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APPENDIX A

FEED RATE OF RICE HUSKS

FEED RATE OF RICE HUSKS

For the design of a 50 kW thermal two-stage gasification system, the first step has to be the calculation of feed or mass flow rate of rice husks. Feed rate of rice husks is calculated from lower heating value and gasification efficiency of rice husks. Lower heating value of rice husks is 14.44 MJ/kg from bomb calorimeter. Gasification efficiency was determined from literature review which gasification efficiency of rice husks under the optimum conditions is estimated at 65% (Yin, X.L. *et al.*, 2002) and gasification efficiency of rice straw in a two-stage gasifier is between 80.8%-84.6% (Yi, S. and Yonghao, L., 2009). Therefore gasification efficiency for this study is defined between 65% and 80% which is 70%.

$$50 \text{ kW thermal} = 50\,\frac{kJ}{s} \times 3600\,\frac{s}{hr}$$

$$= 180000\,\frac{kJ}{hr}$$

$$= 180\,\frac{MJ}{hr}$$

$$\text{Feed rate of rice husks } (\dot{m}_{rice\ husk}) = \frac{180\ MJ}{hr} \times \frac{kg}{14.44\ MJ}$$

$$= 12.47\,\frac{kg}{hr}$$

Feed rate of rice husks at gasification efficiency 70%

$$\text{Feed rate of rice husks } (\dot{m}_{rice\ husk}) = \frac{12.47\ kg}{0.70\ hr}$$

$$= 17.8\,\frac{kg}{hr}$$

From the previous calculation, result of feed rate of rice husks is on dry basis. In the operation, there is moisture contained in rice husks which has to include for determination of feed rate of rice husk. Moisture content of rice husks was measured by weigh a sample, then heat to 105 °C for 1-4 hours to find bone dry weight. The moisture content wet basis (MCWB) and moisture content in rice husks were calculated by

Equations (A.1) and (A.2). Then feed rate of rice husks on wet basis was calculated by Equations (A.3).

$$MCWB = 100 \text{ X (Initial weight - Dry weight)/(Initial weight)} \quad (A.1)$$

$$\dot{m}_{moisture} = \dot{m}_{rice\ husk,dry\ basis} \times (\%moisture/100) \quad (A.2)$$

$$\dot{m}_{rice\ husk,wet\ basis} = \dot{m}_{rice\ husk,dry\ basis} + \dot{m}_{moisture} \quad (A.3)$$

From weight of initial and dry weight to measure moisture content, moisture content was 9%.

Calculation of feed rate of rice husks on wet basis

$$\dot{m}_{moisture} = 17.8\,\frac{kg}{hr} \text{ x } (9/100)$$

$$= 1.6\,\frac{kg}{hr}$$

$$\text{Feed rate of rice husks } (\dot{m}_{rice\ husks,wet\ basis}) = 17.8\,\frac{kg}{hr} + 1.6\,\frac{kg}{hr}$$

$$= 19.4\,\frac{kg}{hr}$$

Therefore feed rate of rice husks on wet basis of this system was 19.4 kg/hr.

APPENDIX B

FEED RATE OF AIR

FEED RATE OF AIR

Ambient air was used as the fluidizing and the gasifying medium for this study. To calculate the air flow rate, the stoichiometric air required for rice husks was calculated. Raw biomass generally contains moisture and ash. Apart from hydrogen and oxygen in the biomass, the moisture also contributes to the composition of the producer gas and the composition of ash is not gasified to producer gas. Therefore, these have to be taken into consideration. The feed rate dry ash free biomass is calculated as follows:

<u>Basis:</u> 100 kg rice husks

wt. rice husks = wt. rice husks $_{(d.a.f.)}$ + wt. moisture + wt. ash

100 kg = 65.2 kg + 9 kg + 25.8 kg

$$\dot{m}_{(d.a.f.rice\ husks)} = \frac{100\ kg\ rice\ husks}{65.2\ kg\ rice\ husks_{(d.a.f.)}}$$

$$\dot{m}_{(d.a.f.rice\ husks)} = 1.53\ \frac{kg\ rice\ husks}{kg\ rice\ husks_{(d.a.f.)}}$$

Calculation Stoichiometric Air Required for Rice Husks Gasification

<u>Basis:</u> 100 kg rice husks $_{(d.a.f.)}$

Element	wt.	kmol	RxN	O_2 required
C	45.9	3.825	$C + O_2 \rightarrow CO_2$	3.825
H	6.2	3.1	$H_2 + \frac{1}{2}O_2 \rightarrow H_2O$	1.55
O	47.3	2.96	$O_2 \rightarrow O_2$	-1.48
N	0.6	0.04	$\frac{1}{2}N_2 + O_2 \rightarrow NO_2$	0.04

The stoichiometric O_2 required for rice husks $_{(d.a.f.)}$ equals to 3.935 kmol.

The stoichiometric air required for rice husks $_{(d.a.f.)}$:

$$\dot{m}_{dry\ air} = \text{wt. } O_2 + \text{wt. } N_2$$

$$\dot{m}_{dry\ air} = (3.935 \times 32\ kg\ O_2) + (3.935 \times 3.76 \times 28\ kg\ N_2)$$

$$\dot{m}_{dry\ air} = 540.2 \frac{kg\ air}{100\ kg\ rice\ husks_{(d.a.f.)}}$$

$$\dot{m}_{dry\ air} = 5.4 \frac{kg\ air}{kg\ rice\ husks_{(d.a.f.)}}$$

The stoichiometric air required for rice husks:

$$\dot{m}_{dry\ air} = \frac{5.4\ kg\ air}{kg\ rice\ husk_{s(d.a.f.)}} \times \frac{kg\ rice\ husks_{(d.a.f.)}}{1.53\ kg\ rice\ husks}$$

$$\dot{m}_{dry\ air} = 3.53 \frac{kg\ air}{kg\ rice\ husks}$$

Equivalence ratio (ER) is defined as the air-to-fuel weight ratio divided by the air-to-fuel weight ratio of stoichiometric ratio. The ER is significant when evaluating gasification processes since the gas composition, gas yield and heating value can be viewed as a function of the ER. The ER for biomass gasification in practice varies from 0.20 to 0.40. In case of rice husk gasification, the maximum of the combustible gas components and the higher heating value occur at the ER of 0.25 (Mansaray, K.G. *et al.*, 1999). Consequently, the dry air mass flow rate was calculated by Equations (B.1).

$$ER = \frac{(weight\ of\ air/weight\ of fuel\ )_{act}}{(weight\ of\ air/weight\ of fuel)_{stoi}} \qquad (B.1)$$

$$0.25 = \frac{weight\ of\ air/19.4}{3.53}$$

$$\dot{m}_{dry\ air} = 17.12 \frac{kg}{hr}$$

Generally, there is moisture (water vapor) contained in the ambient air. The amount of moisture contained in dry air could be calculated by Equations (B.2). While the humidity ratio (ω) came from a psychometric chart by assuming the average dry bulk temperature of 30°C and the average relative humidity (ϕ) at 60% for air in Thailand (ω @ 30°C and ϕ at 60% $= 0.016 \frac{kg}{kg}$). The air mass flow rate and air volume flow rate were calculated by Equations (B.3) and (B.4).

$$\dot{m}_{moisture} = \dot{m}_{dry\ air} \times \omega \qquad (B.2)$$

$$\dot{m}_{air} = \dot{m}_{dry\ air} + \dot{m}_{moisture} \qquad (B.3)$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} \quad ; \rho_{air} @ 30°C = 1.66 \frac{kg}{m^3} \tag{B.4}$$

Calculation of air volume flow rate:

$$\dot{m}_{moisture} = 17.12 \frac{kg}{hr} \text{ x } 0.016 \frac{kg}{kg}$$

$$\dot{m}_{moisture} = 0.27 \frac{kg}{hr}$$

$$\dot{m}_{air} = 17.12 \frac{kg}{hr} + 0.27 \frac{kg}{hr}$$

$$= 17.39 \frac{kg}{hr}$$

$$\dot{V}_{air} = \frac{17.39\ kg}{hr} \text{ x } \frac{m^3}{1.66\ kg} \text{ x } \frac{hr}{3600\ s}$$

$$\dot{V}_{air} = 0.0029 \frac{m^3}{s}$$

Therefore, air volume flow rate of this system which consists of pyrolysis stage and char gasifier stage is 0.0029 m^3/s.

