BACHELOR THESIS – ME141502

STUDY ANALYSIS OF ORGANIC RANKINE CYCLE SYSTEM AS POWER GENERATION IN CEMENT PLANT

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FACULTY OF MARINE TECHNOLOGY
INSTITUT TEKNOLOGI SEPULUH NOPEMBER
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ANALISIS STUDI SISTEM ORGANIC RANKINE CYCLE SEBAGAI PEMBANGKIT LISTRIK DI PABRIK SEMEN

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Dosen Pembimbing:
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JURUSAN TEKNIK SISTEM PERKAPALAN
FAKULTAS TEKNOLOGI KELAUTAN
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APPROVAL FORM

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Submitted to Comply One of The Requirement to obtain a Bachelor of Engineering Degree
on
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S-1 Program Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

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ABSTRACT

Cement plant produce large amount of heat source in cement making process, due to inefficiency of system there still waste heat available in form of flue gas that can be utilize. Flue gas in cement plant can be utilized as alternative power generation. With the 200-300^{o}C temperature output range of flue gas from suspension preheater and air quenching cooler (AQC) in cement plant, organic rankine cycle (ORC) can be suitable option for alternative power generation. ORC is development of rankine cycle, the different is the working fluid in ORC using refrigerant. In cement plant that produce 8466 TPD kiln production, used flue gas from suspension preheater to dry raw material and produce 163888 m^3/h flue gas from AQC that still not utilized. Flue gas with 235^{o}C temperature from AQC can utilized for power generation purpose using ORC system. Waste heat recovery calculation carried out to know the potential recovery. Operating condition of the ORC system will determine power produced that can be generated and ORC components calculated and selected according to the operating condition of the system. Using R141b as working fluid with 8 bar pressure and 110^{o}C temperature inlet to turbine, power produced by turbine is 666 kW. For the components, evaporator and condenser use shell and tube heat exchanger, with evaporator heat transfer area is 676.49 m^2 while condenser has 740.729 m^2 of heat transfer area. And for working fluid pump it needs 16.392 Kw power to pump R141b back to evaporator.

Keywords : Cement plant, Flue gas, Power generation, Organic Rankine Cycle, Heat exchanger

ANALISIS STUDI SISTEM ORGANIC RANKINE CYCLE SEBAGAI PEMBANGKIT LISTRIK DI PABRIK SEMEN

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ABSTRAK

Pabrik semen memghasilkan sumber panas dengan jumlah yang banyak selama proses pembuatan semen, dikarenakan inefisiensi sistem terdapat panas buang yang tersisa dalam bentuk gas buang yang dapat dimanfaatkan. Gas buang di pabrik semen dapat dimanfaatkan sebagai alternatif pembangkitan listrik. Dengan *range* temperatur 200-300°C gas buang dari *preheater* dan *air quenching cooler* (AQC) di pabrik semen, *organic rankine cycle* (ORC) cocok untuk digunakan sebagai alternatif pembangkitan listrik. ORC adalah pengembangan dari *rankine cycle*, yang membedakan adalah ORC menggunakan refrigerant sebagai fluida kerja. Di pabrik semen yang memproduksi kiln 8466 TPD, menggunakan gas buang dari *preheater* untuk pengering material awal dan menghasilkan 163888 m^3/h gas buang dari AQC yang belum digunakan. Gas buang dengan suhu 235°C dapat dimanfaatkan untuk pembangkitan lisrik menggunakan sistem ORC. Perhtiungan *waste heat recovery* dilakukan untuk mengetahui potensi yang dapat dimanfaatkan. Kondisi operasi dari sistem ORC akan mementukan daya yang dapat dihasilkan dan komponen ORC dihitung dan dipilih berdasarkan kondisi operasi dari sistem. Menggunakan fluida kerja R141b dengan tekanan 8 bar dan suhu 110°C pada *inlet* turbin, menghasilkan daya sebesar 666 kW. Untuk komponen - komponennya, *evaporator* dan *condenser* menggunakan s*hell and tube heat exchanger*, dengan area perindahan panas *evaporator* sebesar 676.49 m^2 sedangkan area perpindahan panas *condenser* 740.729 m^2. Dan pompa fluida kerja membutuhkan daya sebesar 16.392 kW untuk memompa R141b kembali ke *evaporator*.

Kata Kunci : Pabrik Semen, Gas buang. Pembangkitan listrik, *Organic Rankine Cycle*, *Heat exchanger*

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CHAPTER I
INTRODUCTION

1.1 Background

Nowdays industries sector develop rapidly along with development of technology that support the industries. And development of industries itself need energy to continue the working process. Every process in the industries from the starting process of producing something from raw material until raw material become product that has added value use energy in the making process. So energy is very important to support production activity in industries. Cement industry also including industry that need large amount of energy for transforming raw material into cement, due to all process mostly used machine to control the production and also need thermal energy. There are some steps conducted in cement making process from the delivery raw material to the factory from the quarry until the packing process and deliver the cement to the consumer. And below the process of cement making in cement industry will be explained.

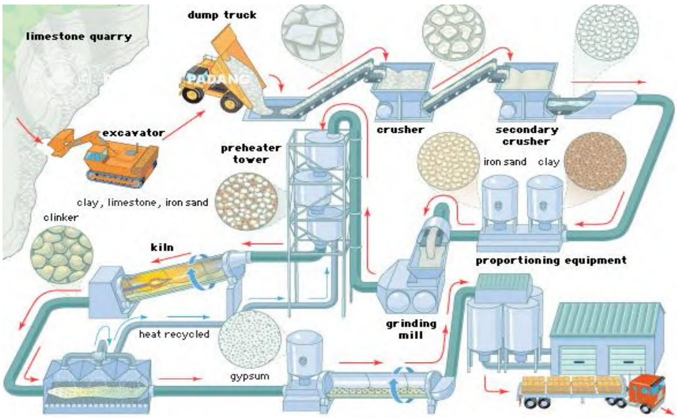

Figure 1.1 Cement making Process
(Source :www.britannica.com/technology/cement-building-material)

Generally, there are 5 main steps in cement making process, and the first step is raw material preparation. The raw material will be collected according to cement making composition. And the raw material consists of, limestone ($CaCO_3$), Clay (SiO_2 and Al_2O_3), Silica Sand that contain SiO_2, Iron Ore (Fe_2O_3) and other material in small amount. After all material collected raw material will come into the crusher, usually there are two crushers, first will crush limestone into baseball size than in the second crusher will limestone will crusehed into size of gravel.

And the second step is blending and grinding process. After material already crushed the raw material will be scaled appropriately regarding requirement of cement contain, and blended into proper proportion according laboratory analysis in the factory. Blended material will come into grinding wheel and become small blend material. After that the step is clinker production. Clinker is red hot particle that come from $2CaO + SiO_2$, so it will become Ca_2SiO_4. In this process blended material will come through chamber tower that called suspension preheater and and got heated by hot gases, the hot gases itself come from the kiln process. After that raw material will come into kiln (hot rotating furnace) that placed horizontally sloped that built from steel lined with firebrick and has cylindrical firebrick. The kiln itself rotate 1-3 rpm, and give intense heat to the blended material approximately up to 1870^{o}C. Then this material that already through kiln called clinker.

After that the clinker get cooled by air in air quenching cooler (AQC) and become gray powder Then the clinker come through ball mill that filled with steel ball. The ball mill rotates and the clinker that goes through it will break into powder because pressed by steel ball, this step called cement grinding. Then the last step is packing of cement, the cement will conveyed into silos for shipping for large scale or packed into small scale for local use.

From cement making process above (Raw material preparation, Blending and Grinding, Clinker production, Cement grinding, Packing). Known that cement production process need large amount of energy.

And in the cement making process, the there will be waste heat in form of flue gas that will be available. Usually flue gas will go through stack and wasted to environment with around 200-350^{o}C temperature. Flue gas usually come from suspension pre heater and air quenching cooler of cement plant. In some cement plant suspension preheater flue gas used to dry raw material of cement and coal while air quenching cooler flue gas usually out to the stack. The available flue gas can be utilized for another purpose that is as alternative power generation. In cement plant waste heat recovery technology that used as alternative power generation already established since 1970 that developed by Kawasaki Heavy Industry (KHI). Using organic rankine cycle (ORC) as system to generate alternative power generation can be suitable for cement plant flue gas which has output temperature 200-350^{o}C.

Organic rankine cycle is development from rankine cycle, the components consist of evaporator, turbine, condenser and pump. Working fluid in ORC use refrigerant that has lower boiling point than water. So with the temperature flue gas in cement plant, refrigerant can be in superheated condition to rotate turbine blade and generating power as alternative power generation in cemen plant.

1.2 Statement of Problems

Since many flue gas in cement making process (in suspension preheater and air quenching cooler) is not utilize yet, it will usually just wasted to stack then to environment. With 200-350^{o}C actually the flue gas can be used for another purpose, for example in some cement plant of preheater flue gas used to dry raw material in raw grinding process.

When the flue gas no utilize yet, it can be used for another purpose like alternative power generation with ORC. With ORC that use refrigerant as working fluid, as mention before that refrigerant that has lower boiling point than water, which can make the working fluid in superheated phase with 200-350^{o}C.

The alternative power generation concept using ORC is, flue gas will evaporate working fluid in ORC closed cycle and superheated working fluid will expand to turbine. From expansion of working fluid turbine will rotate and drive the shaft generator and transmit the power to generator and producing alternative power generation in cement plant

The process of implementation starts with analyze the potential flue gas in the cement plant in preheater and air quenching cooler. If the flue gas source already used, it can’t be made for alternative power generation due to the temperature already decreasing. Then calclulation of potential power of flue gas conducted after that the operating condition in ORC system created, to ensure the operating condition is right (the phase of refrigerant is suitable for ORC) simulating through soconducted from that value of power produced will be known too. Then the component of ORC (evaporator, turbine, condenser and pump) can be designed with manual calculation.

And from the statement we can summarize into several question:

1. Is there any cement plant potential that not yet utilize in cement plant?
2. How much energy that can be generated using ORC utilizing the flue gas recovery?
3. What are the requirements of the organic rankine cycle components?

1.3 Research Limitation

The research limitation on this bachelor thesis are:

1. System condition in equilibrium state
2. Not including analysis of the piping
3. Not including economical calculation

1.4 Research Objectives

The research limitation on this bachelor thesis are:

1. Investigating availability of flue gas potential in cement plant.

2. Calculating potential alternative power generation and power produced can be generated from flue gas in cement plant.
3. Calculating and selecting appropriate component for the ORC system with manual calculation.

1.5 Research Benefits

1. Understanding waste heat potential in cement plant to make alternative power generation using ORC system.
2. Optimizing waste resource in cement plant to make alternative power generation.

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CHAPTER II
LITERATURE STUDY

2.1 Waste Heat Recovery

Waste heat arise from inefficiencies of equipment and thermodynamic limitation on equipment. And waste heat can be use as other purpose in many way, recovering industrial waste heat can be achieved by numerous methods. The heat can be reused in the same process or transfer into another process. Or we can use the waste heat to preheat combustion air or feedwater in boiler. By doing preheating in the feedwater before the feed water come into the boiler, the amount of energy required to heat the water can be reduced. In another way heat can be transfer into another process like heat exchanger, it could be used to transfer heat from combustion exhaust gases to hot air needed for drying material. Such methods for recovering waste heat can help facilities reduce their fossil fuel consumption and pollutant emissions even operating cost. These are some example of waste heat and recovery sources and optional purpose to utilize it:

Table 2.1 Example of Waste Heat Sources and Utilization

Waste Heat Source		Uses for Waste Heat
1. Combustion Exhaust :		1. Combustion Air Preheating
	1. Boiler Feedwater Preheating	2. Boiler Feedwater Preheating
	2. Power Generation	3. Power Generation
	3. Steam generation	4. Steam Generation
	4. Space Heating	5. Space Heating
	5. Water Preheating	6. Water Preheating
2. Process off Gases		
	Steel Electric arc Furnace	
	Alumunium Reverberatory Furnace	
3. Cooling Water Form		
	Furnaces	
	Internal Combustion Engines	

(Source : Waste Heat Recovery : Technology and Opportunities in U.S. Industry)

2.2 Organic Rankine Cycle (ORC)

As mentioned by Kusumaning, et al, (2015) (Organic rankine cycle (ORC) is close cycle that circulated, the cycle use refrigerant as working fluid. The cycle convert heat energy into electricity, and even the heat is in low temperature condition organic rankine cycle can work too, due to the refrigerant can vaporize in low temperature. The advantage of using this ORC cycle are, it can work in low temperature and the simplicity of the system, the system normally consists of condenser, pump, evaporator and turbine, but in some case there is another heat exchanger like recuperator to increase the efficiency of the system, and using thermal oil to conduct electricity to the refrigerant.

Ozdil, et al (2015) state that Organic Rankine Cycle is generally preferred for the processes having low temperature like $T < 150^{o}$ C. This process named as Organic Rankine Cycle owing to usage of the organic fluid as working fluid instead of water and high pressure steam. The organic fluid that is used in ORC, has high molecular weight liquid with low boiling temperature than water. Energy is the measurement of the maximum useful work that can be obtained in the system. Therefore, it has become more important subject than energy in order to specify the useful work. Moreover, the exergy can be called as irreversibility in thermodynamic point of view. Because of the irreversibility, exergy can be consumed or destroyed in the processes. The consumption of the exergy rate in a process is directly related with the entropy generation.

ORC is a close cycle that used refrigerant as a work fluid to drive the turbine. The working fluid will change from liquid phase into vapor phase because of the heat from exhaust gas in evaporator. Exhaust gas in this case will come from waste of kiln process in the cement making process. After the turbine spin it connected with winding and generate electricity. And working fluid will come into the condenser so it will condense by water and become liquid phase again. After that the working fluid pumped again into the evaporator. So the ORC system consist of 4 main components, that is:

1. Evaporator: Evaporator here used to change the phase of fluid from liquid into the vapor by give heat to the fluid from exhaust gas. And the vapor formed will move upward into the turbine.
2. Turbine: Turbine is a driving machine that change potential energy into kinetic energy then change into mechanical energy with the spin that driven by vapor fluid. The vapor fluid generate work by expanding in the turbine and will move into the condenser.
3. Condenser: Condenser is like evaporator it used to exchange the heat, but in this case the heat in vapor will absorb by water and the water will change into liquid phase again.
4. Pump: Pump used to move the fluid to another place that has different height. And the fluid used usually is in liquid phase. In this case the working fluid from condenser will suct and discharge buy pump into the evaporator again, so that's the close cycle of ORC will work circulatory.

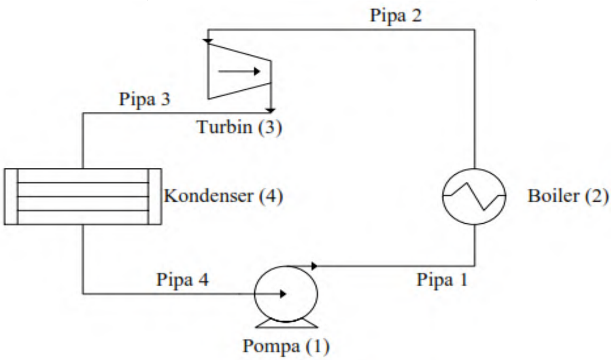


Figure 2.1 Rankine Cycle Model
(Source :Perancangan Sistem Kogenerasi Pada Pabrik Semen PT HOLCIM Indonesia Cilacap Plant)

Bardani, (2015) describe There are four step of work in organic rankine cycle for the power generation. This cycle operate ideally and not involve internal irreversibility. The cycles of the organic rankine cycle are:

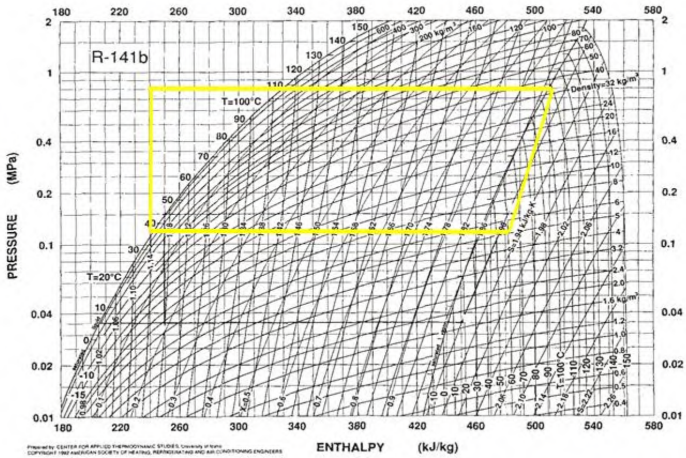


Figure 2.2 P-h Diagram of Working Fluid in ORC
(Source :ENSC 388 : Engineering Thermodynamics and Heat Transfer)

From the picture above we can describe process for every component in rankine cycle:

1. Compression (1-2): Pump will increase the pressure of working fluid and send the working fluid to evaporator. In pump the process condition is adiabatic (Q = 0).
2. Evaporation (2-3): In this process working fluid receives heat by isobaric (assumed that pressure inlet evaporator same as pressure outlet evaporator) from heat source that will used to increase temperature from pump to evaporator and changing in working fluid phase to superheated vapor.
3. Expansion (3-4): Working fluid that already in superheated vapor expand and rotate the turbine blade to transmit to shaft generator then to generator to produce power. When the working fluid expand the temperature is decreasing and the pressure as well.

4. Condensation (4-1): In condenser the process is also isobaric like in evaporator. Working fluid will change phase into liquid again and pumped to the evaporator.

2.3 Organic Rankine Cycle in Cement Plant

The previous research will be reviewing some studies and researchs that already done, in making power generation in cement plant by ORC system as well.

2.3.1 Previous Research

In the cement making process there are thermal energy in forms of hot air from air quenching cooler (AQC) and flue gas from in Suspension Preheater (SP) at temperature of 300°C-400°C are wasted in cement plant. Therefore, the waste energy can be utilized through application of power plant via waste heat recovery system. There are some advantages of using waste heat recovery systems in cement industry according to Confederation of Indian Industry (CII), there are:

1. Reduce Fuel consumption
2. Efficiency of heat energy and electricity
3. Bring down specific energy consumption of plant
4. Provide economic competitive advantage in the market
5. Mitigate the emission of green house gases which are affecting the environment.

And heat recovery potential in cement plant will be depend on some aspect like:

1. Moisture content in the raw material
2. Number of cyclone or stage in suspension preheater
3. Efficiency of cyclone, etc

Number of preheater stage in cement plant or the number of cyclone will affect the thermal energy consumption and waste heat recovery potential. More stage means higher consumption of thermal energy and lower waste heat recovery potential. The selection of preheater cyclone itself is depend on heat requirement of coal mill and raw mill, efficiency of cooler, restriction on the preheater tower height etc.

Beside the waste gas from the clinker from kiln so the clinker temperature is not too high and not risky to the other equipment suspension preheater there is another area that has waste gas excess, the cooler exhaust where air from cooler mostly vented into the atmosphere. Basic function of cooler is to recover the heat from hot like cement grinding. Rapid cooling from air also improves quality of clinker itself and its grind ability.

From Jayaraman's, (2009) report theoretically about 540 kcal is required to evaporate one kg water from raw material. Although practically in vertical roller mills requires 900 to 1100 kcal of heat per kg of moisture and ball mills required 750 to 850 kcal of heat / kg of moisture due to loss in mill outlet gas, radiation loss and ingress of false air. Following graphs below gives raw meal requirement and heat required in Mkcal/hour for various kiln production rates at different moisture level with following assumption:

1. Raw meal to clinker factor is 1,55
2. Heat requirement of 950 kcal/kg of water for raw mill
3. Raw mill running 22 hours/day

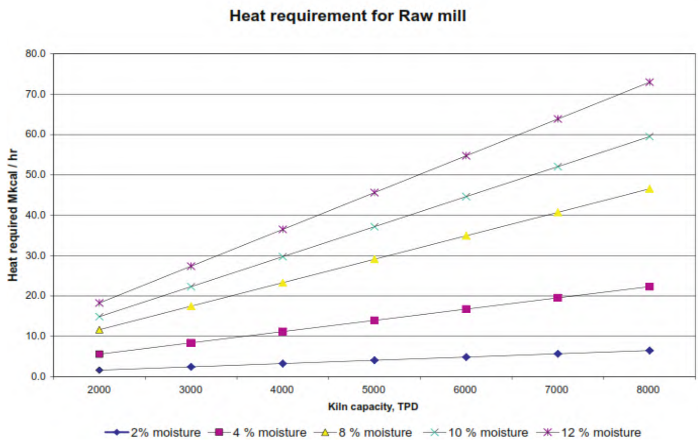


Figure 2.3 Heat Requirement for Raw Mill
(Source: Manual on Waste Heat Recovery in Indian Cement Industry)

The graph shows the kiln capacity in ton per day (TPD) and the heat requirement for drying raw material. And from the graph we can conclude that higher contain of moisture in material it will need higher heat to drying the material. And every kiln capacity has different heat requirement too, bigger capacity of kiln of course will lead into higher need of heat to drying the raw material. But according to the manual fact that 3000 TPD kiln of 4 stage has heat availability of 23,8 MKcal/hour (from graph), cement plant can handle moisture content of 10% and there wont be any excess heat available. On the other hand, if limestone moisture is less than 4% heat requirement will be around 5,6 Mkcal/hour and hence 18,2 Mkcal/hr (23,8-5,6) will be available for coal mill and waste heat recovery.

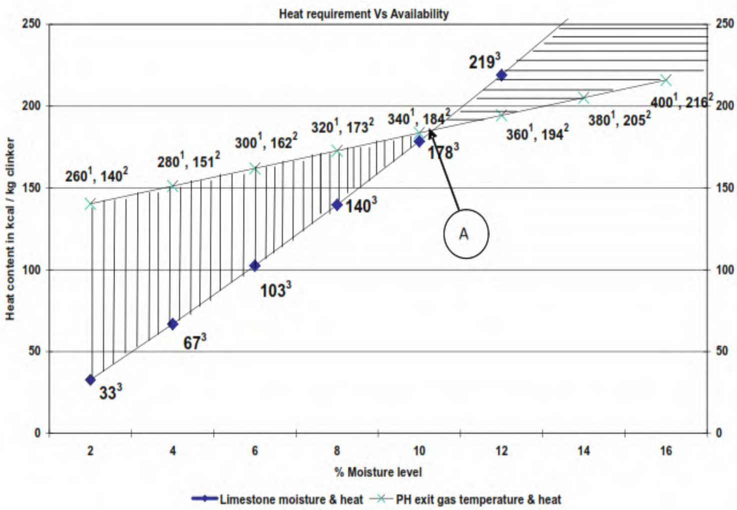


Figure 2.4 Heat Requirement vs Availability of Waste Heat Graph
(Source :Manual on Waste Heat Recovery in Indian Cement Industry)

From the number 1 above number determine the output gas temperature in celcius, number 2 above number determine the heat content available in the preheater gas and number 3 as shown in the graph represent the moisture level in kcal/ kg clinker.

