Investigation of Organic Rankine Cycle Using Different Working Fluids for Conversion of Low Grade Heat Sources

By

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Dissertation submitted on
Requirement for the Degree of Study (Hons)
(Mechanical Engineering)

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Perak

CERTIFICATION OF APPROVAL

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Approved,

____

(Dr. Aklilu Tesfamichael Baheta)

UNIVERSITI TEKNOLOGI PETRONAS
BANDAR SERI ISKANDAR, PERAK
January 2016

CERTIFICATION OF ORIGINALITY

This is to satisfy that I am responsible for the work submitted in this project, that the original work is my own except as specified in the reference and acknowledgement, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Khuganeswaran Jayakumar

ABSTRACT

This research paper entitled Investigation of Organic Rankine Cycle using different working fluids for conversion of low grade heat sources, is studied in Final Year Project I and Final Year Project II. This paper is reviewed on type of Rankine cycles that produces electrical power of heat sources, including conventional Rankine cycle and especially organic Rankine cycle. Water as the common working fluid for Rankine cycle when the heat input is from high grade heat sources, example coal and fuel. High molecular weight working fluids example, R22, R134a and R141b are ideal for Organic Rankine Cycle, this is because is boiling point of those working fluids are relatively very smaller compared to water. Besides that, ORC uses low grade heat sources like, solar and waste heat energy. The main purpose of this research project is to identify suitable organic working fluid which gives better work net, back work ratio and efficiency. First, working fluids that have boiling temperature above 220K is chosen to ease the evaluation on selecting best working fluids. Secondly, the working fluids are divided into three groups, isentropic, wet and dry working fluids. Furthermore, constraints of this project is introduced in Methodology and thermophysical properties is plotted as planned for Final Year Project I. as for the second part of Final Year Project, thermodynamic modelling of Organic Rankine Cycle is modelled in Engineering Equation Software. From the result, R290, the working fluid from isentropic group is chosen as the best working fluid in terms of Work Net giving a value of 54.61kJ/kg, efficiency giving a value of 15% and back work ratio, being the highest in isentropic fluid group giving a value of 42.75.

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CHAPTER 1

INTRODUCTION

1.1 Background

Demand of fossil fuels, as primary energy source is exceeding year by year exponentially due to modernisation and technology. Heavy industries, conventional buildings and residential areas' end usage of power is increasing, as this leads to a severe environmental impacts. According to figure below Malaysia stands at 30th among the list of countries that consumes a greater amount of energy each year by 112,000,000,000 kWh, according to data from Central Intelligence Agency (CIA) World FACT Book [1]. Figure 1.1 below may appeared to be convincing when compared to the highest energy consumer, China, by 4,831,000,000,000 kWh per year, but very vital steps need to be taken and research to reduce the fossil fuel consumption. Generation of electricity carried out from the primary energy sources fossil fuel through numerous ways, and one of it, the most common way of harvest electrical energy from thermal energy is vapour power cycle.

25	POLAND	137,500,000,000	2011 EST.
26	EGYPT	129,400,000,000	2011 EST.
27	SWEDEN	128,100,000,000	2011 EST.
28	NETHERLANDS	116,800,000,000	2013 EST.
29	ARGENTINA	114,200,000,000	2011 EST.
30	MALAYSIA	112,000,000,000	2012 EST.

Figure 1.1: Malaysia's standing in energy consumption among world countries

The vapour power cycles can be categorized into steam engines, which are mobile, and power plants which are stationary. Example of steam engines that use vapour power cycle are First steam-powered road vehicle by Joseph Cugnot at year 1770, Direct-steam powered Steam Locomotive Train at year 1942 and Indirect-steam powered train Shinkansen, Japan at year 2013 [2]. All these are vapour, also known as steam driven applications through vapour power cycle. Besides that, power plants which are basically stationary and produces electricity through vapour power cycle, example is Taichung Power Plant, the world's largest coal-fired power station which is also to be the highest CO_2 emitter in the world [3]. Thus, power generated using vapour power cycle is highly powered by non-renewable energy sources.

Looking into vapour power cycle, an ideal cycle for vapour power cycle is Rankine cycle. Rankine cycle is the main cycle that converts heat energy into electrical and is being used to drive all vapour power machines. With the rising issue of environmental impacts due to fossil fuels taken into consideration, studies on usage of low grade heat sources, also known as renewable energy sources such as geothermal, solar and biomass have widely carried out in higher energy consuming countries. Taiwan for instance, using Ocean Thermal Energy Conversion (OTEC) and solar energy are become inevitable option for them [4]. Vapour power cycle using low grade heat to produce heat energy and cooperated with organic working fluids is known as Organic Rankine Cycle. Conventional Rankine cycle uses water as its working fluid which is now being studies through ORC.

A medium of working fluid is highly needed to transfer the energy gained from combustion of fossil fuels or heat sources from renewable energy. Water is a conventional working fluid for Rankine cycle. But when it comes to Organic Rankine Cycle, higher molecular weight fluids which are known as organic working fluids like R134a, R152a and R22, is needed.

1.2 Problem statement

Water as the heat transfer medium in Rankine cycle, is chosen based on several factors. First, water is abundance in the earth and it is chemical inert. Water is not prone to attacking the pipes, turbines, pumps and equipment in general and is non-reactive. Water, boiling point of 100°C can easily being heated above the boiling point when it passes through boiler and achieve a vapour phase for turbine power generation using high grade heat sources[6]. The problem arises when the usage of low grade heat energy into the working fluids, because of the thermal energy input from low grade heat sources is not sufficient and efficient enough for water, to be heated above its boiling point. Thus to mitigate this problem organic working fluids are used because of their high molecular mass and boiling point which is much lower temperature than water-steam phase change. For instance, Refrigerant-11(R11), Refrigerant-12(R12) and Ammonia are type of organic working fluids [5]. Before justifying the usage of working fluids, selection of proper working fluids highly needed and it is the main objective. Working fluids has different thermophysical properties involves, thermal conductivity, specific heat, latent heat of vaporization and specific volume. Apart from this properties, mechanical parameters like work net, efficiency and back work ratio are also considered in this research project.

1.3 Objective

Replacement of water with high molecular mass working fluid, also known as organic working fluid, is the main approach being carried out in this project. Objective of this project is given below:

1. To investigate and compare the organic working fluids in terms of their performance.
2. To suggest on the best organic working fluid for Organic Rankine Cycle.