APPENDIX C

BIOMASS FEEDING SYSTEM

BIOMASS FEEDING SYSTEM

The characteristic of rice husks is a bulk solid. A screw conveyor was selected because screw is a bulk material handling device capable of handling a great variety of materials that have good flow abillity. One of the primary advantages of the screw conveyor is the number of feed inlets and discharge opending that can be provided for. Typical applications include grain storage, feed mills, cereal processing, and chemical plants. Conveyors can also be adapted for volume control. When utilized in this form they are called screw feeder and are attached to the bottom of bins, hoppers, bag dumps, storage piles, and so on. These units are designed to regulate the flow of material to the downsteam equipment. Toxic materials can easily be handled with a screw conveyor because the enclosed trough can be built tight enough to contain the toxic dusts or vapors and reduce personnel hazards. On the other hand, materials that must be free from contaminants may also be handled because the enclosed screw conveyor trough protects the materials from outside influences.

A screw or spiral feeder consists of metal flighting mounted on a shaft. The mounted flighting is suspended within a U-shaped, round, or flared trough driven by an electric motor through a speed reducer or chain drive. Various designs of screw feeders are available and are fully covered in manufacturers' catalogs. Standard full pitch is suitable for handling fine free-flowing materials where the feed opending is limitted to 1-1 ½ pitches. If the diameter of the screw is uniform the feed of the material will come from only a portion of a slotted inlet and not across the entire length.

Screw feeder conveyor pitch can be varied increasing in the direction of flow to provide a more uniform discharge across a slotted bin opening. Another variation to accomplish the same effect is to utilize a tapered flight diameter. However, in order to ensure the best uniform bin discharge across a slotted bin opening requires a combination of tapered flight diameter and increasing pitch.

C.1 BULK MATERIAL CHARACTERISTICS

A study was made to define the characteristics of bulk materials in terms which are readily recognized. These characteristics and terms are indicated in the Material Classification Code Chart as APPENDIX C.1. The material properties classified follow by the Conveyor Equipment Manufacturers Association (CEMA), first of all, the letter first means classified according to particle size. Then followed by the two numbers, The first number means the properties of flowability. The second number means abrasiveness of material.

In determining the flowability of a material two considerations must be studied. One is the angle of slide, and the other is the internal angle of friction of the material. The angle of slide may be determined by tilting the plane carrying a quantity of material. By measuring the angle at which the material slides, a sliding friction factor can be determined. The internal angle of friction can be determined through the shear cell test. Any changes in moisture, temperature, particle size, or corrossion characteristics of the material will affect flowability which will in turn affect horsepower and other design principles.

C.2 DESIGN CONSIDERATION

C.2.2 Conveyor Size and Speed

To determine the conveyor size and speed, it is first necessary to establish the degree of trough loading. After the degree of trough loading has been determined, refer to APPENDIX C.2 for a cross-sectional loading area. Proper performance of a screw conveyor assumes the operation is controlled with volumetric feeders and the material is uniformly fed into the conveyor housing at all time.

For further determining capacity in nonstandard units, the following equation may be used.

$$m_s = \frac{\rho\pi(D^2-d^2)kpN}{4} \quad (C.1)$$

Where m_s= mass flow rate of material

ρ = density of material

D = diameter of screw

d = diameter of shaft

k = factor of trough loading

p = pitch of screw

N = speed of the screw

The size of a screw conveyor depends not only on a capacity required but on the size and proportion of the lumps in the material to be handled. Also, the characteristic of a lump needs to be considered and whether or not it will break up in transit as it is being conveyed. The allowable lump size in the screw conveyor is a function of the radial clearance between the outside diameter of the central pipe and the inside radius of the screw conveyor trough, as well as the portion of lumps in any given mix.

Calculation of screw conveyor

1. Find the material class of rice husks from material table (First letter & last two numbers). This determines trough loading, factor of trough loading = 0.45
2. Determine mass flow rate and density of rice husk:

   mass flow rate of rice husk = 19.4 $\frac{kg}{hr}$ = 0.32 $\frac{kg}{min}$

   density of rice husk = 120 $\frac{kg}{m^3}$
3. Define size of shaft, diameter of shaft = 0.020 m
4. Define speed of the screw, N = 150 rpm
5. Define pitch, pitch = diameter of screw
6. Calculate diameter of screw

$$m_s = \frac{\rho\pi(D^2-d^2)kpN}{4}$$

$$0.32\frac{kg}{min} = \frac{120\,kg/m^3\,\pi\left(D^2 - 0.02\,m^2\right)0.45 \times D \times 150\,rpm}{4}$$

D = 0.0404 m

According to clearance between the screw and the radius of the inside of the screw trough is around 12 – 15 mm, thus the diameter of the inside of the screw trough is calculated as follow:

the diameter of the inside of the screw trough = 0.041 + 0.013 + 0.013

= 0.067 m

As screw trough is designed as pipe, the selected internal diameter of pipe is equal or higher than 0.067 m. The selected pipe is NPS 2½ with SCH 5 which the outside diameter is 0.07302 m, the internal diameter is 0.0688 m, and the wall thickness is 2.108 mm.

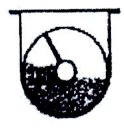
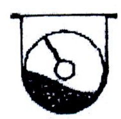
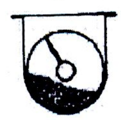
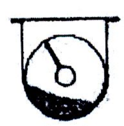
APPENDIX C.1

MATERIAL CLASSIFICATION CODE CHART

Major Class		Material Characteristics Included		Code Designation
Density	Bulk Density, Loose			Actual Lbs./cu.ft.
Size	Very Fine	No. 200 Sieve (.0029") And Under		A_{200}
		No. 100 Sieve (.0059") And Under		A_{100}
		No. 40 Sieve (.016") And Under		A_{40}
	Fine	No. 6 Sieve (.132") And Under		B6
	Granular	½" Ander Under		$C_{½}$
		3" And Under		D_3
		7" And Under		D_7
	Lumpy*	16" And Under		D_{16}
		Over 16" To Be Specified X = Actual Maximum Size		D_x
	Irregular	Stringy, Fibrous, Cylindrical, Slabs, Etc.		E
Flowability	Very Free Flowing-Flow Function > 10			1
	Free Flowing-Flow Function > 4 But < 10			2
	Average Flowability-Flow Function > 2 But < 4			3
	Sluggish-Flow Function < 2			4
Abrasiveness	Mildly Abrasive		Index 1-17	5
	Moderately Abrasive		Index 18-67	6
	Extremely Abrasive		Index 68-416	7