From the graph above we can conclude where point A determine the intersection point between output temperature compare with heat content and moisture level. When the moisture level of raw material up to 12% there will be no waste heat gas available and the system will need extra heat to raw mill. But if the moisture level around 10% even less than that there will be waste heat available for recovery. In cement plant heat available that suitable for ORC that only need low temperature from gas can obtained from clinker cooler exhaust that can be recovered effectively by installing waste heat power plant like ORC. Moreover, the ORC is a cycle that environmentally friendly and well suited to work with lower temperatures to generate electricity on continuous basis without interfering clinker production process. Low temperature heat is converted into useful work to generate electricity.

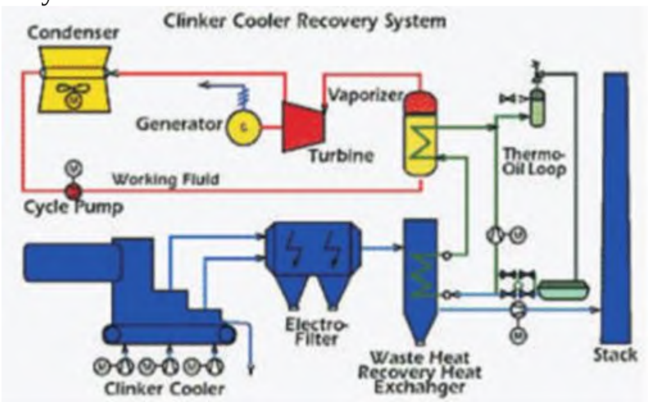


Figure 2.5 Clinker Cooler Recovery System
(Source: Manual on Waste Heat Recovery in Indian Cement Industry)

Prasat, (2010) in his presentation about “4 MW Waste Heat Recovery Power Plant” in UltraTech Cement Limited on 2010, from the data said that on design data of technical system from waste heat recovery power generation (WHRPG) in UltraTech Cement the system designed to generate 4 MW gross power from the clinker cooler as heat source with mass flow rate capacity 485 ton/ hour and 320°C temperature. This system use thermal oil as conductor of heat from waste heat to ORC and use pentane as working fluid. Input temperature and pressure to the turbin is set 204 °C and 23,8 bar.

From Bardani, (2015) describe that exhaust gas from cement production, produce from the excess combustion of fuel in “kiln” and the hot air that come from cooling process of clinker. From the study mentions that the waste heat from the exhaust gas is 320-340° C. The exhaust gas generally is wasted and in some industry it use to reduce water content in raw material of cement production and fuel. The Waste Heat Recovery utilizing begin in 1970 by Kawasaki Heavy Industry that use rankine cycle.

In his research report Gorbanteko, et al, (2014) state that there are some factor that influence the project payback using waste heat recovery power generation in cement plant. WHR installation will be taking cost in design, engineering process, construction and commissioning process, and also buying of component and the supporting system will be, higher production of WHR system will have smaller dollar cost per kilowatt of generation capacity and vice versa. In 2013 sample calculation made for Chinese WHR with typical 5000 tpd clinker production, 9000 installed WHR capacity and 7200 working hours annually the payback period of the waste heat recovery will be around 4 years.

2.3.2 Effect of Installation of ORC on Existing Plant

Flue gas that come to the ORC system will heat the refrigeorant, and after that flue gas will going to stack. In evaporator when flue gas going in there will be friction in there that will make pressure drop to the flue gas, the flue gas pressure drop must not be too high so flue gas still can be going to the stack. The pressure different flue gas in the bottom of stack and in the top can be calculated with this following formula:

$$\Delta P = C \times a \times h \left(\frac{1}{T_o} - \frac{1}{T_i}\right) \qquad 2.1$$

Where :

ΔP	= Pressure different (Pa)
C	= 0.0342
a	= Athmospheric Pressure (Pa)
T_o	= Outside Temperature (K)
T_i	= Inside Temperature (K)

Actually the pressure drop that happen in evaporator wont be affect the existing system due to the pressure different in the bottom and top of the stack is very low, for the 200 m height of stack with the temperature in the bottom 100°C and 35°C the pressure different is only 0.0038 bar according to the calculation. So the ORC wont be affect the existing plant. This due to hot gas has lower density that make the gas lighter than outside gas and causes natural flow of flue gas to the stack.

2.4 Heat Transfer

Bergman, et al, (2011) state that heat transfer is exchange of thermal energy between two substances or medium. Heat transfer will occur when there is temperature different between the two substances or medium, lower temperature substances will absorb thermal energy from the higher temperature substances until the equilibrium of the thermal energy reached. There are three kind of heat transfer that is, conduction, convection and radiation.

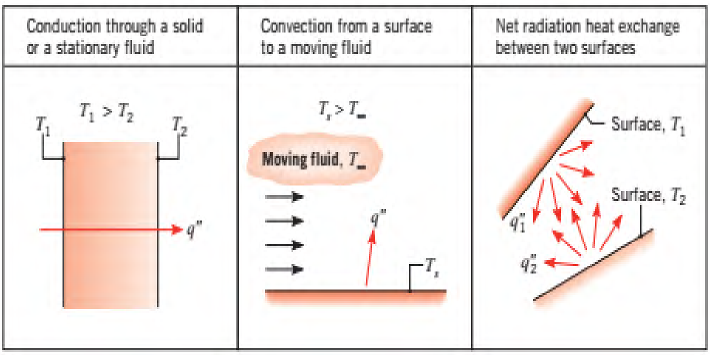


Figure 2.6 Type of Heat Transfer
(Source : Fundamentals of Heat and Mass Transfer)

2.4.1 Conduction

Conduction is transfer of thermal energy from the higher temperature substance to the lower temperature substance due to the interaction between the substances, or it can be said that there is contact between the substances so the heat will get transferred because of the contact of between substances. All material or substance has their own thermal conductivity value, this refer to the ability of the substance to transfer heat when there is contact with another substance, usually conduction will be efficient in solid particle due to molecules in solid substance is tighter than in the liquid and gas. The equation of conduction is expressed by:

$$q = -kA\frac{\Delta T}{x} \qquad 2.2$$

Where :

q = Conductive heat flux (W)

k = Thermal Conductivity (W/m.K)

A = Area of Heat Transferred (m^2)

ΔT = Temperature Different (K)

x = Thickness of heat transferred (m)

2.4.2 Convection

Convection usually occur in gas and liquid substances due to the particle in the substances is not tight like solid. Convection is the heat transfer due to the molecular motion of the fluid when there is the temperature different in bounding surface of the fluid. The movement of the fluid itself can occur due to force from something that usually called forced convection and can move due to density different that usually called free convection. The convection can be expressed by:

$$q = h\,A\,\Delta T \qquad 2.3$$

Where :

q = Convective heat flux (W)

h = Thermal Conductivity (W/ m^2.K)
A = Area of Heat Transferred (m^2)
ΔT = Temperature Different (K)

2.4.3 Radiation

Radiation is type of heat transfer that emitted by substances that has thermal energy. The heat transferred without medium, the heat can be transmitted to the empty space like the sun that transfer heat to the earth. This due to the electromagnetic wave can move through the any space without any medium required. Radiation can be calculated with :

$$q = \varepsilon . A . \sigma (Ts^4 - Tsur^4) \quad 2.4$$

Where :
q = Energy of Radiation (W)
ε = Emissivity
σ = Stefan Boltzman constant (W/ m^2.K^4)
A = Area of Heat Transferred (m^2)
T_s = Absolute temperature of surface (K)
T_{sur} = Absolute temperature surrounding (K)

2.5 Working Fluid Selection

Haibibzadeh, et al, (2014) in his engineering journal mention that working fluid is on ORC system usually using organic fluid that usually called refrigerant that has boiling point below the water, for example R134a that has boiling point at -26.3^{o}C, R245fa that has boiling point 15.2^{o}C and R141b that has boiling point 32^{o}C at 1 atm pressure, all of these refrigerant has the boiling point below water boiling point which is 100^{o}C, and it means that refrigerant can be evaporated or superheated with not so high heat source. In ORC the heat source usually the temperature is around 100-300^{o}C due to usually the heat source is waste heat or exhaust gas. So refrigerant in here will be suitable for ORC itself.

Usually to choose refrigerant there some consideration, like the fouling rate of the refrigerant, corrosiveness, toxicity and flammability of refrigerant, the refrigerant should has as few as possible those factor to minimize risk of loss. And also the working fluid should be safety for the environment, this factor can be rated with the Ozone Depletion Potential (ODP) and Global Warming Potential (GWP). ODP is relative amount of degradation to ozone layer that can caused by the refrigerant. And GWP is measurement of how much a given mass of refrigerant contributes to global warming. The CO^2 is the GWP reference, the GWP of carbon dioxide is 1.

The refrigerant usually groups into three according to the slope of saturation of vapour curves in temperature entropy diagram. The fluids having positive slope are dry fluid, the fluids having negative fluids are wet fluid and isentropic fluid the slope is infinite or it can be said that the vapor curve fall in vertical line. From the research that compared wet, isentropic and dry working fluid the ORC, showed that dry and isentropic fluid more likely suitable as working fluid for the ORC cycle due to dry or isentropic fluids are superheated after the expansion process while wet fluid will come into two phase or saturated after the fluid goes through turbine that can make droplets and damage the turbine especially turbine’s blade.

From 12 different working fluids that analyzed. The results show that R123 has highest rate thermal efficiencies in dry refrigerant, while R14b and R142 has the highest rate of thermal efficiencies in isentropic refrigerant and wet refrigerant.

And based from this engineering journal, the refrigerant choosed for this ORC system in this bachelor thesis is R141b (1,1-Dichloro-1-1fluoroehtane) which is isentropic refrigerant that has $C_2H_3Cl_2F$ chemical formula and also non flammable refrigerant. With 32^{o}C boiling point in atmospheric pressure, it will be suitable for the ORC system in cement plant that using air quenching cooler as the source of flue gas.

Here are some physical properties and environmental data of compared working fluid:

Tabel 2.2 Physical and Environmental Data of Some Refrigerant

Working Fluid	Type Of Fluid	Critical Temp. (ºC)	Boiling Point (ºC)	Critical Pressure (Mpa)	GWP (100 year)	ODP
R143a	Wet	84.04	-47.2	3.76	0	52
R152a	Wet	66.05	-24	4.52	0	1.4
R290	Wet	44.1	-42.1	4.25	0	0.041
R717	Wet	17.03	-33.3	11.33	0	0.01
R124	Isentropic	136.48	-12	3.62	0.02	5.8
R134a	Isentropic	102.03	-26.1	4.06	0	14
R141b	Isentropic	204.2	32	4.21	0.12	9.3
R142b	Isentropic	100.5	-9.1	4.06	0.07	17.9
R123	Dry	152.93	27.8	3.66	0.02	1.3
R245fa	Dry	134.05	15.1	3.65	0	7.6
R600	Dry	58.12	-0.5	3.8	0	0.018
R600a	Dry	58.12	-11.7	3.63	0	0.019
R601a	Dry	72.15	27.8	3.38	0	0.01

(Source : Thermodynamic analysis of different working fluids used in organic rankine cycle for recovering waste heat from GT-MHR)

2.6 Heat Exchanger

The use of heat exchanger is very important in many industrial process like in refrigeration system, cryogenic, heat recovery, alternate fuel, etc. Some of heat exchanger commonly used in our daily life like air conditioner and water heater. In ORC system there are two heat exchanger, that is evaporator and condenser. Heat exchanger is device to transfer thermal energy from two fluid that has different temperature. In heat exchanger the fluid is not mix, the fluid separated in different area.

Thulukkanam, (2013) said there are many types of heat exchanger, and it is have been classified into several category according to, construction, transfer process, degrees of surface compactness, flow arrangements, pass arrangements, phase of process fluids and heat transfer process.

And in here the explanation will be focused on the classification heat exchanger according to the construction. According to constructional detail heat exchanger classified into :

1. Tubular heat exchanger: Double pipe, shell and tube, coiled tube
2. Plate heat exchanger: Gasketed, brazed, welded, spiral, panel coil
3. Extended surface heat exchanger: Tube fin, plate fin

2.6.1 Tubular Heat Exchanger

Tubular heat exchanger consists of three type that is double pipe, shell and tube, coiled tube as mentioned before.

Double pipe heat exchanger consist of two tube pipe that usually form in U-bend design. The flow of this heat exchanger is countercurrent, usually the purpose is for small duties with the area of heat transfer less than 300 ft^2 and can be use for high pressure and temperatures. The advantage is the flexibility of the double pipe that can be added or removed as required and the design is easy and requires low inventory of spares.

Shell and tube heat exchanger is most common used heat exchanger. The heat exchanger called workhorse of industrial process heat transfer. Shell an tube usually used as oil cooler, feedwater heater, evaporator, condenser, etc. Major components of shell and tube heat exchanger are tubes, shell, baffles, front head, rear head and nozzles. The size of this heat exchanger is very varies from the small heat exchanger up to big heat exchanger that has 2 meter shell diameter, it can be operate from minus temperature up to 500^oC with the operating working pressure up to 600 bar. The design of this type of heat exchanger is flexible and robust and it easy to maintain and repair. But the disadvantages of this heat exchanger if there is limit of space in the industry that will used due to it require large site area for installation.

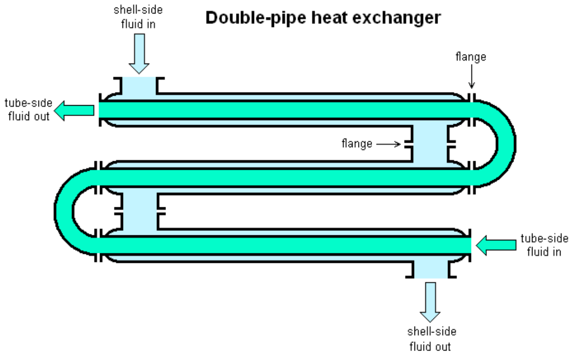


Figure 2.7 Type of Heat Transfer
(Source : http://1.bp.blogspot.com)

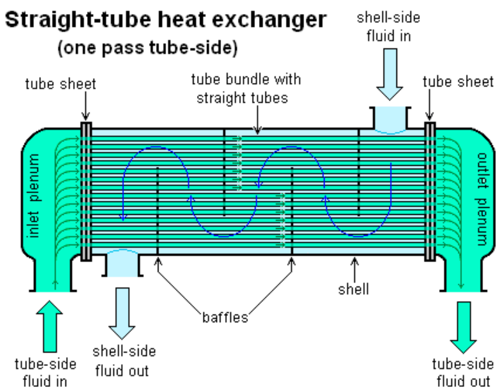


Figure 2.8 Shell and Tube Heat Exchanger
(Source : https://upload.wikimedia.org/)

Coiled tube heat exchanger construction involves winding alarge number of small bore ductile tubes in helix fashion around central core tube, with each exchanger containing many layer of tubes along both principal and radial axes. This heat exchanger advantages it is good when dealing with low temperature applications where simultaneous heat transfer between more than two streams is desired. But due to small bore tubes on both sides it can be permit mechanical cleaning of the heat exchanger.

Figure 2.9 Coiled Tube Heat Exchanger I
(Source Heat Exchanger Design Handbook)

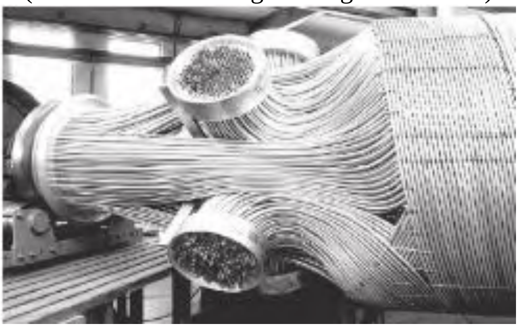


Figure 2.10 Coiled Tube Heat Exchanger II
(Source Heat Exchanger Design Handbook)

2.6.2 Plate Heat Exchanger

Plate heat exchangers are less used than tubular heat exchangers but it has some important advantages like overall weight of set is less than the tubular heat exchanger also it is fits in less space, overall heat transfer coefficient is more than the tubular and it is easy to install. The type of this heat exchangers are gasketed, brazed, welded, spiral, panel coil. But this type of heat exchanger has some limitation like the maximum operating pressure is only 25 bar and maximum temperature is 160°C. And also it can be used for every fluid due to the gasketed cannot handle corrosive or aggressive media, and this plate heat exchanger cannot handle particulates that are larger than 0.5 mm.

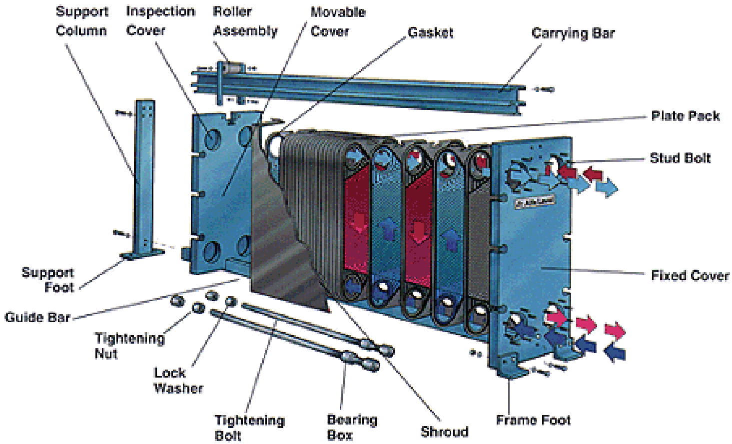


Figure 2.11 Plate Heat Exchanger
(Source http://www.separationequipment.com/)

2.6.3 Extended Surface Heat Exchanger

Extended surface heat exchanger also usually called with fin heat exchanger. There are two types of this heat exchanger, finned tube and plate fin heat exchanger. When the fluid heat transfer coefficient is quite low, this fin heat exchanger can be solution due to it will increase the heat transfer rate through convection of fin. For the tube finned heat exchanger longitudinal fins apply on ths heat exchanger which generally used in condensing applications and for viscous fluids in double pipe heat exchanger. The tube layout pattern is mostly staggered, the application of this finned tube heat exchanger usually for condenser and evaporator of air conditioners, radiator for internal combustion engine, charge air coolers and intercooler for cooling supercharged engine intake air.

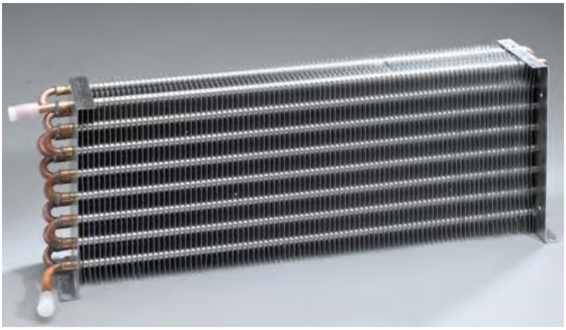


Figure 2.12 Finned Tube Heat Exchanger
(Source : http://img.hisupplier.com/)

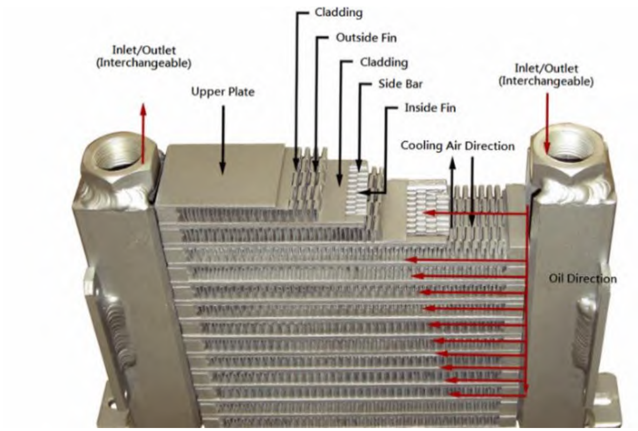


Figure 2.13 Plate Tube Heat Exchanger
(Source: http://www.coolbit.com.tw/)

And the other type is plate fin heat exchanger, this type use plates and finned chamber to transfer heat between fluids. Plate fin surface commonly use in gas to gas exchanger applications. And it is offer high area densities and designed for low pressure applications with operating pressure only up to 10 bar and the operating temperature from -162-150^{o}C. This type of heat exchanger has very high thermal effectiveness, the 95% and above effectiveness is common.

2.7 Design of Shell and Tube Heat Exchanger

As said before that shell and tube is most common heat exchanger used because of its versatility that can be used in any condition and situation. And in here will be explained about how to make preliminary design of heat exchanger. Usually the design standard of shell and tube using TEMA standard. TEMA standard applicable for shell and tube heat exchanger that has shell diameter not more than 60 inch (1524 mm). And before step further into the formula of the design, construction design (Tubes, Baffle and Shell) of the heat exchanger will be explained first.

2.7.1 Tubes

Tubes is circular cross section like pipe but smaller in diameter than pipe. Usually the geometrical variable like, tube outside diameter (OD), tube inside diameter (ID), tube length (L), tube pitch (PT), tube layout pattern is determined first. There are two type of tube mostly, straight tube for the single pass of heat exchanger and U-tubes for two pass of heat exchanger.

Tube diameter specified by OD and wall thickness (t_w), smaller diameter of tubes will lead into higher heat transfer coefficients but if the fluid is hard to clean it is more likely to use larger diameter of tubes and larger diameter also make tube side pressure drop smaller too. The TEMA standard tubes size usually (6.35, 9.53, 12.7, 15.88, 19.05, 22.23, 25.4, 31.75, 38.1, and 50.8 in mm), while the most popular of tubes used is 9.53 and 19.05 mm.

For the tube length, most economical exchanger is possible with small shell diameter and long tubes, however in many literature said in preliminary design, the ratio between length of tube and shell inside diameter is 5-10, and for the best practice is 8. But the most important is the heat exchanger can adjust the available space in the site.

Tube pitch is the distance between center of tube to the other center of tube. Close pitch will lead into increased shell side heat transfer and surface compactness while larger pitch will make the pressure drop is smaller and give easiness for cleaning the heat exchanger, The minimum ratio of tube pitch is 1.25 of tubes outside diameter.

