348) \times 0.7685 = 14.530 \text{ lb/lb oil}$$

$$\text{SO}_2 = 0.100 \text{ lb/lb oil}$$

$$\text{H}_2\text{O} = 1.152 \text{ lb/lb oil}$$

Weight of dry flue gas per lb oil

$$= \text{CO}_2 + \text{O} + \text{N} + \text{SO}_2$$

$$= 3.194 + 1.130 + 14.530 + 0.100$$

$$= 18.954 \text{ lb/lb oil}$$

Weight of water vapor per lb oil

$$= \text{H}_2\text{O}$$

$$= 1.152 \text{ lb/lb oil}$$

Inlet flue gas humidity

$$\begin{aligned} &= \frac{\text{weight of water vapor per lb oil}}{\text{weight of dry flue gas per lb oil}} \\ &= (1.152 \text{ lb/lb}) / (18.954 \text{ lb/lb}) \\ &= 0.0608 \text{ (lb water vapor) / (lb dry gas)} \end{aligned}$$

* Total duty : Q_T

In this case, the total duty is the same as Q_g .

$$\begin{aligned} Q_T (=Q_g) &= (\text{gas flow rate}) \times (\text{specific heat}) \times (\text{temp. diff.}) \\ &= (124.7 \text{ lb/hr}) \times (0.255 \text{ Btu/lb}^\circ\text{F}) \times (189^\circ\text{F}) \\ &= 6010 \text{ Btu/hr} \end{aligned}$$

$$\begin{aligned} * \text{ gas flow rate} &= (\text{weight of dry gas and water vapor per} \\ &\quad \text{lb oil}) \times (\text{fuel flow rate}) \\ &= (20.106 \text{ lb/lb oil}) \times (6.203 \text{ lb/hr}) \\ &= 124.7 \text{ lb/hr} \end{aligned}$$

* Liquid heating duty (sensible) : Q_L

$$\begin{aligned} Q_L &= (\text{water flow rate}) \times (\text{specific heat of water}) \\ &\quad \times (\text{temp. diff. between inlet and outlet water}) \\ &= (240.4 \text{ lb/hr}) (0.997 \text{ Btu/lb}^\circ\text{F}) (20^\circ\text{F}) \\ &= 4794 \text{ Btu/hr} \end{aligned}$$

* Humidification duty : Q_h

$$\begin{aligned} Q_h &= Q_T - Q_L \\ &= 6010 \text{ Btu/hr} - 4794 \text{ Btu/hr} \\ &= 1216 \text{ Btu/hr} \end{aligned}$$

* Outlet gas humidity:

Water gained in outlet gas

$$\begin{aligned} &= Q_h / (\text{Heat of vaporization of water}) \\ &= (1216 \text{ Btu/hr}) / (970.3 \text{ Btu/lb}) \\ &= 1.253 \text{ lb/hr} \end{aligned}$$

Weight of water humidified in the outlet gas per lb oil

$$\begin{aligned} &= (\text{water gained in outlet gas}) / (\text{fuel flow rate}) \\ &= (1.253 \text{ lb/hr}) / (6.203 \text{ lb/hr}) \\ &= 0.202 \text{ lb/lb oil} \end{aligned}$$

Outlet flue gas humidity

$$\begin{aligned} &= \frac{\text{water vapor per lb oil} + \text{water humidified per lb oil}}{\text{dry flue gas per lb oil}} \\ &= \frac{1.152 \text{ lb/lb} + 0.202 \text{ lb/lb}}{18.954 \text{ lb/lb}} \\ &= 0.0714 \text{ lb/lb} \end{aligned}$$

* Outlet water flow rate

$$\begin{aligned} &= \text{water flow rate} - \text{water gained in outlet gas} \\ &= 240.37 \text{ lb/hr} - 1.25 \text{ lb/hr} \\ &= 241.62 \text{ lb/hr} \end{aligned}$$

* Liquid mass velocity : L

$$\begin{aligned}
 L &= \frac{\text{water flow rate at outlet}}{\text{column cross section}} \\
 &= (241.62 \text{ lb/hr}) / \left(\frac{\pi}{4} \times 0.7^2 \text{ ft}^2 \right) \\
 &= 627.8 \text{ lb/hr ft}^2
 \end{aligned}$$

* Gas mass velocity : G

$$\begin{aligned}
 G &= \frac{\text{gas flow rate}}{\text{column cross section}} \\
 &= (124.7 \text{ lb/hr}) / \left(\frac{\pi}{4} \times 0.7^2 \text{ ft}^2 \right) \\
 &= 324.0 \text{ lb/hr ft}^2
 \end{aligned}$$

* Log mean temperature difference: Δt_m

$$\begin{aligned}
 \Delta t_m &= \frac{(Th_1 - Tc_2) - (Th_1 - Tc_1)}{\ln [(Th_1 - Tc_2)/(Th_1 - Tc_1)]} \\
 &= \frac{(112 - 93) - (301 - 113)}{\ln [(112 - 93)/(301 - 113)]} \\
 &= 73.7^\circ \text{F}
 \end{aligned}$$

With above calculated values the 5 equations are solved as follows:

U_a is first determined from Equation 2.2.

$$\begin{aligned}
 U_a &= Q_T / (V_T \Delta t_m) \\
 &= \frac{6010 \text{ Btu/hr}}{\left(\frac{\pi}{4} \times 0.7^2 \times 2.4 \text{ ft}^3 \right) (73.7^\circ \text{F})} \\
 &= 88.3 \text{ Btu/hr ft}^2 \text{ }^\circ \text{F}
 \end{aligned}$$

where, V_T is the contact volume of spray column.

Since $1/h_{ga} \approx 0$ is assumed compared with $1/h_{ga}$ value, Equation 2.4 is reduced to:

$$\begin{aligned} h_{ga} &= Ua/a \\ &= (88.3 \text{ Btu/hr ft}^3 \text{ }^\circ\text{F}) / a \end{aligned}$$

Now using the above h_{ga} , Equation 2.5 and 2.6 are solved simultaneously.

$$\begin{aligned} Co &= N_A C_{gA} / (h_{ga} V_T) & a &= Co / (1 - e^{-Co}) \\ &= \frac{(-1.253) (0.42)}{(88.3/a) \left(\frac{\pi}{4} \times 0.7^2 \times 2.4 \right)} & &= \frac{-0.00645 a}{1 - e^{+0.00645 a}} \\ &= -0.00645 a & \therefore a &= 0.9968 \end{aligned}$$

where, N_A is the rate of vaporization of liquid or water gained in outlet flue gas. Since mass and heat are transferred in opposite direction, N_A has a negative value and a has a value less than 1.0 as mentioned in Section 2.2.

With known value of a , h_{ga} is determined.

$$\begin{aligned} h_{ga} &= Ua/a \\ &= (88.3 \text{ Btu/hr ft}^3 \text{ }^\circ\text{F}) / 0.9968 \\ &= 88.6 \text{ Btu/hr ft}^3 \text{ }^\circ\text{F} \end{aligned}$$

Finally the device correction factor of the spray column 'C' is determined from Equation 2.14.

$$C = \frac{h_{ga} Z_T^{0.5}}{G^{0.8} L^{0.4}}$$

$$= \frac{(88.6) (2.4)^{0.5}}{(324.0)^{0.8} (627.8)^{0.4}}$$

$$= 0.1023$$

The humidification duty, Q_h , has a positive value which confirms that the assumption of the gas cooling with coolant vaporization was correct.

C.2 Spray column with required capacity

Using Equation 4.2 for gas phase heat transfer coefficient, a spray column with required capacity can be obtained easily.

There are 3 major equations, i.e. Equations 2.2, 2.7 and 4.2. The design procedure is the same as that in Appendix C.1. First, preliminary quantities in the above 3 equations are obtained from design information, then the diameter of the spray column is checked by Equation 2.13. Finally the column height of the spray column is obtained by solving the above 3 main equations simultaneously.

Design example :

Flue gases are to be cooled from 410°F to 135°F by a direct contact spray column with 95°F coolant water. A domestic oil furnace is used at 1.15 GPH of fuel consumption rate with 30% excess air. Water vapor condensation at column outlet is chosen as 70% for this example. The diameter of the column is 0.7 ft. Determine the column height.

Calculations :

In Section 3.2.3, inlet flue gas humidity was obtained as 0.0596 lb/lb for the combustion with 30% excess air. The flow rates of dry gas and water vapor in inlet gas are calculated as follows:

$$\begin{aligned}
 \text{Dry gas flow rate} &= (\text{fuel consumption rate}) \\
 &\quad \times (\text{dry gas per lb oil}) \\
 &= (1.15 \times 7.130 \text{ lb/hr}) \times (18.333 \text{ lb/lb}) \\
 &= 158.52 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 \text{Water vapor flow rate} &= (\text{fuel consumption rate}) \\
 &\quad \times (\text{water vapor per lb oil}) \\
 &= (1.15 \times 7.130 \text{ lb/hr}) \times (1.152 \text{ lb/lb}) \\
 &= 9.45 \text{ lb/hr}
 \end{aligned}$$

The fuel consumption rate, the dry gas per lb oil, and the water vapor per lb oil were calculated in Section 3.2.3.