1.4 Scope of the study

The study on investigation of ORC using different working fluid for low grade heat sources is carried out via understanding and identifying type of working fluids, as the first step. As for this project, performance of ORC is studied based on solar and waste heat recovery having temperature range from 50˚C being the lowest temperature at condensing zone to 150˚C being the highest temperature in turbine inlet zone. Working fluids which have boiling point higher than 220K is chosen to ease the evaluation on selecting best working fluid. Usage of organic working fluid like R-290, R-141b, R-123 and R-134a known as isentropic fluids, R-22, R-143a, R-152a and R-125 known as wet fluids and R-600, R-600a, R-113 and R-114 known as dry fluids, is taken into the study. The thermodynamic modelling for this research project is carried out in Engineering Equation Solver (EES). Therefore, issues regarding material selections, component configuration and heat transfer performances of the evaporator and condenser are not considered in this study.

CHAPTER 2

LITERATURE REVIEW

2.1 Low Grade Heat Sources

Primary energy sources, like fossil fuels, being the highest demand now, is needed to generate electrical and heat energy for human activities. Approach of low grade heat sources, like, geothermal, solar, and biomass are explained and its usage in ORC are significant.

Apart from this, in small biomass plants, the ORC is the ideal choice due to its high efficiency, availability and ability to follow load dependent on fuel supply [14, 15]. Since it is a small scale system, it is suitable for small scale demand side, and even it is commercially available at medium 100-1500kW output energy [16]. This technology is rapidly increasing due to its low carbon footprint and cost effective.

Besides that, solar energy is one of the vital sources used for ORC system. The primary of all energy, sun's energy, is extracted through solar system and has been common in nowadays world. Development of solar powered ORC, has become significant and many modification has been done. Low grade heat energy of below 300˚C, is viable through ORC for generation of electrical energy [16].

Furthermore, recovery heat, waste heat is the best method of controlling excessive non-renewable energy usages. The demand for waste heat recovery is continuously growing under the rising commitment of the industry to reduce energy consumption, operational costs and carbon emissions [14]. Stated that usage of waste heat recovery ORC has return of investment (ROI) 2 years less than conventional one [13]. Development of waste recovery ORC that gives higher output power 2MW using medium temperature level 300˚C is manufactured by Siemens and is commercially used [10].

2.2 Organic Rankine Cycle

Mitigation of the previous problem in Carnot cycle, comes the Rankine cycle, as shown in Figure 2.1 below, its schematic diagram. An Organic Rankine Cycle (ORC), simply means original Rankine cycle where it working fluids are high molecular mass organic working fluids, such as isobutene, n-pentene and aromatic hydrocarbons [8]. The Rankine cycle was invented by W. Rankine in the 1850's and is widely used for boiler steam power generation systems [11]. The Figure 2.1, explains water enter pump at state 1 and is compressed isentropically (process 1-2) to the operating pressure of the boiler, then being heated by boiler (process 2-3) and outputted as superheated vapour at state 3. Condenser exit temperature, turbine exit quality, system efficiency, turbine inlet temperature and pressure are among those parameter that needed to be taken into consideration.

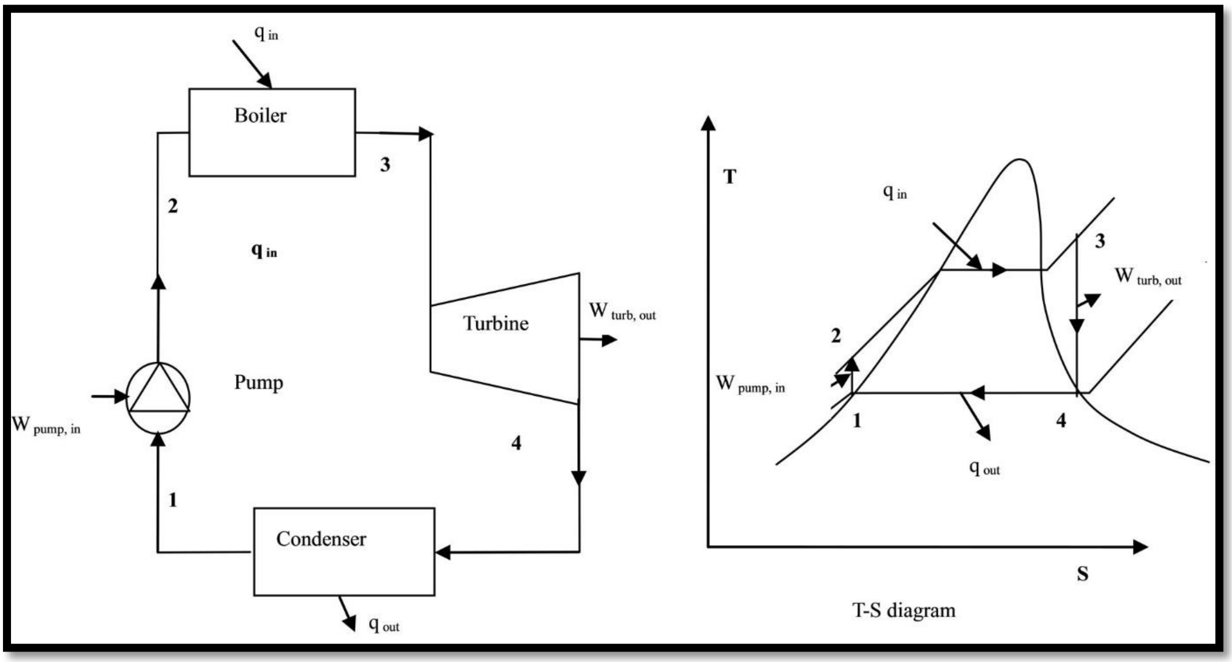

Figure 2.1: Schematic diagram and T-s diagram for Rankine Cycle

2.3 Working Fluids

A working fluid must provide an adequate chemical stability in the desired temperature range. Working fluids in and thermodynamic system involves only three, dry, isentropic and wet fluids depending on the slope of saturation vapour curve on a T-s diagram, shown in Figure 2.2. As the gradient of the slope shows positive, simply means the working fluid is dry, example is pentene. Gradient of the slope that gives approximate zero which is vertical gradient is basically isentropic fluid like R-141b and for negative gradient it is wet fluids like water [8]. Studies shows that with the usage benzene as the working fluid over a medium temperature range, 37.5kW of electrical power is generated [13].