APPENDIX C.2

SCREW CONVEYOR CAPACITIES AND LOADING CONDITIONS

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
				[illegible]	[illegible]
A-15 A-25 B-15 B-25 C-15 C-25	45% 	6	165	368	2.23
		9	155	1,270	8.20
		12	145	2,820	19.40
		14	140	4,370	31.20
		16	130	6,060	46.70
		18	120	8,120	67.60
		20	110	10,300	93.70
		24	100	16,400	164.00
A-35 E-35 A-45 E-45 B-35 B-45 C-35 C-45 D-15 D-25 D-35 D-45 E-15 E-25	**Non-Abrasive Materials 30% A** 	6	120	180	1.49
		9	100	545	5.45
		12	90	1,160	12.90
		14	85	1,770	20.80
		16	80	2,500	31.20
		18	75	3,380	45.00
		20	70	4,370	62.50
		24	65	7,100	109.00
A-16 D-16 A-26 D-26 A-36 D-36 A-46 D-46 B-16 E-16 B-26 E-26 B-36 E-36 B-46 E-46 C-16 C-26 C-36 C-46	**Abrasive Materials 30% B** 	6	60	90	1.49
		9	55	300	5.45
		12	50	645	12.90
		14	50	1,040	20.80
		16	45	1,400	31.20
		18	45	2,025	45.00
		20	40	2,500	62.50
		24	40	4,360	109.00
A-17 D-17 A-27 D-27 A-37 D-37 A-47 D-47 B-17 E-17 B-27 E-27 B-37 E-37 B-47 E-47 C-17 C-27 C-37 C-47	15% 	6	60	45	0.75
		9	55	150	2.72
		12	50	325	6.46
		14	50	520	10.4
		16	45	700	15.6
		18	45	1,010	22.5
		20	40	1,250	31.2
		24	40	2,180	54.6

*Maximum recommended R.P.M.

APPENDIX C.3

NOMINAL PIPE SIZE: NPS ⅛ to NPS 3½

NPS	DN	OD [in (mm)]	Wall thickness [in (mm)]							
			SCH 5	SCH 10s/10	SCH 30	SCH 40s/40 /STD	SCH 80s/80 /XS	SCH 120	SCH 160	XXS
⅛	6	0.405 (10.29)	0.035 (0.889)	0.049 (1.245)	0.057 (1.448)	0.068 (1.727)	0.095 (2.413)	—	—	—
¼	8	0.540 (13.72)	0.049 (1.245)	0.065 (1.651)	0.073 (1.854)	0.088 (2.235)	0.119 (3.023)	—	—	—
⅜	10	0.675 (17.15)	0.049 (1.245)	0.065 (1.651)	0.073 (1.854)	0.091 (2.311)	0.126 (3.200)	—	—	—
½	15	0.840 (21.34)	0.065 (1.651)	0.083 (2.108)	—	0.109 (2.769)	0.147 (3.734)	—	—	0.294 (7.468)
¾	20	1.050 (26.67)	0.065 (1.651)	0.083 (2.108)	—	0.113 (2.870)	0.154 (3.912)	—	—	0.308 (7.823)
1	25	1.315 (33.40)	0.065 (1.651)	0.109 (2.769)	—	0.133 (3.378)	0.179 (4.547)	—	—	0.358 (9.093)
1¼	32	1.660 (42.16)	0.065 (1.651)	0.109 (2.769)	0.117 (2.972)	0.140 (3.556)	0.191 (4.851)	—	—	0.382 (9.703)
1½	40	1.900 (48.26)	0.065 (1.651)	0.109 (2.769)	0.125 (3.175)	0.145 (3.683)	0.200 (5.080)	—	—	0.400 (10.160)
2	50	2.375 (60.33)	0.065 (1.651)	0.109 (2.769)	0.125 (3.175)	0.154 (3.912)	0.218 (5.537)	0.250 (6.350)	0.343 (8.712)	0.436 (11.074)
2½	65	2.875 (73.02)	0.083 (2.108)	0.120 (3.048)	0.188 (4.775)	0.203 (5.156)	0.276 (7.010)	0.300 (7.620)	0.375 (9.525)	0.552 (14.021)
3	80	3.500 (88.90)	0.083 (2.108)	0.120 (3.048)	0.188 (4.775)	0.216 (5.486)	0.300 (7.620)	0.350 (8.890)	0.438 (11.125)	0.600 (15.240)
3½	90	4.000 (101.60)	0.083 (2.108)	0.120 (3.048)	0.188 (4.775)	0.226 (5.740)	0.318 (8.077)	—	—	0.636 (16.154)

APPENDIX D

SHELL AND TUBE HEAT EXCHANGER DESIGN

SHELL AND TUBE HEAT EXCHANGER DESIGN

D.1 DESIGN PROCEDURE

D.1.1 Select Type of Heat Exchanger

Various front and rear head types and shell types have been standardized by TEMA (Tubular Exchanger Manufacturers Association). They are identified by an alphabetic character and explained, as shown in APPENDIX D.1.

D.1.2 Allocation of Streams

A decision must be made as to which fluid will flow through the tubes and which will flow through the shell. This system used shell and tube heat exchanger for heat transfer between produced gas and air. Produced gas, which is the mixture of carbon monoxide, hydrogen, methane and other gases, is flowed inside the tube because ease to mechanical cleaning, while air is flowed inside the shell. The properties of fluid which should flow in the tube are fouling fluid, corrosive fluid, high pressure (>1000 kPa), hot fluid, higher heat transfer coefficient, higher flow rate and lower viscosity.

D.1.3 Outlet Temperature of Produced Gas

The produced gas is cooled in a shell and tube heat exchanger before filtered by the baghouse filter. The inlet temperature is estimated 600°C. Consideration of the outlet temperature is determined by the tar dew point, because the outlet temperature of the produced gas is necessary higher than dew point to prevent the tar condensation that can cause pipes to plug and reduce the effectiveness of heat exchanger.

Tars are generally defined as organic compounds present in the gas, excluding gaseous hydrocarbon. Tars are formed from the main components of biomasss that is cellulose, hemicellulose, and lignin, which first form primary pyrolysis products. These primary products are converted into secondary products, such as phenol and olefin, as the temperature increased. A further increase in temperature generates aromatic compounds with aliphatic substituents. Finally, polyaromatic hydrocarbons (PAHs), such as naphthalene and pyrene, are formed. Consideration of tar dew point is related to the classification system of tar and concentration. Tar classification is comprised of six classes (see Table D.1). This classification system was developed in particular to provide easy insight in the general composition of tar. The concentration of the individual tar components in the syngas is used for predict the tar dew point. An illustration of the relation between the tar dew point and tar concentrations is provided by Figure D.1. Condensation curves are given for the individual tar classes (as defined in Table D.1). Downdraft gasifiers produce the cleanest gases with tar loading typically less than 1000 mg/Nm^3. However, in commercial operation downdraft units often have loadings in excess of 1 g/Nm^3 and the reported highest amount of tar is 5000 mg/Nm^3 (Milne *et al.*, 1998). The major component of exit tar is naphthalene which is in the class 4 tar (Gerun *et al.*, 2008).