Now a heat balance is made as follows:

* Heat balance:

	Inlet Gas at 410°F (lb/hr)	Outlet Gas at 135°F (lb/hr)
Dry Gas	158.52	158.52
Water Vapor	9.45	2.84
<hr/> Total	<hr/> 167.97	<hr/> 161.36

$$p_g \quad (\text{lb/ft}^3) \quad 0.0437$$

$$c_g \quad (\text{Btu/lb}^\circ\text{F}) \quad 0.259 \text{ avg}$$

$$\begin{aligned}
 \text{Water condensed} &= (\text{water vapor flow rate}) \\
 &\quad \times (\text{condensation rate}) \\
 &= (9.45 \text{ lb/hr}) \times (0.7) \\
 &= 6.62 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 \text{Water vapor flowrate in outlet gas} &= (\text{water vapor flow rate}) \\
 &\quad \times (1 - \text{condensation rate}) \\
 &= (9.45 \text{ lb/hr}) \times (1 - 0.7) \\
 &= 2.84 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 Q_g, \text{ sensible gas cooling duty} &= (\text{total inlet gas flow rate}) \\
 &\quad \times (\text{average specific heat of gas}) \\
 &\quad \times (\text{temperature difference}) \\
 &= (167.97 \text{ lb/hr}) \times (0.259 \text{ Btu/lb}^\circ\text{F}) \times (275^\circ\text{F}) \\
 &= 11964 \text{ Btu/hr}
 \end{aligned}$$

$$\begin{aligned}
 Q_c, \text{ condensation duty} &= (\text{water condensed at outlet}) \\
 &\quad \times (\text{heat of vaporization of water}) \\
 &= (6.62 \text{ lb/hr}) \times (970.3 \text{ Btu/lb}) \\
 &= 6423 \text{ Btu/hr}
 \end{aligned}$$

$$\begin{aligned}
 Q_T (= Q_L), \text{ total duty or water heating duty} &= Q_g + Q_c \\
 &= 11964 \text{ Btu/hr} + 6423 \text{ Btu/hr} \\
 &= 18387 \text{ Btu/hr}
 \end{aligned}$$

* Maximum outlet water temperature : (refer to Section 2.1)

(Inlet gas temperature : 410°F
 Inlet gas humidity : 0.0596 lb/lb

→ Adiabatic saturation temp. = 135°F (from Humidity Chart)

∴ Maximum outlet water temp. = 135°F

* Coolant water circulation rate

$$= \frac{(\text{Total duty})}{(\text{specific heat of water}) \times (\text{temp. diff. of water})}$$

$$\begin{aligned}
 &= \frac{18387 \text{ Btu/hr}}{(0.997 \text{ Btu/lb}^\circ\text{F}) \times (135-95)^\circ\text{F}} \\
 &= 461.5 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 * \Delta t_m &= \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln [(T_{h2} - T_{c2}) / (T_{h1} - T_{c1})]} \\
 &= \frac{(135-95) - (410-135)}{\ln [(135-95) / (410-135)]} \\
 &= 121.9^\circ\text{F}
 \end{aligned}$$

* Gas mass velocity : G

$$\begin{aligned}
 G &= \frac{\text{gas flow rate}}{\text{column cross section area}} \\
 &= (167.97 \text{ lb/hr}) / (0.385 \text{ ft}^2) \\
 &= 436.3 \text{ lb/hr ft}^2
 \end{aligned}$$

Liquid mass velocity : L

$$\begin{aligned}
 L &= \frac{\text{water flow rate at outlet}}{\text{column cross section area}} \\
 &= (468.1 \text{ lb/hr}) / (0.385 \text{ ft}^2) \\
 &= 1215.8 \text{ lb/hr ft}^2
 \end{aligned}$$

$$\begin{aligned}
 (* \text{ Water flow rate at outlet} &= \text{water circulation rate} \\
 &\quad + \text{water condensed} \\
 &= 461.5 \text{ lb/hr} + 6.6 \text{ lb/hr} \\
 &= 468.1 \text{ lb/hr})
 \end{aligned}$$

* Checking design gas velocity with maximum allowable gas velocity :

$$\begin{aligned}
 v_{\max} &= 0.20 \left(\frac{P_1 - P_2}{P_2} \right)^{1/2} \quad (2.13) \\
 &= 0.20 \left(\frac{61.8}{0.0437} \right)^{1/2} \\
 &= 7.52 \text{ ft/sec}
 \end{aligned}$$

$$\begin{aligned}
 v_{\text{design}} &= \frac{\text{inlet gas flow rate}}{\text{column cross section area}} \\
 &= (1.011 \text{ ft}^3/\text{sec}) / (0.385 \text{ ft}^2) \\
 &= 2.63 \text{ ft/sec}
 \end{aligned}$$

$$\begin{aligned}
 (* \text{ Gas flow rate} &= (168.0 \text{ lb/hr}) \times (1 \text{ ft}^3 / 0.04614 \text{ lb}) \\
 &= 3641 \text{ ft}^3/\text{hr} \\
 &= 1.011 \text{ ft}^3/\text{sec}.)
 \end{aligned}$$

The assumed column diameter, 0.7 ft, is acceptable because v_{design} is lower than $v_{\max}$. Now the column height, Z_T , is obtained from the following 5 equations.

* Column height :

$$V_T = \frac{Q_T}{U_a \Delta t_m} \quad (2.2)$$

$$U_a = \frac{1}{1/h_{1a} + (1/a h_{ga})(Q_g/Q_T)} \quad (2.4)$$

$$a = \frac{C_o}{1 - e^{-C_o}} \quad (2.5)$$

$$C_o = \frac{N_A C_{g,A}}{h_{ga} V_T} \quad (2.6)$$

$$h_{ga} = \frac{0.276 G^{0.8} L^{0.4}}{Z_T^{1.7}} \quad (4.2)$$

The Equation 2.2 defines a volumetric heat transfer coefficient and Equations 2.4 to 2.6 represent the gas cooling with partial condensation. In Equation 2.4, $1/h_{ga}$ is ignored compared with $1/h_a$. Now N_A in Equation 2.6 is the rate of condensation of water in flue gas. Since mass and heat are transferred in the same direction, N_A has a positive value and α has a value larger than 1.0. Equation 4.2 is an experimental correlation.

The above five equations with five unknowns, i.e. V_T , U_a , α , Co , and h_{ga} can be solved simultaneously. Computer program can treat solving of these equations easily. Refer to Appendix D.2 for computer program.

Referring to the computer program result, Z_T is 3.6 ft. Therefore, if the column height is increased from 2.4 ft to 3.6 ft, a suitable amount of latent heat can be obtained.

2 of/de 2

MicroD

C.3 Packed column with GEMPEK 4A as packing

There are 4 major equations, i.e. Equations 2.2, 2.7, 2.15 and 2.16. The design procedure is the same as that of a spray column with required capacity in Appendix C.2. First, preliminary quantities in the main 4 equations are obtained from design information, then the diameter of the packed column is checked by the generalized pressure drop correlation in Figure 2.7. Finally the column height of the packed column is obtained by solving the 4 main equations simultaneously.

Design example :

Flue gases are to be cooled from 450°F to 100°F by a direct contact packed column with 90°F coolant water. A domestic oil furnace is used at 1 Gal/hr of fuel consumption rate with 30% excess air. Water vapor condensation at column outlet is chosen as 70% for example. The diameter of the column is 0.7ft. Determine the column height. GEMPAK 4A is used as packing material.

* GEMPAK 4A :

Surface area : $137.1 \text{ ft}^2/\text{ft}^3$

Density : $14.5 \text{ lb}/\text{ft}^3$

Free space : 97.2%

Packing factor : 32

Calculations :

The required data for the 4 main equations are first calculated as the same way as in Appendix C.2. In Section 3.2.3, inlet flue gas humidity was obtained as 0.0596 lb/lb for the combustion with 30% excess air. The flow rates of dry gas and water vapor in inlet gas are calculated as follows :

$$\begin{aligned}
 \text{Dry gas flow rate} &= (\text{fuel consumption rate}) \\
 &\quad \times (\text{dry gas per lb oil}) \\
 &= (7.130 \text{ lb/hr}) \times (19.333 \text{ lb/lb}) \\
 &= 137.84 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 \text{Water vapor flow rate} &= (\text{fuel consumption rate}) \\
 &\quad \times (\text{water vapor per lb oil}) \\
 &= (7.130 \text{ lb/hr}) \times (1.152 \text{ lb/lb}) \\
 &= 8.21 \text{ lb/hr}
 \end{aligned}$$

The fuel consumption rate, the dry gas per lb oil and the water vapor per lb oil were calculated in Section 3.2.3.