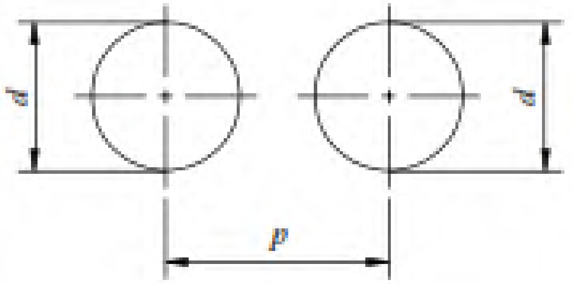


Figure 2.14 Tube Pitch and Outside Diameter
(Source : Heat Exchanger Design Handbook)

Tube layout arrangement designed to make many tubes possible within shell to achieve maximum heat transfer area. There are four types of tube layout patterns that is, 30^o (Triangular), 45^o (Rotated square), 60^o (Rotated triangular) and 90^o (Square). 30^o and 60^o provide compact arrangement of flow, better shellside heat trans fer coefficient and also stronger tube sheet, but the disadvantege of the tube layout it is hard to make mechanical cleaning due to the lanes of tube inaccessible rather than the 45^o and 90^o but it can be accessible. with mechanical cleaning and also water jet cleaning and in the next page is the figure of that four type tube lay out:

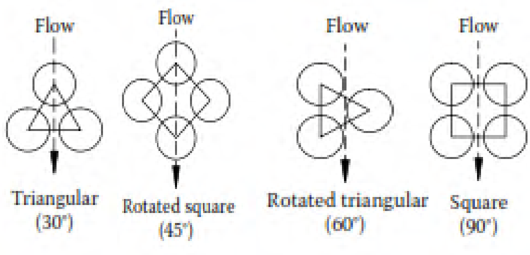


Figure 2.15 Tube Layout
(Source: Heat Exchanger Design Handbook)

2.7.2 Baffle

Baffle usually shaped like circular disk with the baffle hole to allow tube through the baffle. Baffle used to control the direction of the shellside flow, preventing tubes from shagging and excessive vibration and also increase effectiveness of heat transfer from shell to tube but on the other side baffle can increase pressure drop in the shell side too.

To determine baffle we considering the baffle spacing (Lbc) an the baffle cut (Bc). Baffle spacing is space between one baffle to another baffle. The range of baffle spacing vary from 0.2-1 shell diameter. For the maximum baffle spacing can be known from TEMA Table RCB 4.52. and baffle cut is percentage of cut baffle according to the shell diameter. Baffle cut vary from 20-49% of shell diameter and the most being used is 20-25%.

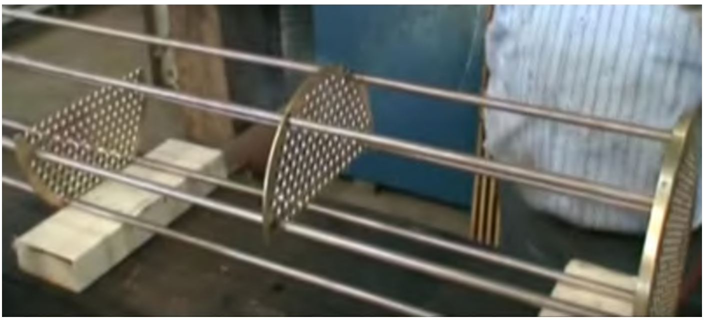


Figure 2.16 Baffle Cut
(Source: Heat Exchanger Design Handbook)

2.7.3 Shell

Shell manufactured in a large range of standard sizes, materials and thickness. Large sizes of shell fabricated from plate by rolling. Cost of shell is much more than cost of tubes. It s found that more economical heat exchanger usually designed with small shell diameter and maximum shell length.

2.7.4 Heat Transfer Rate

Shell To design the heat exchanger, the heat rate required must be specified first, this can be specified with:

$$Q = \dot{m} \times Cp \times \Delta T \qquad 2.5$$

Where :

Q	= Total heat (W)
$\dot{m}$	= Mass flow rate (kg/s)
Cp	= Specific heat (Kj/kg.C)
ΔT	= Temperature different (C)

And to calculate another side flowreate, this equation can be used:

$$Q_1 = Q_2 \qquad 2.6$$

If there is no changing phase in the process the Q can be assume as equation 2.4. But if there is changing phase like in evaporator or condenser the equation will be:

$$Q_1 = \dot{m}((Cp_1 * \Delta T_1) + (\Delta h) + (Cp_2 * \Delta T_2)) \qquad 2.7$$

Where :

Q	= Total heat (W)
$\dot{m}$	= Mass flow rate (kg/s)
Cp	= Specific heat (Kj/kg.C)
ΔT	= Temperature different (C)
Δh	= Latent heat different (Enthalpy) (Kj/kg)

2.7.5 Log Mean Temperature Difference

Log Mean Temperature Difference (LMTD) is average temperature different between hot and cold fluid in the heat exchanger. Larger value of LMTD mean higher heat transferred to each fluid.

There two types of stream flow in shell and tube heat exchanger, parallel current flow and and counter current flow. To achieve greater amount of total heat, the counter current flow can be used.

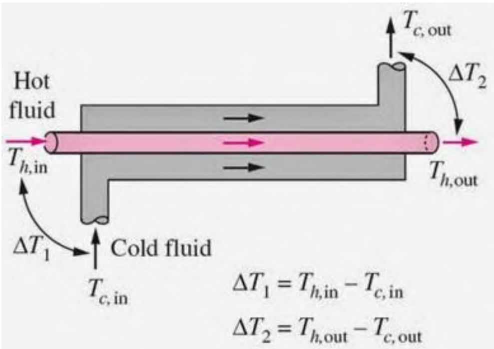


Figure 2.17 Parallel Current Flow
(Source : www. me-mechanicalengineering.com)

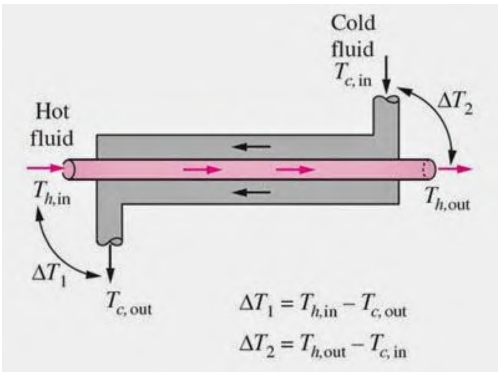


Figure 2.18 Counter Current Flow
(Source : www. me-mechanicalengineering.com)

For parallel current flow the equation of LMTD is:

$$\Delta LMTD = \frac{(Tho - Tco) - (\mathrm{Thi} - \mathrm{Tci})}{ln\frac{(Thi - Tco)}{(Thi - Tci)}} \qquad 2.8$$

And for Counter current flow the equation will be:

$$\Delta LMTD = \frac{(Thi - Tco) - (\mathrm{Tho} - \mathrm{Tci})}{ln\frac{(Thi - Tco)}{(Tho - Tci)}} \qquad 2.9$$

Where :

LMTD = Log mean temperature difference (K)
Thi = Temperature hot in (K)
Tho = Temperature hot out (K)
Tci = Temperature cold in (K)
Tco = Temperature cold out (K)

2.7.6 LMTD Correction Factor

After that LMTD calculated, calculation of LMTD correction factor (F) conducted to know the correction factor of LMT, usually the value is 0.8. and to calculation of F is:

$$F = \frac{\left(\frac{\sqrt{R^2+1}}{R-1}.\, ln\left(\frac{1-x}{1-RX}\right)\right)}{ln\left(\frac{\frac{2}{x} - 1 - R + \sqrt{R^2+1}}{\frac{2}{x} - 1 - R - \sqrt{R^2+1}}\right)} \qquad 2.10$$

R and X value is:

$$R = \frac{Thi - Tho}{Tco - Tci} \qquad 2.11$$

$$X = \frac{1 - \left(\frac{RP-1}{P-1}\right)^{\frac{1}{N}}}{R - \left(\frac{RP-1}{P-1}\right)^{\frac{1}{N}}} \qquad 2.12$$

For P the equation is:

$$P = \frac{Tco - Tci}{Tho - Tci} \qquad 2.13$$

Where :

N	= Number of tube pass
Thi	= Temperature hot in (K)
Tho	= Temperature hot out (K)
Tci	= Temperature cold in (K)
Tco	= Temperature cold out (K)

2.7.7 Required Heat Transfer Area

After that LMTD calculated, calculation of LMTD correction factor (F) conducted to know the correction factor of LMTD, usually the value is 0.8. and to calculation of F is:

To calculate required heat transfer area the equation is:

$$A = \frac{Q}{UF\Delta Tm} \qquad 2.14$$

Where :

A	= Heat transfer area (m^2)
Q	= Total heat (W)
U	= Overall heat transfer coefficient ($W/m^2.K$)
F	= LMTD correction factor
ΔTm	= LMTD (K)

To determine the overall heat transfer coefficient value (U), we must has geometry data, and calculating the heat transfer coefficient in tube side and shell side. So for the first design the U value can be assume from the typical overall heat transfer coefficient table. Than later the assume U value replaced with the actual U value.

2.7.8 Actual Heat Transfer Area

Actual heat transfer area at least has same value with the required heat transfer area or bigger than it, but if it is to big there will be more heat loss in heat exchanger. Actually to design the actual heat transfer area, the trial and error must be conducted in here. First assume geometry data than calculate the tube side and shell side heat transfer coefficient after that we can find the overall heat transfer coefficient (U) value, and if the value is not near the assumption value, then try with another geometry data until the value error of U is close (0-5%).

To calculate it the step the shell diameter should be assume first, as mention before, for preliminary design shell diameter (Ds) can be obtained from making ratio with the length of tube (L), Thulukkanam, (2013) state that the ratio between L/Ds is 5-10, and for best practice is 8. And the length of tube can be assumed from the large of room or site that will used for the heat exchanger.

After that calculate the bundle to shell clearance (Lbb) with:

$$Lbb = 12 + 0.005Ds \tag{2.15}$$

Then calculate the outer tube limit diameter (Dotl)

$$Dotl = Ds - Lbb \tag{2.16}$$

After that calculating bundle diameter of tube (Dctl) which can be calculated with:

$$Dctl = Dotl - OD \tag{2.17}$$

Where :

Ds	= Shell diameter (mm)
Lbb	= Bundle to shell clearance (mm)
Dotl	= Outer tube limit diameter (mm)
Dctl	= Bundle diameter of tube (mm)
OD	= Outside diameter of tube (mm)

Then calculate the tube pitch (PT) and determine tube layout constant (C1). As mention before that tube pitch is 1.25 OD. For the constant of C1 it is given by :

$C1 = 0.86\ for\ \theta_{tp} = 30°\ while$

$C1 = 1\ for\ \theta_{tp} = 45°\ and\ 90°$

Then the number of tube (Nt) can be calculated with:

$$Nt = \frac{0.78Dctl^2}{C1.PT^2} \quad 2.18$$

Where :

Nt = Number of tube

PT = Tube Pitch (mm)

C1 = Tube layout constant

And actual heat transfer area can be expressed with:

$$A = \pi.Ds.L.Nt \quad 2.19$$

Where :

A = Actual heat transfer area (m^2).

2.7.9 Tube Side Heat Transfer Coefficient

To calculate tube side heat transfer coefficient first Reynold number (Re) must be calculated, Reynold number will determine the type of the fluid flow, is it laminar or turbulence. Fluid that has Reynold number below 2300 will categorized as laminar flow and fluid that has Reynold number more than 4000 categorized as turbulence flow, and in between them categorized as transition flow. Reynold number can be calculated by this given formula:

$$Re = \frac{4\dot{m}}{\pi.ID.\mu.Nt} \quad 2.20$$

Where :

$\dot{m}$ = Mass flow rate (kg/s)

ID = Inside diameter (m)
μ = Dynamic viscocity (Pa.s)
Nt = Number of tube

Then calculate the Prandtl number (Pr). Prandtl number is dimensionless number that define as the ratio momentum diffusivity to thermal diffusivity. Prandtl number can be calculated with this following formula:

$$Pr = \frac{\mu . Cp}{k} \tag{2. 21}$$

Where :
Cp = Specific Heat (kJ/kg)
k = Thermal conductivity of fluid (m)
μ = Dynamic viscocity (Pa.s)
Nt = Number of tube (W/m.K)

From Dittus-Boelter equation the tube side heat transfer coefficient can be calculated as this following:

$$\frac{h_t . ID}{k} = 0.0243 . Re^{0.8} . Pr^n \tag{2. 22}$$

Where :
ht = Tubeside heat transfer coefficient (W/m^2.K)
n = 0.3 for cooling process and 0.4 for heating process

2.7.10 Shell Side Heat Transfer Coefficient

Shell side heat transfer coefficient, is calculated in ideal condition first, than later will be calculated with some correction factor in the shell, so here it can be said it is ideal shell side heat transfer coefficient.

And to calculate it first, calculate the shell side mass velocity (Gs). The fomula can be expressed with:

$$G_s = \frac{\dot{m}_{s.}}{S_m} \qquad 2.23$$

Where :
Gs = Shell side mass velocity (kg/m^2.s)
$\dot{m}_{s.}$ = Shell fluid mass flow rate (kg/s)
Sm = Shell side crossflow area (m^2)

And for Sm the calculation is :

$$S_m = Lbc\left[Lbb + \frac{Dctl}{TP}(TP - OD)\right] \qquad 2.24$$

Where :
Lbc = Baffle spacing (m)

Then calculate the dimensionless number (Re and Pr) with the shell side data, for prandtl number the formula is same with the tube side for the Reynold number the formula given by:

$$R_e = \frac{OD.G_s}{\mu} \qquad 2.25$$

After that calculate ideal Colburn J factor (Ji) for the shell side, from Thome, (2004) in engineering data book III state that ji can be determine from Bell Delaware curve, but for more specific result can be calculated with this following equation:

$$j_i = a_1\left(\frac{1.33}{\frac{Ltp}{OD}}\right)^a Re^{a_2} \qquad 2.26$$

Where a is

$$a = \frac{a_{3.}}{1 + 0.14Re^{a_4}} \qquad 2.27$$

And for the a_1,a_2,a_3 and a_4 values is given in this following table:

Tabel 2.3 Physical and Environmental Data of Some Refrigerant

Layout	Re	a_1	a_2	a_3	a_4
30°	10^5-10^4	0.321	-0.388	1.450	0.519
	10^4-10^3	0.321	-0.388		
	10^3-10^2	0.593	-0.477		
	10^2-10	1.360	-0.657		
	<10	1.400	-0.667		
45°	10^5-10^4	0.370	-0.396	1.930	0.500
	10^4-10^3	0.370	-0.396		
	10^3-10^2	0.730	-0.500		
	10^2-10	0.498	-0.656		
	<10	1.550	-0.667		
90°	10^5-10^4	0.370	-0.395	1.187	0.370
	10^4-10^3	0.107	-0.266		
	10^3-10^2	0.408	-0.460		
	10^2-10	0.900	-0.631		
	<10	0.970	-0.667		

(Source : Engineering Data Book III)

Than before calculating ideal shell side heat transfer coefficient, viscocity correction factor calculated with:

$$\emptyset s = \left(\frac{\mu_t}{\mu_w}\right)^{0.14} \qquad 2.28$$

Where :
$\emptyset s$ = Shell side viscocity correction factor
μ_t = Dynamic viscocity at tube temperature average (Pa.s)
μ_w = Dynamic viscocity at wall temperature average (Pa.s)

For the gas being heated the viscocity correction factor is 1. And to determine viscocity of tube and wall first, calculate the tube average temperature and wall average temperature, and then find the viscocity at that specific temperature given from calculation.

$$T_{tube\ av} = \frac{inlet\ temp. + outlet\ temp.}{2} \quad 2.\ 29$$

$$T_w = \frac{1}{2}\left(\frac{T_{shell\ av} + T_{shell\ av}}{2}\right) \quad 2.\ 30$$

Then ideal shell side heat transfer coefficient (hi) is:

$$h_i = \frac{j_{i.}Cp.Gs.(\emptyset_s)^n}{Pr^{2/3}} \quad 2.\ 31$$

Where :

Ji = Ideal Colburn J factor
Cp = Specific heat (kJ/kg.K)
Gs = Shell side mass velocity (kg/m^2.s)
$\emptyset s$ = viscocity correction factor
Pr = Prandtl number

2.7.11 Shell Side Heat Transfer Coefficient Factor

The flow fraction for each stream is found by knowing the corresponding flow areas and flow resistance. The heat transfer coefficient for ideal crossflow is then modified for the presence of each stream through correction factor with Bell-Delaware method. The heat transfer in the shell side then modified for the presence of each stream through correction factors. Then the shell side heat transfer coefficient (hs) is:

$$h_s = h_i.J_c.J_1 J_b J_s.J_r \quad 2.\ 32$$

Where :

hi = Ideal shellside heat transfer coefficient (W/m^2.K)
Jc = Segmental baffle window correction factor
J1 = Baffle leakage effects correction factor
Jb = Bundle bypass effects correction factor
Js = Unequal baffle spacing at inlet and outlet correction factor

Jr= Adverse temperature gradient correction factor in laminar flow

The value of Jc for well designed heat exchanger is 1 and can increases to 1.15 for small baffle cut. To calculate Jc first, calculate upper centriangle of baffle cut (θctl), and it can be expressed with:

$$\theta_{ctl} = 2cos^{-1}\left[\frac{Ds}{Dctl}\left(1-\frac{2Bc}{100}\right)\right] \qquad 2.33$$

Then (θctl) used to calculate fraction number of tubes in the baffle window (Fw) :

$$Fw = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi} \qquad 2.34$$

After that calculate pure crossflow between baffle cut (Fc) with this following formula given :

$$Fc = 1 - 2Fw \qquad 2.35$$

Then Jc will be :

$$J_c = 0.55 + 0.72F_c \qquad 2.36$$

Where :

θ_{ctl}	= Centriangle of baffle cut (°)
Bc	= Baffle cut (%)
Fw	= Fraction number of tubes in the baffle window
Fc	= Pure crossflow between baffle cut tips

To give simple explanation of segmental baffle geometry, the picture here will give an explanation, where Fc is described.

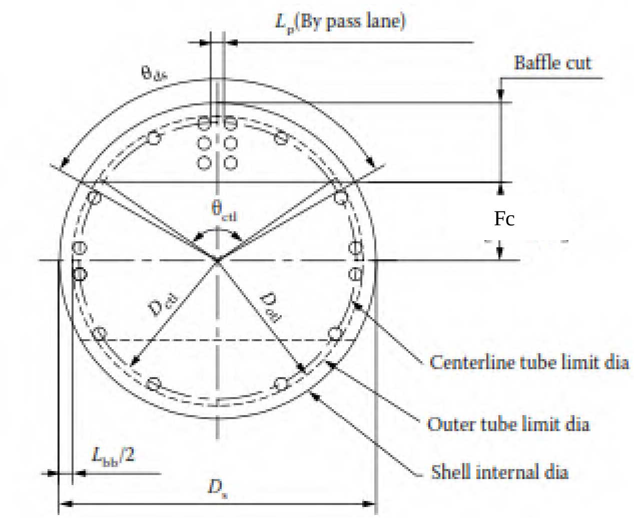


Figure 2.19 Segmental Baffle Geometry
(Source : Heat Exchanger Design Handbook)

Baffle leakage effects correction (J1) factor design should not less than 0.6. This leakage will become higher when the distance between baffle is too close that makes excessive fraction of the flow. Before calculate J1, the Shellside crossflow area (Sm), Shell to baffle leakage area (Ssb) and Tube to baffle hole leakage area (Stb) must be calculated, and the Sm already determine when Gs calculated, so in here just the Ssb and Stb will be determine, and the formula of those two are:

$$S_{sb} = \pi Ds \frac{Lsb}{2} \left(\frac{2\pi - \theta ds}{2\pi} \right) \tag{2. 37}$$

Where Lsb is the cleareance between Ds and Bc and θds is shell to baffle leakage area that can be calculated with :

$$L_{sb} = 3.11 + 0.004DS \tag{2. 38}$$

$$\theta_{ctl} = 2cos^{-1}\left(1 - \frac{2Bc}{100}\right) \tag{2. 39}$$

Where :
Ssb = Shell to baffle leakage area (m^2)
θ_s = Baffle window angle (^{o})
Lsb = Clearance between shell diameter and baffle cut (mm)

For the Stb the formula is :

$$S_{tb} = \frac{\pi}{4}[(OD + Ltb)^2 - OD^2]Nt(1 - Fw) \tag{2. 40}$$

Where :
Stb = Tube to baffle hole leakage area (m^2)
Ltb = Diameter clearance between OD and baffle hole (mm), (according to TEMA standard the value is 0.4 or 0.8)

After that correlational parameter calculated (rs and rlm) with :

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}} \tag{2. 41}$$

$$r_{lm} = \frac{S_{sb} + S_{tb}}{S_m} \tag{2. 42}$$

Then J1 will be

$$J_1 = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2rlm} \tag{2. 43}$$

For bundle bypass flow correction factor (Jb) usually the value is above 0.9 if the clearance between shell and tube bundle is small, to increase the correction factor, sealing strip can be add.

To calculate Jb first, calculate the bypass area between shell and tube bundle within one baffle (Sb), this can be expressed with:

$$S_b = Lbc(Ds - Dotl + Lpl) \qquad 2.44$$

Then calculate ratio of bypass area to the overall crossflow area (Fsbp) with :

$$F_{sbp} = \frac{S_b}{S_m} \qquad 2.45$$

Where :
Fbsp = Ratio of bypass area to the overall crossflow area
Sb = Bypass area between shell and tube bundle (m^2)

And then determine the number of sealing strips (Ns) (in pairs) in one baffle spacing. After that calculate the number of tube rows crossed between baffle tips (Ntcc)

$$N_{tcc} = \frac{Ds}{Lpp}\left(1 - 2\frac{Bc}{100}\right) \qquad 2.46$$

Where Lpp is 0.866 x PT for 30° tube pitch and equal to PT for other tube pitch. After that calculate the ratio between Ns and Ntcc (rss).

$$r_{ss} = \frac{Ns}{Ntcc} \qquad 2.47$$

And Jb can be calculated with this given formula :

$$J_b = exp\left\{-C_{bh}.F_{sbp}\left[1 - (2.r_{ss})^{1/3}\right]\right\} \qquad 2.48$$

Where :
Cbh = 1.25 for Re < 100 and 1.35 for Re > 100
Fsbp = Ratio of bypass area to the overall crossflow area
Rss = Ratio between Ns and Ntcc

Js is the correction factor for unequal baffle spacing at inlet and outle nozzlet, in this preliminary design better assumed that the inlet and outlet baffle spacing is same. And for the same spacing the value of Js is 1.