Table D.1 Tar Classification System (Bergman *et al.*, 2002)

Class	Type	Examples
1	GC undetectable tars.	biomass fragments, heaviest tars (pitch)
2	Heterocyclic compounds. These are components that generally exhibit high water solubility.	phenol, cresol, quinoline, pyridine
3	Aromatic components. Light hydrocarbons, which are important from the point view of tar reaction pathways, but not in particular towards condensation and solubility.	toluene, xylenes, ethylbenzene (excluding benzene)
4	Light polyaromatic hydrocarbons (2-3 rings PAHs). These components condense at relatively high concentrations and intermediate temperatures.	naphthalene, indene, biphenyl, antracene
5	Heavy polyaromatic hydrocarbons (^{3}4-rings PAHs). These components condense at relatively high temperature at low concentrations.	fluoranthene, pyrene, crysene
6	GC detectable, not identified compounds.	unknowns

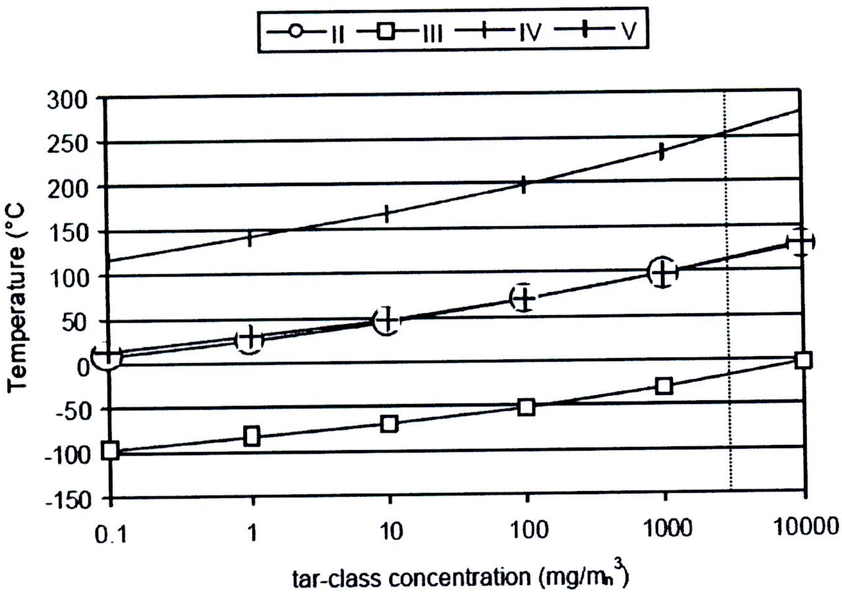


Figure D.1 The tar dew point of the different tar classes in relation to the concentration (Bergman *et al.*, 2002)

Tar dew point from condensation curves shows that class 4 starts to condense around 250°C at 5000 mg/Nm^3 of tar concentration. Therefore the outlet temperature of produced gas is defined as 250°C which is coped with the entire tar classification.

D.1.4 Outlet Temperature of Air

The hot air from heat transfer of produced gas is used as a preheat air for two-stage gasification to prevent the heat lost of the system. The outlet temperature of the air should be the highest temperature possible and related with temperature effectiveness (P) and heat capacity rate ratio (R) to determine LMTD correction factor (F). Therefore, the outlet temperature of air which is the highest temperature possible is 350°C.

D.1.5 Mass Flow Rate of Produced Gas

Mass flow rate of produced gas was considered from mass balance. It was assumed from the of conservation of mass that, when biomass was gasified with air in the two-stage gasifier, all the mass in the rice husks and the gasifying air was converted into gasification products. From the steady flow and the steady state condition, the mass balance of the rice husks gasification with air can be written as follows.

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}$$

$$\dot{m}_{rice\ husks} + \dot{m}_{air} = \dot{m}_{rice\ husks\ ash} + \dot{m}_{produced\ gas}$$

$$\dot{m}_{produced\ gas} = \dot{m}_{rice\ husks} + \dot{m}_{air} - \dot{m}_{rice\ husks\ ash}$$

$$\dot{m}_{produced\ gas} = 17.8\frac{kg}{hr} + 15.71\frac{kg}{hr} - 5.4\frac{kg}{hr}$$

$$\dot{m}_{produced\ gas} = 28.11\frac{kg}{hr}$$

$$\dot{m}_{produced\ gas} = 0.0078\frac{kg}{s}$$

D.1.6 Mass Flow Rate of Air

Mass flow rate of air is considered from heat balance of produced gas (hot stream) and air (cold stream). Hot stream is cooled from 600°C to 250°C. Cold stream is heated from 30°C, which is ambient temperature, to 350°C.

Heat balance

Hot stream; $Q = \dot{m}_h C_{p,h} \Delta T$

Cold stream; $\dot{m}_c = \frac{Q}{C_{p,c}\Delta T}$

D.1.7 Parameters to be Considered

For the design of shell and tube heat exchanger, there are many limiting parameters to be considered as follows.

- Tube pass
- Tube outside diameter: considered from (APPENDIX D.2)
- Tube length $D_i/d_o > 15$
- Tube layout $D_i/L = 1/5 - 1/15$
- Tube pitch and clearance $P_T/d_o = 1.25 - 1.5$
- Tube count
- Baffle $B/D_i = 0.4 - 0.6$

D.1.8 Size Estimation

In a size estimation, U, $C_{p,c}$, $C_{p,h}$, and the inlet and outlet temperatures are specified, and the heat transfer area (A_s) is to be determined. Then the value of A_s is interpreted in term of tube length and number of tube. Steps to estimating the size is shown below.

- Estimate U

- Cal LMTD

$$\Delta T_{lm} = \frac{(\Delta T_1 - \Delta T_2)}{ln(\Delta T1/\Delta T2)}$$

-Find F from APPENDIX D.3

$$P = \frac{(t_o - t_i)}{(T_i - t_i)} = 0.614$$

$$R = \frac{(T_i - T_o)}{(t_o - t_i)} = 0.914$$

F = 0.52

- Estimate heat transfer area

$$A_s = \frac{Q}{U_f(LMTD)F} = 2.50$$

- Interpret the value of A_s in term of tube length and number of tubes

$$N_t = \frac{A_s}{\pi d_o L_t}$$

Table D.2 shows the relation of tube length and shell internal diameter. Tube length was determined and was calculated the number of tubes. APPENDIX D.4 gives the number of tubes and shell internal diameters. Then the ratio of tube length and shell internal dimeter is calculated. The tube length with the ratio of tube length and shell internal dimeter was selected within the range of 5-15.

Table D.2 Relation of tube length and shell internal diameter

L	N_t (cal)	N_t (table)	D_i (in)	D_i (m)	L/D_i
1	62.76	76	15.25	0.38735	2.582
2	31.38	56	13.25	0.34	5.882
3	20.92	45	12	0.3048	9.843

D.1.9 Rating

For the rating process, all the preliminary geometrical calculations must be carried out as the input into the heat transfer and the pressure drop correlations. When the heat exchanger is available, then all the geometrical parameters are also known. In the rating process, the other two basic calculations are the calculations of heat transfer coefficients and the pressure drops for each stream specified. The heat duty is fixed for this study, then the result from the rating is the length of the heat exchanger required to satisfy the fixed heat duty of the heat exchanger. Then the pressure drops for both stream in the heat exchnager are calculated. Steps to rating of shell and tube heat exchanger is shown below.