Now heat balance is made as follows :

* Heat balance :

	Inlet Gas at 450°F (lb/hr)	Outlet Gas at 100°F (lb/hr)
Dry Gas	137.84	137.84
Water Vapor	8.21	2.46
<hr/>	<hr/>	<hr/>
Total	146.05	140.30
P_g (lb/ft ³)	0.0437	
c_g (Btu/lb°F)		0.259 avg

Water condensed = (water vapor flow rate)

x (condensation rate)

= (8.21 lb/hr) x (0.7)

= 5.75 lb/hr

Water vapor content in outlet gas = (water vapor flow rate)

x (1 - condensation rate)

= (8.21 lb/hr) x (1 - 0.7)

= 2.46 lb/hr

Q_g, sensible gas cooling duty = (total inlet gas flow rate)

x (average specific heat of gas)

x (Temperature difference)

= (146.05 lb/hr) x (0.259 Btu/lb°F) x (350°F)

= 13239 Btu/hr

Q_c, condensation duty = (water condensed at outlet)

x (heat of vaporization of water)

= (5.75 lb/hr) x (970.3 Btu/lb)

= 5579 Btu/hr

Q_T (= Q₁), total duty or water heating duty = Q_g + Q_c

= 13239 Btu/hr + 5579 Btu/hr

= 18818 Btu/hr

* Maximum outlet water temperature : (refer to Section 2.1)

(Inlet gas temperature : 450°F

Inlet gas humidity : 0.0596 lb water vapor/lb dry gas

→ Adiabatic saturation temp. = 140°F (From Humidity Chart)

∴ Maximum outlet water temp. = 140°F

* Coolant water circulation rate

$$\begin{aligned}
 &= \frac{\text{(total duty)}}{\text{(specific heat of water)} \times \text{(water temp. difference)}} \\
 &= \frac{18818 \text{ Btu/hr}}{(0.997 \text{ Btu/lb}^\circ\text{F}) \times [(140 - 90)^\circ\text{F}]} \\
 &= 377.5 \text{ lb/hr}
 \end{aligned}$$

$$\begin{aligned}
 * \Delta t_m &= \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln [(T_{h2} - T_{c2}) / (T_{h1} - T_{c1})]} \\
 &= \frac{(100 - 90) / (450 - 140)}{\ln [(100 - 90) / (450 - 140)]} \\
 &= 87.4^\circ\text{F}
 \end{aligned}$$

* Gas mass velocity : G

$$\begin{aligned}
 G &= \frac{\text{gas flow rate}}{\text{column cross section area}} \\
 &= \frac{146.05 \text{ lb/hr}}{0.385 \text{ ft}^2} \\
 &= 379.4 \text{ lb/hr ft}^2
 \end{aligned}$$

* Liquid mass velocity : L

$$L = \frac{\text{water flow rate at outlet}}{\text{column cross section area}}$$

$$= \frac{383.3 \text{ lb/hr}}{0.385 \text{ ft}^2}$$

$$= 995.6 \text{ lb/hr ft}^2$$

(* water flow rate at outlet

= coolant water circulation rate

+ water condensed

$$= 37.50 \text{ lb/hr} + 5.8 \text{ lb/hr}$$

$$= 383.3 \text{ lb/hr}$$

Now column gas capacity is compared with its limit value based on the generalized pressure drop correlation (see Section 2.4.2).

The flow parameter shown in Figure 2.7 is first calculated.

$$\frac{L}{G} \left(\frac{\rho_g}{\rho_l} \right)^{1/2} = \frac{995.6}{379.4} \left(\frac{0.0437}{61.8} \right)^{1/2} = 0.0698$$

From the same figure, for the flow parameter 0.0698 :

$$\frac{G_F^2 F_p (\mu_l / 2.42)^{0.2}}{g \rho_g \rho_l} = 0.157$$

where, G_F = max. allowable gas velocity for packing (lb/sec ft²)

F_p = Packing factor

μ_l = viscosity of liquid (lb/ft hr) = 1.43 for water at 100°F

g = conversion constant, 32.2 (lb-mass ft/lb-force sec²)

ρ_g = density of gas (lb/ft³) = 0.0437 for dry gas at 450°F

ρ_l = density of liquid (lb/ft³) = 61.8 for water at 100°F

Rearranging the above equation for G gives :

$$G_F^2 = \frac{(0.157) (32.2) (0.0437) (61.8)}{F_p (1.43/2.42)^{0.2}} = \frac{15.17}{F_p}$$

The packing factor for GEMPAK 4A is 32, given by Table 3.2.

Substituting 32 into F_p in the above equation :

$$G_F = \sqrt{\frac{15.17}{32}} = 0.689 \text{ lb/sec ft}^2$$

$$= 2479 \text{ lb/hr ft}^2$$

Therefore the maximum allowable gas mass velocity for this packed column is 2479 lb/hr ft². Now comparing gas mass flow rate, G , with G_F , the percent flooding is determined.

$$\text{Percent flooding} = \frac{G}{G_F} = \frac{379.4 \text{ lb/hr ft}^2}{2479 \text{ lb/hr ft}^2} = 0.1530 = 15\%$$

The pressure drop can also be found in the Figure 2.7, Substituting $G_F = G/0.1530$ into the equation for G_F^2 .

$$\frac{G^2 F_p (\mu_l/2.42)^{0.2}}{\rho_g \rho_l} = 0.00368$$

In the Figure 2.17, the pressure drop is found about 0.04 in of water per foot of column height.

Normally the packed column is designed to operate at the highest economic pressure drop, to ensure good liquid and gas distribution. The pressure drop is directly dependent on packing factor. Recommended design values of pressure drop for absorbers and strippers are between 0.18 and 0.60 in of water per foot of column height [15]. But these values are rather suitable for the classic packed column using packing material like Raschig rings or Berl saddles.

Since GEMPAK 4A, which is a recently developed packing material, has high free surface, the above recommended pressure drop value may not be very useful. Therefore the pressure drop 0.04 in of water per foot of column height is maintained in this design. However, if a compact and more pressure drop column is desired, this can be done by decreasing the column diameter. For ultimate design of a packed column, see Section 4.2.2.

Now the column height, Z_T , is obtained by solving the following 4 main equations.

$$V_T = \frac{Q_T}{(Ua)_{avg} \Delta t_m} \quad (2.2)$$

$$Ua = \frac{1}{\frac{1}{h_L a} + \left(\frac{1}{a h_g a} \right) \left(\frac{Q_g}{Q_T} \right)} \quad (2.7)$$

$$a = \frac{Co}{1 - e^{-Co}} \quad (2.5)$$

$$Co = \frac{N_A c_{g,A}}{h_{g,a} V_T} \quad (2.6)$$

$$h_{g,a} = \left(\frac{Sc_g}{Pr_g} \right)^{2/3} \frac{c_g G}{Hg} \quad (2.15)$$

$$Hg = \frac{\psi Sc_g^{0.5} D^n}{(L f_1 f_2 f_3)^m} \cdot \left(\frac{Z_T}{10} \right)^{1/3} \quad (2.16)$$

The Equation 2.2 defines a volumetric heat transfer coefficient in terms of column contact volume, total cooling duty and logarithmic temperature difference. Equation 2.7 is applied to gas cooling with partial condensation. The term $1/h_{l,a}$ is ignored compared with $1/h_{g,a}$. N_A in Equation 2.6 is the rate of condensation of water in flue gas. Since mass and heat are transferred in the same direction, N_A has a positive value and a has a value larger than 1.0. Equations 2.15 and 2.16 are experimental correlations developed for packing materials of Raschig rings and Berl saddles.

Necessary packing material data, i.e. constants in Equations 2.15 and 2.16, are adopted from the known values of Raschig rings or Berl saddles which have experimental data in the wide range of their sizes (see Table 2.2). Contact surface area should be a main factor in selecting the proper data. Comparing the properties of GEMPAK 4A in Table 3.2 with those of Raschig rings in Table 2.2, 1/2 in Raschig rings has similar contact surface area. Therefore the data for 1/2 in Raschig

rings in Table 2.2 are used.

Now there are 6 equations with 6 unknowns, i.e. U_a , $h_g a$, a , Co , Hg , and Z_T . Therefore the equations are solved simultaneously. Computer program can treat solving of these equations easily. Refer to Appendix D.3 for computer program.

Referring to the computer program result, Z_T was obtained as 2.4 ft.