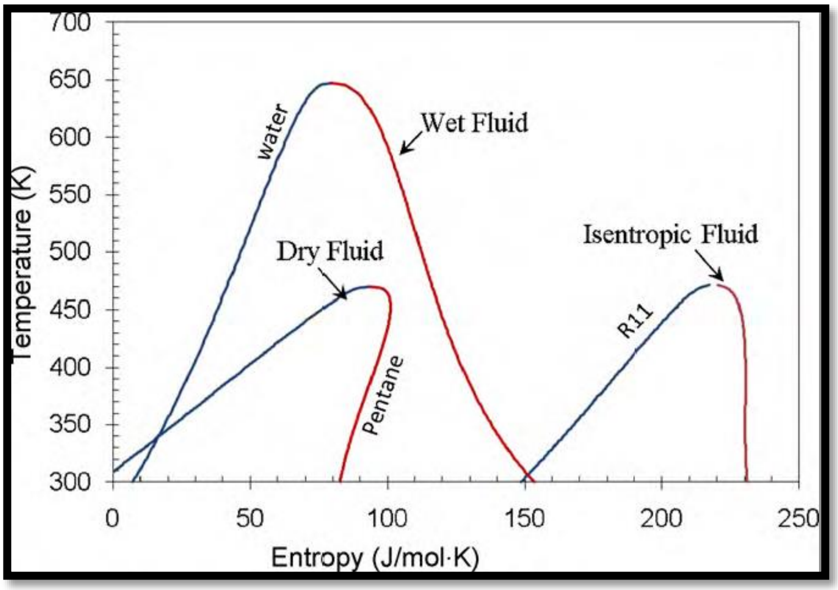

Figure 2.2: T-s diagram for Wet, Isentropic and Dry fluids

Most of the organic fluids have a lower boiling point compared to water as mentioned earlier in Problem Statement. This property makes organic fluid need a lower heat source temperature then water to evaporate for turbine inlet. Figure 2.3 below shows the T-s diagram for water and some organic working fluids for ORC. The positive and infinite slopes have enormous advantages for turbo machinery expanders. These working fluids leave the expander as superheated vapour and eliminate the corrosion risk in case of using turbo machinery expanders. Furthermore, there is no need for overheating the vapour before entering the expander, and a smaller and cheaper heat exchanger can be used [9].

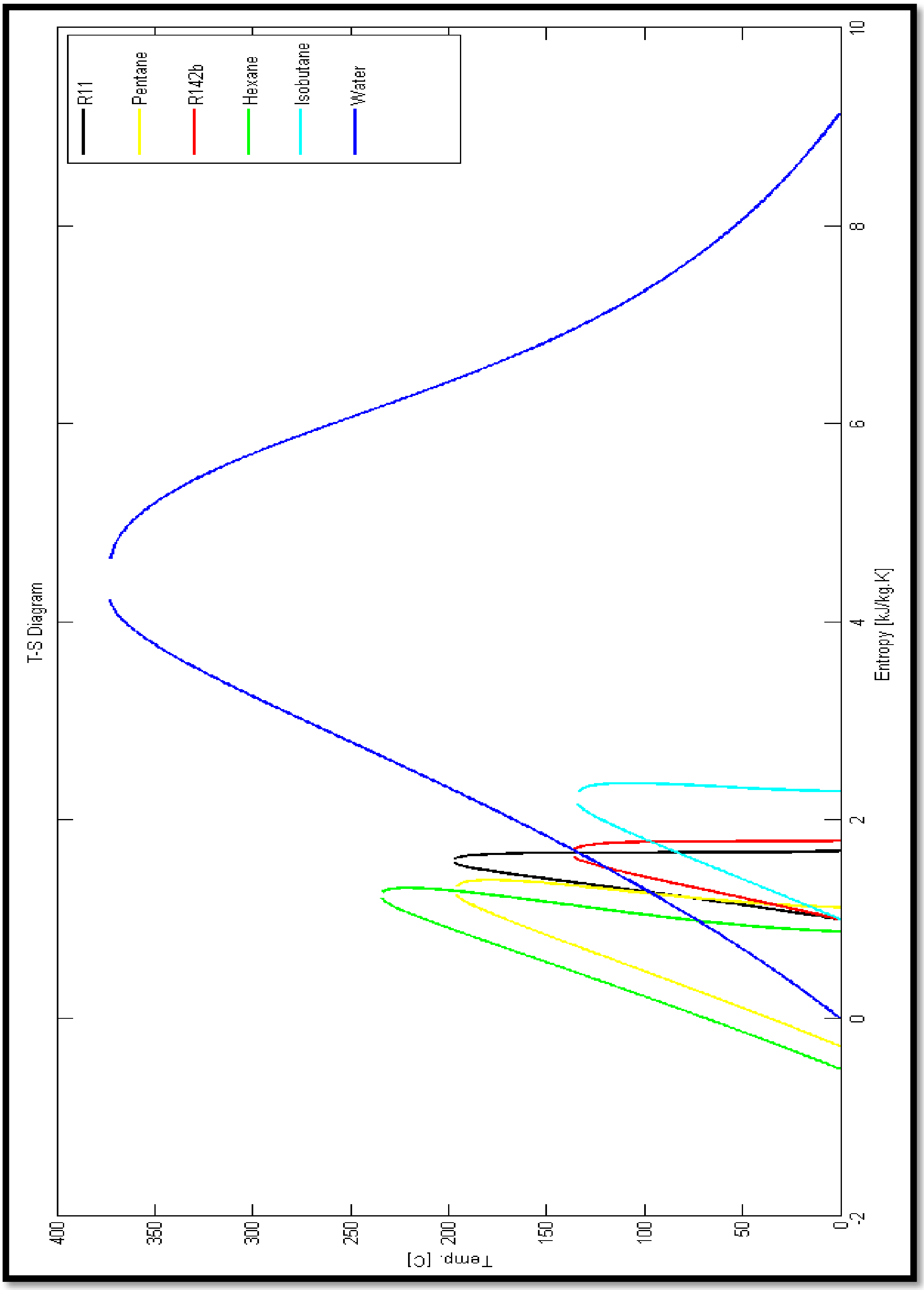

Figure 2.3: T-s diagram for water and some organic working fluids for ORC

Selection of working fluids is dependant of several factors, one of it is toxicity of working fluid, where low toxicity level fluid need to be chosen. In terms of chemical stability of the fluid and compatibility with materials in contact, unlike water, organic working fluids undergo chemical deterioration and decomposition at higher temperature, thus for the stability concern, its maximum operating temperature must be limited. Plus, boiling temperature of the fluids is also important, given very low boiling temperature requires a suitable condenser as it requires lower condenser temperature and high boiling temperature requires high heat input from boiler. Besides that, flash point criteria is vital, as higher flash point should be selected to avoid flammability. Apart from that, lower specific heat and higher latent heat should be selected for low load for the condenser and to raise efficiency of heat recovery. Ozone depletion potential (ODP), global warming potential (GWP) and atmospheric lifetime (ALT) factors, needed controlled at safe zone for environmental aspects of working fluids[8].

Table 2.1: shows the thermophysical data of isentropic working fluids.

Isentropic Working Fluids				
Working Fluid	R123	R141b	R-290	R-134a
Critical Point Temperature (K)	456.8	477.4	369.8	374.25
Critical Point Pressure (Mpa)	3.668	4.249	4.247	4.059
Triple Point Temperature (K)	166	169.9	85.48	169.85
Boiling Temprature (K) @ 1atm	300.9	305.2	231.1	246.79
Latent Heat of Vapourization, hfg (kJ/kg) @ 1atm	170.6	223.5	425.8	217

Table 2.2: shows the thermophysical data of wet working fluids.