The selected shell and tube heat exchanger has the following parameters:

Shell internal diameter	D_i = 13.25 in. (=0.34 m)
Number of tubes	N_t = 56
Tube diameter	d_o = 0.0254 m, d_i = 0.02362 m
Baffle spacing	B = 0.1360 m, baffle cut 25%
Pitch size	P_t = 0.0318 m
Number of tube passes	N_p = 2

The properties of the shell side air at $\frac{350+30}{2}$ = 190°C (463.15 K) are:

$\rho = 0.7633$ kg/m^3

$C_p = 1023.52$ J/kg·K

$\mu = 2.53 \times 10^{-5}$ N·s/m^2

$k = 0.038$ W/m·K

$Pr = 0.682$

Properties of tube side produced gas at $\frac{600+250}{2}$ = 425°C (698.15 K) are:

$\rho = 0.4837$ kg/m^3

C_p = 1119.23 J/kg·K

μ = 3.09 x 10^{-5} N·s/m^2

k = 0.072 W/m·K

Pr = 0.69

For a square pitch layout,

$$D_e = \frac{4(P_T^2 - \pi d_o^2/4)}{\pi d_o} = \frac{4[0.0318^2 - \pi(0.0254^2/4)]}{\pi(0.0254)} = 0.0251 \text{ m}$$

$$C = P_T - d_o = 0.0318 - 0.0254 = 0.0064 \text{ m}$$

$$A_{c,s} = \frac{D_i C B}{P_T} = \frac{(0.34\,m)(0.0064\,m)(0.136\,m)}{0.0318\,m} = \dot{0}.00925 \text{ m}^2$$

$$V = \frac{m}{\rho A_{c,s}} = \frac{0.0092\,kg/s}{(0.7633\,kg/m^3)(0.00925\,m^2)} = 1.3033 \text{ m/s}$$

$$Re = \frac{\rho D_e V}{\mu} = \frac{(0.7633 kg/m^3)(0.0251 m)(1.3033 m/s)}{2.53 \text{ x } 10^{-5} \text{ N·s/m2}} = 987.21$$

$$T_w = \frac{1}{2}\left(\frac{t_i+t_o}{2} + \frac{T_i+T_o}{2}\right) = \frac{1}{2}\left(\frac{600+250}{2} + \frac{30+350}{2}\right) = 307.5°\text{C } (580.65 \text{ K})$$

From APPENDIX D.5, at the approximate wall temperature of 580.65 K,

μ_w = 2.95 x 10^{-5} N·s/m^2

Since Re < 2300, the flow is laminar.

Using Sieder's correlation which 0.48 < Pr < 16700,

$4.4 < (\mu/\mu_w) < 9.75$,

$(RePrd/L)^{\frac{1}{3}}(\mu/\mu_w)^{0.14} > 2$,

$$Nu = 1.86(RePrd/L)^{1/3}(\mu/\mu_w)^{0.14}$$

$$= 3.805$$

$$h_o = \frac{Nuk}{D_e} = \frac{(3.805)(0.038)}{0.0314} = 5.75 \text{ W/m}^2\text{·K}$$

For the tube side heat tranfer coefficient,

$$A_{c,t} = \frac{\pi d_i^2 N_t}{4N_p} = \frac{\pi(0.02362)^2(56)}{4(2)} = 0.0123 \text{ m}^2$$

$$V = \frac{m}{\rho A_{c,t}} = \frac{0.0078\ kg/s}{(0.4837\ kg/m^3)(0.0123\ m^2)} = 1.31 \text{ m/s}$$

$$Re = \frac{\rho d_i V}{\mu} = \frac{(0.4837\ kg/m^3)(0.02362\ m)(1.31\ m/s)}{3.09 \text{ x } 10^{-5} \text{ N·s/m2}} = 486.59$$

Since Re < 2300, the flow is laminar.

Using Sieder's correlation,

$$Nu = 1.86(RePrd/L)^{1/3}(\mu/\mu_w)^{0.14}$$

$$= 2.945$$

$$h_i = \frac{Nuk}{D_e} = \frac{(2.945)(0.072)}{0.0251} = 8.92 \text{ W/m}^2\text{·K}$$

To calculate the overall heat transfer coefficient,

$$U = \frac{1}{\frac{1}{h_o}+\frac{1}{h_i}} = \frac{1}{\frac{1}{5.75}+\frac{1}{8.92}} = 3.5 \text{ W/m}^2\text{·K}$$

To determine the shell side pressure drop,

$$\Delta P_s = \frac{fV_s^2 \rho D_i(N_b+1)}{2D_e \emptyset_s}$$

$$f = exp(0.576 - 0.19 lnRe)$$

$$= exp(0.576 - 0.19 ln 987.21) = 0.48$$

$$\emptyset_s = \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{2.53\times10^{-5}}{2.95\times10^{-5}}\right)^{0.14} = 0.9787$$

$$N_b = \frac{L}{B} - 1 = \frac{2}{0.25} - 1 = 7$$

$$\Delta P_s = \frac{(0.48)(1.3033)^2(0.7633)(0.0314)(7+1)}{2(0.0251)(0.9787)} = 3.18 \text{ Pa} = 0.000461 \text{ psi}$$

Since 0.000354 < 5, the shell side pressure drop is acceptable.

For tube length,

$$Q = \dot{m}C_p\Delta T = (0.0078\text{kg/s})(1119.23\ \text{J/kg}\cdot\text{K})(600\text{-}250)\text{K} = 3055.5\ \text{W}$$

$$A_s = \frac{Q}{U_f(\Delta T_{lm})F}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)} = \frac{(600-350)-(250-30)}{ln((600-350)/(250-30))} = 234.7\ \text{K}$$

$$A_s = \frac{3055.5\ W}{(3.54\ \text{W/m}^2\cdot\text{K})(234.7\ K)(0.52)} = 7.07\ \text{m}^2$$

$$L_t = \frac{A_s}{\pi d_o L_t} = \frac{7.07\ m^2}{\pi(0.0254\ \text{m})(56)} = 1.58\ \text{m}$$

This is rounded-off to 1.6 m.