APPENDIX D COMPUTER PROGRAMS

D.1 Device correction factor 'C' for spray column

Flow chart

Nomenclature

ALPA:	Correction factor for simultaneous heat and mass transfer (Eq. 2.5)
C:	Device correction factor for spray column
CD:	Amount of carbon dioxide (%)
DTM:	Log mean temperature difference ($^{\circ}\text{F}$)
EX:	Amount of excess air
FFR:	Fuel flow rate (GPH)
G:	Gas mass velocity (lb/hr ft^2)
GIT:	Flue gas inlet temp. ($^{\circ}\text{F}$)
GOT:	Flue gas outlet temp. ($^{\circ}\text{F}$)
HGA:	Gas phase volumetric heat transfer coefficient ($\text{Btu/hr ft}^3 \text{F}$)
HI:	Inlet gas humidity (lb/lb)
HO:	Outlet gas humidity (lb/lb)
L:	Liquid mass velocity (lb/hr ft^2)
QH:	Humidification duty (Btu/hr)
QL:	Liquid heating duty (Btu/hr)
QT:	Total cooling duty (Btu/hr)
WFR:	Water flow rate (GPH)
WIT:	Inlet water temperature ($^{\circ}\text{F}$)
WOT:	Outlet water temperature ($^{\circ}\text{F}$)

```

" <<< DEVICE CORRECTION FACTOR 'C' FOR SPRAY COLUMN >>>
20 INPUT "WATER FLOW RATE (GPH), FUEL FLOW RATE (GPH) : "; WFR, FFR
30 INPUT "CARBON DIOXIDE (%): "; CD
40 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) : "; GIT, GOT
50 INPUT "WATER INLET, OUTLET TEMP (DEGREE F) : "; WIT, WOT
70 WFR=8.289*WFR : REM Water flow rate unit to lb/hr
80 FFR=7.12*FFR : REM Fuel flow rate unit to lb/hr
100 EX=15.5/CD-1 : REM Excess air
110 REM ----- Inlet gas humidity -----
120 H1=3.194+14.026*EX*.2315+14.126*(1+EX)*.7685+.1
130 H2=1.152
140 HI=H2/H1 : REM Inlet gas humidity
150 REM ----- Heat transfer duties -----
160 QT=(H1+H2)*FFR*.255*(GIT-GOT)
170 QL=WFR*.997*(WOT-WIT)
180 QH=QT-QL : REM Humidification duty
190 GW=QH/970.3 : REM Gained water in the outlet gas
200 WW=GW/FFR : REM Weight of water humidified
210 HU=(H2+WW)/H1
220 L=(WFR-GW)/(3.141592/4*.7^2)
230 G=(H1+H2)*FFR/(3.141592/4*.7^2)
240 DTM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
250 REM ----- Calculation of HGA -----
260 UA=QT/((.9236*DTM))
270 U1=(GW*.42)/(UA*.9236)
280 ALPA=LOG(1+U1)/U1
290 HGA=UA/ALPA
300 C=2.4*.5*HGA/(G^.8*L^.4)
310 PRINT "-----"
320 PRINT "*** Liquid heating duty (QL, BTU/HR) : "; QL
330 PRINT "*** Humidification duty (QH, BTU/HR) : "; QH
340 PRINT "*** Total cooling duty (QT, BTU/HR) : "; QT
350 PRINT
360 PRINT "*** Liquid mass velocity (L, LB/HR-SQ.FT) : "; L
370 PRINT "*** Gas mass velocity (G, LB/HR-SQ.FT) : "; G
380 PRINT
382 PRINT "*** Inlet gas humidity (HI, LB/LB) : "; HI
384 PRINT "*** Outlet gas humidity (HO, LB/LB) : "; HO
386 PRINT
390 PRINT "*** Correction factor (ALPA) : "; ALPA
400 PRINT "*** Gas phase vol heat transfer coeff (HGA, BTU/HR-CU.FT-F) : "; H
410 PRINT
420 PRINT "***** CORRECTION FACTOR for HGA (C) : "; C

```

Sample Output

run

DEVICE CORRECTION FACTOR 'C' FOR SPRAY COLUMN >>>
 WATER FLOW RATE (GPH), FUEL FLOW RATE (GPH) : ? 29,.87
 CARBON DIOXIDE (%) : ? 11.5
 FLUE GAS INLET, OUTLET TEMP (DEGREE F) : ? 301,112
 WATER INLET, OUTLET TEMP (DEGREE F) : ? 93,113

 *** Liquid heating duty (QL, BTU/HR) : 4793.197
 *** Humidication duty (QH, BTU/HR) : 1247.907
 *** Total cooling duty (QT, BTU/HR) : 6041.104

 *** Liquid mass velocity (L, LB/HR-SQ.FT) : 621.2758
 *** Gas mass velocity (G, LB/HR-SQ.FT) : 325.7081

 *** Inlet gas humidity (HI, LB/LB) : 6.045603E-02
 *** Outlet gas humidity (HO, LB/LB) : 7.133667E-02

 *** Correction factor (ALPA) : .9967176
 *** Gas phase vol heat transfer coeff (HGA, BTU/HR-CU.FT-F) : 88.99973

 **** CORRECTION FACTOR for HGA (C) : .1027822
 OK

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LPT1 7TRON 8TROFF9KEY 0SCREEN

run

DEVICE CORRECTION FACTOR 'C' FOR SPRAY COLUMN >>>
 WATER FLOW RATE (GPH), FUEL FLOW RATE (GPH) : ? 29,.93
 CARBON DIOXIDE (%) : ? 11.5
 FLUE GAS INLET, OUTLET TEMP (DEGREE F) : ? 365,128
 WATER INLET, OUTLET TEMP (DEGREE F) : ? 97,125

 *** Liquid heating duty (QL, BTU/HR) : 6710.476
 *** Humidication duty (QH, BTU/HR) : 1387.315
 *** Total cooling duty (QT, BTU/HR) : 8097.791

 *** Liquid mass velocity (L, LB/HR-SQ.FT) : 620.9024
 *** Gas mass/velocity (G, LB/HR-SQ.FT) : 348.1707

 *** Inlet gas humidity (HI, LB/LB) : 6.045603E-02
 *** Outlet gas humidity (HO, LB/LB) : 7.177179E-02

 *** Correction factor (ALPA) : .9962404
 *** Gas phase vol heat transfer coeff (HGA, BTU/HR-CU.FT-F) : 86.18191

 **** CORRECTION FACTOR for HGA (C) : 9.437969E-02
 OK

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LPT1 7TRON 8TROFF9KEY 0SCREEN

D.2 Spray column with required capacity

Flow chart

Nomenclature

ALPA: Correction factor for simultaneous heat and mass transfer (Eq.2.5)
 CC: Condensation rate desired (%)
 CG: Specific heat of gas (Btu/lb°F)
 DTM: Log mean temp. difference (°F)
 EX: Amount of excess air
 FFR: Fuel flow rate (GPH)
 G: Gas mass velocity (lb/hr ft²)
 GD: Inlet gas density (lb/ft³)
 GIT: Flue gas inlet temp. (°F)
 GOT: Flue gas outlet temp. (°F)
 HGA: Gas phase volumetric heat transfer coefficient (Btu/hr ft³°F)
 HI: Inlet gas humidity (lb/lb)
 ID: Column diameter (ft)
 L: Liquid mass velocity (lb/hr ft²)
 QC: Condensation duty (Btu/hr)
 QG: Sensible gas cooling duty (Btu/hr)
 QT: Total duty (Btu/hr)
 VD: Design gas velocity (ft/sec)
 VT: Maximum allowable gas velocity (ft/sec)

WIT: Inlet water temperature (°F)
 WOT: Adiabatic saturation temperature of inlet gas (°F)
 WR: Coolant water circulation rate (lb/hr)
 ZT: Column height (ft)

Computer Program

```

10 PRINT "  SPRAY COLUMN WITH REQUIRED CAPACITY >>>"
20 INPUT "FUEL FLOW RATE (GPH) : "; FFR
30 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) : "; GIT, GOT
40 INPUT "WATER INLET TEMP (DEGREE F) : "; WIT
50 INPUT "AMOUNT OF EXCESS AIR (%): "; EX
60 INPUT "DESIRED CONDENSATION (%): "; CC
70 INPUT "INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : "; GD, CG
80 INPUT "COLUMN DIAMETER (FT) : "; ID
100 FFR=7.13*FFR : REM Fuel flow rate unit to lb/hr
110 EX=EX/100 : CC=CC/100 : PI=3.141592
120 REM ----- Inlet gas humidity -----
130 H1=3.194+14.026*EX*.2315+14.026*(1+EX)*.7685+.1
140 H2=1.152
150 H1=H2/H1 : REM Inlet gas humidity
160 REM ----- Heat transfer duties -----
170 G1=H1*FFR : REM Weight of dry flue gas
180 W1=H2*FFR : REM Weight of water in inlet flue gas
190 WC=W1*CC : REM Weight of condensed water
200 QG=(G1+W1)*CG*(GIT-GOT)
210 QC=WC*970.3 : REM Condensation duty
220 QT=QG+QC : REM Total duty
230 REM ----- Outlet water temperature -----
240 PRINT "Inlet gas temp (degree F) : "; GIT; " , Humidity (lb/lb) : "; H1
250 INPUT "ADIABATIC SATURATION TEMP (DEGREE F) : "; WOT
260 REM ----- Water circulation rate -----
270 WR=QT/((.997*(WOT-WIT)) : REM Water circulation rate
280 REM ----- Checking with maximum gas velocity -----
290 VT=.2*((61.8-GD)/GD)^.5
300 VD=(G1+W1)/(GD*3600)
310 IF VD > VT THEN PRINT "GAS VELOCITY TOO BIG, ADJUST COLUMN DIA":END
320 REM ----- Calculation of column height -----
330 G=(G1+W1)/(PI*ID^2/4)
340 L=(WR+WC)/(PI*ID^2/4)
350 DTM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
360 ZT=.5
370 HGA=.276*G^.8*L^.4/(ZT^1.7)
380 CQ=(WC*.42)/(HGA*PI*ID^2*ZT/4)
390 ALFA=CQ/(1-2.71828^(-CQ))
400 UA=ALFA*HGA*QT/QG
410 TZ=QT/(UA*DTM*PI*ID^2/4)
420 IF ABS(TZ-ZT)>.05 THEN ZT=ZT+.05 : GOTO 370
465 PRINT "-----"
470 PRINT "*** Sensible cooling duty (QG, BTU/HR) : "; QG
480 PRINT "*** Condensation duty (QC, BTU/HR) : "; QC
490 PRINT "*** Total duty (QT, BTU/HR) : "; QT
500 PRINT
510 PRINT "*** Max outlet water temp (WOT, F) : "; WOT
520 PRINT "*** Water circulation rate (WR, LB/HR) : "; WR
530 PRINT
540 PRINT "*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : "; HGA
550 PRINT "*** Correction factor (ALFA) : "; ALFA
560 PRINT
570 PRINT "***** COLUMN HEIGHT (ZT, FT) : "; ZT