Jr is correction factor for adverse temperature gradient in laminar flow and it is just apply for the flow with Reynold number below 100. If the Reynold number above 100 the value is 1.

$$J_r = \frac{1.51}{((Ntcc + Ntcw) + (Nb + 1)^{0.18})} \qquad 2.\ 49$$

2.7.12 Overall Heat Transfer Coefficient

Here the actual value of overall heat transfer coefficient (U) will be calculated, the result of the calculation will be close with the assumed value of U in the first time. And the calculation of U is :

$$U = \frac{1}{\frac{1}{hs} + Rf_s + \left(\frac{t_w}{k_w}\right).\left(\frac{A_o}{A_m}\right) + \left(Rf_t + \frac{1}{ht}\right)\frac{A_o}{A_i}} \qquad 2.\ 50$$

Where :

U	= Overall heat transfer coefficient (W/m^2.K)
hs	= Shell side heat transfer coefficient (W/m^2.K)
ht	= Tube side heat transfer coefficient (W/m^2.K)
Rfs	= Fouling factor of shell fluid (m^2.K/W)
Rft	= Fouling factor of tube fluid (m^2.K /W)
kw	= Thermal conductivity of heat exchanger material (W/m.K)
tw	= Tube wall thickness (W/m^2.K)
A_o/A_i	= Outside diameter/ inside diameter (m^2)
Am	= mean wall heat transfer area (m^2)

2.7.13 Tube Side Pressure Drop

The tube side pressure drop equation using Kern method is:

$$\Delta Pt = \frac{f.Gt^2.L.np}{5.22x10^{10}.ID.Sg.\emptyset t} \qquad 2.51$$

Where :

f = Friction factor from moody diagram
Gt = Tube side mass velocity (kg/m^2.s)
L = Length of tube (m)
Np = Number of tube pass
ID = Inside diameter of tube (m)
Sg = Specific gravity of tube fluid
$\emptyset t$ = Fluid of tube viscocity correction factor

The allowable limit of pressure drop is 10 psi.

2.7.14 Shell Side Pressure Drop

The formula for the shell side pressure drop is:

$$\Delta Ps = \frac{f.Gs^2.Ds.(nb+1)}{5.22.10^{10}.de.Sg.\emptyset s} \qquad 2.52$$

Where :

f = Friction factor from moody diagram
Gs = Shell side mass velocity (kg/m^2.s)
Ds = Shell diameter (m)
Nb = Number of baffle
de = diameter equivalent (m)
Sg = Specific gravity of shell fluid
$\emptyset s$ = Fluid of shell viscocity correction factor

Allowable limit of pressure drop in shell side is 10 psi as well.

2.8 Turbine

Turbine is one of main part in ORC, in turbine, the refrigerant that already in superheated phase will be expand in turbine, the expansion of refrigerant will give energy to turbine to rotate shaft generator then later transmit the energy from shaft to generator and produced the power. The refrigerant that already in expand will decrease in temperature and pressure.

To choose turbine in ORC the consideration is capacity of the turbine, maximal pressure and temperature that can handle by the turbine. To calculate the power output by the turbine this following equation can be used:

$$Wt = \dot{m}.(h_2 - h_1) \qquad 2.\ 53$$

Where :

Wt = Power of Turbine (W)

$\dot{m}$ = Mass flow rate of refrigerant (kg/ s)

h = enthalphy (Kj/kg)

P-h diagram can be used for the value of enthalpy at specific temperature and pressure of the refrigerant, bigger different between the enthalpy in the inlet and outlet of the turbine the power produce of the turbine will be higher too. For consideration also in the outlet when pressure drop occur in the turbine it is important to make the refrigerant phase stay in superheater, if the refrigerant become saturated there will be droplets of the refrigerant that can damage the turbine.

2.9 Pump

Pump used to move fluid from one place to another place, the fluid that through the pump is liquid. The fluid will come into inlet suction of pump and go to the discharge nozzle, the using of pump usually due to the height different. But in organic rankine cycle usually the main requirement is to increase the pressure of fluid and send it back to the evaporator, higher pressure that can achieve by the pump, will lead into higher power production of the turbine.

Sularso, (2000) mention that when selecting pump, the thing that must be known first is the capacity of the flow or flowrate of the fluid and head minimum required for pumped the fluid then the pumping power will be known from that. Beside that other requirement for selecting pump is suction and discharge condition, type of fluid, work hour of pump and installation location of the pump.

2.9.1 Pump Capacity

Pump capacity is capability of pump to move fluid in particular time unit, the unit usually is meter cubic per hour (m^3/h) or galloon per minute (gpm). To calculate pump capacity it can be expressed with :

$$Q = \frac{\dot{m}}{\rho} \qquad 2.54$$

Where :
Q = Capacity of pump (m^3/s)
$\dot{m}$ = Mass flow rate of fluid (kg/ s)
ρ = Density of fluid (kg/m^3)

2.9.2 Pump Head

There are four type of head in pump, that is, pressure head, velocity head, static head and friction head.

Pressure head caused by pressure different in inlet suction and discharge nozzle, it can be calculate with:

$$H_p = \frac{\Delta P}{\rho . g} \qquad 2.55$$

Where :
H_p = Pressure head (m)
ΔP = Pressure different (discharge-suction) (Pa)
g = Gravitational acceleration (m/s^2)

velocity head caused by velocity different, the formula for velocity head is:

$$H_v = \frac{v_2^2}{2.g} - \frac{v_1^2}{2.g} \qquad 2.56$$

Where :
H_v = Velocity head (m)
g = Gravitational acceleration (m/s^2)

Static head caused by height different, and it can be calculated by this given formula :

$$H_s = h_2 - h_1 \qquad 2.57$$

Where :
H_s = Static head (m)
h_2 = Height at point 2 (m)
h_1 = Height at point 1 (m)

And friction head caused by fluid friction with the pipe and also fitting if there is any. The head consist of major losses and minor losses, and for the calculation is:

Major losses

$$H_{fmajor} = \lambda . L . \frac{D}{2g} \qquad 2.58$$

Where :
H_{fmajor} = Major losses (m)
λ = Darcy-Wiesbach pipe friction loss
L = Length of pipe (m)
D = Pipe diameter (m)
g = Gravitational acceleration (m/s^2)

Minor losses

$$H_{fminor} = \Sigma n. k. \frac{v^2}{2g} \qquad 2.59$$

Where :

H_{fmajor} = Minor losses (m)
Σn = Sum of Fitting
k = Minor loss coefficient
D = velocity of fluid (m/s)
g = Gravitational acceleration (m/s^2)

Total friction head will be :

$$H_f = H_{fminor} + H_{fmajor} \qquad 2.60$$

And total head of pump will be :

$$H = H_p + H_v + H_s + H_f \qquad 2.61$$

Where :
H = Total head of pump (m)

2.9.3 Pumping Power

For given values of density, flow rate and head, the power input required by pump can be expressed by:

$$P = \frac{\rho. Q. g. H}{\eta} \qquad 2.62$$

Where :
P = Pumping power (kW)
ρ = Density of fluid (kg/m^3)
Q = Flow rate (m^3/h)
g = Gravitational acceleration (m/s^2)
H = Total head (m)
η = Pump Efficiency

CHAPTER III
METHODOLOGY

3.1 Literature Study

Literature study conducted to collect the source learning material that will support the research of this bachelor thesis. The learning material source is for this bachelor thesis come out from book, journal, paper, presentation, other bachelor thesis that connected with the study about, organic rankine cycle, waste heat recovery, refrigerant characteristic, heat exchanger etc.

3.2 Collecting Data

Collecting data conducted in PT. Semen Indonesia (Persero), Tuban. The data is about flue gas that available in the plant which is from preheater and air quenching cooler, then find out about the utilization of that flue gas. Then the data about the flue gas itself (flowrate, density, specific heat, temperature, pressure and flue gas composition) and also the space that available in the plant.

3.3 Flue Gas Data Analysis

After flue gas data obtained, then find out whether the flue gas already used or not. In the cement plant, flue gas in the suspension preheater already used for drying raw material purpose when the flue gas leave the psuspension preheater and based on calculation from Confederation of Indian Industry (CII), the flue gas available from preheater is too few for recovery purpose. And for the air quenching cooler, some of flue gas already used to ensure that the cement is already dry. And there still many flue gas go through the stack. And from this the flue gas can be used for alternative power generation.

3.4 Refrigerant Selection

As mention before in ORC the working fluid is refrigerant. And there are some characteristic of refrigerant must be considered beforechoosing the refrigerant. The main consideration is the thermal efficiency of refrigerant, higher thermal efficiency of refrigerant will make higher heat transfer rate and will impact to power generated. Other characteristic like critical pressure, critical temperature, boiling point is considered too and also environmental factor like ozone depletion potential (ODP) and global warming potential (GWP) must be considerd.

3.5 Determining Operating Condition

Operating condition must be suitable for ORC, in chapter 2 already mention that the refrigerant chosen is R141b. So the operating condition must be suitable for R141b, the R141b must be in superheated phase when left the evaporator and get into liquid again after going through the condenser.

3.6 ORC System Calculation

After that the ORC system calculated, first calculate the evaporator heat transfer and also condenser hea transfer and in this step the power produce by the cycle can be known too.

3.7 Simulation

Hysys simulation used to assume best operating condition in the cycle, we input some data parameter like temperature, pressure, mass flow rate or phase of the liquid that already known based on degree of freedom calculation.

3.8 ORC Components Manual Calculation

The organic rankine cycle system consist of evaporator, turbine, condenser and pump. And using manual calculation the design (geometry data) of heat exchanger (evaporator and condenser will be conducted), the flowchart below will describe how to make preliminary design of shell and tube heat exchanger. And for selecting the pump capacity and head of pump need to be known. And for chooing turbine, the operating condition is enough to choosing it, the temperature in and out of turbine, pressure in and out of turbine and also power produced of the turbine.

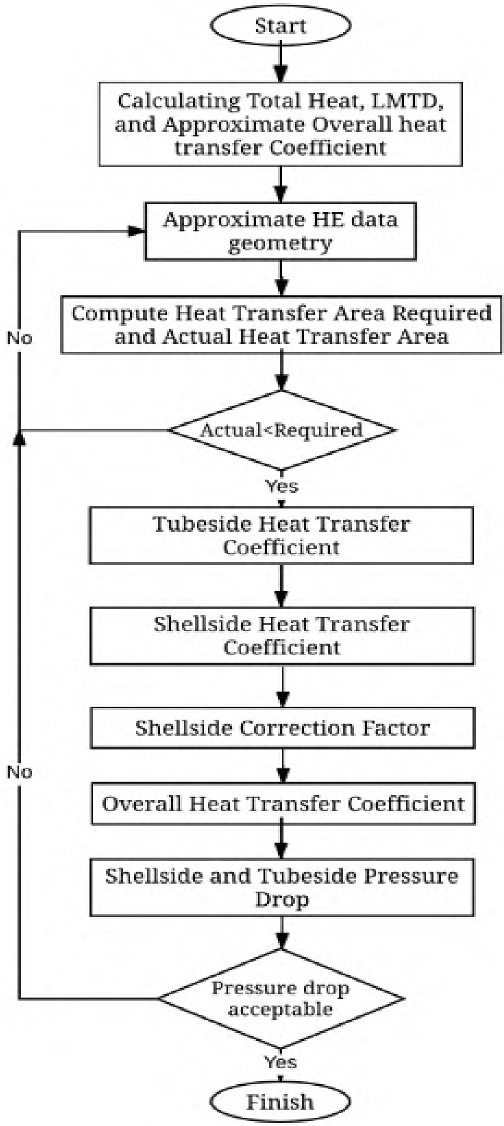


Figure 3.1 Shell and Tube Heat Exchanger Preliminary Design Flowchart

And for selecting pump the step is :

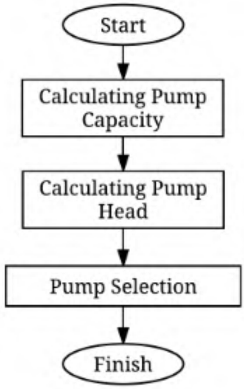


Figure 3.2 Pump Selection Flowchart

3.9 Conclusion and Suggestion

After all calculation and selecting component conducted from conclusion and suggestion the state of problems will be answered here and the suggestion based on correction to make this bachelor thesis better and can be continued by others.

3.10 Methodology Flowchart

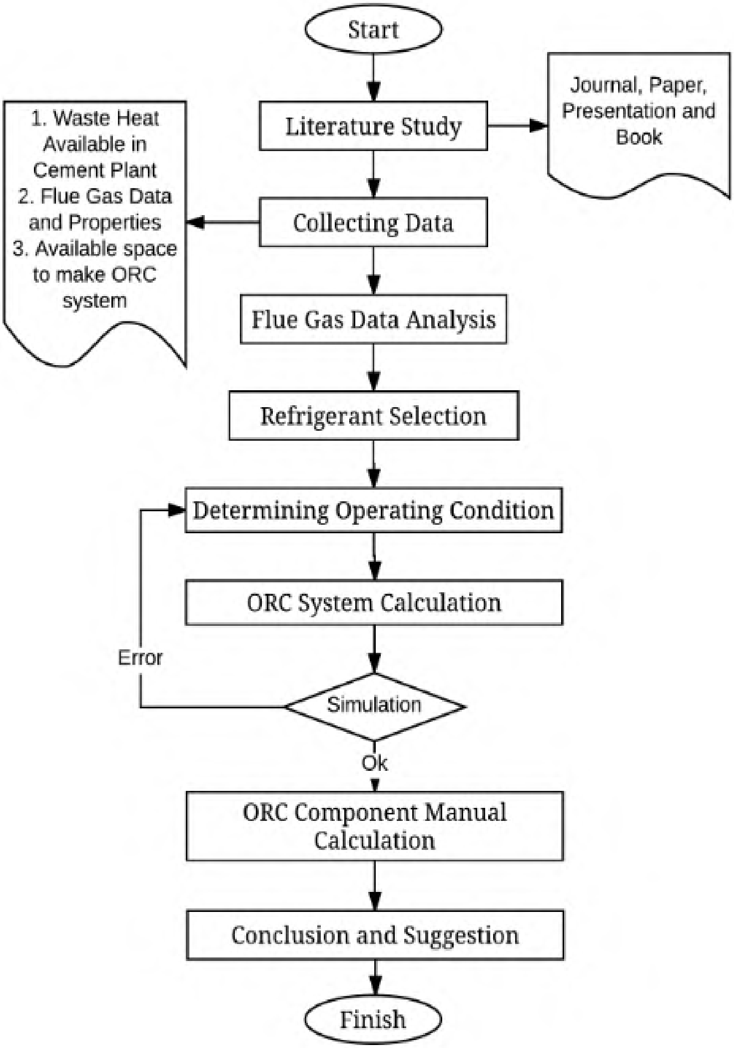


Figure 3.3 Bachelor Thesis Flowchart Processing

CHAPTER IV
ANALYSIS AND CALCULATION

In this chapter analysis and calculation based on the data that already got from the cement plant will be conducted. First calculation of potential waste heat from the cement plant will be known, then arrangement of ORC operating condition and also design calculation of ORC component will be carried out.

4.1 General Data

The data got from PT. SEMEN INDONESIA (PERSERO) Tbk.

Table 4.1 Operation Data of Cement Plant

Date Process Operation Date	31st January 2015
Temperature Ambient	34^{o}C
Capacity of Kiln	8466 Ton/Day
Density of Air	1.29 kg/m^3
Number Stage of Preheater	4
Limestone Moisture	5.74%
Raw Coal Moisture	10.85%
Raw Mill Running Hours	24 hours/day
Kiln Running Days	335 days/annum
Specific Heat Consumption	760 kcal/kg clinker
Raw Material to Clinker Factor	1.55
Heat Requirement in Raw Mill & Coal Mill	950 kcal/kg water
Calorific Value of Coal	4800 kcal/kg coal
Coal Mill Running Hours	24 hours/day

And here are the heat source data that available on the plant is from the preheater and from air quenching cooling and this is the data of gas that through preheater and air quenching cooling.

Table 4.2 Heat Source from Preheater String 1

Area	Parameter	Quantity	Unit
Exit of Preheater String 1	Velocity	9.48	S.m/s
	Specific Heat	0.2462	kcal/kgC
	Temperature	401.79	^{o}C
	Pressure	530.13	mmWG
	O2	3.5	%
	CO	247.5	ppm
	CO2	15.34	%
	Nox	144.5	ppm
	Diameter	3.55	m
	Flow Rate	727749.4	m3/hr

Table 4.3 Heat Source from Preheater String 2

Area	Parameter	Quantity	Unit
Exit of Preheater String 2	Velocity	9.48	S.m/s
	Specific Heat	0.2462	kcal/kgC
	Temperature	401.79	^{o}C
	Pressure	530.13	mmWG
	O2	3.5	%
	CO	247.5	ppm
	CO2	15.34	%
	Nox	144.5	ppm
	Diameter	3.55	m
	Flow Rate	587880	m3/hr

Table 4.4 Heat Source from Air Quenching Cooler

Area	Parameter	Quantity	Unit
Cooler Excess Air	Velocity	8.81	m/s
	Specific Heat	0.237736	kcal/kgC
	Temperature	235	^{o}C
	Pressure	27.54	mmWG
	O2	21.0	%
	CO	0	ppm
	CO2	0	%
	Nox	0	ppm
	Diameter	3	m
	Flow Rate	381135	m3/hr

4.2 Available Waste Heat Calculation

As mention in the chapter II the available waste heat in the preheater is based on the moisture content of the material that used and the temperature from the exit of preheater due to the waste heat from preheater in cement plant used to reduce water content in the raw material and from calculation based on Confederation of Indian Industry the waste heat available in the suspension preheater is only 2.2 Watt (The calculation of waste heat recovery potential on suspension preheater is attached on ATTACHMENT).

Due to the waste heat available in the preheater is very few (2.2 W) the heat from the preheater will not use to generate power using ORC instead the waste heat from air quenching cooling will be used.

Mostly the heat from the air quenching cooler will go to stack without using it, even though the temperature of the air quenching cooling can be use for other purpose that need heat. In this cement plant as mention before some of flue gas from air quenching cooler is used for ensure that the cement is already dry, while the others go through stack. And in this bachelor thesis will used air quenching cooler flue gas that goes through stack.

First the mass flowrate of flue gas is calculated

$\dot{m} = \rho \; x \; V$
$= 1.29 \; x \; 127044$
$= 163888$ kg/h
$= 45.5245$ kg/s

From thatWaste heat recovery potential on air quenching cooler can be known:

$Q = \dot{m} \; x \; Cp \; x \; \Delta T$
$= 45.5245 \; x \; 0.99066 \; x \; (235\text{-}100)$
$= 6088.399$ kJ/s

Waste available on air quenching cooler
= 6.08 MW

4.3 Refrigerant Selection and Determining Operating Condition

Refrigerant selection and operating condition is connected each other. The operating condition must be appropriate for the refrigerant, for example the operating condition must make the refrigerant in superheated phase when leaving evaporator then when it is expanding in turbine refrigerant should not changing phase into saturated, due to it can damage the turbine.

For this research 4 refrigerant compared (R141b, n-pentane, R11 and R114). And based on simulation R141b will generated most power compared to the other with the same operating condition, the simulation is attached in attachment 7.

After refrigerant already selected and operating condition determined, flow rate of refrigerant, power produced in the turbine and heat rate requirement in condenser can be known, with this following calculation:

1. Calculation of R141b mass flow rate ($\dot{m}$)

$Q_1 = Q_2$

$6088.399 = \dot{m}.\mathrm{Cp}_{liquid}.\Delta T + \dot{m}.\mathrm{h} + \dot{m}.\mathrm{Cp}_{gas}.\Delta T$

$$\dot{m} = \frac{Q}{\left(\mathrm{Cp}_{liquid}.\Delta T\right) + \mathrm{h} + \left(\mathrm{Cp}_{gas}.\Delta T\right)}$$

$$\dot{m} = \frac{6088.399}{273.584}$$

$\dot{m} = 22.25425$ kg/s

2. Power of Turbine (WT)

And after flow rate of refrigerant known, power produce by turbine can be determine with:

$$WT = \dot{m}.(h_{in} - h_{out})$$
$$= 22.25425(511.07 - 481.14)$$
$$= 666.0697 \text{ kJ/s}$$

3. Heat Transfer Rate in Condenser

$$Q = \dot{m}.\mathrm{Cp}_{liquid}.\Delta T + \dot{m}.h + \dot{m}.\mathrm{Cp}_{gas}.\Delta T$$
$$= 22.6(0.86x(20.36) + 217.07 + 1.17x(10.14))$$
$$= 5488.304 \text{ kJ/s}$$

4.4 ORC Component Selection

In the Chapter 2 already mention that ORC system is consists of evaporator, turbine, condenser and pump. In here the calculation for design and selection of all component will be conducted.

4.4.1 Evaporator

Here are the step to make the preliminary design of shell and tube evaporator.

1. Determining shell and tube flow

As mentioned in chapter 3 the evaporator will be shell and tube heat exchanger which is has shell and tube side, to determine fluid in the shell and tube side, this formula will be used

$$2Th_o \geq Tc_i + Tc_o = hot\ fluid\ on\ the\ shell\ side$$
$$2Tc_o \leq Th_i + Th_o = cold\ fluid\ on\ the\ shell\ side$$

And from the operating condition

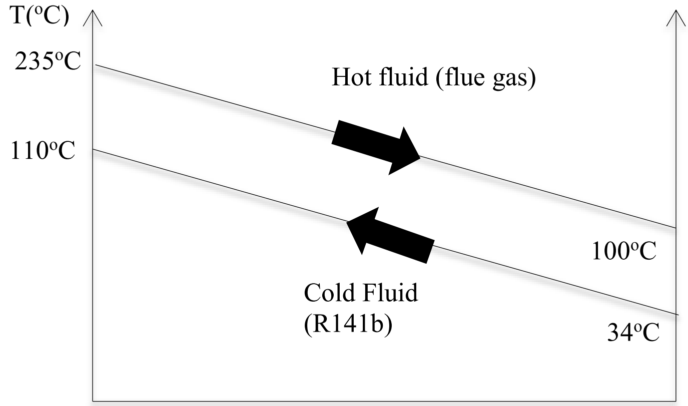


Figure 4.1 Evaporator Operating Condition

$$2x100 \geq 34 + 110$$
$$= 200 \geq 144$$

So the hot fluid (flue gas) will be in the shell side of shell and tube heat exchanger and the R141b will be on the tube side.