Wet Working Fluids				
Working Fluid	R-152a	R-143a	R-22	R-125
Critical Point Temperature (K)	386.4	346.15	369.3	339.17
Critical Point Pressure (Mpa)	4.52	3.761	4.989	3.618
Triple Point Temperature (K)	154.6	161.34	115.7	172.55
Boiling Temprature (K) @ 1atm	249.1	225.55	232.3	224.79
Latent Heat of Vapourization, hfg (kJ/kg) @ 1atm	329.4	226.6	233.9	164.2

Table 2.3: shows the thermophysical data of dry working fluids.

Dry Working Fluids				
Working Fluid	R-600	R-600a	R-113	R-114
Critical Point Temperature (K)	425.1	408.13	487.3	418.9
Critical Point Pressure (Mpa)	3.796	3.64	3.439	3.289
Triple Point Temperature (K)	134.9	-	-	179
Boiling Temperature (K) @ 1atm	272.6	261.45	320.8	276.9
Latent Heat of Vapourization, hfg (kJ/kg) @ 1atm	385	364.8	144.5	136.1

Thermophysical properties of dry, wet and isentropic refrigerants, as per listed in the scope of this project, is tabulated in the Table 2.1, 2.2 and 2.3 above. The tabulated data is imported into Final Year Project II to ease the study on selection of working fluids suitable for ORC operation. Besides, Engineering Equation Solver software, and sources from internet were used to generate the thermophysical data for various refrigerants, according to type [17-19].

CHAPTER 3

METHODOLOGY

3.1 Mathematical Analysis

Mathematical approach to calculate the efficiency of the ORC system is vital. Work input in pump and output in turbine is important to determine the back work ratio. Figure 2.2 is chosen as the reference assuming ideal cycle. Below is the formula to find work done in the pump:

$$w_{12} = (\, p_2 + \; p_2 \,)v_1 \tag{1}$$

$$(\; v_1 \text{ and } v_2 \text{ is same, as the State 2 is liquid, assuming isentropic})$$

$$h_2 = \; h_1 + \; w_{12}, \tag{2}$$

$$h_3 = \text{properties at Temperature 3 and Pressure 3}$$

Heat transfer in the boiler of the ORC, is calculated using formula below:

$$q_{23} = (h_3 - \; h_2) \tag{3}$$

$$h_4 = \text{properties Pressure 4 and Temperature 4, assuming isentropic} \tag{4}$$

The essential parameter to equate the efficiency is where, the work done by turbine need to be calculated. The overall efficiency of the system is given as well:

$$w_{34} = h_3 - h_4 \tag{4}$$

Overall Efficiency : $\eta_{overall} = \frac{w_{34} - w_{12}}{q_{23}}$ (5)

Back work ratio parameter is also considered to be an evaluation of the ORC performance and the formula as follows:

Back work ratio, BR = $\frac{w_{34}}{w_{12}}$ (6)

In addition to the mathematical approach of the system, a thermodynamic modelling via a proper series of equation is important, involving actual and ideal Organic Rankine cycle. Before that, modelling of such thermodynamic system involves certain constraint that needed to be fixed. Below are the constraints to be fixed:

1. Selection of working fluids which has boiling point above 220K, the reason is to have a clear and efficient way of comparing its performance on ORC system [20].
2. The condensing temperature is fixed at 50˚C, also known as the lowest temperature T_L of the system. The ambient temperature of Malaysia is 30 ˚C to 37 ˚C, thus condensing temperature is chosen above the ambient temperature for heat transfer purpose. The system temperature range between boiler exit and turbine exit can be from 50˚C or lower to 150˚C or higher for waste heat and solar energy usage. Generally the maximum pressure range that a boiler operate is between 0.2MPa to 2.0MPa. The scope is smaller than the boiling temperature above, thus consideration of boiling pressure is vital. Thus, the highest pressure of 1.5MPa and lowest of 0.6MPa is fixed in this operation.
3. Efficiency of turbine is set to 0.8. The reason is to find out the actual enthalpy, h_3 entropy at the turbine exit compares with the ideal one, given h_{3s} and s_{3s}. the formula for efficiency of turbine is [20]:

$$\eta_{turbine} = \frac{h_2 - h_3}{h_2 - h_{3s}} \tag{7}$$

4. The efficient of the pump is set to 0.8 as well, which is to find out the actual enthalpy, h_1 and entropy at pump exit given h_{1s} and s_{1s}.

$$\eta_{pump} = \frac{h_{4s} - h_1}{h_4 - h_1} \tag{8}$$

5. The energy and work calculation throughout the system is based on specific energy and work done denoted as kJ/kg, thus mass flow rate is ignored in the modelling.

3.2 Thermodynamic modelling of ORC in Engineering Equation Solver (EES)

Based on the equation on ideal ORC and actual ORC cycle from the constraints above, the series of equation is being written in the Engineering Equation Solver (EES) software. The series of equation is used for all the 12 working fluids from three type of group to generate the result on various parameters. Below are the equation code and its comment about the variables being used:

{Constraints"}

{Turbine inlet/Boilet Outlet}

T_2 = 150 [C] "Temperature inlet to turbine/ boiler exit"

P_2=1.5[MPa] "Pressure at turbine inlet"

{Quality is superheated}

eff_turbine = 0.8 "Efficiency of turbine fixed"

{Condenser inlet/Turbine Outlet}

P_3=0.6 [MPa] "Condensing Pressure"

{Pump Inlet/Condenser Outlet}

T_4 = 50 [C] "Condensing Temperature"

P_4=P_3 "Pressure on turbine exit is same throughout the condensing process"

eff_pump = 0.8 "Efficiency of pump"

{Boiler inlet/Pump outlet}

P_1= P_2 "Pressure on pump exit is same as boiler exit"

"Organic Rankine Cycle modelling"

{Working Fluids}

```
F$='Working Fluids Name'

h_4=ENTHALPY(F$, T=T_4, x=0)            "Enthalpy at pump inlet"
s_4=ENTROPY(F$, T=T_4, P=P_4)           "Entropy at pump inlet"
s_1=ENTROPY(F$,T=T_4,x=0)               "Entropy at boiler inlet"
h_1s=ENTHALPY(F$,s=s_1,P=P_1)           "Isentropic enthalpy at boilet inlet"
h_1=((h_1s-h_4)/eff_pump)+h_4           "Actual Enthaply at boilet inlet"
h_2=ENTHALPY(F$, T=T_2, P=P_2)          "Enthalpy at turbine inlet"

s_2=ENTROPY(F$,T=T_2,P=P_2)             "Entropy at turbine inlet"
h_3s=ENTHALPY(F$,s=s_2,P=P_3)           "Isentropic enthalpy at condenser inlet"
X_3_quality = QUALITY(F$,h=h_3,P=P_3)"Quality of steam at condenser inlet"
X_4_quality = QUALITY(F$,h=h_4,P=P_4)"Quality of steam at condenser outlet"
h_3=h_2 - ((h_2 - h_3s)*eff_turbine)    "Actual enthalpy at turbine exit"

T_3=TEMPERATURE(F$,h=h_3,P=P_3)         "Temperature at condenser inlet"
```