```

Sample Output

<<< SPRAY COLUMN WITH REQUIRED CAPACITY >>>

FUEL FLOW RATE (GPH) : ? 1.15

FLUE GAS INLET, OUTLET TEMP (DEGREE F) : ? 410,135

WATER INLET TEMP (DEGREE F) : ? 95

AMOUNT OF EXCESS AIR (%) : ? 30

DESIRED CONDENSATION (%) : ? 70

INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : ? .0437,.259

COLUMN DIAMETER (FT) : ? .7

Inlet gas temp (degree F) : 410 ,Humidity (lb/lb).: .063017

ADIABATIC SATURATION TEMP (DEGREE F) : ? 135

*** Sensible cooling duty (QG, BTU/HR) : 11348.93

*** Condensation duty (QC, BTU/HR) : 6415.698

*** Total duty (QT, BTU/HR) : 17764.62

*** Max outlet water temp (WOT, F) : 135

*** Water circulation rate (WR, LB/HR) : 445.452

*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : 68.80006

*** Correction factor (ALPA) : 1.015058

**** COLUMN HEIGHT (ZT, FT) : 3.499998

Ok

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LPT1 7TRON 8TROFF9KEY 0SCREEN

<<< SPRAY COLUMN WITH REQUIRED CAPACITY >>>

FUEL FLOW RATE (GPH) : ? 1.15

FLUE GAS INLET, OUTLET TEMP (DEGREE F) : ? 410,135

WATER INLET TEMP (DEGREE F) : ? 95

AMOUNT OF EXCESS AIR (%) : ? 41

DESIRED CONDENSATION (%) : ? 70

INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : ? .0437,.259

COLUMN DIAMETER (FT) : ? .7

Inlet gas temp (degree F) : 410 ,Humidity (lb/lb) : 5.811244E-02

ADIABATIC SATURATION TEMP (DEGREE F) : ? 134

*** Sensible cooling duty (QG, BTU/HR) : 12249.97

*** Condensation duty (QC, BTU/HR) : 6415.698

*** Total duty (QT, BTU/HR) : 18665.67

*** Max outlet water temp (WOT, F) : 134

*** Water circulation rate (WR, LB/HR) : 480.0471

*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : 71.80326

*** Correction factor (ALPA) : 1.014025

**** COLUMN HEIGHT (ZT, FT) : 3.599998

Ok

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LPT1 7TRON 8TROFF9KEY 0SCREEN

D.3 Packed column with GEMPAK 4A as packing

Flow chart

Nomenclature

ALPA: Correction factor for simultaneous heat and mass transfer (Eq.2.5)

CC: Condensation rate desired

CG: Specific heat of gas (Btu/lb^oF)

DTM: Log mean temp. difference (°F)

EX: Amount of excess air

FFR: Fuel flow rate (GPH)

G: Gas mass velocity (lb/hr ft²)

GD: Inlet gas density (lb/ft³)

GIT: Flue gas inlet temp. (°F)

GOT: Flue gas outlet temp. (°F)

HGA: Gas phase volumetric heat transfer coefficient (Btu/hr ft³ °F)

HI: Inlet gas humidity (lb/lb)

ID: Column diameter (ft)

L: Liquid mass velocity (lb/hr ft²)

QC: Condensation duty (Btu/hr)

QG: Sensible gas cooling duty (Btu/hr)

QT: Total duty (Btu/hr)

WIT: Inlet water temp. (°F)

WOT: Adiabatic saturation temp. of inlet gas (°F)

WR: Coolant water circulation rate (lb/hr)

ZT: Column height (ft)

Computer Program

```

10 PRINT "<<< PACKED COLUMN WITH GEMPAK 4A AS PACKING >>>"
20 INPUT "FUEL FLOW RATE (GPH) :";FFR
30 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) :";GIT,GOT
40 INPUT "WATER INLET TEMP (DEGREE F) :";WIT
50 INPUT "AMOUNT OF EXCESS AIR (%) :";EX
60 INPUT "DESIRED CONDENSATION (%) :";CC
70 INPUT "INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) :";GD,CG
80 INPUT "COLUMN DIAMETER (FT) :";ID
100 FFR=7.13*FFR :REM Fuel flow rate unit to lb/hr
110 EX=EX/100 :CC=CC/100 :PI=3.141592
120 REM ----- Inlet gas humidity -----
130 H1=3.174+14.026*EX*.2315+14.026*(1+EX)*.7685+.1
140 H2=1.152
150 HI=H2/H1 :REM Inlet gas humidity
160 REM ----- Heat transfer duties -----
170 G1=H1*FFR :REM Weight of dry flue gas
180 W1=H2*FFR :REM Weight of water in inlet flue gas
190 WC=W1*CC :REM Weight of condensed water
200 QG=(G1+W1)*CG*(GIT-GOT)
210 QC=WC*970.3 : REM Condensation duty
220 QT=QG+QC : REM Total duty
230 REM ----- Outlet water temperature -----
240 PRINT "Inlet gas temp (degree F) : ";GIT;" ,Humidity (lb/lb) : ";HI
250 INPUT "ADIABATIC SATURATION TEMP (DEGREE F) :"; WOT
260 REM ----- Water circulation rate -----
270 WR=QT/((.997*(WOT-WIT)) : REM Water circulation rate
320 REM -----Calculation of column height -----
330 G=(G1+W1)/(PI*ID^2/4)
340 L=(WR+WC)/(PI*ID^2/4)
350 DTM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
355 H0=(1-CC)*HI
360 ZT=1.1
370 A1=110*.6*.5*.7*1.24*10^(-.333)/(.98*.6*L^.6)
380 A2=.872*.667*CG*G
390 B1=WC*.42/(PI*ID^2/4)
400 B2=B1/((A2/A1)*ZT^.667)
410 B3=(1-2.71828^(-B2))/B2
420 B4=B3*QG*(A1/A2)/(PI/4*ID^2*DTM)
430 C1=ZT
440 C2=B4^1.5
450 IF ABS(C1-C2) > .1 THEN ZT=ZT+.1 : GOTO 370
462 HGA=(A2/A1)*ZT^(-.333) : ALPA=1/B3
465 PRINT "-----"
470 PRINT "*** Sensible cooling duty (QG, BTU/HR) : ";QG
480 PRINT "*** Condensation duty (QC, BTU/HR) : ";QC
490 PRINT "*** Total duty (QT, BTU/HR) : ";QT
500 PRINT
510 PRINT "*** Max outlet water temp (WOT, F) : ";WOT
520 PRINT "*** Water circulation rate (WR,LB/HR) : ";WR
530 PRINT
540 PRINT "*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : ";HGA
550 PRINT "*** Correction factor (ALPA) : ";ALPA
560 PRINT
570 PRINT "**** COLUMN HEIGHT (ZT, FT) : ";ZT

```

Sample Output

```

PACKED COLUMN WITH GEMPAK 4A AS PACKING >>>
FUEL FLOW RATE (GPH) : 2.1
FLUE GAS INLET, OUTLET TEMP (DEGREE F) : 450, 100
WATER INLET TEMP (DEGREE F) : 90
AMOUNT OF EXCESS AIR (%) : 30
DESIRED CONDENSATION (%) : 70
INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : .0437, .259
COLUMN DIAMETER (FT) : .7
Inlet gas temp (degree F) : 450 , Humidity (lb/lb) : .063017
ADIABATIC SATURATION TEMP (DEGREE F) : 140
-----
*** Sensible cooling duty (QG, BTU/HR) : 12560.08
*** Condensation duty (QC, BTU/HR) : 5578.867
*** Total duty (QT, BTU/HR) : 18138.94

*** Max outlet water temp (WOT, F) : 140
*** Water circulation rate (WR, LB/HR) : 363.8705

*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : 152.1178
*** Correction factor (ALPA) : 1.008619

**** COLUMN HEIGHT (ZT, FT) : 2.4
(0)

11LIST 2RUN 3LOAD 4SAVE 5CONT 6,"LPT1 7TRON 8TROFF 9KEY 0SCREEN