2. Data and Properties of Working Fluid and Flue Gas :

Table 4.5 Shell Side Fluid Properties

Shell Side Fluid :	Flue Gas	Vapor (no changing phase in the evaporator)
Name	Quantity	Dimension (SI unit)
Temperature in	235 508.15	°C K
Temperature out	100 373.15	°C K
Mass Flow Rate	45.5245	Kg/s
Density	1.29	Kg/m^3
Specific Heat	0.99066	KJ/kg.K
Thermal Conductivity	0.0358	W/m.K
Dynamic Viscocity	0.00002477	Pa.s
Fouling Factor	0.001	m^2.K/W

Table 4.6 Tube Side Fluid Properties

Tube Side Fluid :	R141b	Liquid to Vapor (changing phase)
Name	Quantity	Dimension (SI unit)
Temperature in	34 307.15	^{o}C K
Temperature out	110 383.15	^{o}C K
Mass Flow Rate	22.24325	Kg/s
Density	1137.4	Kg/m^3
Specific Heat	1.1018	KJ/kg.K
Thermal Conductivity	0.078098	W/m.K
Dynamic Viscocity	0.0001907	Pa.s
Fouling Factor	0.00018	m^2.K/W

3. Evaporator Data Geometry Assumption

Table 4.7 Evaporator Geometry Data

Tube Outside Diameter (OD)	0.0254 m
Tube Inside Diameter (ID)	0.0214 m
Tube Pitch (PT)	30^{o}
Shell Diameter (Ds)	1.084 m
Length of Tube	8.8 m
Baffle Spacing	0.976 m
Number of Baffle	9
Baffle Cut	25%

4. Log Mean Temperature Different (LMTD)

To design shell and tube heat exchanger, it is necessary to approximate the sizing, which is already mention in chapter 2. And the step is:

Due to the type of flow in this operating condition is counter current flow, LMTD formula is:

$$\Delta LMTD = \frac{(Thi - Tco) - (\text{Tho} - \text{Tci})}{ln\frac{(Thi - Tco)}{(Tho - Tci)}}$$

$$\Delta LMTD = \frac{(508.15 - 383.15) - (373.15 - 307.15)}{ln\frac{(508.15 - 383.15)}{(373.15 - 307.15)}}$$

= 92.3811

5. LMTD Correction Factor

And after that the correction factor of LMTD can be calculated with:

$$F = \frac{\left(\frac{\sqrt{R^2 + 1}}{R - 1}.ln\left(\frac{1 - x}{1 - RX}\right)\right)}{ln\left(\frac{\frac{2}{x} - 1 - R + \sqrt{R^2 + 1}}{\frac{2}{x} - 1 - R - \sqrt{R^2 + 1}}\right)}$$

where :

$$R = \frac{Thi - Tho}{Tco - Tci}$$

$$X = \frac{1 - \left(\frac{RP - 1}{P - 1}\right)^{\frac{1}{N}}}{R - \left(\frac{RP - 1}{P - 1}\right)^{\frac{1}{N}}}$$

And for P

$$P = \frac{Tco - Tci}{Tho - Tci}$$

So:

$$R = \frac{508.15 - 373.15}{383.15 - 307.15} = 1.77632$$

$$P = \frac{383.15 - 307.15}{508.15 - 307.15} = 0.37811$$

$$X = \frac{1 - \left(\frac{(1.77632x0.37811) - 1}{0.37811 - 1}\right)^{1}}{1.77632 - \left(\frac{1.77632x0.37811 - 1}{0.37811 - 1}\right)^{1}} = 0.37811$$

From that the value of LMTD correction factor is

$$F = \frac{\left(\frac{\sqrt{1.77632^2 + 1}}{1.77632 - 1} . ln\left(\frac{1 - 0.37811}{1 - (1.77632x0.37811)}\right)\right)}{ln\left(\frac{\frac{2}{0.37811} - 1 - 1.77632 + \sqrt{1.77632^2 + 1}}{\frac{2}{0.37811} - 1 - 1.77632 - \sqrt{1.77632^2 + 1}}\right)}$$

$$F = 0.79185$$

6. Assuming Overall Heat Transfer Coefficient

And after that the overall heat transfer coefficient(U) assumed. For shell and tube heat exchanger that has low pressure gas in the shell side (up to 1 bar) and low viscocity liquid (R141b) in the tube side, the value of U is : 99 W/m^{2}K

7. Heat Transfer Area Required for Evaporator

So from that the total heat transfer area that require to fulfill the heat transfer will be known with :

$$Q = UAF\Delta Tm$$

$$6088406 = 99xAx0.79185x92.3811$$

$$A = 666.21728\, m^2$$

8. Actual Heat Transfer Area of Evaporator

After that we assume the data geometry of the evaporator, to assume it this formula can be used :

$$A = \pi . d . L . Nt\ (m^2)$$

Where Nt (number of tube) is :

$$Nt = \frac{0.78Dctl^2}{C1.PT^2}$$

And Dctl, TP and C1 can be found by :
Dctl :

$$Lbb = 12 + 0.005Ds\ (mm)$$
$$Lbb = 12 + 0.005x1084$$

Then

$$Dotl = Ds - Lbb\ (mm)$$
$$Dotl = 1084 - 17.42$$

From that

$$Dctl = Dotl - OD\ (mm)$$
$$Dctl = 1066.58 - 25.4$$
$$Dctl = 1041.18\ mm$$

From the Heat Exchanger Design Handbook that already mention in chapter II, the minimum ratio of PT (Tube Pitch) is 1.25x of outside diameter value, so

$$PT = 1.25 + 25.4$$
$$Lbb = 31.75mm$$

C1 which is the tube layout constant can be determined with:

$$C1 = 0.86\ for\ \theta_{tp} = 30°\ while$$
$$C1 = 1\ for\ \theta_{tp} = 45°\ and\ 90°$$

Then the Nt is

$$Nt = \frac{0.78x1041.18^2}{0.86x31.75^2}$$

$$Nt = 963.488895$$

The number of tube can be made into 963. After that the A value calculated with previous formula. For the length of the tube

(L) of preliminary design of shell and tube heat exchanger, the value of L usually, is 5-10 according to L/D ratio and the suggested value of the ratio is 8. So for example when the length of tube is 8000 mm the diameter of shell will be 1000 mm for the L/D ratio 8.

And from many trial and error experiment for the suitable value for this heat exchanger, the L of the evaporator is 8.8 m (8800mm) while the diameter of shell is 1084 mm. Which makes the L/D ratio is 8.11808

Then the actual design of A value is

$$A = \pi x 0.0254 x 8.8 x 963$$

$$A = 676.49925\ m^2$$

With this data geometry we can fulfill the requirement of evaporator needed which is 666.21728 m^2.

9. Tube side Heat Transfer Coefficient

To calculate the tube side heat transfer coefficient (h_t) the Reynold number and prandtl number mus be determined first. Where the Reynold number (Re) is :

$$Re = \frac{4\dot{m}}{\pi . ID . \mu . Nt}$$

$$Re = \frac{4x22.25245}{\pi x 0.021 x 0.0002641.963}$$

$$Re = 5208.7658$$

And for the Prandtl Number (Pr) is :

$$Pr = \frac{\mu . Cp}{k}$$

$$Pr = \frac{0.0002641x1166.5}{0.088153}$$

$$Pr = 4.88775$$

Then, for the Reynolds number that varies from 2,100 to 10,000, Dittus-Boelter equation can be used :

$$\frac{h_t.ID}{k} = 0.0243.Re^{0.8}.Pr^n$$

Where n is 0.4 for heating and 0.3 for cooling

$$\frac{h_t.0.0214}{0.088153} = 0.0243.5208^{0.8}.4.88775^{0.4}$$

$$h_t = 191.73188\frac{W}{m^2K}$$

10. Shell side Heat Transfer Coefficient

And for calculation of ideal shell side heat transfer coefficient where the fluid that flow through this shell is flue gas the formula is given by

$$h_i = \frac{j_{i.}Cp.Gs.(\emptyset_s)^n}{Pr^{2/3}}$$

First we calculate the shell side mass velocity (Gs) to get the Reynold number of the flow in the shell side. Gs can be calculated by

$$G_s = \frac{\dot{m}_{s.}}{S_m},$$

Where Sm is shell side crossflow area which is calculated by:

$$S_m = Lbc\left[Lbb + \frac{Dctl}{TP}(Tp - OD)\right]$$

$$S_m = 976\left[17.42 + \frac{1041.18}{31.75}(31.75 - 25.4)\right]$$

$$S_m = 219001\, mm^2 = 0.219\, m^2$$

Then side shell mass velocity is:

$$G_s = \frac{45.5245}{0.219}$$
$$G_s = 207.874 \frac{kg}{m^2 s}$$

For the Reynold Number (Re) and Prandtl Number (Pr) :

$$R_e = \frac{OD.G_s}{\mu} \quad , \quad \text{Pr} = \frac{\mu.Cp}{k}$$

$$R_e = \frac{0.0254x207.874}{0.00002499} \quad , \quad \text{Pr} = \frac{0.00002499x1.0208}{0.0363}$$

$$R_e = 237847 \quad , \quad \text{Pr} = 0.70275$$

After that J_i can be calculated, J_i is ideal Colburn J factor for the shellside that can be determine from appropriate bell Delaware curve or with this calculation:

$$j_i = a_1 \left(\frac{1.33}{\frac{Ltp}{OD}} \right)^a Re^{a_2}$$

Where a is

$$a = \frac{a_3}{1 + 0.14Re^{a_4}}$$

And the values of a_1,a_2,a_3 and a_4 can be determine in the table 2.3 from chapter 2

$$a = \frac{1.45}{1 + 0.14(237847^{0.519})}$$

$$a = 0.01659$$

Then

$$j_i = 0.321x\left(\frac{1.33}{\frac{0.03175}{0.0254}}\right)^{0.01659} x237847^{-0.388}$$

$$j_i = 3.29379$$

The term of $(\emptyset_s)^n$ is the viscocity correction factor, for the gas being cooled the viscocity correction factor value is equal to 1 So the ideal shell side heat transfer coefficient will be

$$h_i = \frac{3.29379x1.0208x207.874x1}{0.70275^{2/3}}$$

$$h_i = 1002.98339\,W/m^2K$$

11. Shell side Heat Transfer Correction Factor

As described in the previous chapter that in the shell side area, the flow fraction for each stream is found by knowing the corresponding flow areas and flow resistance. The heat transfer coefficient for ideal crossflow is then modified for the presence of each stream through correction factor with Bell-Delaware method.

$$h_s = h_i.J_c.J_1J_bJ_s.J_r$$

- Jc

Formula of Segmental baffle window correction factor (Jc) for the baffle cut between 15-45% is :

$$J_c = 0.55 + 0.72F_c$$

And the step to calculate it is:

Calculate upper centriangle of baffle cut (θctl), and the formula is expressed by:

$$\theta_{ctl} = 2cos^{-1}\left[\frac{Ds}{Dctl}\left(1 - \frac{2Bc}{100}\right)\right]$$

$$\theta_{ctl} = 2cos^{-1}\left[\frac{1084}{1041.18}\left(1 - \frac{2Bc}{100}\right)\right]$$

$$\theta_{ctl} = 117.259924°$$

Then fraction number of tubes in the baffle window (Fw) calculated with:

$$Fw = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi}$$

$$Fw = \frac{117.259924}{2\pi} - \frac{sin117.259924}{2\pi}$$

$$Fw = 0.18424$$

After Fw calculated Pure crossflow between baffle cut (Fc) can be calculated :

$$Fc = 1 - 2Fw$$

$$Fc = 1 - 2x0.18424$$

$$Fc = 0.63151$$

Then Jc will be :

$$J_c = 0.55 + 0.72x0.63151$$

$$J_c = 1.004687$$

- J1

For correction factors for baffle leakage effects for heat transfer (J1), the formula is :

$$J_1 = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2rlm}$$

The step to calculate begin with calculating the, Shellside crossflow area (Sm), Shell to baffle leakage area (Ssb) and Tube to baffle hole leakage area (Stb).

Where the $S_m = 219001\, mm^2 = 0.219\, m^2$ as calculated before

For Sb

$$S_{sb} = \pi Ds \frac{Lsb}{2}\left(\frac{2\pi - \theta ds}{2\pi}\right)$$

Lsb is the clearance between Ds and baffle diameter, that expressed with:

$$L_{sb} = 3.11 + 0.004DS$$

$$L_{sb} = 3.11 + (0.004x1084)$$

$$L_{sb} = 7.436\,mm$$

While shell to baffle leakage area (θds) is:

$$\theta_{ctl} = 2cos^{-1}\left(1 - \frac{2Bc}{100}\right)$$

$$\theta_{ctl} = 2cos^{-1}\left(1 - \frac{2.25}{100}\right)$$

$$\theta_{ctl} = 120°$$

And then ssb will be

$$S_{sb} = \pi x1084 \frac{7.436}{2}\left(\frac{2\pi - 120}{2\pi}\right)$$

$$S_{sb} = 8403.58095\,mm^2 = 0.00840358\,m^2$$

For Stb

$$S_{tb} = \frac{\pi}{4}[(OD + Ltb)^2 - OD^2]Nt(1 - Fw)$$

Ltb is diameter clearance between tube outside diameter and baffle hole. TEMA standards specify the value 0.8 or 0.4. In here 0.8 value take for less correction factor of J1.

Then

$$S_{tb} = \frac{\pi}{4}[(25.4 + 0.8)^2 - 25.4^2]963(1 - 0.18424316)$$

$$S_{tb} = 25479.5262\,mm^2 = 0.025479\,m^2$$

After that correlational parameter calculated (rs and rlm) with :

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}}$$

$$r_s = \frac{0.00840358}{0.00840358 + 0.02547953}$$

$$r_s = 0.24801683$$

$$r_{lm} = \frac{S_{sb} + S_{tb}}{S_m}$$

$$r_{lm} = \frac{0.00840358 + 0.02547953}{0.22024026}$$

$$r_{lm} = 0.15384611$$

Then J1 will be

$$J_1 = 0.44(1 - 0.248) \ + [1 - 0.44(1 - 0.154)]e^{-2.2(0154.)}$$

$$J_1 = 0.686122848$$

- Jb

The formula of correction factor for bundle bypass effects for heat transfer (Jb) is:

$$J_b = exp\{-C_{bh}.F_{sbp}[1 - (2.r_{ss})^{1/3}]\}$$

Where Cbh is 1.25 Re < 100 and 1.35 for Re > 100

Then for the calculation of ratio of bypass area to the overall crossflow area (Fsbp) can be calculated from

$$S_b = Lbc(Ds - Dotl + Lpl)$$

Where Lpl equal to 0 for all standard calculation .

$$S_b = 976(1084 - 1066.58 + 0)$$

$$S_b = 17001.92\, mm^2 = 0.01700192\, m^2$$

Then

$$F_{sbp} = \frac{S_b}{S_m}$$

$$F_{sbp} = \frac{0.01700192}{0.22024026}$$

$$F_{sbp} = 0.077197$$

After that for the rss calculation start to determine the number of sealing strip (Ns) and calculate number of tube rows crossed between baffle tips (Ntcc). Assumed that Ns is 1 pair.

And for Ntcc

$$N_{tcc} = \frac{Ds}{Lpp}\left(1 - 2\frac{Bc}{100}\right)$$

Where Lpp is 0.866x PT for 30^{o} tube pitch, so:

$$N_{tcc} = \frac{1084}{27.4955}\left(1 - 2\frac{25}{100}\right)$$

$$N_{tcc} = 19.70909$$

The number of tube rows rounded into 20

So Jb is:

$$J_b = exp\{-1.35x0.077197[1 - (2x0.05)^{1/3}]\}$$

$$J_b = 0.96904$$

- Js

Js is the heat transfer correction factor for unequal baffle spacing at inlet and outlet, in this preliminary design, assumed that the inlet and outlet baffle spacing is same, and because of that reason the Js value is equal to 1.

- Jr

Jr is heat transfer correction factor for adverse temperature gradient in laminar flow. So this correction factor only applies in laminar flow. Due to the flow in this shell is turbulent the value of Jr is equal to 1.

Than shellside heat transfer coefficient is

$$h_s = 1002.98339x1.004687x0.686122848x0.96904x1x1$$

$$h_s = 606.202635\ W/m^2K$$

12. Overall Heat Transfer Coefficient (U)

$$U = \frac{1}{\frac{1}{hs} + Rf_s + \left(\frac{t_w}{k_w}\right).\left(\frac{A_o}{A_m}\right) + \left(Rf_t + \frac{1}{ht}\right)\frac{A_o}{A_i}}$$

Where
tw = 0.004 m
A_o = OD = 0.0254 m
A_i = ID = 0.0214 m
A_m = OD+ID = 0.0468 m

So the overall heat transfer coefficient is:

$$= \frac{1}{\frac{1}{606.2} + 0.001 + \left(\frac{0.004}{65}\right).\left(\frac{0.0254}{0.0468}\right) + \left(0.00018 + \frac{1}{191.7}\right)\frac{0.0254}{0.0214}}$$

$$U = 99.2857095\ W/m^2K$$

13. Tube side Pressure Drop

Kern method use to calculate the pressure drop in shell side and tube side of this evaporator. The formula for calculate the shell tube pressure drop is :

$$\Delta Pt = \frac{f.Gt^2.L.np}{5.22x10^{10}.ID.Sg.\emptyset t}$$

F is the friction factor that can be determine from the moody diagram with the Reynold number divide with the relative rougness of pipe (ε/d). From that the value of friction factor is: 0.004

Gt is tube side mass velocity (kg/m^2.s), to calculate Gt the step is:

Calculating tube side flow area (At)

$$A_t = \frac{\pi}{4}.ID^2.Nt$$

$$A_t = \frac{\pi}{4}.0.0214^2.963$$

$$A_t = 0.34651\, m^2$$

Then

$$G_t = \frac{\dot{m}_{t.}}{A_t}$$

$$G_t = \frac{22.25245}{0.34651}$$

$$G_t = 64.2184 \frac{kg}{m^2 s}$$

The specific gravity (Sg) of R141b when the temperature is 34^{o}C is: 1.216

For $\emptyset s$ which is viscocity correction factor, the calculation is:

$$\emptyset s = \left(\frac{\mu_t}{\mu_w}\right)^{0.14}$$

To determine the viscocity correction vactor, first it need to calculate the tube average temperature, and average wall temperature.

Tube average temperature is :

$$T_{tube\ av} = \frac{inlet\ temp. + outlet\ temp.}{2}$$

$$T_{av} = \frac{307.15 + 383.15}{2} = 345.15K$$

Wall temperature is :

$$T_w = \frac{1}{2}\left(\frac{T_{shell\,av} + T_{shell\,av}}{2}\right)$$

$$T_w = \frac{1}{2}\left(\frac{440.65 + 345.15}{2}\right) = 392.9\,K$$

Then we determine the viscocity of R141b in the tube average and wall temperature

$$\mu_t = 0.00024908\,Pa.s$$

$$\mu_w = 0.00015719\,Pa.s$$

Then $\emptyset s$ equal to

$$\emptyset s = \left(\frac{0.00024908}{0.00015719}\right)^{0.14}$$

$$\emptyset s = 1.06657$$

Next step is, due to the unit of pressure drop in Kern (1965) is in pounds per square inch (psi) which is british unit, the unit must be converted first from SI into the british.

Table 4.8 Converting unit to british 1

L	8.8 meter	28.8714 Feet
Gt	64.2184 kg/m^2.s	47350.64093 lb/ft^2.hr
ID	0.0214 meter	0.0702099 Feet

$$\Delta Pt = \frac{0.004x47350.64093^2.\,28.87.1}{5.22x10^{10}.\,0.07021.1.216.1.06657}$$

$$\Delta Pt = 0.05447372\,psi = 3.75583x10^{-3}bar$$

14. Shell side Pressure Drop

For the shell side pressure drop the formula is :

$$\Delta Ps = \frac{f.Gs^2.Ds.(nb+1)}{5.22.10^{10}.de.Sg.\emptyset s}$$

The friction factor (f) value for Reynold number = 237847.015 and relative rougness of pipe (ε/d)= 0.0018 is 0.023 Gs is shell side mass velocity (kg/m^2.s), that have been calculated before and the value is: 207.874 kg/m^2.s

The specific gravity (Sg) of Air is 1

For $\emptyset s$ which is viscocity correction factor, as mention before for the gases being cooled, the value is 1

And for diameter equivalent (de) can be calculated with this following formula :

$$de = 4\ \frac{\left(PT^2 - \frac{\pi OD^2}{4}\right)}{\pi OD}$$

$$de = 4\ \frac{\left(0.03175^2 - \frac{\pi x 0.0254^2}{4}\right)}{\pi x 0.0254}$$

$$de = 0.02988\, m$$

Due to the unit of pressure drop in Kern (1965) is in pounds per square inch (psi) which is british unit, the unit must be converted first from SI into british unit.

Table 4.9 Converting unit to british 2

Ds	1.084 meter	3.556 Feet
Gs	207.874 kg/m^2.s	153,273.118 lb/ft^2.hr
de	0.02988 meter	0.09803 Feet

$$\Delta Ps = \frac{0.004x47350.64093^2x3.556x(9+1)}{5.22x10^{10}x0.09803x1x1}$$

$$\Delta Ps = 3.75599\, psi = 0.20684\, bar$$

15. Specification of Evaporator

Table 4.10 Specification of Evaporator

Tube Outside Diameter (OD)	0.0254 m
Tube Inside Diameter (ID)	0.0214 m
Tube Pitch (PT)	30^o
Length of Tube	8.8 m
Number of Tube	963
Number of Tube Pass	1
Shell Diameter (Ds)	1.084 m
Baffle Spacing	0.976 m
Number of Baffle	9
Baffle Cut	25%
Evaporator Material	Cu-Ni (k=65)

4.4.2 Turbine

With the operating condition that already set from the P-h diagram, it is already known that the power produced from the turbine is 666.0697 watt. And to choose the turbine the setting parameter is inlet temperature to the turbine, inlet pressure to the turbine and the maximum power output from the turbine.