{Results}

```
W_turbine=h_2 - h_3                     "Work of turbine"
W_pump= h_1-h_4                         "Work of pump"

W_net=W_turbine-W_pump                  "Work net output"
eff_overall=W_net/(h_2-h_1)             "Efficiency of the ORC system"
Back_work_ratio=W_turbine/W_pump        "Back Work Ratio"
```

3.3 Flowchart of Research Project

The main objective of this project is to research and investigate on the proper working fluids that are suitable for higher ORC efficiency. The vital approach of this project is made through steps of research planning and it is shown in Figure 3.1 below, the detailed flowchart of the project overall for Final Year Project 1 and Final Year Project 2.

Phase I

During this Phase I, a detailed study on the literature review of this research project is carried out. Journals, research papers and internet sources has been collected and reviewed.

Phase II

During this Phase II of this research project, low grade heat sources information and numerical data is collected starting solar and waste heat. Working fluids, all three types, dry, wet and isentropic as specified in the Scope of the Project is taken and thermophysical properties is tabulated and plotted in Result. The Phase II is this project is completed during Final Year Project II.

Phase III

The last phase III, now, is involved all the stimulation and modelling of ORC using Engineering Equation Solver. As for the last part, after getting information from the phase II process, selection of suitable working fluids for each low grade heat sources is done. Design of ORC system for low grade heat sources is carried out in Engineering Equation Solver (EES) software. Evaluation and compilation of results and information for final report is be carried out during the end of this phase III, and the research work is finished.

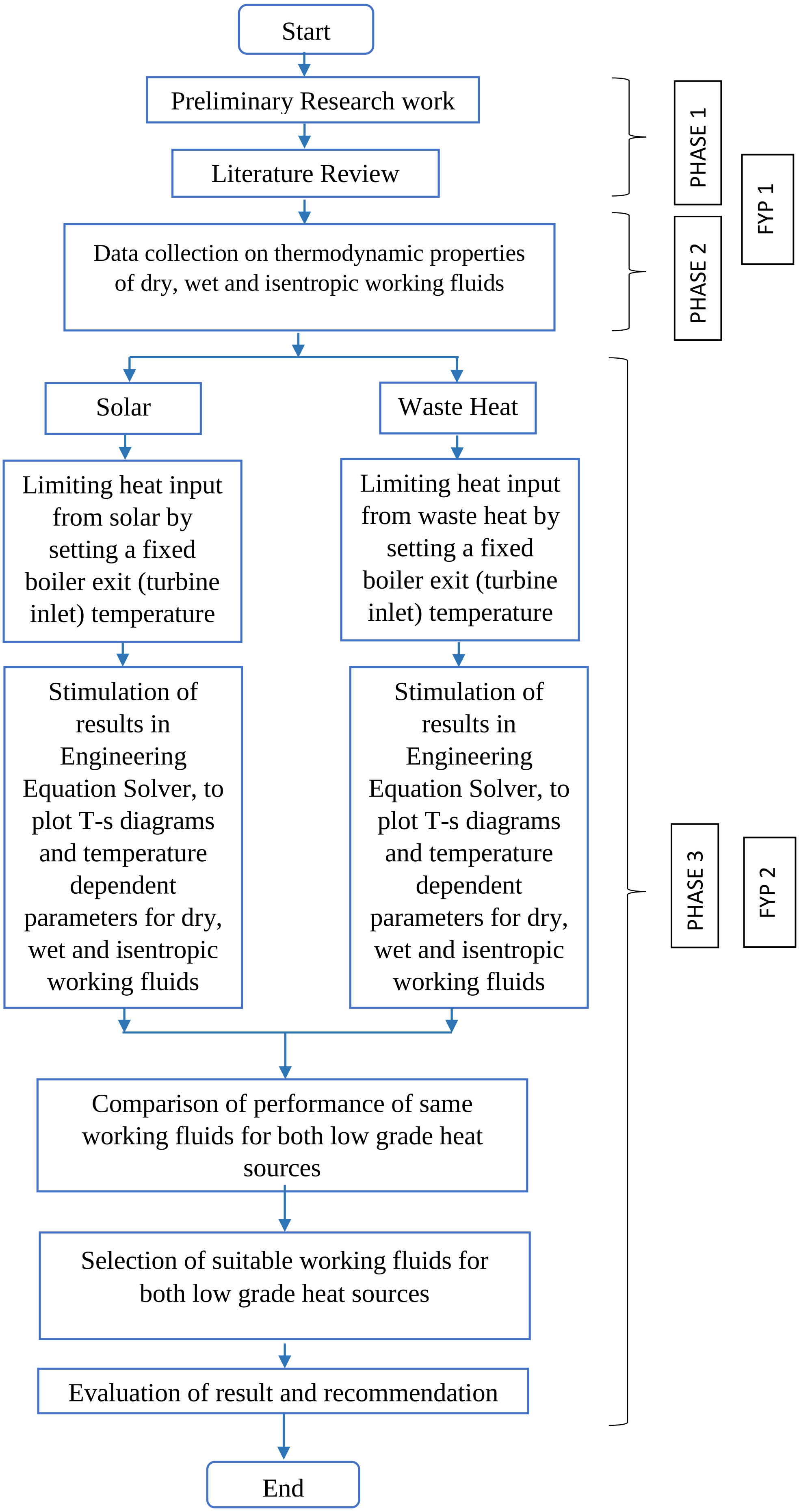

Figure 3.1: Flowchart of Research Project

3.4 Gantt Chart and milestones

According to the updated time flow, extended proposal is targeted to complete within its due week 7. Furthermore, the last milestone of FYP1 which is the Final Interim Report is completed by the end of week 13. Table 3.1 and Table 3.2 shows the Gantt Chart of this research project.