To calculate the tube side pressure drop,

$$\Delta P_t = \left(\frac{4fLN_p}{d_i} + 4N_p\right)\frac{fV_t^2}{2}$$

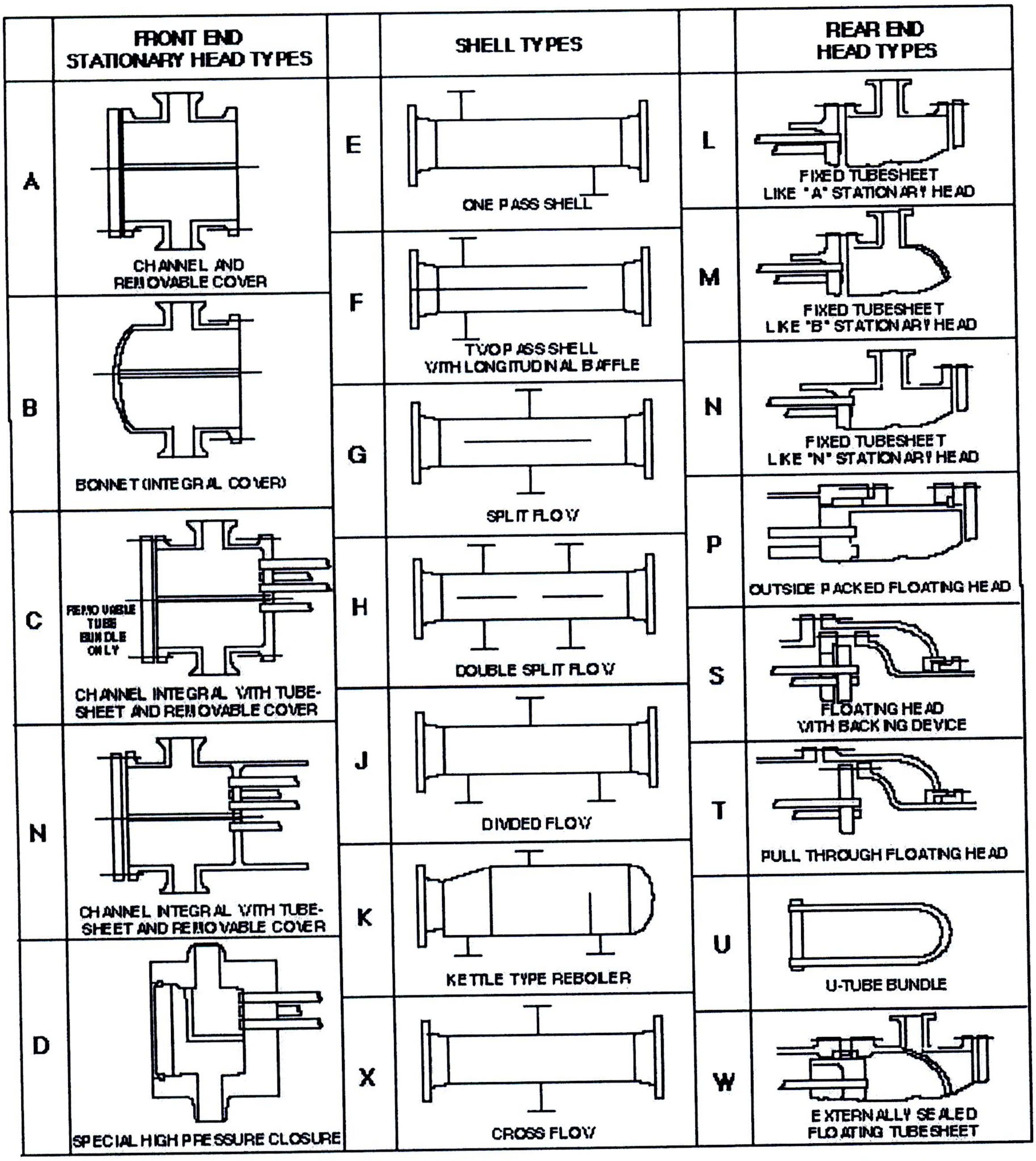
$$f = 0.0035 + \frac{0.265}{Re^{0.42}} = 0.0035 + \frac{0.265}{486.59^{0.42}} = 0.023$$

$$\Delta P_t = \left(\frac{4(0.023)(1.6\ m)(2)}{(0.02362\ \text{m})} + 4(2)\right)\frac{(0.023)(1.31\ m/s)^2}{2} = 0.31\ \text{Pa} = 0.000045\ \text{psi}$$

From side estimation and rating of shell and tube heat exchnger calculation above, the length of shell and tube heat exchanger between tube sheets is 1.6 m.

APPENDIX D.1

TEMA HEAT EXCHANGER LAYOUT DESIGNATION



The first letter of the three-letter TEMA-type designation describes the heat exchanger front-end or stationary head type. The first letter is selected from the five types that are illustrated in the first column.

The second letter of the three letter TEMA-type designator describes the heat exchanger shell, and it is selected from the seven types that are shown in the middle column. This letter may be omitted in a specification or proposal if a shell is not included in the equipment that is to be purchased (for example, if the specification or a proposal is to be used for the purchase of a replacement tube bundle with tubesheet[s], only).

The third letter of the three letter TEMA-type designator describes the heat exchanger rear-end or floating-head type, and it is selected from the eight types that are shown in the right-hand column.

Considerations for Selecting Exchanger Component Options

There are five stationary head types that are used in shell-and-tube exchangers. Table-1 lists the considerations that are used to select the appropriate stationary head type for specific applications. Table-2 lists the considerations that are used to select the appropriate shell design. Selection criteria for the rear end are listed in Table-3.

Table-1 Selection Consideration for Stationary Head

Type	Description	Selection Consideration
A	Channel and Removable Cover	The most common type of head that is used in shell-and-tube heat exchangers. Used with fixed tubesheet, U-tube, and floating head exchangers. In most cases, the bundle is removable for mechanical cleaning.
B	Bonnet or Removable Channel with Integral Cover	Normally used only for low-fouling tubeside services. Used with fixed tubesheet, U-tube, and floating head exchangers. Less expensive than Type A head.
C	Channel Integral with Tubesheet and Removable Cover	Used with some types of fixed tubesheet exchangers and reboilers.
N	Channel Integral with Tubesheet, Shell, and Removable Cover	Shellside fluid must be relatively low-fouling so that chemical cleaning can be used. Not recommended for use with U-tube or floating head exchangers because of maintenance difficulties.
D	Special High Pressure Closure	Special high pressure head that is used when the tubeside design pressure exceeds approximately 10340 kPa (1500 psi).

Table-2 Selection Consideration for Shell Design

Type	Description	Selection Consideration
E	Single Pass	The most common shell design.
F	Two Pass	Two pass affords slightly better heat transfer than single pass because two passes on the shell side more closely approximates counter-current flow. In order to avoid an excessively thick longitudinal baffle, two pass should not be used with a shellside pressure drop greater than approximately 70 kPa (10 psi). Shellside temperature range should be limited to 175°C (350°F) to avoid both excessive heat leakage through the baffle and thermal stress in the baffle, the shell, and the tubesheet.
G	Split Flow	Typically used in condensing and boiling services to reduce pressure drop and to enhance heat transfer duty.
H	Double Split Flow	
J	Divided Flow Shell	
X	Cross Flow	
K	Kettle-Type Reboiler	Typically used for boiling/vaporizing services. The large shell promotes heat transfer and vapor disengagement.

Table-3 Selection Consideration for Rear Ends

Type	Selection Consideration
L	Used with fixed tubesheet exchangers when the tubes must be cleaned mechanically.
M and N	Used, if necessary, with fixed tubesheet exchangers when the tubes can be chemically cleaned.
P	Not recommended because of the tendency of packed joints to leak. Type P heads should never be used with shellside hydrocarbons or toxic fluids.
S and T	Removable bundle designs. The floating head in an S-type exchanger has a split backing ring that reduces shell diameter requirements and that maintains high thermal efficiency. For maintenance reasons, generally Type T head is preferred, which allows the bundle to be more easily removed.
U	Used with U-tube bundle where tubeside does not need mechanical cleaning. Typically, a formed head is used on the shell, although a bonnet-type head can be used also.
W	Uses a packed joint to separate the tubeside and shellside fluids. Not recommended because of tendency of packed joints to leak.