And from that where the operating condition of this ORC system, where the inlet temperature is 110 ^oC and the inlet pressure is 8 bar with the power output 666.0697 kilowatt. Siemens SST 060 chosen for this ORC system. And the specification of the turbine is given below :

1. Power output up to 0.75 MW
2. Inlet pressure up to 65 bar
3. Intlet temperature up to 480 ^oC
4. Speed up to 24.900 rpm driven machine
5. Dimensions : Length : 1.5, Width : 2.5, Height : 2.5 (in m)

Figure 4.2 Siemens SST 060 Turbine
(Source: Predesigned Steam Turbine)

4.4.3 Condenser

The step to design the condenser is same like the evaporator due to the condenser is shell and tube heat exchanger as well. This part will only show the summary of the condenser calculation.

And for the operating condition of the condenser is:

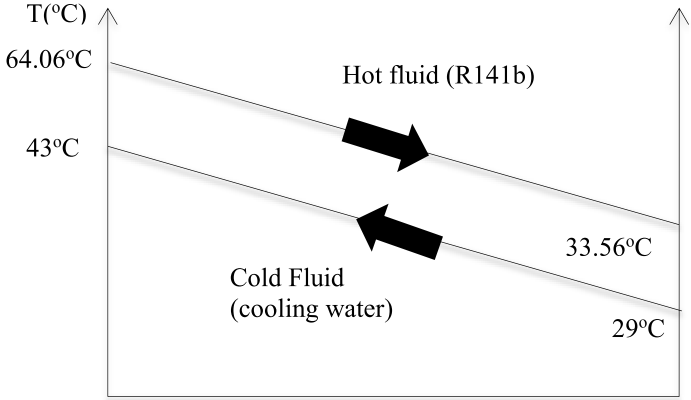


Figure 4.3 Condenser Operating Condition

$$2x43 \leq 64.06 + 33.56$$
$$= 86 \geq 97.62$$

So the cold fluid (cooling water) will be in the shell side of shell and tube heat exchanger and the R141b will be on the tube side.

1. Data and Properties of Working Fluid:

Table 4.11 Shell Side Fluid Properties

Shell Side Fluid :	Cooling Water	Liquid (no changing phase in the Condenser)
Name	Quantity	Dimension (SI unit)
Temperature in	29 302.15	°C K
Temperature out	43 316.15	°C K
Mass Flow Rate	93.7786	Kg/s
Density	997	Kg/m^3
Specific Heat	4.18	KJ/kg.K
Thermal Conductivity	0.607	W/m.K
Dynamic Viscocity	0.0008	Pa.s
Fouling Factor	0.00020	m^2.K/W

Table 4.12 Tube Side Fluid Properties

Tube Side Fluid :	R141b	Vapor to Liquid (changing phase)
Name	Quantity	Dimension (SI unit)
Temperature in	64.06 337.21	°C K
Temperature out	33.56 306.71	°C K
Mass Flow Rate	22.2425	Kg/s
Density	1137.4	Kg/m^3
Specific Heat	1.1018	KJ/kg.K
Thermal Conductivity	0.078098	W/m.K
Dynamic Viscocity	0.0001907	Pa.s
Fouling Factor	0.00018	m^2.K/W

2. Condenser Data Geometry Assumption

Table 4.13 Condenser Geometry Data

Tube Outside Diameter (OD)	0.0254 m
Tube Inside Diameter (ID)	0.0214 m
Tube Pitch (PT)	30^o
Shell Diameter (Ds)	1.12 m
Length of Tube	9 m
Baffle Spacing	1 m
Number of Baffle	9
Baffle Cut	25%
Material of Evaporator	Alumunium Brass

3. Summary of Condenser Calculation

Table 4.14 Calculation Summary

LMTD	10.7839
F (LMTD correction factor)	0.822586
Assumption of U	850 $W/m^2.K$
Heat Transfer Area required (A)	727.8812 m^2
Assumed length of Tube (L)	9 m
Number of tubes (Nt)	1031
Actual heat transfer area (A)	740.7293 m^2
Tube side heat transfer coerfficient (ht)	1863.058 $W/m^2.K$
Shell side Heat Transfer Coefficient	1500.431 $W/m^2.K$
Shell side heat transfer correction factor =	Jc = 1.028437
	J1 = 0.94627
	Jb = 0.833333
	Js = 1
	Jr = 1
ideal side Heat Transfer Coefficient (hs)	1141.235 $W/m^2.K$
Overall heat transfer coefficient (U)	858.2355 $W/m^2.K$
Tubeside Pressure Drop	0.268896 bar
Shellside Pressure Drop	0.67086 bar

4. Specification of Condenser

Table 4.15 Specification of Condenser

Tube Outside Diameter (OD)	0.0254 m
Tube Inside Diameter (ID)	0.0214 m
Tube Pitch (PT)	30°
Length of Tube	9 m
Number of Tube	1031
Number of Tube Pass	1
Shell Diameter (Ds)	1.12 m
Baffle Spacing	1 m
Number of Baffle	9
Baffle Cut	25%
Evaporator Material	Alumunium Brass (k=100)

4.4.4 Pump

To select pump for the refrigerant, capacity and head of the pump must be calculate first, than also the pumping power of the pump. Ant below is the step and equation to select pump.

First the capacity is calculated with:

$$Q = \frac{\dot{m}}{\rho}$$

$$Q = \frac{22.25245}{1216.2}$$

$$Q = 0.0183\ m^3/s = 65.8681\ \frac{m^3}{h} = 289.997\ gpm$$

After that calculate the total head of the pump, and before that, pressure head, velocity head, static head and friction head must be calculated

Pressure head:

$$H_p = \frac{\Delta P}{\rho . g}$$

$$H_p = \frac{8.027 - 1.417x(10^5)}{1216.2x9.81}$$

$H_p = 55.4588\ m$

velocity head is:

$$H_v = \frac{v_2^2}{2.g} - \frac{v_1^2}{2.g}$$

$H_v = 0\ m$ Due to the assumption, there is no velocity

Static head is:

$H_s = h_2 - h_1$

$H_s = 0\ m$ Due to the ORC system assume installed horizontally. And there is no height different.

And friction head caused by fluid friction with the pipe and also fitting if there is any. To calculate friction pipe diameter and the length of pipe in the suction and discharge pipe must be determined first.

In PT. SEMEN INDONESIA (PERSERO) Tbk. that planned to make WHRPG already provide room for the WHPRG component, the dimension of the room is 35m long and 19m width.

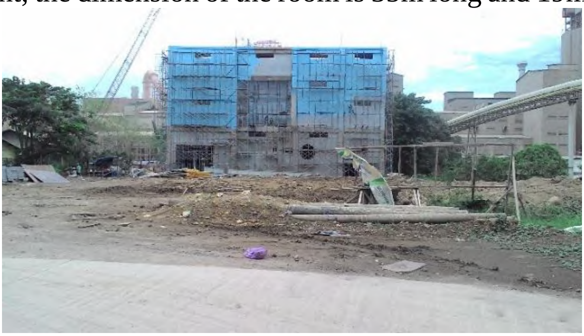


Figure 4.4 Construction for WHPRG Room on 13th February 2016

Assumed that distance of pipe between condenser to pump is 20m long and between pump to condenser 60 m. And for pipe diameter it is assumed the velocity of the liquid is 0.78 m/s according to comparison system (JFE engineering). So to calculate pipe inside diameter the equation is:

$$D = \sqrt{\frac{Q.4}{\pi.v}}$$

$$D = \sqrt{\frac{0.0183x4}{\pi x 0.78}}$$

$$D = 0.172\, m = 172.864\, mm$$

And according to JIS for the cast iron pipe for 172.86 mm categorized to:

Table 4.16 Pipe Diameter Data

Outside Diameter	190.7 mm
Inside Diameter	185.4 mm
Thickness	5.3 mm

From that spec of pipe the velocity of the fluid will be 0.678 m/s using same equation above.

The friction head consist of major losses and minor losses, and for the calculation is:

Major losses caused by long of the pipe in the suction side and in the discharge side, it is assumed that the suction side pipe long 20m while in the discharge side 60m. And the calculation of major losses can use this following formula:

Suction side:

$$H_{fmajor\ sucyion} = \lambda.L.\frac{D}{2g}$$

Where:

$$\lambda = 0.02 + \frac{0.0005}{D}$$

$$\lambda = 0.02 + \frac{0.0005}{0.1854}$$

$$\lambda = 0.022697$$

Then total length of head major is 80m

$$H_{fmajor} = 0.022697x80\frac{0.1854}{2x9.8}$$

$$H_{fmajor} = 0.0171756\,m$$

Minor losses

$$H_{fminor} = \Sigma n.k.\frac{v^2}{2g}$$

It assumed that there are 3 90^o elbow, 2 SDRNV in the suction side of the pump and 9 90^o elbow, 6 SDNRV in the discharge side of the pump and also 1 filter before the pump. And for minor losses calculation, the calculation in the suction and discharge side is merged

Fitting	n	k	n x k
Elbow 90^o	12	1	12
SDNRV	8	3	24
Filter	1	2.5	2.5
Total k			38.5

$$H_{fminor} = 15x\frac{0.678^2}{2x9.81}$$

$$H_{fminor} = 0.9020\,m$$

And the total Head friction value of the pump is the addition of Head friction major and minor, and the value will be:

$$H_{ftotal} = H_{fmajor} + H_{fminor}$$

$$H_{ftotal} = 0.0.0171756 + 0.9020$$

$$H_{ftotal} = 0.9191756\,m$$

And total head of pump will be:
$H = H_p + H_v + H_s + H_f$
$H = 55.4588 + 0 + 0 + 0.9191756$
$H = 56.3779\ m = 185ft$

After that for calculating pumping power of pump this given equation can be used:

$$P = \frac{\rho.Q.g.H}{\eta}$$

$$P = \frac{22.25245x9.8x56.3779}{75\%}$$

$$P = 16.392\ kW$$

With the given data calculation (capacity, head and pumping power), the Buffalo Pump is chosen. There 13 pump size available of this type, and one of those type (I) is suitable for the pump specification.

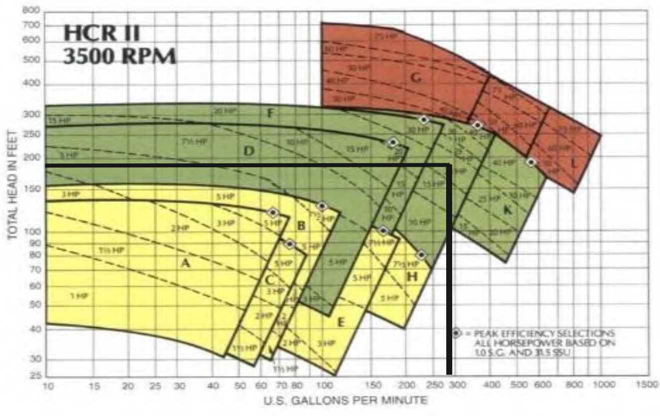


Figure 4.5 Pump Performance Chart
(Source : www.buffalopumps.com)

Table 4.17 Pump Specification

Brand	Buffalo Pump
Type	Centrifugal Pump
Type	Can O Matic Refrigerant Pump HCR II
Capacity	Up to 1500 GPM
Head	Up to 700 Ft
Power	25 HP = 18.64 kW
rpm	3500 rpm

Attachment 1:
Waste Heat Recovery Potential In Preheater
Based on calculation from Confederation of Indian Industry, available waste heat in preheater is:

1. Heat in Preheater Gas

Preheater String 1:

$Q = \dot{m}.\mathrm{Cp}.\mathrm{T}$

= 0.877 × 0.2515 × 401.79

=88.621 kcal/kg clinker

Preheater String 2

$Q = \dot{m}.\mathrm{Cp}.\mathrm{T}$

= 0.887 × 0.2462 × 396.79

= 86.651 kcal/kg clinker

2. Heat Required for Raw Mill

a. Raw Mill Capacity = Capacity of Kiln × Raw Meal Factor × (Running Hour of Raw Mill/day)

= 8466 × 1.55 × 24/24

= 13122 Ton/Day

= 546.76 Ton/Hour

b. Moisture in Raw Mill = Raw Mill Capacity × (100/(100-moisture of raw material)) – Raw Mill Capacity

= 546.76 × (100/(100-5.74)) – 546.76

= 33.295 Ton/Hour

= 60.895 kg water/ Ton clinker

c. Heat Requirement for Raw Mill = Moisture in Raw Mill × Heat Requirement in Raw Mill/1000

= 60.985 × 950/1000

= 57.851 kcal/kg clinker

3. Heat Required for Coal Mill (Coal Requirement)

a. Specific coal consumption = Specific Heat Consumption / Calorific Value of Coal

= 760 / 4800

= 0.1583 kg coal/kg clinker

b. Coal Mill Capacity : Specific Coal Consumption x Coal Consumption x Running Hours of Coal Mill/Day

= 0.1583 × 366.57 × (24/24)

= 58.04 Ton/Hour

4. Moisture Evaporation in Coal Mill = Coal Mill Capacity x (100/(100-Coal Moisture)) – Coal Mill Capacity

= 58.04 (100/ (100-11)) – 58.04

= 7.1735 Ton/Hour

= 123.6 kg water / Ton Coal

5. Heat Requirement For Coal Mill = Moisture in Coal Mill × Heat Requirement in Coal Mill/1000

= 123.6× 950/1000

= 117.42

6. Waste Heat Available in Preheater = Heat in Preheater (String 1+ String 2) – Heat Required in (Raw Mill + Coal Mill)

= (88.621+86.651) – (57.851+117.42)

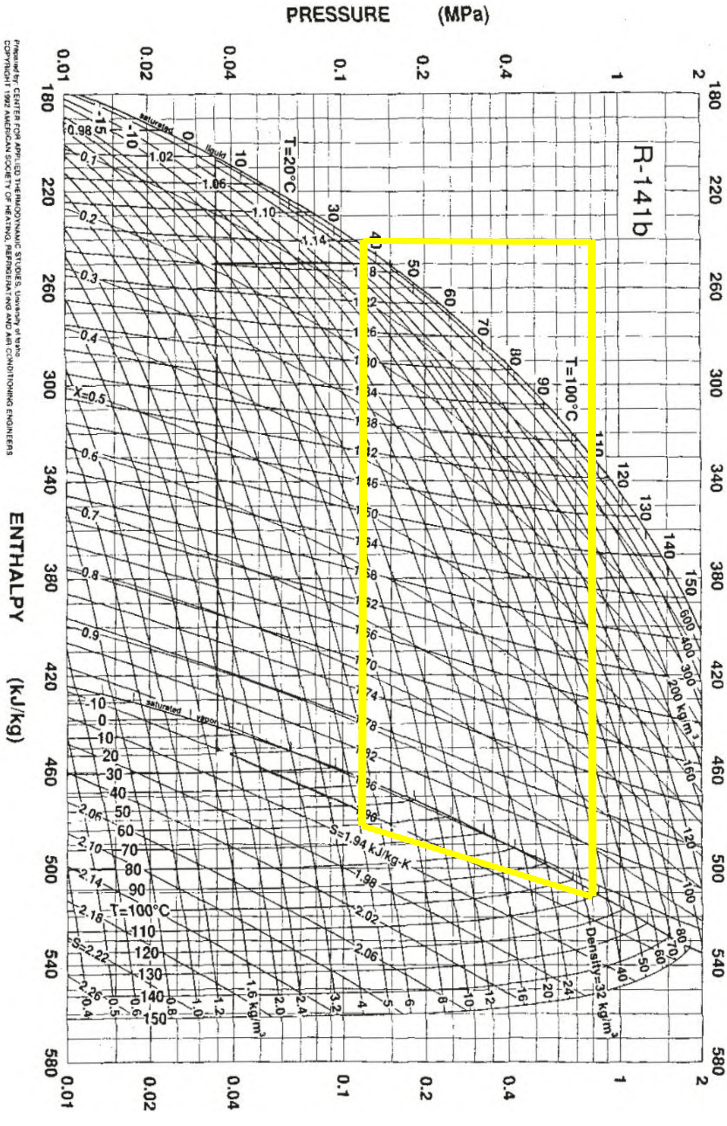
= 0.0054 kcal/kg clinker

= 45.87 kcal/day

= 0.0022 kW = 2.2 W

Attachment 2:

P-h diagram of R141b, and operating condition of ORC



Attachment 3:
R141b Properties Table

R141b Liquid and Vapor Phase Table

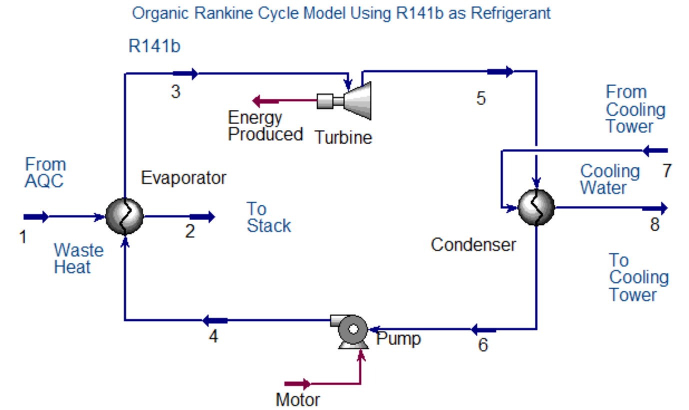
Table Thermodynamics Properties of R141b in Liquid Phase

R141b	Properties Table	Liquid Phase			
Temp (C)	Density (kg/m3)	Specific Volume (m3/kg)	Enthalpy (kJ/kg)	Cp (kJ/kg.C)	Viscosity (Pa.s)
33.56	1217.1	0.00082165	238.34	1.1658	0.00037116
34	1216.2	0.00082224	238.94	1.1665	0.00036937
43.772	1196.7	0.00083564	250.3	1.1814	0.00033236
64.06	1154.6	0.0008661	274.79	1.2167	0.00026958
100	1072.5	0.00093238	319.98	1.2983	0.00018991
108.46	1051.3	0.00095117	331.08	1.3227	0.00017516
110	1047.4	0.0009547	333.12	1.3274	0.00017260

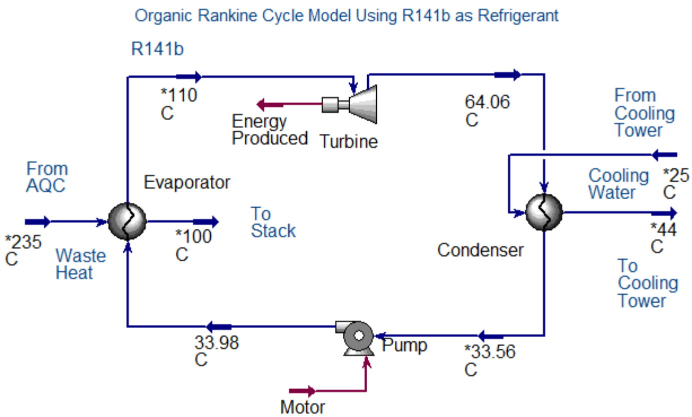
Table Thermodynamics Properties of R141b in Vapor Phase

R141b	Properties Table	Liquid Phase			
Temp (C)	Density (kg/m3)	Specific Volume (m3/kg)	Enthalpy (kJ/kg)	Cp (kJ/kg.C)	Viscosity (Pa.s)
33.56	5.1057	0.19586	460.40	0.81438	9.4535e-06
34	5.1790	0.19309	460.70	0.81544	9.4671e-06
43.772	7.0289	0.14227	467.37	0.83937	9.7708e-06
64.06	12.475	0.080159	481.14	0.89204	1.0414e-05
100	29.678	0.033695	504.80	1.0007	1.1657e-05
108.46	35.670	0.028035	510.12	1.0312	1.1986e-05
110	36.861	0.027129	511.07	1.0371	1.2049e-05

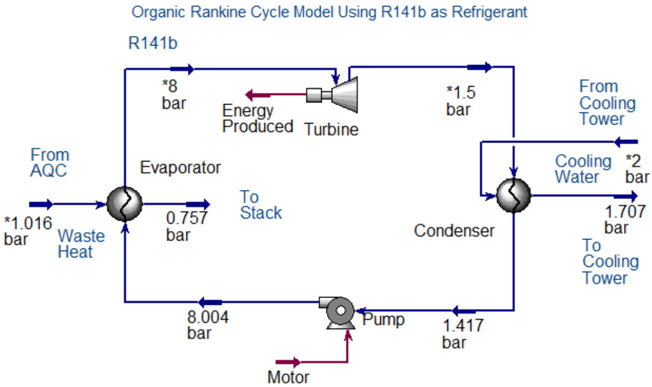
Attachment 4:
Hysis Simulation



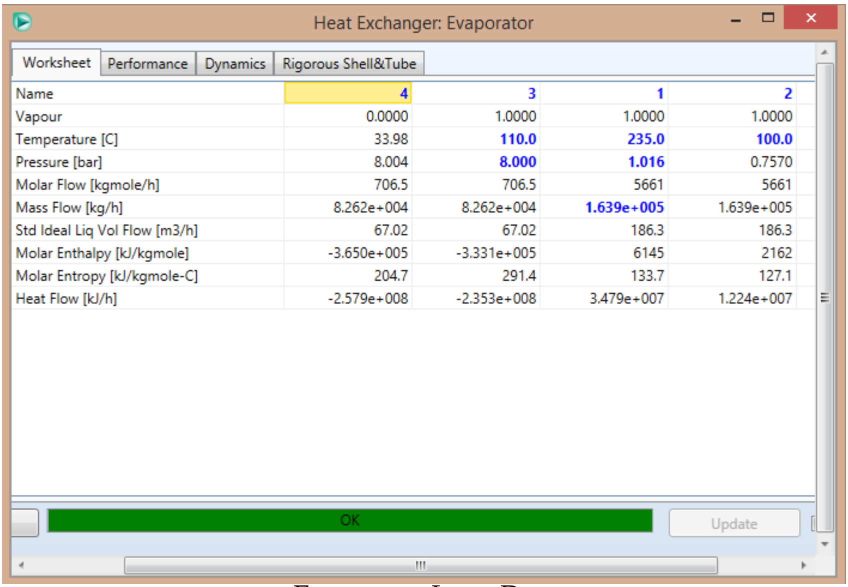
Organic Rankine Cycle System with R141b as working fluid