```

```

PACKED COLUMN WITH GEMPAK 4A AS PACKING >>>
FUEL FLOW RATE (GPH) : 2.1
FLUE GAS INLET, OUTLET TEMP (DEGREE F) : 365, 101
WATER INLET TEMP (DEGREE F) : 95
AMOUNT OF EXCESS AIR (%) : 30
DESIRED CONDENSATION (%) : 93
INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : .0437, .259
COLUMN DIAMETER (FT) : .7
Inlet gas temp (degree F) : 365 , Humidity (lb/lb) : .063017
ADIABATIC SATURATION TEMP (DEGREE F) : 135
-----
*** Sensible cooling duty (QG, BTU/HR) : 9473.887
*** Condensation duty (QC, BTU/HR) : 7411.924
*** Total duty (QT, BTU/HR) : 16885.81

*** Max outlet water temp (WOT, F) : 135
*** Water circulation rate (WR, LB/HR) : 423.4156

*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : 169.2003
*** Correction factor (ALPA) : 1.010749

**** COLUMN HEIGHT (ZT, FT) : 2.3
(0)

11LIST 2RUN 3LOAD 4SAVE 5CONT 6,"LPT1 7TRON 8TROFF 9KEY 0SCREEN

```

D.4 Packed column, result

Flow chart

Nomenclature

CD: Amount of carbon dioxide (%)
 CG: Specific heat of gas (Btu/lb^oF)
 EX: Amount of excess air
 FFR: Fuel flow rate (GPH)
 G: Gas mass velocity (lb/hr ft²)
 GD: Inlet gas density (lb/ft³)
 GIT: Flue gas inlet temp. (°F)
 GOT: Flue gas outlet temp. (°F)
 HC: Condensation rate at gas outlet (%)
 HI: Inlet gas humidity (lb/lb)
 Ho: Outlet gas humidity (lb/lb)
 ID: Column diameter (ft)
 L: Liquid mass velocity (lb/hr ft²)
 QC: Condensation duty (Btu/hr)
 QG: Sensible gas cooling duty (Btu/hr)
 QT: Total duty (Btu/hr)
 WIT: Inlet water temp. (°F)
 WFR: Coolant water flow rate (GPH)
 WOT: Outlet water temp. (°F)

```

10 PRINT " << PACKED COLUMN, RESULT >>>"
20 INPUT "WATER FLOW RATE (GPH), FUEL FLOW RATE (GPH) : "; WFR, FFR
30 INPUT "CARBON DIOXIDE (%) : "; CD
40 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) : "; GIT, GOT
50 INPUT "WATER INLET, OUTLET TEMP (DEGREE F) : "; WIT, WOT
55 INPUT "INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTB/LB-F) : "; GD, CG
60 INPUT "COLUMN DIAMETER (FT) : "; ID
65 PI=3.141592
70 WFR=8.289*WFR : REM Water flow rate unit to lb/hr
80 FFR=7.13*FFR : REM Fuel flow rate unit to lb/hr
100 EX=15.5/CD -1 : REM Excess air
110 REM ----- Inlet GAS HUMIDITY -----
120 H1=1.194 +14.062*EX*.2315 +14.062*(1+EX)*.7685 +.1
130 H2=1.152
140 HI=H2/H1 : REM Inlet gas humidity.
150 REM ----- Heat transfer duties -----
160 QG=(H1+H2)*FFR*CG*(GIT-GOT)
170 QT=WFR*.977*(WOT-WIT)
180 QC=QT-QG : REM Condensation duty
190 GW=QC/770.3 : REM Gained water in the outlet water
200 WW=GW/FFR : REM Weight of water condensed per lb-oil
210 HO=(H2-WW)/H1 : REM Outlet gas humidity
215 HC=100*(HI-HO)/HI
220 L=(WFR+GW)/(3.141592/4*ID^2)
230 G=(H1+H2)*FFR/(3.141592/4*ID^2)
240 DIM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
250 WR=WFR+GW
400 PRINT "-----"
410 PRINT "*** Exess air (EX,%) : "; EX
420 PRINT "*** Gas mass velocity (G, LB/HR SQ-FT) : "; G
430 PRINT "*** Liquid mass velocity (L, LB/HR SQ-FT) : "; L
440 PRINT "*** Outlet water temperature (WOT, DEGREE F) : "; WOT
450 PRINT "*** Inlet gas humidity (HI, LB/LB) : "; HI
460 PRINT "*** Outlet gas humidity (HO, LB/LB) : "; HO
470 PRINT "*** Condensation (HC, %) : "; HC
480 PRINT "*** Sensible cooling duty (QG, BTU/HR) : "; QG
490 PRINT "*** Condensation duty (QC, BTU/HR) : "; QC
500 PRINT "*** Total duty (QT, BTU/HR) : "; QT

```

Sample Output.

RUN

<<< PACKED COLUMN, RESULT >>>

WATER FLOW RATE (GPH); FUEL FLOW RATE (GPH) : ? 44.6,1

CARBON DIOXIDE (%) : ? 12.2

FLUE GAS INLET, OUTLET TEMP (DEGREE F) : ? 370,102

WATER INLET, OUTLET TEMP (DEGREE F) : ? 94,133

INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTB/LB-F) : ? .0437,.259

COLUMN DIAMETER (FT) : ? .7

*** Exess air (EX,%) : .2704919

*** Gas mass velocity (G, LB/HR SQ-FT) : 353.055

*** Liquid mass velocity (L, LB/HR SQ-FT) : 973.8575

*** Outlet water temperature (WOT, DEGREE F) : 133

*** Inlet gas humidity (HI, LB/LB) : 6.434208E-02

*** Outlet gas humidity (HO, LB/LB) : 2.443191E-02

*** Condensation (HC, %) : 62.02811

*** Sensible cooling duty (QG, BTU/HR) : 9431.108

*** Condensation duty (QC, BTU/HR) : 4943.523

*** Total duty (QT, BTU/HR) : 14374.63

OK

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LPT1 7TRON 8TROFF9KEY 0SCREEN

D.5 Packed column, predicted

Flow chart

Nomenclature

CC: Condensation rate desired (%)
 CG: Specific heat of gas (Btu/lb°F)
 DTM: Log mean temperature difference (°F)
 EX: Amount of excess air (%)
 FFR: Fuel flow rate (GPH)
 G: Gas mass velocity (lb/hr ft²)
 GD: Inlet gas density (lb/ft³)
 GIT: Flue gas inlet temp. (°F)
 GOT: Flue gas outlet temp. (°F)
 HI: Inlet gas humidity (lb/lb)
 HO: Outlet gas humidity (lb/lb)
 ID: Column diameter (ft)
 L: Liquid mass velocity (lb/hr ft²)
 QC: Condensation duty (Btu/hr)
 QG: Sensible gas cooling duty (Btu/hr)
 QT: Total duty (Btu/hr).
 WIT: Inlet water temp. (°F)
 WOT: Outlet water temp. (°F)
 WR: Coolant water circulation rate (GPH)
 ZT: Column diameter (ft)

```

10 PRINT "PACKED COLUMN, PREDICTED >>>"
20 INPUT "FUEL FLOW RATE (GPH) : "; FFR
30 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) : "; GIT, GOT
40 INPUT "WATER INLET TEMP (DEGREE F) : "; WIT
50 INPUT "AMOUNT OF EXCESS AIR (%) : "; EX
60 INPUT "DESIRED CONDENSATION (%) : "; CC
70 INPUT "INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : "; GD, CG
80 INPUT "COLUMN DIAMETER (FT), HEIGHT (FT) : "; ID, ZT
100 FFR=7.185*FFR : REM Fuel flow rate unit to lb/hr
110 EX=EX/100 : CC=CC/100 : PI=3.141592
120 REM ----- Inlet gas humidity -----
130 H1=3.054+14.124*EX*.2315+14.124*(1+EX)*.7685+.02
140 H2=1.195
150 HI=H2/H1 : REM Inlet gas humidity
160 REM ----- Heat transfer duties -----
170 G1=H1*FFR : REM Weight of dry flue gas
180 W1=H2*FFR : REM Weight of water in inlet flue gas
190 WC=W1*CC : REM Weight of condensed water
200 QG=(G1+W1)*CG*(GIT-GOT)
210 QC=WC*970.3 : REM Condensation duty
220 QT=QG+QC : REM Total duty
230 WOT=120
240 WR=QT/((.997*(WOT-WIT)) : REM Water flow-rate
250 DTM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
260 G=(G1+W1)/(PI*ID^2/4)
270 L=(WR+WC)/(PI*ID^2/4)
280 HO=(1-CC)*HI
300 HD=110*.6^1.5*ID^1.24*(ZT/10)^.333/((.98*L)^.6)
310 HGA=.872^1.667*CG*G/HD
320 CO=WC*.42/(HGA*PI*ID^2*ZT/4)
330 ALPA=CO/(1-2.71828^(-CO))
340 UA=ALPA*HGA*QT/QG
350 TZ=QT/(UA*DTM*PI*ID^2/4)
360 IF ABS(ZT-TZ)>.1 THEN WOT=WOT+.5 : GOTO 240
370 WR=WR/8.289
400 PRINT "-----"
410 PRINT "*** Excess air (EX, %) : "; EX
420 PRINT "*** Gas mass velocity (G, LB/HR SQ-FT) : "; G
430 PRINT "*** Liquid mass velocity (L, LB/HR SQ-FT) : "; L
440 PRINT "*** Outlet water temperature (WOT, DEGREE F) : "; WOT
450 PRINT "*** Inlet gas humidity (HI, LB/LB) : "; HI
460 PRINT "*** Outlet gas humidity (HO, LB/LB) : "; HO
470 PRINT "*** Water flow rate (WR, GPH) : "; WR
480 PRINT "*** Sensible cooling duty (QG, BTU/HR) : "; QG
490 PRINT "*** Condensation duty (QC, BTU/HR) : "; QC
500 PRINT "*** Total duty (QT, BTU/HR) : "; QT