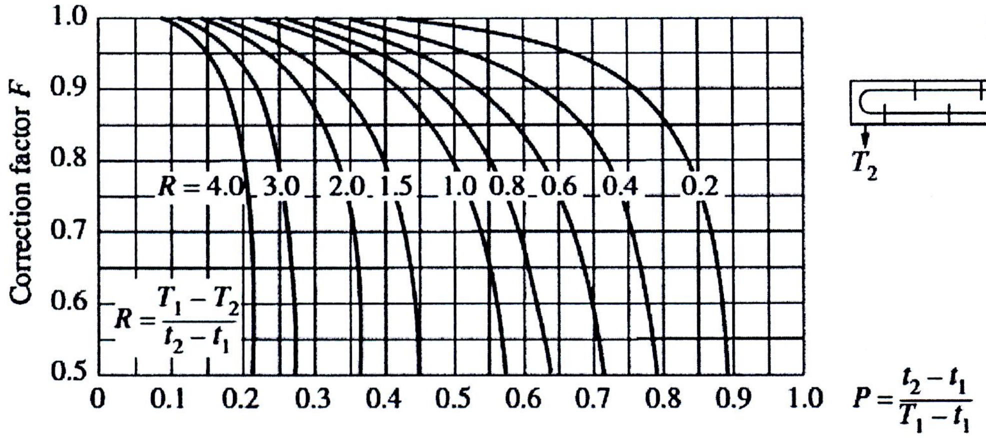
APPENDIX D.2

TUBE DIMENSIONAL DATA

Tube OD (in)	BWG	Thickness	Tube ID (in)
0.5	12	0.109	0.282
	14	0.083	0.334
	16	0.065	0.370
	18	0.049	0.402
	20	0.035	0.430
0.75	10	0.134	0.482
	11	0.120	0.510
	12	0.109	0.532
	13	0.095	0.560
	14	0.083	0.584
	15	0.072	0.606
	16	0.065	0.620
	17	0.058	0.634
	18	0.049	0.652
1	10	0.134	0.732
	11	0.120	0.760
	12	0.109	0.782
	13	0.095	0.810
	14	0.083	0.834
	15	0.072	0.856
	16	0.065	0.870
	17	0.058	0.884
	18	0.049	0.902
1.25	10	0.134	0.982
	11	0.120	1.010
	12	0.109	1.032
	13	0.095	1.060
	14	0.083	1.084
	15	0.072	1.106
	16	0.065	1.120
	17	0.058	1.134
	18	0.049	1.152

APPENDIX D.3

LMTD CORRECTION FACTOR (shell and tube heat exchanger with one-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes)



APPENDIX D.4

TUBE-SHELL LAYOUTS

1 inch OD on 1.25 inch Square pitch					
Shell ID (in.)	1 pass	2 pass	4 pass	6 pass	8 pass
8	21	16	14	0	0
10	32	32	26	24	0
12	48	45	40	38	36
13.25	61	56	52	48	44
15.25	81	76	68	68	64
17.25	112	112	96	90	82
19.25	138	132	128	122	116
21.25	177	166	158	152	148
23.25	213	208	192	184	184
25	260	252	238	226	222
27	300	288	278	268	260
29	341	326	300	294	286
31	406	398	380	368	358
33	465	460	432	420	414
35	522	518	488	484	472
37	596	574	562	544	532
39	665	644	624	612	600

APPENDIX D.5

THERMOPHYSICAL PROPERTIES: Air at 1 atm

T (K)	ρ density (kg/m^3)	c_p specific heat (kJ/Kg-K)	μ viscosity (10^{-7}N-s/m²)	ν kinematic viscosity (10^{-6}m^2/s)	k thermal conductivity (10^{-3}W/m-K)	α thermal diffusivity (10^{-6}m^2/s)	Pr Prandtl number
300	1.1614	1.007	184.6	15.89	26.3	22.5	0.707
350	0.995	1.009	208.2	20.92	30.0	29.9	0.700
400	0.8711	1.014	230.1	26.41	33.8	38.3	0.690
450	0.7740	1.021	250.7	32.39	37.3	47.2	0.686
500	0.6964	1.030	270.1	38.79	40.7	56.7	0.684
550	0.6329	1.040	288.4	45.57	43.9	66.7	0.683
600	0.5804	1.051	305.8	52.69	46.9	76.9	0.685
650	0.5356	1.063	322.5	60.21	49.7	87.3	0.690
700	0.4975	1.075	338.8	68.10	52.4	98	0.695
750	0.4643	1.087	354.6	76.37	54.9	109	0.702
800	0.4354	1.099	369.8	84.93	57.3	120	0.709
850	0.4097	1.110	384.3	93.80	59.6	131	0.716
900	0.3868	1.121	398.1	102.9	62.0	143	0.720
950	0.3666	1.131	411.3	112.2	64.3	155	0.723
1000	0.3482	1.141	424.4	121.9	66.7	168	0.726
1100	0.3166	1.159	449	141.8	71.5	195	0.728
1200	0.2902	1.175	473	162.9	76.3	224	0.728
1300	0.2679	1.189	496	185.1	82	238	0.719
1400	0.2488	1.207	530	213	91	303	0.703

APPENDIX E

GAS CLEANING DEVICES

GAS CLEANING DEVICES

E.1 CYCLONE

Cyclones are usually designed with geometric similarity such that the ratios of the dimensions remain constant at different diameters, and this dimensions can be expressed in term of the body diameter.

E.1.1 Prediction of Pressure Drop

Pressure drop is an important parameter because it relates directly to operating costs. Higher efficiencies for a given cyclone can be obtained by higher inlet velocities, but this also increases the pressure drop, and a trade-off must must be made. Shepherd and Lapple found empirically that pressure drop depends inversly on exit diameter squared (Buonicore and Davis, 1992):

$$H_v = K\frac{H_c B_c}{D_e^2} \tag{E.1}$$

Where H_v = the number of velocity heads

K = an empirical constant wiith a value of 16 for a tangential inlet cyclone and 7.5 for one with an inlet vane

The pressure drop is as follows (Buonicore and Davis, 1992):

$$\Delta P = \frac{1}{2}\rho_g V_g^2 H_v \tag{E.2}$$

Where ΔP = pressure drop, Pa (N/m^2)

ρ_g = gas density, kg/m^3

V_g = gas velocity, m/s

Pressure is a function of the square of inlet velocity, so too high a velocity will cause excessive pressure drop. On the other hand, too low a velocity will cause a low efficiency. A very high velocity would also actually decrease efficiency because of increased turbulence and saltation/reentrainment of particles. The recommended minimum gas velocity for conveying medium density dust is 15 m/s, and for heavy dust(metal turnings) is 25 m/s. Common ranges of pressure drops are as follows (Reed and Das, 1988):

Low-efficiency cyclones	2-4 inches water
Medium-efficiency cyclones	4-6 inches water
High-efficiency cyclones	8-10 inches water

E.1.2 Advantages and Disadvantages of Cyclone

Advantages

1. Low cost of construction.
2. Relatively simple equipment with few maintenance problems.
3. Relatively low operating pressure drop in the range of approximately 2 to 6 inches water column.
4. Temperature and pressure limitations imposed only by the materials of construction used.
5. Dry collection and disposal.
6. Relatively small space requirements.

Disadvantages

1. Relatively low overall particulate collection efficiencies, especially on particulates below 10 μ in size.
2. Inability to handle tacky materials.

E.1.3 Cyclone Design

1. Calculate the gas flow rate

A 50 kW thermal require 40 Nm^3/h of producer gas, which corresponds to a gas energy output of 180 MJ/h. The volume of gas at the cyclone inlet temperature of 600°C will be

$$40\ Nm^3/h\ \frac{873.15\ K}{273.15\ K} = 127.86\ m^3/h$$

2. Calculate the body diameter, D_c

From the proportion of standard cyclone dimensions, cyclone inlet width, B_c, equal to $D_c/4$ and the cyclone inlet height, H_c, equal to $D_c/2$, then substitute these variable into the cyclone inlet area, A, as Equation (E.3) to calculate the body diameter.

$$v = \frac{Q}{A} \tag{E.3}$$

where $A = H_c \times B_c$; $H_c = D_c/2, B_c = D_c/4$

substitute H_c and B_c in Equation (E.3):

$$v = \frac{Q}{D_c/2 \times D_c/4}$$

$$v = \frac{8 \times Q}{D_c^2}$$

$$D_c^2 = \frac{8 \times Q}{v}$$

The body diameter equal to 0.1192 m which is rounded-off to 0.12 m.