Table 3.1: Gantt Chart of this research project for FYP1

No	Project Activity	FINAL YEAR PROJECT 1													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Preliminary Research Work	■	■	■	■	■	■	■	■						
2	Proposal Defence									■	■				
3	Phase I											•			
4	Interim Report												■	■	
5	Phase II														•
6	Phase III starts														■

■	Timeline for each task (Weekly)
•	Milestones of this research project

Table 3.2: Gantt Chart of this project for FYP2

No	Project Activity	FINAL YEAR PROJECT 2													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Stimulation of ORC performance in EES	■	■	■	■										
2	Design of ORC for Solar and Waste heat					■	■	■	■	■	■				
3	Evaluation of result											■	■		
4	Phase III													•	
5	Final Report													■	■

■	Timeline for each task (Weekly)
•	Milestones of this research project

CHAPTER 4

RESULTS

4.1 Temperature dependant Thermophysical Properties

Temperature dependant Thermophysical properties were plotted using EES software, with respect to the temperature of 50℃, lowest temperature and 150℃, highest temperature of the system. The data was plotted through 5 iterations between the temperature and pressure from 0.6MPa to 1.5MPa. Figure 4.1a, 4.1b, 4.1c and 4.1d shows the thermophysical properties of isentropic fluids. Those figures shows the plots on specific heat capacity, thermal conductivity, specific volume and latent heat at turbine outlet temperature range between 25˚C to 150˚C. Based on the Figure 4.1a below the R290 and R134a shows a linear increase in thermal conductivity with temperature whereas R141b and R123 exhibits decreasing value with temperature.

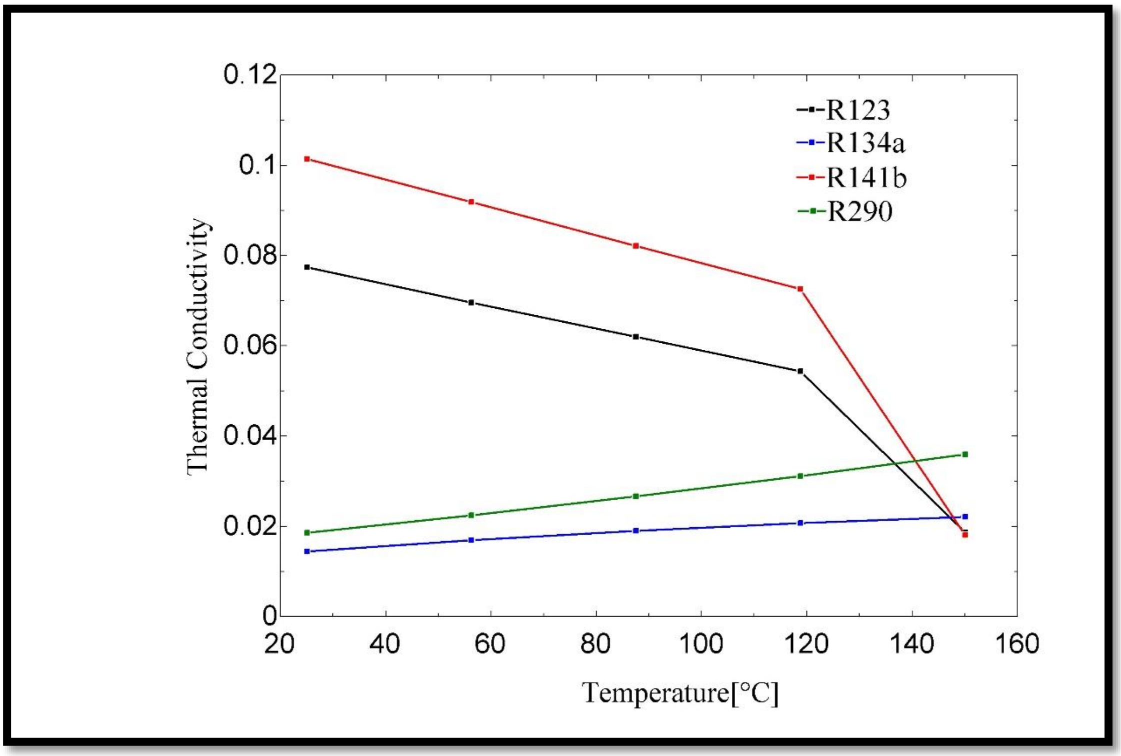

Figure 4.1a: Thermal Conductivity of Isentropic Fluid

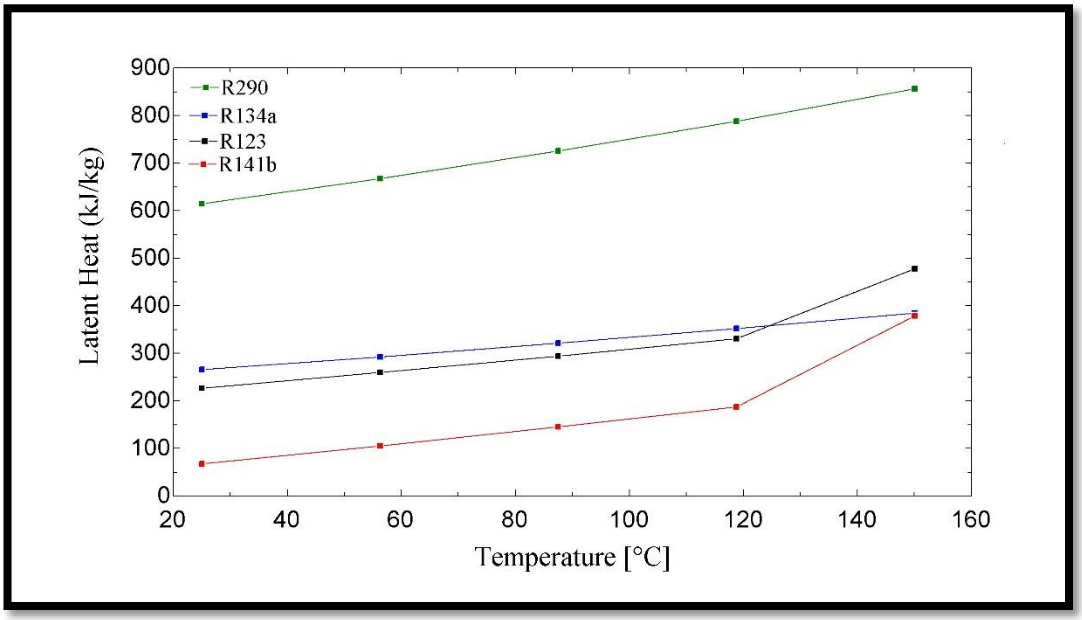

Figure 4.1b: Latent Heat of vaporization for Isentropic Fluid

It shows that, in the Figure 4.1b above, R290 has the highest latent heat of vaporization meaning that the fluids need higher heat input for a phase change. A greater latent heat of vaporization is better for higher work done on turbine.