Temperature Operating Condition

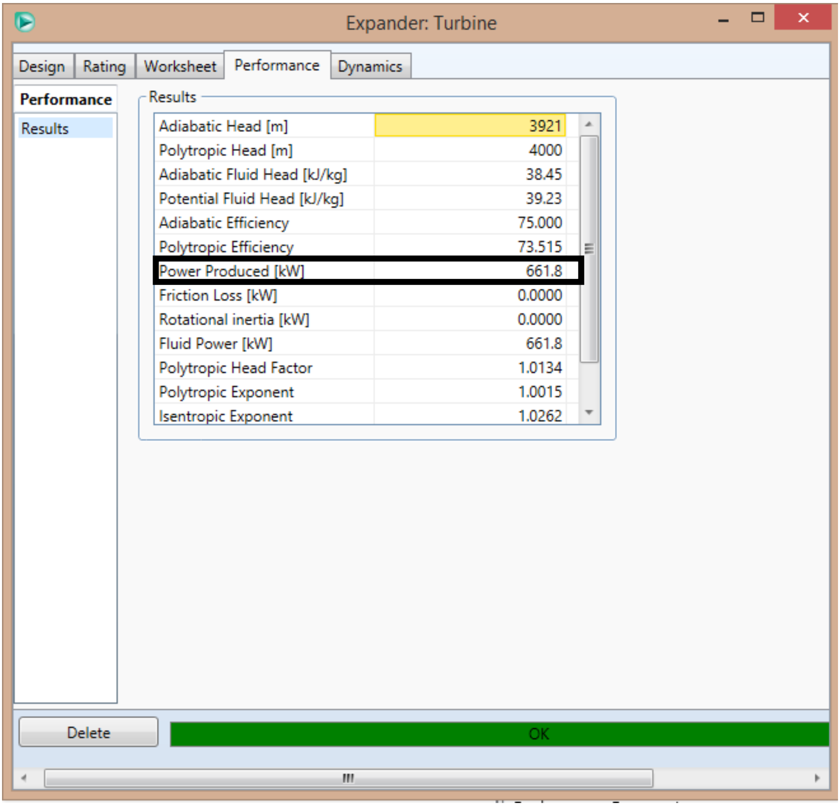


Pressure Operating Condition

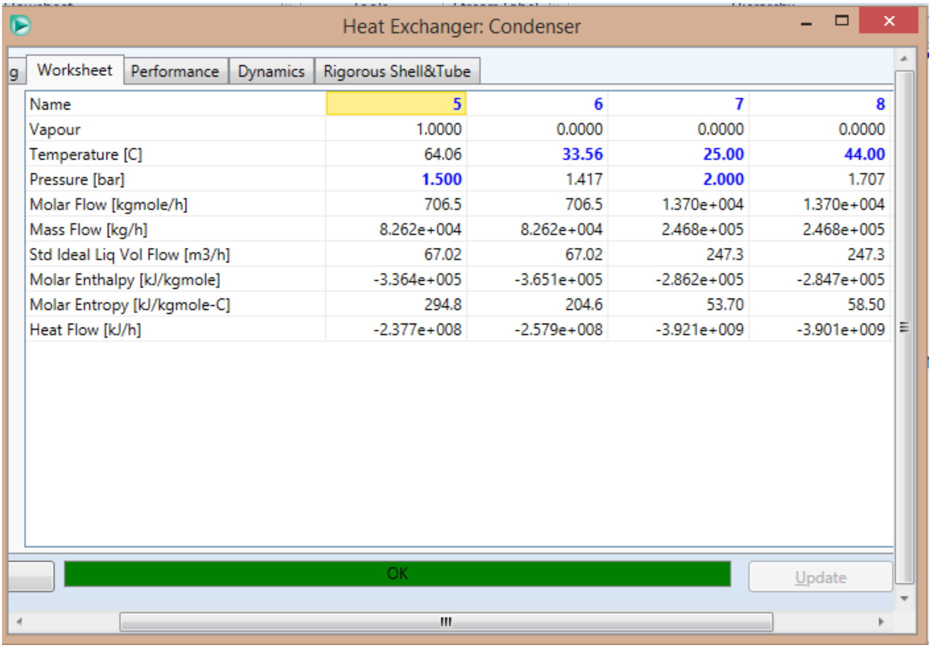


Name	4	3	1	2
Vapour	0.0000	1.0000	1.0000	1.0000
Temperature [C]	33.98	110.0	235.0	100.0
Pressure [bar]	8.004	8.000	1.016	0.7570
Molar Flow [kgmole/h]	706.5	706.5	5661	5661
Mass Flow [kg/h]	8.262e+004	8.262e+004	1.639e+005	1.639e+005
Std Ideal Liq Vol Flow [m3/h]	67.02	67.02	186.3	186.3
Molar Enthalpy [kJ/kgmole]	-3.650e+005	-3.331e+005	6145	2162
Molar Entropy [kJ/kgmole-C]	204.7	291.4	133.7	127.1
Heat Flow [kJ/h]	-2.579e+008	-2.353e+008	3.479e+007	1.224e+007

Evaporator Input Data

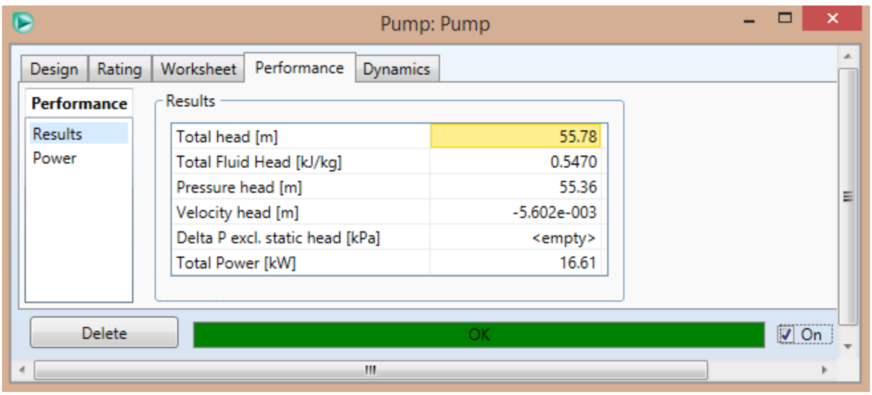


Turbine Power Produced



Name	5	6	7	8
Vapour	1.0000	0.0000	0.0000	0.0000
Temperature [C]	64.06	33.56	25.00	44.00
Pressure [bar]	1.500	1.417	2.000	1.707
Molar Flow [kgmole/h]	706.5	706.5	1.370e+004	1.370e+004
Mass Flow [kg/h]	8.262e+004	8.262e+004	2.468e+005	2.468e+005
Std Ideal Liq Vol Flow [m3/h]	67.02	67.02	247.3	247.3
Molar Enthalpy [kJ/kgmole]	-3.364e+005	-3.651e+005	-2.862e+005	-2.847e+005
Molar Entropy [kJ/kgmole-C]	294.8	204.6	53.70	58.50
Heat Flow [kJ/h]	-2.377e+008	-2.579e+008	-3.921e+009	-3.901e+009

Condenser Input Data



Total head [m]	55.78
Total Fluid Head [kJ/kg]	0.5470
Pressure head [m]	55.36
Velocity head [m]	-5.602e-003
Delta P excl. static head [kPa]	<empty>
Total Power [kW]	16.61

Pump Power Requirement

Attachment 5:

Condenser Calculation

1. Log Mean Temperature Different (LMTD)

$$\Delta LMTD = \frac{(Thi - Tco) - (\mathrm{Tho} - \mathrm{Tci})}{ln\frac{(Thi - Tco)}{(Tho - Tci)}}$$

$$\Delta LMTD = \frac{(337.36 - 316{,}15) - (306.71 - 302.15)}{ln\frac{(337.36 - 316.15)}{(306.71 - 302.15)}}$$

= 10.7839

2. LMTD Correction Factor

$$F = \frac{\left(\frac{\sqrt{R^2+1}}{R-1}.\, ln\left(\frac{1-x}{1-RX}\right)\right)}{ln\left(\frac{\frac{2}{x} - 1 - R + \sqrt{R^2+1}}{\frac{2}{x} - 1 - R - \sqrt{R^2+1}}\right)}$$

where :

$$R = \frac{Thi - Tho}{Tco - Tci}$$

$$X = \frac{1 - \left(\frac{RP-1}{P-1}\right)^{\frac{1}{N}}}{R - \left(\frac{RP-1}{P-1}\right)^{\frac{1}{N}}}$$

And for P

$$P = \frac{Tco - Tci}{Tho - Tci}$$

So:

$$R = \frac{337.36 - 306.71}{316.15 - 302.15} = 2.178571$$

$$P = \frac{316.15 - 306.71}{337.36 - 306.71} = 0.309508$$

$$X = \frac{1 - \left(\frac{(2.178571x0.37811) - 1}{0.309508 - 1}\right)^1}{1.77632 - \left(\frac{2.178571xP - 1}{0.309508 - 1}\right)^1} = 0.309508$$

From that the value of LMTD correction factor is

$$F = \frac{\left(\frac{\sqrt{2.178571^2 + 1}}{2.178571 - 1} . ln\left(\frac{1 - 0.309508}{1 - (2.178571x0.309508)}\right)\right)}{ln\left(\frac{\frac{2}{0.309508} - 1 - 2.178571 + \sqrt{2.178571^2 + 1}}{\frac{2}{0.309508} - 1 - 2.178571 - \sqrt{2.178571^2 + 1}}\right)}$$

$$F = 0.822586$$

3. Assuming Overall Heat Transfer Coefficient
U is : 850 W/m^{2}K

4. Heat Transfer Area Required for Evaporator
$Q = UAF\Delta Tm$

$$5488304 = 850xAx0.822586x10.7839$$

$$A = 727.8812\ m^2$$

5. Actual Heat Transfer Area of Evaporator
After that we assume the data geometry of the evaporator, to assume it this formula can be used :

$$A = \pi . Ds. L. Nt\ (m^2)$$

Where Nt is:

$$Nt = \frac{0.78Dctl^2}{C1.PT^2}$$

And Dctl, TP and C1 can be found by:

Dctl:

$$Lbb = 12 + 0.005Ds\ (mm)$$
$$Lbb = 12 + 0.005x1120$$

Then

$$Dotl = Ds - Lbb\ (mm)$$
$$Dotl = 1120 - 17.6$$

From that

$$Dctl = Dotl - OD\ (mm)$$
$$Dctl = 1102.4 - 25.4$$
$$Dctl = 1070.65\ mm$$

From the Heat Exchanger Design Handbook that already mention in chapter II, the minimum ratio of PT (Tube Pitch) is 1.25x of outside diameter value, so

$$PT = 1.25 + 25.4$$
$$Lbb = 31.75mm$$

C1 which is the tube layout constant can be determined with:

$$C1 = 0.86\ for\ \theta_{tp} = 30°\ while$$
$$C1 = 1\ for\ \theta_{tp} = 45°\ and\ 90°$$

Then the Nt is

$$Nt = \frac{0.78x1070.65^2}{0.86x31.75^2}$$
$$Nt = 1031.344 = 1031$$

is 5-10 according to L/D ratio and the suggested value of the ratio is 8.

9 m (8800mm) while the diameter of shell is 1120 mm. Which makes the L/D ratio is 8.03

Then the actual design of A value is

$$A = \pi x 0.0254 x 9 x 1031$$

$$A = 740.7293\ m^2$$

With this data geometry we can fulfill the requirement of evaporator needed which is 513.346 m^2.

6. Tube side Heat Transfer Coefficient

Where the Reynold number (Re) is:

$$Re = \frac{4\dot{m}}{\pi . ID . \mu . Nt}$$

$$Re = \frac{4x22.25245}{\pi x 0.021 x 0.00001041 x 1031}$$

$$Re = 175087$$

And for the Prandtl Number (Pr) is :

$$Pr = \frac{\mu . Cp}{k}$$

$$Pr = \frac{0.000010414x892.04}{0.012846}$$

$$Pr = 0.72316$$

$$\frac{h_t . ID}{k} = 0.0243 . Re^{0.8} . Pr^n$$

$$\frac{h_t . 0.0214}{0.012846} = 0.0243 . 175087^{0.8} . 0.72316^{0.3}$$

$$h_t = 1863.058\ W/m^2K$$

7. Shell side Heat Transfer Coefficient

$$h_i = \frac{j_{i.}Cp.Gs.(\emptyset_s)^n}{Pr^{2/3}}$$

First we calculate the shell side mass velocity (Gs) to get the Reynold number of the flow in the shell side. Gs can be calculated by

$$G_s = \frac{\dot{m}_{s.}}{S_m},$$

Where Sm is shell side crossflow area which is calculated by:

$$S_m = Lbc\left[Lbb + \frac{Dctl}{TP}(Tp - OD)\right]$$

$$S_m = 1000\left[17.6 + \frac{1070.65}{31.75}(31.75 - 25.4)\right]$$

$$S_m = 259537\, mm^2 = 0.259538\, m^2$$

Then side shell mass velocity is:

$$G_s = \frac{93.7786}{0.259538}$$

$$G_s = 361.3295\frac{kg}{m^2 s}$$

For the Reynold Number (Re) and Prandtl Number (Pr) :

$$R_e = \frac{OD.G_s}{\mu} \quad , \quad \mathrm{Pr} = \frac{\mu.Cp}{k}$$

$$R_e = \frac{0.0254x361.329}{0.000772} \quad , \quad \mathrm{Pr} = \frac{0.00089011x4181.6}{0.607}$$

$$R_e = 11888.3 \quad , \quad \mathrm{Pr} = 5.3138$$

$$j_i = a_1 \left(\frac{1.33}{\frac{Ltp}{OD}} \right)^a Re^{a_2}$$

Where a is

$$a = \frac{a_3}{1 + 0.14Re^{a_4}}$$

And the values of a_1,a_2,a_3 and a_4 can be determine in the table 2.3 from chapter 2

$$a = \frac{1.45}{1 + 0.14(11888^{0.519})}$$

$$a = 0.075349$$

Then

$$j_i = 0.321x \left(\frac{1.33}{\frac{0.03175}{0.0254}} \right)^{0.07839} x11888^{-0.388}$$

$$j_i = 12.29367$$

The term of $(\emptyset_s)^n$ is the viscosity correction factor, for the gas being cooled the viscosity correction factor value is equal to 1

$$h_i = \frac{12.29367x1.017x361.3295x1}{5.313^{2/3}}$$

$$h_i = 1863.058\ W/m^2K$$

8. Shell side Heat Transfer Correction Factor

$$h_s = h_i.J_c.J_1J_bJ_s.J_r$$

- Jc

Formula of Segmental baffle window correction factor (Jc) for the baffle cut between 15-45% is :

$J_c = 0.55 + 0.72F_c$

And the step to calculate it is:

Calculate upper centriangle of baffle cut (θctl), and the formula is expressed by:

$$\theta_{ctl} = 2cos^{-1}\left[\frac{Ds}{Dctl}\left(1 - \frac{2Bc}{100}\right)\right]$$

$$\theta_{ctl} = 2cos^{-1}\left[\frac{1120}{1070.65}\left(1 - \frac{2Bc}{100}\right)\right]$$

$$\theta_{ctl} = 113.0947°$$

Then fraction number of tubes in the baffle window (Fw) calculated with:

$$Fw = \frac{\theta_{ctl}}{2\pi} - \frac{sin\theta_{ctl}}{2\pi}$$

$$Fw = \frac{113.0947}{2\pi} - \frac{sin114.0947}{2\pi}$$

$$Fw = 0.16775$$

After Fw calculated Pure crossflow between baffle cut (Fc) can be calculated:

$$Fc = 1 - 2Fw$$

$$Fc = 1 - 2x0.16775$$

$$Fc = 0.664496$$

Then Jc will be :

$$J_c = 0.55 + 0.72x0.664496$$

$$J_c = 1.0028437$$

- J1

For correction factors for baffle leakage effects for heat transfer (J1), the formula is :

$$J_1 = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]e^{-2.2rlm}$$

The step to calculate begin with calculating the, Shellside crossflow area (Sm), Shell to baffle leakage area (Ssb) and Tube to baffle hole leakage area (Stb).

Where the $S_m = 259537\, mm^2 = 0.259538\, m^2$ as calculated before

For Sb

$$S_{sb} = \pi Ds \frac{Lsb}{2}\left(\frac{2\pi - \theta ds}{2\pi}\right)$$

Lsb is the clearance between Ds and baffle diameter, that expressed with:

$$L_{sb} = 3.11 + 0.004DS$$
$$L_{sb} = 3.11 + (0.004x1120)$$
$$L_{sb} = 7.58\, mm$$

While shell to baffle leakage area (θds) is:

$$\theta_{ctl} = 2cos^{-1}\left(1 - \frac{2Bc}{100}\right)$$
$$\theta_{ctl} = 2cos^{-1}\left(1 - \frac{2.25}{100}\right)$$
$$\theta_{ctl} = 120°$$

And then ssb will be

$$S_{sb} = \pi x 11120 \frac{7.58}{2}\left(\frac{2\pi - 120}{2\pi}\right)$$

$$S_{sb} = 8893.867\, mm^2 = 0.00889\, m^2$$

For Stb

$$S_{tb} = \frac{\pi}{4}[(OD + Ltb)^2 - OD^2]Nt(1 - Fw)$$

Ltb is diameter clearance between tube outside diameter and baffle hole. TEMA standards specify the value 0.8 or 0.4. In here 0.8 value take for less correction factor of J1.

$$S_{tb} = \frac{\pi}{4}[(25.4 + 0.8)^2 - 25.4^2]1031(1 - 0.16775)$$
$$S_{tb} = 27830.2\ mm^2 = 0.02783\ m^2$$

After that correlational parameter calculated (rs and rlm) with :

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}}$$
$$r_s = \frac{0.00889}{0.00889\ + 0.02783}$$

$$r_s = 0.242181$$
$$r_{lm} = \frac{S_{sb} + S_{tb}}{S_m}$$
$$r_{lm} = \frac{0.00889\ + 0.02783}{0.259538}$$
$$r_{lm} = 0.158478$$

Then J1 will be

$$J_1 = 0.44(1 - 0.242)\ \ + [1 - 0.44(1 - 0.242)]e^{-2.2(0.158)}$$
$$J_1 = 0.833333$$

- Jb

The formula of correction factor for bundle bypass effects for heat transfer (Jb) is:

$$J_b = exp\left\{-C_{bh}.F_{sbp}\left[1 - (2.r_{ss})^{1/3}\right]\right\}$$

Then for the calculation of ratio of bypass area to the overall crossflow area (Fsbp) can be calculated from

$$S_b = Lbc(Ds - Dotl + Lpl)$$

Where Lpl equal to 0 for all standard calculation.

$$S_b = 1000(1120 - 1102.4 + 0)$$

$$S_b = 17600\, mm^2 = 0.0176\, m^2$$

Then

$$F_{sbp} = \frac{S_b}{S_m}$$
$$F_{sbp} = \frac{0.0176}{0.259538}$$
$$F_{sbp} = 0.07595$$

Assumed that Ns is 1 pair.

And for Ntcc

$$N_{tcc} = \frac{Ds}{Lpp}\left(1 - 2\frac{Bc}{100}\right)$$

Where Lpp is 0.866x PT for 30^o tube pitch, so:

$$N_{tcc} = \frac{1120}{27.4955}\left(1 - 2\frac{25}{100}\right)$$
$$N_{tcc} = 20.36364$$

The number of tube rows rounded into 20

So Jb is:

$$J_b = exp\{-1.35x0.08203[1 - (2x0.055)^{1/3}]\}$$

$$J_b = 0.94627$$

- Js

Js value is equal to 1.

- Jr

Jr is equal to 1.

Than shellside heat transfer coefficient is

$$h_s = 1500.431x1.0028437x0.833333x0.94627x1x1$$

$$h_s = 1141.235\, W/m^2K$$

9. Overall Heat Transfer Coefficient (U)

$$U = \frac{1}{\frac{1}{hs} + Rf_s + \left(\frac{t_w}{k_w}\right).(\frac{A_o}{A_m}) + \left(Rf_t + \frac{1}{ht}\right)\frac{A_o}{A_i}}$$

Where

tw = 0.004 m

A_o = OD = 0.0254 m

A_i = ID = 0.0214 m

A_m = OD+ID = 0.0468 m

So the overall heat transfer coefficient is:

$$= \frac{1}{\frac{1}{1141.24} + 0.00025 + \left(\frac{0.004}{100}\right).(\frac{0.0254}{0.0468}) + \left(0.00018 + \frac{1}{1863.06}\right)\frac{0.0254}{0.0214}}$$

$$U = 858.2355\, W/m^2K$$

10. Tube side Pressure Drop

$$\Delta Pt = \frac{f.Gt^2.L.np}{5.22x10^{10}.ID.Sg.\emptyset t}$$

F is the friction factor that can be determine from the moody diagram with the Reynold number divide with the relative roughness of pipe (ε/d). From that the value of friction factor is: 0.031

Gt is tube side mass velocity (kg/m^2.s), to calculate Gt the step is:

Calculating tube side flow area (At)

$$A_t = \frac{\pi}{4} xID^2xNt$$

$$A_t = \frac{\pi}{4} x0.0214^2x1120$$

$$A_t = 0.37098\, m^2$$

Then

$$G_t = \frac{\dot{m}_{t.}}{A_t}$$

$$G_t = \frac{22.25245}{0.37098}$$

$$G_t = 59.98283 \frac{kg}{m^2 s}$$

The specific gravity (Sg) of R141b when the temperature is 64.06°C is: 1.25

For $\emptyset s$ which is viscosity correction factor, the calculation is:

$$\emptyset s = \left(\frac{\mu_t}{\mu_w}\right)^{0.14}$$

For $\emptyset s$ which is viscosity correction factor, as mention before for the gases being cooled, the value is 1

Due to the unit of pressure drop in Kern (1965) is in pounds per square inch (psi) which is british unit, the unit must be converted first from SI into the british.