3. Calculate the dimension

From the standard cyclone dimensions, the dimension of design cyclone are:

D_c	0.120
$B_c = D_c/4$	0.030
$D_e = D_c/2$	0.060
$H_c = D_c/2$	0.060
$L_c = 2D_c$	0.240
$S_c = D_c/8$	0.015
$Z_c = 2D_c$	0.240
$J_c = D_c/4$	0.030

4. Calculate the cyclone pressure drop

From Equation (E.1 and E.2), the cyclone pressure drop is 1.216 kPa.

E.2 BAGHOUSE FILTER

Baghouse filter removes dust from a gas stream by passing the stream through a porous fabric. Dust particles form a more or less porous cake on the surface of the fabric. It is normally this cake that actually does the filtration. Baghouse filters are usually constructed using many cylindrical bags that hang vertically in the unit. The number of bags can vary from as few as four to more than a thousand depending on the size of the control system.

E.2.1 Filtration Process

The fabric is covered with a dust cake and the dust cake is of continually varying thickness. Once the dust cake has reformed after each cleaning, sieving is probably the dominant mechanism. As particles approach the porous mass of dust that constitutes the cake, they either will strike one or more surface particles or enter a pore. If the particles is larger than the pore it attempts to enter, it will be sieved out. If the particle is smaller than the pore it enter, it will continue traveling through the pore until it touches the pore wall and adheres; or until the pore narrows to dimensions smaller than the particle, causing the particle to be sieved oout; or until the particle passes through the dust pore and a fabric pore and exits on the clean-air side of the air filter._Ordinarily, only one out of 1000, or an even 10000, particles finds its way through the filter.

E.2.2 Pressure Drop

Pressure drop describes the resistance to air flow across the baghouse: the higher the pressure drop, the higher the resistance to air flow. The pressure drop of a system is determined by measuring the difference in total pressure at two points, usually the inlet and outlet. The total system pressure drop can be related to the size of the fan that would be necessary to either push or pull the exhaust gas through the baghouse. A baghouse with a high pressure drop would need more energy or possibly a larger fan to move the exhaust gas through the baghouse. Many different relationships have been used to estimate the pressure drop across a fabric filter. In a baghouse, the total pressure drop is a function of the pressure drop across both the filter and the deposited dust cake. Some pressure losses due to friction also occur as the gas streammoves through the baghouse.

E.2.3 Filtration Velocity: Air-To-Cloth Ratio

The terms filtration velocity and air-to-cloth (A/C) ratio can be used interchangeably. The formula used to express filtration velocity is

$$v_f = \frac{Q}{A_c} \quad \text{(E.4)}$$

Where: v_f = filtration velocity, ft/min (cm/sec)

Q = volumetric air flow rate, ft^3/min (cm^3/sec)

A_c = area of cloth filter, ft^2 (cm^2)

Air-to-cloth ratio is a measure of the amount of gas driven through each square foot of fabric in the baghouse. Typical units used to express the A/C ratio are (ft^3/min)/ft^2 or (cm^3/sec)/cm^2. Comparisons of the cleaning methods are given in Table E.1.

Table E.1 Typical air-to-cloth ratio (filtration velocity) comparisons for three cleaning mechanisms

Cleaning mechanisms	Air-to-cloth ratio		Filtration velocity	
	(cm3/sec)/cm2	(ft3/min)/ft2	cm/sec	ft/min
Shaking	1 to 3:1	2 to 6:1	1 to 3:1	2 to 6:1
Reverse-air	0.5 to 2:1	1 to 4:1	0.5 to 2:1	1 to 4:1
Pulse-jet	1 to 7.5:1	2 to 15:1	1 to 7.5:1	2 to 15:1

Note: These may vary for specific applications.

The A/C ratio (filtering velocity) is a very important factor used in the design and operation of a baghouse. Improper ratios can contribute to inefficient operation of the baghouse. Operating at an A/C ratio that is too high may lead to a number of problems. Very high ratios can cause compaction of dust on the bag resulting in excessive pressure drops. In addition, breakdown of the dust cake could also occur, which in turn results in reduced collection efficiency. The major problem of a baghouse using a very low A/C ratio is that the baghouse will be larger in size, and therefore have a higher capital cost.

E.2.4 Component of Baghouse Filter

Filter Medium and Support

The particle collection surface is composed of the filtering material and a support structure. The baghouse designs contain felted fabric or woven cloth as the filtering medium. Woven filters are made of yard consisting of fibers constructed into fabric with a defenitely repeated pattern. Felted filters are composed of randomly placed fibers that are compressed into a mat and attatched to a loosely woven backing material called a scrim. Felted filters are normally thicker than woven filters. The cloth can be supported at the top and bottom of the bag by metal rings or clasps; or by an internal cage that completely supports the entire bag. Dust is collected on either the inside or outside of the fabric material depending on the baghouse design.

Housing or Shell

Baghouses are constructed as single or compartmental units. The single unit is generally used on small processes that are not in continuous operation such as grinding and paint spraying processes. Compartmental units consist of more than one compartment and are used in continuous operating processes with large exhaust volumes such as electric melt steel furnaces and industrial boilers. Compartmetalized units can have a compartment off-line for bag cleaning and maintenance while the remaining baghouse compartments continue to filter. In both cases, the bags are housed in a shell made of a rigid metal material. Often it is necessary to include insulation with the shell particularly when treating high temperature flue gas. This is done to prevent moisture or acid mist contained in the flue gas from condensing in the unit, thus causing corrosion and rapid deterioration of the baghouse.

Baghouse Filter Material

Bag filters can be made of woven or nonwoven materials. Nonwoven materials can further be divided as felted or membrane. Most bags are either completely or partially made by weaving since nonwoven fabrics are generally attached to a woven base called a *scrim*. **Woven** filters are made of yarn with a definite repeated pattern. **Felted** filters are composed of randomly placed fibers compressed into a mat and attached to loosely woven backing material. A **membrane** filter is a special treatment where a thin, porous membrane (expanded polyfluorocarbon) is bonded to the scrim, or support fabric. Woven filters are generally used with low energy cleaning methods such as shaking and reverse-air. Felted fabrics are usually used with higher energy cleaning systems such as pulse-jet cleaning. Membrane filters were developed in efforts to achieve high efficiency particle capture and to handle flue gas conditions where high moisture and resulting high pressure drop problems frequently occur.

E.2.5 Important Fiber Characteristics

When selecting a fiber for gas filtration, attention must me paid to the following factors that interact and thus must be considered together.

1. Temperature: The fiber must be have a maximum continuous service temperature higher than the normal temperature of the application. If temperature surges above the normal range occure, the ability of the fiber to with stand the expected conditions of surge temperature and duration must be considered.
2. Corrosiveness: The ability of the fiber to resist physical degradation from the expected application levels of acid, alkalies, solvents, or oxidizing agents must be considered.
3. Hydrolysis: Effects of the expected level of humidity must be taken into account.
4. Dimensional stability: If the fiber is expected level of humidity must be taken into account.
5. Cost: As with any enginerring product, the least costly selection that will meet overall requirements is usually the best selection.

Table E.2 lists a number of typical fibers used for fabric filters. The properties of the listed fibers include temperature limits, acid and alkali resistance, abrasion resistance, and relative bag costs.