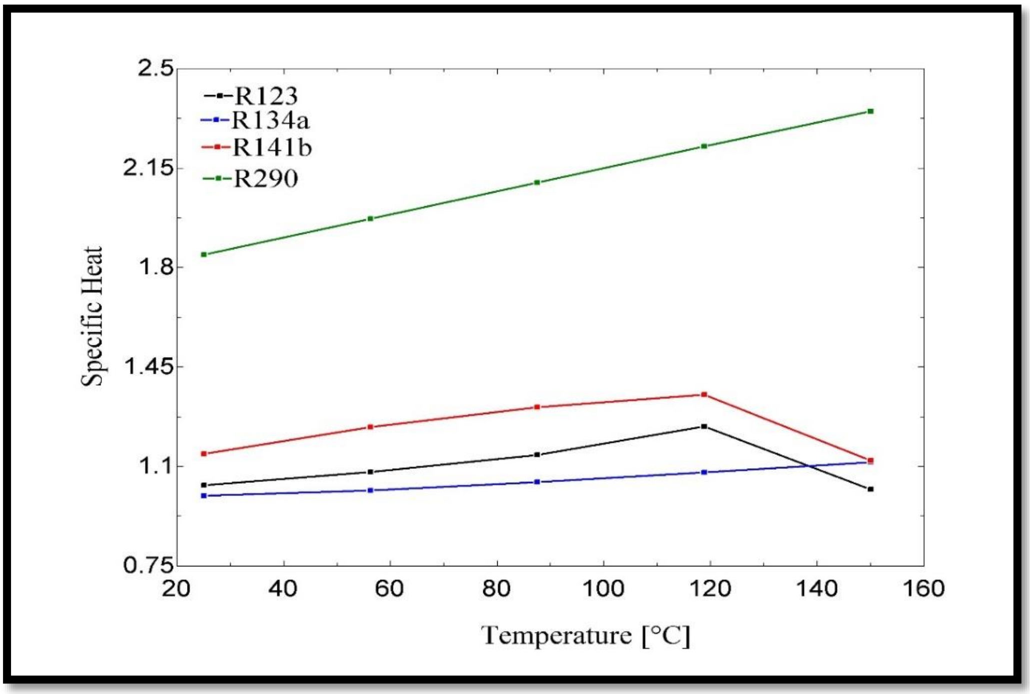

Figure 4.1c: Specific Heat of Isentropic Fluid

From the Figure 4.1c, R290 fluids is also having the highest specific heat which is the ability of storing heat per kg os substance. This favours the work out of the system, as higher specifc heat gives higher work of turbine theoratically.

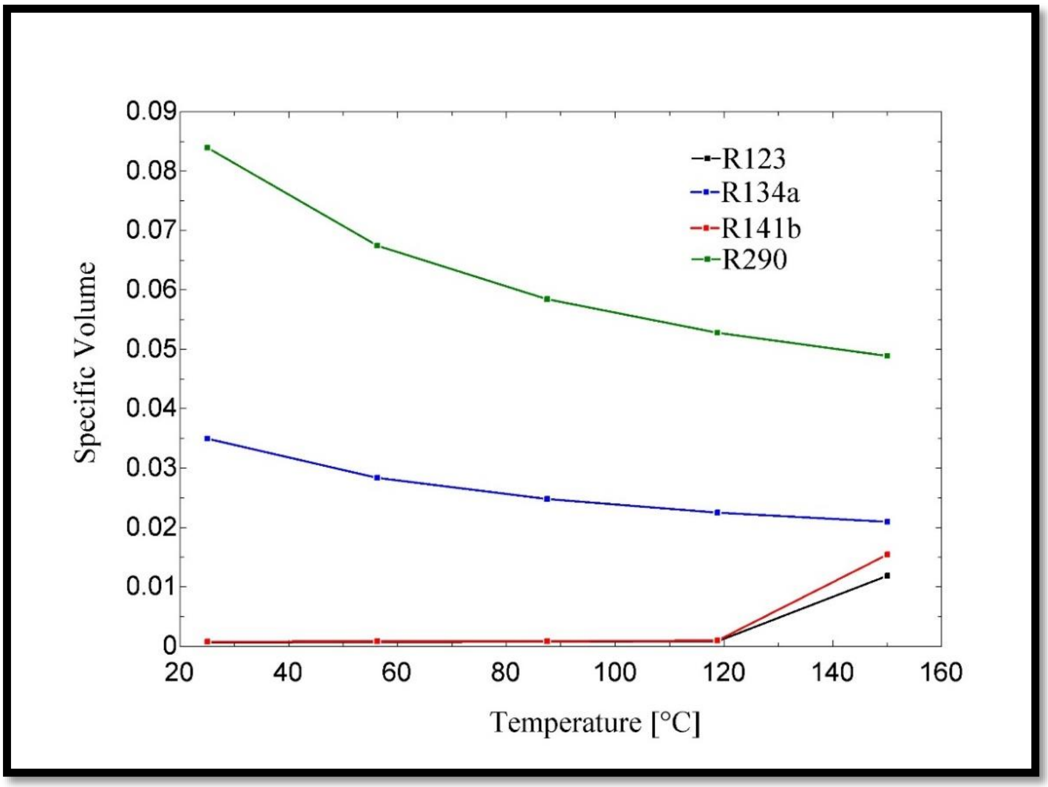

Figure 4.1d: Specific Volume of Isentropic Fluid

The Figure 4.1d, reveals that R290 has highest specific volume giving higher work of pump due to larger volume of fluids to be pump into boiler. This also gives impact of Back Work Ratio parameter, making it lower. Beside the isentropic fluids, Figure 4.2a, 4.2b, 4.2c and 4.2d shows the thermophysical properties of wet fluids. Those figures shows the plots on specific heat capacity, thermal conductivity, specific volume and latent heat at turbine outlet temperature range between 25˚C to 150˚C. Based on the Figure 4.2a below, R152a shows higher specific heat and R22 shows the lowest among wet fluid groups. From the Figure 4.2b below, R152a exhibits the highest specific volume after the temperature 60˚C, and R125 shows the lowest among all the fluids that giving it chances of lower work of pump.