```

Sample Output

RUN

PACKED COLUMN, PREDICTED >>>

FUEL FLOW RATE (GPH) : 1

FLUE GAS INLET, OUTLET TEMP (DEGREE F) : 370,102

WATER INLET TEMP (DEGREE F) : 94

AMOUNT OF EXCESS AIR (%) : 27

DESIRED CONDENSATION (%) : 62.0

INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : .0437,.259

COLUMN DIAMETER (FT), HEIGHT (FT) : .7,2.4

*** Excess air (EX, %) : .27

*** Gas mass velocity (G, LB/HR SQ-FT) : 352.9268

*** Liquid mass velocity (L, LB/HR SQ-FT) : 1025.378

*** Outlet water temperature (WOT, DEGREE F) : 131

*** Inlet gas humidity (HI, LB/LB) : 6.436694E-02

*** Outlet gas humidity (HO, LB/LB) : 2.445944E-02

*** Water flow rate (WR, GPH) : 46.99229

*** Sensible cooling duty (QG, BTU/HR) : 9427.686

*** Condensation duty (QC, BTU/HR) : 4941.282

*** Total duty (QT, BTU/HR) : 14368.97

OK

1LIST 2RUN 3LOAD" 4SAVE" 5CONT 6,"LFT1 7TRON 8TROFF9KEY 0SCREEN

D.6 Packed column design, ultimate design

Flow chart

Nomenclature

ALPA: Correction factor for simultaneous heat and mass transfer (Eq.2.5)
 CC: Condensation rate desired (%)
 CG: Specific heat of gas (Btu/lb°F)
 DTM: Log mean temp. difference (°F)
 EX: Amount of excess air (%)
 FFR: Fuel flow rate (GPH)
 FP: Flow parameter (Fig. 2.7)
 FV: Flooding value interpolated (Fig. 2.7)
 G: Gas mass velocity (lb/hr ft²)
 GD: Inlet gas density (lb/ft³)
 GF: Maximum allowable gas velocity (lb/sec ft²)
 GG: Maximum allowable gas velocity (lb/hr ft²)
 GIT: Flue gas inlet temp. (°F)
 GOT: Flue gas outlet temp. (°F)
 HGA: Gas phase vol: heat transfer coefficient (Btu/hr ft² °F)
 HH: Approach to flooding
 HI: Inlet gas humidity (lb/lb)
 ID: Column diameter (ft)
 L: Liquid mass velocity (lb/hr ft²)
 PF: Packing factor
 QC: Condensation duty (Btu/hr)
 QG: Sensible gas cooling duty (Btu/hr)
 QT: Total duty (Btu/hr)
 WIT: Inlet water Temp. (°F)
 WOT: Adiabatic saturation temp. of inlet gas (°F)
 WR: Coolant water circulation rate (lb/hr)
 ZT: Column diameter (ft)

Computer Program

```

10 PRINT "/// PACKED COLUMN, ULTIMATE DESIGN >>>"
20 INPUT "FUEL FLOW RATE (GPH) : "; FFR
30 INPUT "FLUE GAS INLET, OUTLET TEMP (DEGREE F) : "; GIT, GOT
40 INPUT "WATER INLET TEMP (DEGREE F) : "; WIT
50 INPUT "AMOUNT OF EXCESS AIR (%): "; EX
60 INPUT "DESIRED CONDENSATION (%): "; CC
70 INPUT "INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : "; GD, CG
80 INPUT "INITIAL VALUE OF COLUMN DIAMETER (FT) : "; ID
90 INPUT "MATERIAL PACKING FACTOR : "; PF
100 FFR=7.13*FFR : REM Fuel flow rate unit to lb/hr
110 EX=EX/100 : CC=CC/100 : PI=3.141592
120 REM ----- Inlet gas humidity -----
130 H1=7.194+14.062*EX*.2315+14.062*(1+EX)*.7685+.1
140 H2=1.152
150 HI=H2/H1 : REM Inlet gas humidity
160 REM ----- Heat transfer duties -----
170 G1=H1*FFR : REM Weight of dry flue gas
180 W1=H2*FFR : REM Weight of water in inlet flue gas
190 WC=W1*CC : REM Weight of condensed water
200 QG=(G1+W1)*CG*(GIT-GOT)
210 QC=WC*970.3 : REM Condensation duty
220 QT=QG+QC : REM Total duty
230 REM ----- Outlet water temperature -----
240 PRINT "Inlet gas temp (degree F) : "; GIT; " Humidity (lb/lb) : "; HI
250 INPUT "ADIABATIC SATURATION TEMP (DEGREE F) : "; WOT
260 REM ----- Water circulation rate -----
270 WR=QT/((.997*(WOT-WIT))) : REM Water circulation rate
320 REM ----- Calculation of column height -----
330 G=(G1+W1)/(PI*ID^2/4)
340 L=(WR+WC)/(PI*ID^2/4)
350 DTM=((GOT-WIT)-(GIT-WOT))/LOG((GOT-WIT)/(GIT-WOT))
351 FP=(1/G)*(GD/61.8)^.5
352 GOSUB 600
353 FZ=FV*32.2*GD*61.8/((.591^2*PF))
354 GF=SQR(FZ)
355 GG=GF*3600
356 HH=G/GG
357 IF HH<.44 THEN ID=ID-.01 : GOTO 330
358 IF HH>.46 THEN ID=ID+.01 : GOTO 330
360 ZT=.3
370 A1=110*.6^1.5*.7^1.24*10^((-333)/(.98^1.6*L^1.6))
380 A2=.072^1.667*CG*G
390 R1=WC*.42/(PI*ID^2/4)
400 R2=R1/((A2/A1)*ZT^1.667)
410 B3=(1-2.71828^(-B2))/B2
420 B4=B3*QG*(A1/A2)/(PI/4*ID^2*DTM)
430 C1=ZT
440 C2=B4^1.5
450 IF ABS(C1-C2)>.1 THEN ZT=ZT+.1 : GOTO 370.
462 HGA=(A2/A1)*ZT^(-.333) : ALPA=1/B3

```

```

465 PRINT "-----"
470 PRINT "*** Sensible cooling duty (QG, BTU/HR) : ";QG
480 PRINT "*** Condensation duty (QC, BTU/HR) : ";QC
490 PRINT "*** Total duty (QT, BTU/HR) : ";QT
500 PRINT "*** EX,G,L : ";EX,G,L
520 PRINT "*** Water circulation rate (WR, LB/HR) : ";WR
540 PRINT "*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : ";HGA
550 PRINT "*** Correction factor (ALPA) : ";ALPA
555 PRINT "*** Approach to flooding : ";HH
565 PRINT "*** Column diameter (ID) : ";ID
570 PRINT "**** COLUMN HEIGHT (ZT, FT) : ";ZT
600 REM ----- SUBROUTINE Interpolation of flooding -----
610 S1=2.1793*(FP-.2)*(FP-.08)*(FP-.02)
620 S2=-20.8333*(FP-.4)*(FP-.08)*(FP-.02)
630 S3=63.8021*(FP-.4)*(FP-.2)*(FP-.02)
640 S4=-49.7076*(FP-.4)*(FP-.2)*(FP-.08)
650 FV=S1+S2+S3+S4
660 RETURN

```

Sample Output

FUEL FLOW RATE (GPH) : 1
FLUE GAS INLET, OUTLET TEMP (DEGREE F) : 450, 100
WATER INLET TEMP (DEGREE F) : 90
AMOUNT OF EXCESS AIR (%) : 30
DESIRED CONDENSATION (%) : 70
INLET GAS DENSITY (LB/CU FT), SPECIFIC HEAT (BTU/LB-F) : .0457, .259
INITIAL VALUE OF COLUMN DIAMETER (FT) : .5
MATERIAL PACKING FACTOR : .32
Inlet gas temp (degree F) : 450 , Humidity (lb/lb) : 6.288468E-02
ADIABATIC SATURATION TEMP (DEGREE F) : 140

*** Sensible cooling duty (QG, BTU/HR) : 12584.94
*** Condensation duty (QC, BTU/HR) : 5578.867
*** Total duty (QT, BTU/HR) : 18163.81
*** EX.G.L : .3 1104.774 2945.312
*** Water circulation rate (WR, LB/HR) : 364.3692
*** Gas phase vol heat tran coeff (HGA, BTU/HR-CU.FT-F) : 1318.236
*** Correction factor (ALPA) : 1.009142
*** Approach to flooding : .4498441
*** Column diameter (ID) : .4000001
**** COLUMN HEIGHT (ZT, FT) : .8000001
RETURN without GDSUB in 660
OK

1LIST 2RUN 3LOAD 4SAVE 5CONT 6,"LPT1 7TRON 8TROFF 9KEY 0SCREEN

APPENDIX E Operating lines for cooling of a gas by direct contact with a liquid [19]

The spray column used for the thesis performed humidification in the lower part and dehumidification in the upper part of the column. Because of insufficient column height, the absolute humidities of outlet gas were little higher than those of inlet gas. The gas and water temperatures at inlet and outlet are shown in the following table.