Table Converting unit to british 1

L	9 meter	25.91864 Feet
Gt	59.9828 kg/m^2.s	44227.6 lb/ft^2.hr
ID	0.0214 meter	0.0702099 Feet

$$\Delta Pt = \frac{0.031x44227.6\ ^2x25.9828\ \text{x1}}{5.22x10^{10}.0.07021x1.25x1}$$

$$\Delta Pt = 3.90838\, psi = 0.268896\, bar$$

11. Shell side Pressure Drop

$$\Delta Ps = \frac{f.Gs^2.Ds.(nb+1)}{5.22.10^{10}.de.Sg.\emptyset s}$$

The friction factor (f) value for Reynold number = is 0.02

Gs is shell side mass velocity (kg/m^2.s), that have been calculated before and the value is: 361.3295 kg/m^2.s

The specific gravity (Sg) of Water is 1

$$\emptyset s = 1.01138$$

And for diameter equivalent (de) can be calculated with this following formula:

$$de = 4\ \frac{\left(PT^2 - \frac{\pi OD^2}{4}\right)}{\pi OD}$$

$$de = 4\ \frac{\left(0.03175^2 - \frac{\pi x 0.0254^2}{4}\right)}{\pi x 0.0254}$$

$$de = 0.02988\ m$$

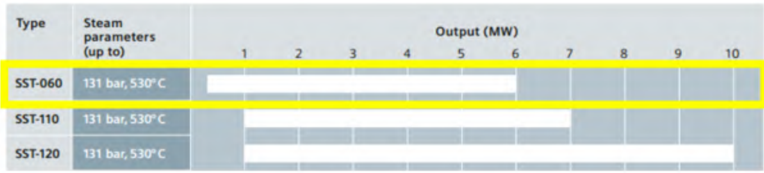
Table Converting unit to british 2

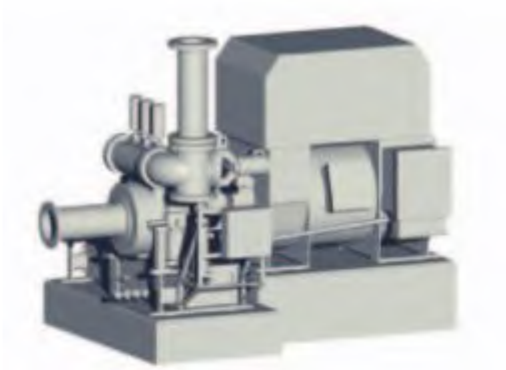
Ds	1.12 meter	3.69 Feet
Gs	361.3295 kg/m^2.s	245820 lb/ft^2.hr
de	0.02988 meter	0.09803 Feet

$$\Delta Ps = \frac{0.02 x 266421.9^2 x 3.69 x (9+1)}{5.22 x 10^{10} x 0.09803 x 1 x 1.01138}$$

$$\Delta Ps = 9.736265\ psi = 0.67086\ bar$$

Attachment 6:
Turbine and Pump Specification
Turbine





SIEMENS SST 060 STEAM TURBINE (formerly known as AFA, CFA or CFR series)
The SST- 060 stands out by their rugged design and renowned reliability even under the most severe operating conditions. Ideal for saturated steam service. Their suitability for use as condensation or back-pressure turbines in combination with various integral gears modules opens up a broad application range.

Technical data

- Power output up to 6 MW
- Inlet pressure up to 65 bar
- Inlet temperature dry saturated steam up to 480 °C
- Speed up to 24.900 rpm driven machine
- Exhaust pressure: back pressure up to 17 bar

• 5 different gearing size
• Typical dimensions:
Length 1.5 m/4.9 ft. × Width 2.5 m/8.2 ft. × Height 2.5 m/8.2 ft.× Weight Turbine including oil reservoir and coupling up to 12 Ton

Features

• Backpressure or condensing type
• Flexible rotor
• Package unit design
• Oil unit integrated in base frame
• Nozzle group control valves available
• Quick-start without pre-heating
• Meet requirements of API 611/612
• Atex version available
• Suitable for Organic Rankine Cycle (ORC) already installed more than 800 applications
• Suitable for gas expansion

Application

• Cogeneration
• Biomass
• Waste to Energy
• Gas Expansion

Pump

Buffalopumps

BUFFALO CAN-O-MATIC® II

Reliable Zero Leakage Pumps

Background

The Can-O-Matic II is a hermetically sealed zero leakage pump with an ANSI standard pump end and a motor with unique tapered spring-loaded wear-compensating bearings. This specialized bearing construction provides a high level of reliability. When operating, pump axial and radial thrust loads within a Can-O-Matic II are balanced, and bearings wear concentrically. Elliptical wear patterns that cause stator-can rupture, as in failure prone sleeve bearing designs, are controlled. Under normal or near-normal operating conditions, a periodic inspection (usually annually) of the Can-O-Matic II bearing system is suggested to insure trouble-free, reliable operation. In liquid handling systems where severe pressure or temperature transients may occur, or suction pressures vary dramatically resulting in "zero" NPSH available to a Can-O-Matic II pump, vibration monitors or proximity probes should be utilized to avoid catastrophic failure due to massive axial thrust forces.

To confirm the built-in reliability of the Can-O-Matic II design, each unit is thoroughly tested before shipment. Each pumping unit is given a hydrostatic test to insure the integrity of materials... a running test to insure performance capability ...and a halogen/mass spectrometer test to insure that the complete unit is hermetically sealed.

It is worth noting the fact that the Buffalo Can-O-Matic II is engineered and manufactured to ANSI Standards. This provides an efficient, low NPSH pump design that is field repairable and easily maintained.

The Buffalo Can-O-Matic II design has been field-proven in thousands of installations, and is backed by 30 years of experience in handling toxic, corrosive and carcinogenic liquids, refrigerants, high temperature water and heat transfer liquids.

Availability/Service

The Can-O-Matic II is a part of Buffalo's "PDQ" Quick Shipment Program, where most popular model pumps are available for short-term shipment. In addition, a complete pump repair service is provided for Buffalo Pump customers.

Engineering Assistance

Buffalo Pumps Sales Engineers throughout the United States have the engineering training and practical field experience necessary for the correct selection and application of Can-O-Matic II Pumps. In addition, they have the full support of Buffalo's home office research and engineering personnel. This in-depth engineering service is vital to assure proper pump application, especially when handling difficult and/or possibly hazardous fluids.

General Pump Specifications

Pump Sizes	13
Gpm	1500
Head	700'
Rpm	1750 & 3500

Working Pressure to 400 psi standard. Higher pressure 600 psi designs available.

Temperature range from −100°F to 250°F standard, −150°F to 400°F available.

Materials of Construction

PART NO.		DUCTILE IRON CAST IRON FITTED	DUCTILE IRON BRONZE FITTED	DUCTILE IRON 316SS FITTED	ALL 316SS
2	CASING	D.I.	D.I.	D.I.	316SS
3	IMPELLER	C.I.	BRZ	316SS	316SS
9	CASING RING		CARBON GRAPHITE		
10	MOTOR RING		CARBON GRAPHITE		
202	MOTOR FLANGE	316SS	316SS	316SS	316SS
205	BEARING HOUSING		316 STAINLESS STEEL		
208	WEAR RING HOUSING	316SS	316SS	316SS	316SS
247	BEARINGS		CARBON GRAPHITE		
211	JOURNALS		316SS HARD FACED TO 500 BRINELL		
216	STATOR CAN		HASTELLOY C		
221	ROTOR CAN		316 STAINLESS		
229	ROTOR SHAFT		316 STAINLESS		
250	SPRINGS		316 STAINLESS		
258	THRUST-O-MATIC ORIFICE		316 STAINLESS		

General Motor Specifications

- Totally enclosed liquid cooled design
- Hastelloy C Stator Can
- 316SS Rotor and Rotor Can
- Class C Insulation System
- Hardened bearing journals
- Thermal sensor
- Liquid tight conduit box with oil expansion dome and pressure relief valve set at 15 PSI
- Dielectric oil filled stator housing

Motor Horsepower Capability

Frame	RPM 1750	RPM 3500
66M	5 HP	7½ HP
66V	10	15
66ZF	20	25
215ZH	30	40
256ZM	50	60
256AB	75	100

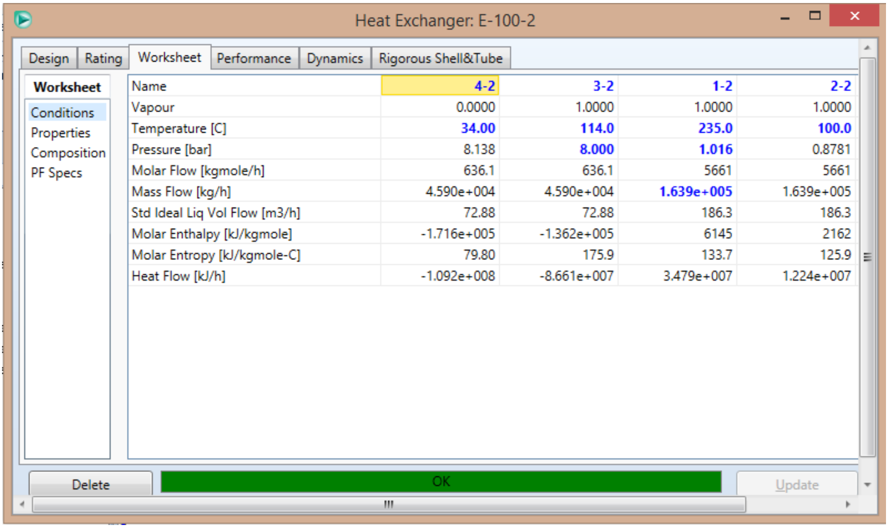
Attachment 7:
Refrigerant Selection for ORC System

In the ORC system R141b chosen as the working fluid in the system, this due to the boiling point and thermal efficiency of R141b appropriate when it is meet flue gas with 235^oC and when it is compared with another refrigerant, R141b produce more power than another refrigerant that compared (n-pentane, R11 and R114) with same operating condition.

And simulation carried out to compare power produce by another refrigerant with same operating condition with R141b. In attachment 4 from simulation R141b generating power as big as 661.8 kW, while in the simulation shown that n-pentane with 34^oC boiling point in atmospheric pressure, produced 608 kW with almost same operating condition, the different is the temperature output is 114^oC, while in the R141b is 110^oC. Then when using R11 as working fluid (R11 boiling point is 22.85^oC) produce 590 kw of power with the same operating condition in the turbine inlet. And R114 cannot be use for this ORC system operating condition due to the boiling point of the refrigerant is too low (4^oC), that will make the system not running, and the power cannot be generated using R114 as working fluid.

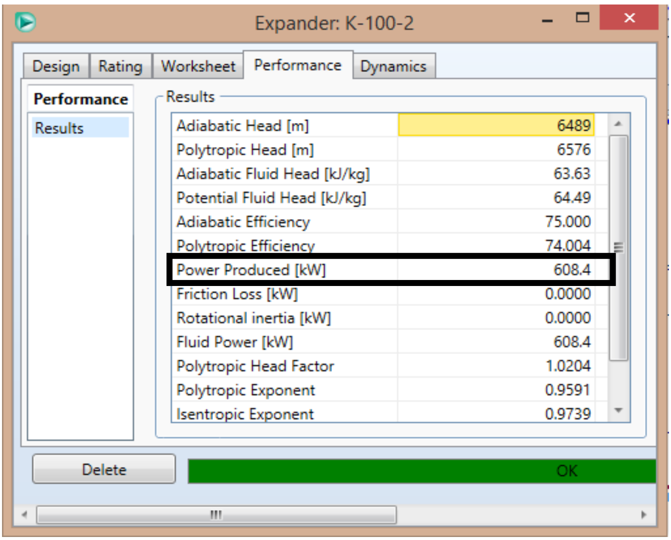
And these are the interface from the software about another power produced of n-pentane, R11 and R114:

n-pentane :



Name	4-2	3-2	1-2	2-2
Vapour	0.0000	1.0000	1.0000	1.0000
Temperature [C]	34.00	114.0	235.0	100.0
Pressure [bar]	8.138	8.000	1.016	0.8781
Molar Flow [kgmole/h]	636.1	636.1	5661	5661
Mass Flow [kg/h]	4.590e+004	4.590e+004	1.639e+005	1.639e+005
Std Ideal Liq Vol Flow [m3/h]	72.88	72.88	186.3	186.3
Molar Enthalpy [kJ/kgmole]	-1.716e+005	-1.362e+005	6145	2162
Molar Entropy [kJ/kgmole-C]	79.80	175.9	133.7	125.9
Heat Flow [kJ/h]	-1.092e+008	-8.661e+007	3.479e+007	1.224e+007

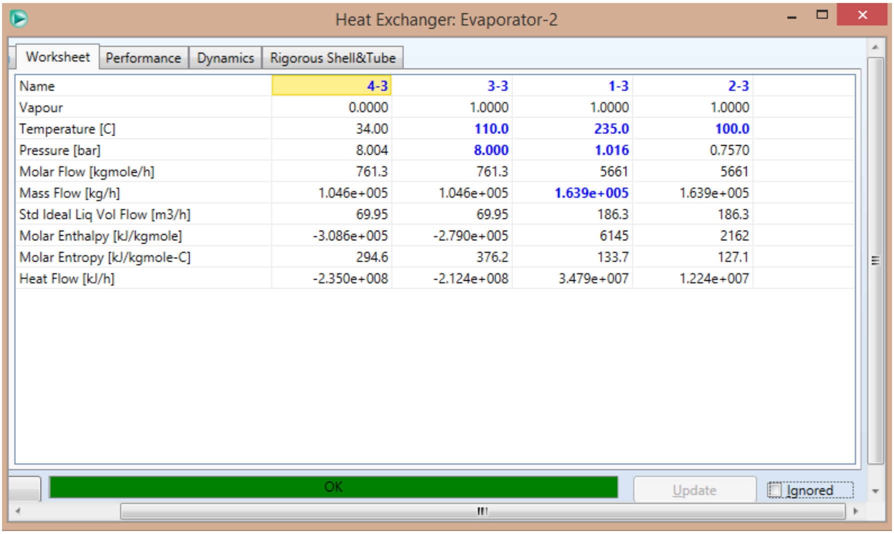
n- pentane Operating Condition in Evaporator



Adiabatic Head [m]	6489
Polytropic Head [m]	6576
Adiabatic Fluid Head [kJ/kg]	63.63
Potential Fluid Head [kJ/kg]	64.49
Adiabatic Efficiency	75.000
Polytropic Efficiency	74.004
Power Produced [kW]	608.4
Friction Loss [kW]	0.0000
Rotational inertia [kW]	0.0000
Fluid Power [kW]	608.4
Polytropic Head Factor	1.0204
Polytropic Exponent	0.9591
Isentropic Exponent	0.9739

Power Produced in The Turbine using n-pentane

R-11 :



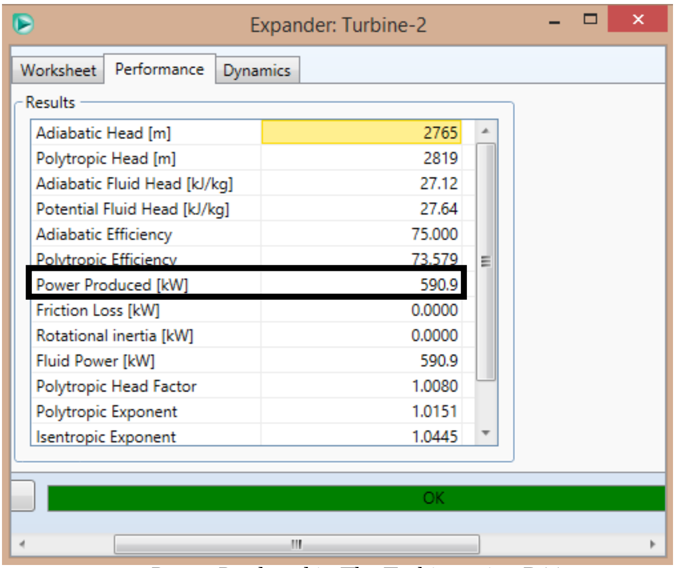
Heat Exchanger: Evaporator-2

Worksheet | Performance | Dynamics | Rigorous Shell&Tube

Name	4-3	3-3	1-3	2-3
Vapour	0.0000	1.0000	1.0000	1.0000
Temperature [C]	34.00	110.0	235.0	100.0
Pressure [bar]	8.004	8.000	1.016	0.7570
Molar Flow [kgmole/h]	761.3	761.3	5661	5661
Mass Flow [kg/h]	1.046e+005	1.046e+005	1.639e+005	1.639e+005
Std Ideal Liq Vol Flow [m3/h]	69.95	69.95	186.3	186.3
Molar Enthalpy [kJ/kgmole]	-3.086e+005	-2.790e+005	6145	2162
Molar Entropy [kJ/kgmole-C]	294.6	376.2	133.7	127.1
Heat Flow [kJ/h]	-2.350e+008	-2.124e+008	3.479e+007	1.224e+007

OK | Update | Ignored

R11 Operating Condition in Evaporator



Expander: Turbine-2

Worksheet | Performance | Dynamics

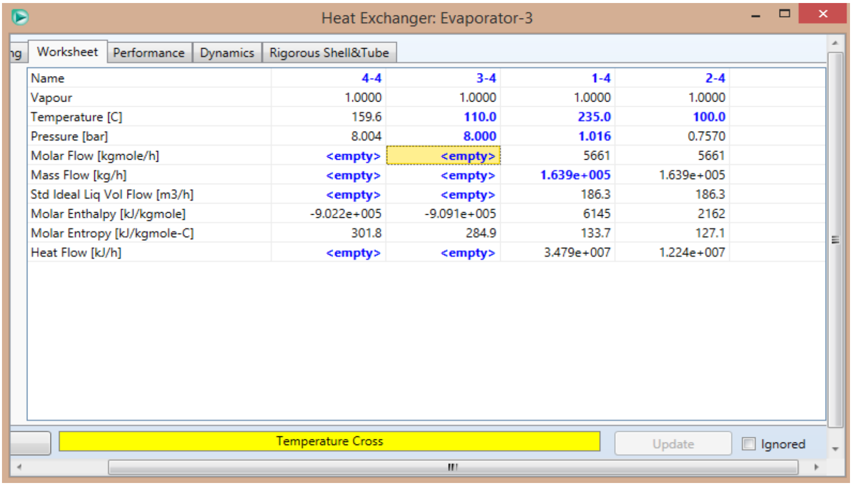
Results

Adiabatic Head [m]	2765
Polytropic Head [m]	2819
Adiabatic Fluid Head [kJ/kg]	27.12
Potential Fluid Head [kJ/kg]	27.64
Adiabatic Efficiency	75.000
Polytropic Efficiency	73.579
Power Produced [kW]	590.9
Friction Loss [kW]	0.0000
Rotational inertia [kW]	0.0000
Fluid Power [kW]	590.9
Polytropic Head Factor	1.0080
Polytropic Exponent	1.0151
Isentropic Exponent	1.0445

OK

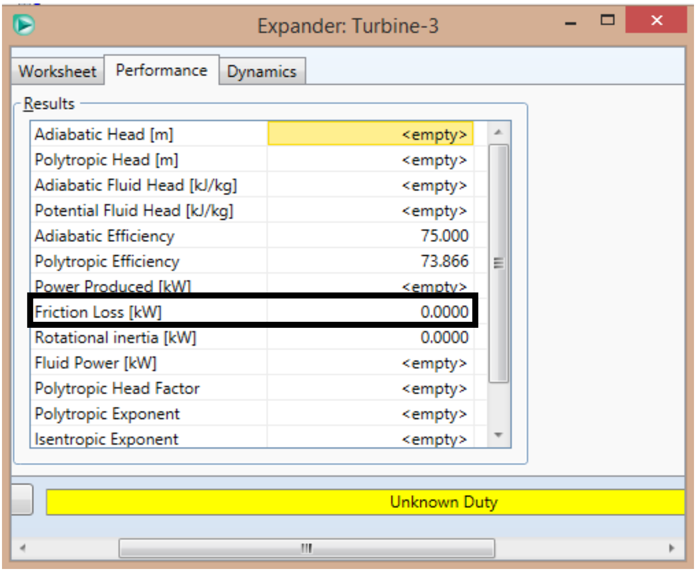
Power Produced in The Turbine using R11

R-114 :



Name	4-4	3-4	1-4	2-4
Vapour	1.0000	1.0000	1.0000	1.0000
Temperature [C]	159.6	110.0	235.0	100.0
Pressure [bar]	8.004	8.000	1.016	0.7570
Molar Flow [kgmole/h]	<empty>	<empty>	5661	5661
Mass Flow [kg/h]	<empty>	<empty>	1.639e+005	1.639e+005
Std Ideal Liq Vol Flow [m3/h]	<empty>	<empty>	186.3	186.3
Molar Enthalpy [kJ/kgmole]	-9.022e+005	-9.091e+005	6145	2162
Molar Entropy [kJ/kgmole-C]	301.8	284.9	133.7	127.1
Heat Flow [kJ/h]	<empty>	<empty>	3.479e+007	1.224e+007

R114 Operating Condition in Evaporator



Adiabatic Head [m]	<empty>
Polytropic Head [m]	<empty>
Adiabatic Fluid Head [kJ/kg]	<empty>
Potential Fluid Head [kJ/kg]	<empty>
Adiabatic Efficiency	75.000
Polytropic Efficiency	73.866
Power Produced [kW]	<empty>
Friction Loss [kW]	0.0000
Rotational inertia [kW]	0.0000
Fluid Power [kW]	<empty>
Polytropic Head Factor	<empty>
Polytropic Exponent	<empty>
Isentropic Exponent	<empty>

Power Produced in The Turbine using R114

CHAPTER 5
CONCLUSION AND SUGGESTION

This chapter will describe about conclusion from the data analysis and calculation. And also suggestion that can be considered to develop this topic for future research will be described as well.

5.1 Conclusion

1. In PT. SEMEN INDONESIA (PERSERO) Tbk, flue gas from preheater already used for raw grinding purpose, and flue gas in air quenching cooler used in cement grinding to ensure that the cement already fully dried, but there is still many flue gas from air quenching cooler go through stack due to the need in cement grinding not so high. So the flue gas from air quenching cooler that go through stack can be used for alternative power generation using ORC.
2. power produce in the turbine, higher pressure that can be achieve can make power produce will be higher too. In this case from the 8 bar and 110°C inlet to turbine and 1.5 bar and 64.06°C in outlet of turbine with the mass flow rate 22.25245 kg/s can produce 666 kW of power.
3. Shell and tube heat exchanger are the most used HE in the world and it is easier to maintain and used. But the shell and tube will spend large enough space if the requirement is high. The evaporator with the 6 MW requirement need 8.8 meter long and 1.084 meter of shell diameter, and for 5.5 MW requirement condenser need 9 meter long and 1.12 meter shell diameter. Due to it consume large place the choosing of shell and tube HE must be considered too.

5.2 Suggestion

1. Further investigation could be vary the ORC operating condition, when the pressure and temperature different is higher power produced for alternative power generation could be higher too.

2. Type of refrigerant can be compared to know different effect and power output in the cycle.
3. Refrigerant in the turbine should not be changing phase into saturated. It can damage the turbine blades, the refrigerant selection type should be dry refrigerant or isentropic refrigerant that wont changing phase in the expansion process in the turbine. And the turbine should suit with the ORC like Siemens SST 060 series.

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- **Motto: Your faith must be greater than your fear**