Table E.1 Gas and Water Temperature at Inlet and Outlet for Spray Column

Data No.	Temperature (°F)				Gas Humidity (lb/lb)	
	Gas		Water		Inlet w_{gi}	Outlet w_{go}
	Inlet T_{gi}	Outlet T_{go}	Inlet T_{li}	Outlet T_{lo}		
1	301	112	93	113	0.060	0.071
2	365	128	97	125	0.060	0.072
3	364	128	83	122	0.059	0.086
4	355	114	91	118	0.076	0.076
5	382	121	92	123	0.066	0.082
6	410	135	95	127	0.058	0.093
7	368	117	93	119	0.067	0.077
8	390	124	94	123	0.063	0.079
9	410	135	97	127	0.060	0.092

The cooling of a hot dry gas with water in a direct contact column raises a special problem since at the bottom of the column the equilibrium water vapor pressure is greater than its partial pressure in the gas and evaporation occurs. The vapor condenses in the upper section of the column where both the gas and liquid temperatures are lower. In the design of a column operating under these conditions it is necessary to carry

out a stepwise calculation of the temperatures and humidities at a series of points to determine the point at which evaporation ceases and condensation commences.

In any short section of the column, the change in temperature and humidity of the gas may be calculated from the equation :

$$\frac{dw}{dT} = \frac{w_g - w_l}{T_g - T_l} \quad (E.1)$$

where, w_g = humidity of gas

w_l = humidity of saturated gas at water temperature

T_g = temperature of gas

T_l = temperature of water contacting gas locally

The temperature difference $T_g - T_l$ is positive, but the humidity difference $w_g - w_l$ may be positive or negative.

Figure E.1 Equilibrium and Operating Lines

The figure above shows equilibrium and operating lines for the cooling of a gas by direct contact with a liquid. The humidity was plotted against temperature as shown in the above figure. The curve AB is the saturation line, point C represents the temperature and humidity of the gas at the inlet and D the state of the gas leaving the column. The operating line joining these points is obtained by drawing a series of tangents, CE, FG, etc., each of which has a slope dw/dT calculated by substituting the appropriate temperatures and humidities in the Equation E.1.

Now using the above method, the slopes of the operating line for the experiment can be drawn at gas inlet and outlet points. Therefore the operating lines can be obtained roughly. Following figures show the plotting of operating lines with the experimental data in Table E.1.

Figure E.2 Operating Line with Experimental Data (Data No.1)

Figure E.3 Operating Line with Experimental Data (Data No. 6)

APPENDIX F Calculations for heat recovery curve of No.2 oil fired boiler.

In order to draw the heat recovery curve of No.2 oil fired boiler, the same assumptions as those of natural gas fired boiler mentioned in Section 1.2 are made. That is, complete combustion in boiler is assumed with no excess air and the outlet gas of heat exchange column is assumed to be saturated.

For inlet flue gas temperature 300°F and inlet flue gas humidity 0.0819 lb/lb (see Section 3.2.3), the dew point is found in a humidity chart (Figure 2.3) as 119°F .

Therefore the heat recovered per lb oil is calculated as follows :

$$\begin{aligned} Q_s &= G c_p \Delta t & (\text{F.1}) \\ &= (15.225 \text{ lb gas/lb oil}) \times (0.259 \text{ Btu/lb}^{\circ}\text{F}) \times (300 - t) (^{\circ}\text{F}) \\ &= 3.943 \times (300 - t) (\text{Btu/lb}) \end{aligned}$$

$$\begin{aligned} Q_L &= \lambda W_c & (\text{F.2}) \\ &= 970.3 \times W_c (\text{Btu/lb}) \end{aligned}$$

$$Q_T = Q_s + Q_L \quad (\text{F.3})$$

where, Q_s = sensible heat recovered (Btu/lb)

Q_L = latent heat recovered (Btu/lb)

Q_T = total heat recovered (Btu/lb)

G = weight of flue gas generated per lb oil

= 15.225 lb/lb (see Section 3.2.3)

c_p = average-specific heat of flue gas

$$= 0.259 \text{ Btu/lb}^\circ\text{F}$$

t = outlet flue gas temperature ($^\circ\text{F}$)

λ = heat of vaporization of water = 970.3 Btu/lb

W_c = amount of water condensed per lb oil (lb water/lb oil)

W_c is determined at each temperature below dew point in a humidity chart. The amount of water vapor in flue gas per lb oil was calculated as 1.152 lb in Section 3.2.3. For example, the amount of water condensed, ~~at~~ at 104°F is calculated as follows :

$$W_c = 1.152 \text{ (lb water vapor/lb oil)}$$

$$\times \frac{(0.0819 - 0.0510) \text{ (lb/lb)}}{0.0819 \text{ (lb/lb)}}$$

$$= 0.435 \text{ (lb/lb)}$$

where, 0.0819 lb/lb is flue gas inlet humidity and 0.0510 lb/lb is saturation humidity at 104°F .

Therefore following table can be obtained in order to draw Figure 4.6.

Table F.1 Heat Recovery for No.2 Oil Fired Boiler

Temp. t (°F)	Sensible heat recovered Q_s (Btu/lb-oil)	Latent heat recovered Q_L (Btu/lb-oil)	Total heat recovered Q_T (Btu/lb-oil)
119	714	0	714
104	773	422	1195
86	843	736	1579
68	915	899	1814
50	986	1009	1995

REFERENCES

1. BETT Research Report, Heat Recovery From Flue Gases in Office and Apartment Buildings, Energy, Mines and Resources Canada, 1984, pp.7-9.
2. Fair, J.R., Designing Direct-Contact Coolers/Condensers, Chemical Engineering, June 12, 1972, pp.91-100.
3. Kiang, Y.H., Waste Energy Utilization Technology, Marcel Dekker Inc., 1981, pp.173-196.
4. Goldstick, R.J., Thumann, A., Principle of Waste Heat Recovery, Fairmont Press, 1986, p.138.
5. Same reference [4], p.95.
6. Azbel, D., Heat Transfer Applications in Process Engineering, Noyes Publications, 1984, pp.342-356.
7. Same reference [4], pp.132-134.
8. Lydersen, A.L., Mass Transfer in Engineering Practice, John Wiley & Sons Inc., 1983, pp.203-204.
9. Fair, J.R., Process Heat Transfer by Direct Fluid-Phase Contact, AIChE Symposium Series, 1972, No.118/68, pp.1-11.

10. Fair, J.R., Design of Direct-Contact Gas Coolers, Petro/Chem Engineer, August, 1961, pp. 203-210.

11. Mehta, K.C., and Sharma, M.M., British Chemical Engineering, No.11, 1400; No.12, 1556, 1970.

12. Fair, J.R., Steinmeyer, D.E., Penney, W.R., and Brink, J.A., Liquid-Gas System, Chemical Engineer's Handbook 5th/e, McGraw Hill, 1973, pp.18-51.

13. Cornell, D., Knapp, W.G., and Fair, J.R., Mass transfer Efficiency -Packed Column- Part I, Chemical Engineering Progress, Vol.56, No.7, July 1960, pp.68-74.

14. Eckert, J.S., Selecting the Proper Distillation Column Packing, Chemical Engineering Progress, Vol.66, No.3, March 1970, pp.39-44.

15. Coulson, M.M., Richardson, J.F., and Sinnott, R.K., Chemical Engineering, Vol.6, Pergamon Press, 1983, pp.488-491.

16. Ganapathy, V., Steam Plant Calculations Manual, Marcel Dekker Inc. 1984, pp.53-54.

17. Ganapathy, V., Applied Heat Transfer, Penn Well Books, 1982, pp.22-27.

18. Same reference [4], p.139.

19. Norman, W.S., Absorption, Distillation and Cooling Towers, John Wiley and Sons Inc, 1962, pp.293-294.

20. Coulson, J.M., Richardson, J.F., Backhurst, J.R., and Harker, J.H., Chemical Engineering, Vol.2, Pergamon Press, 1978, pp.150-151.

21. Leonard, C.M., and Maleev, V.L., Heat Power Fundamentals, Pitman, 1956, pp.303-315, p.299.

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