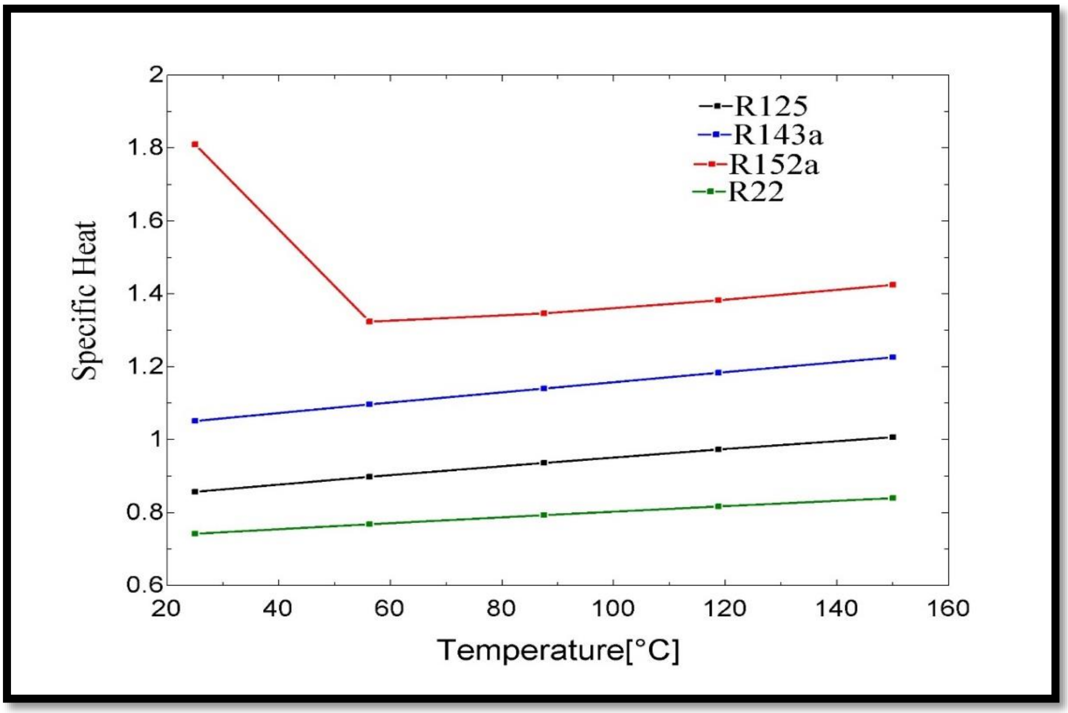

Figure 4.2a: Specific Heat of Wet Fluids

From the Figure 4.2c below shows, R152a fluids has the highest thermal conductivity among all the fluids in that group. This gives a positive impact on work net of the system.

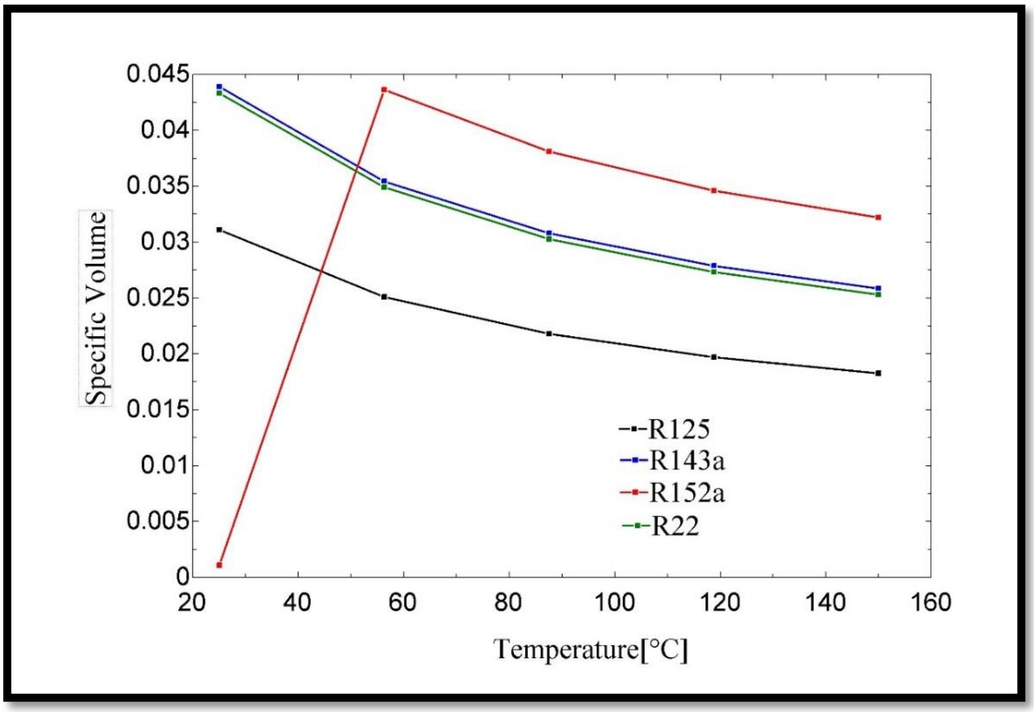

Figure 4.2b: Specific Volume of Wet Fluids

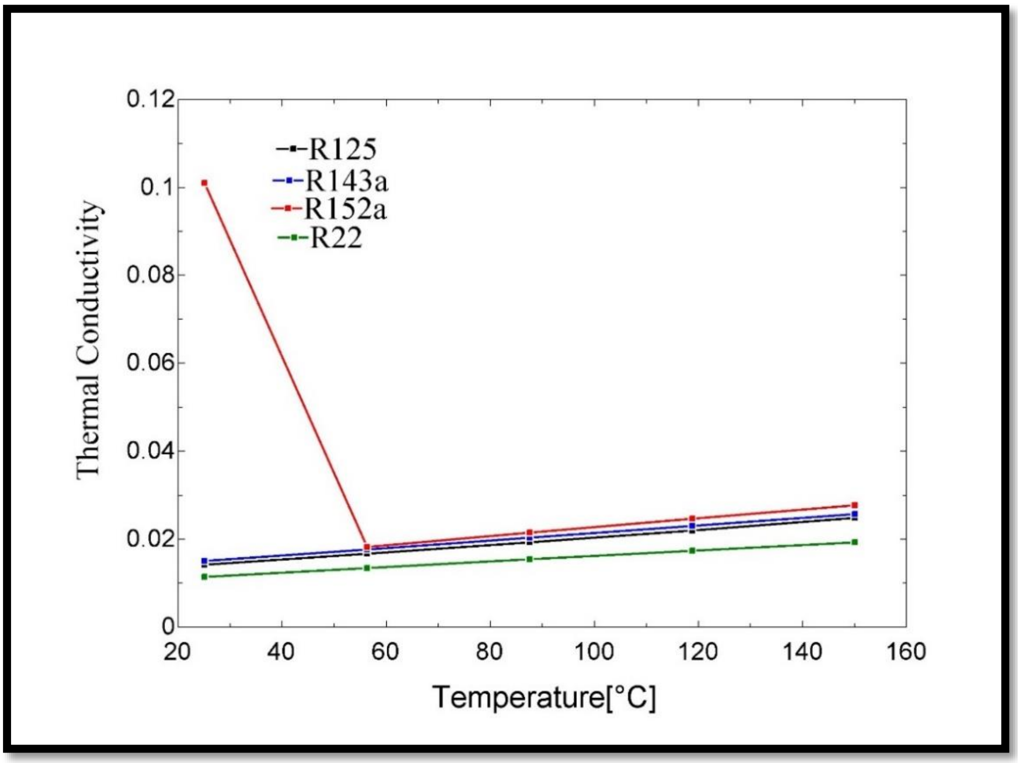

Figure 4.2c: Thermal Conductivity of Wet Fluids

The Figure 4.2d exhibits on the latent heat of vaporization, resulting R152a giving the highest latent heat that also favours on the work out from the turbine.

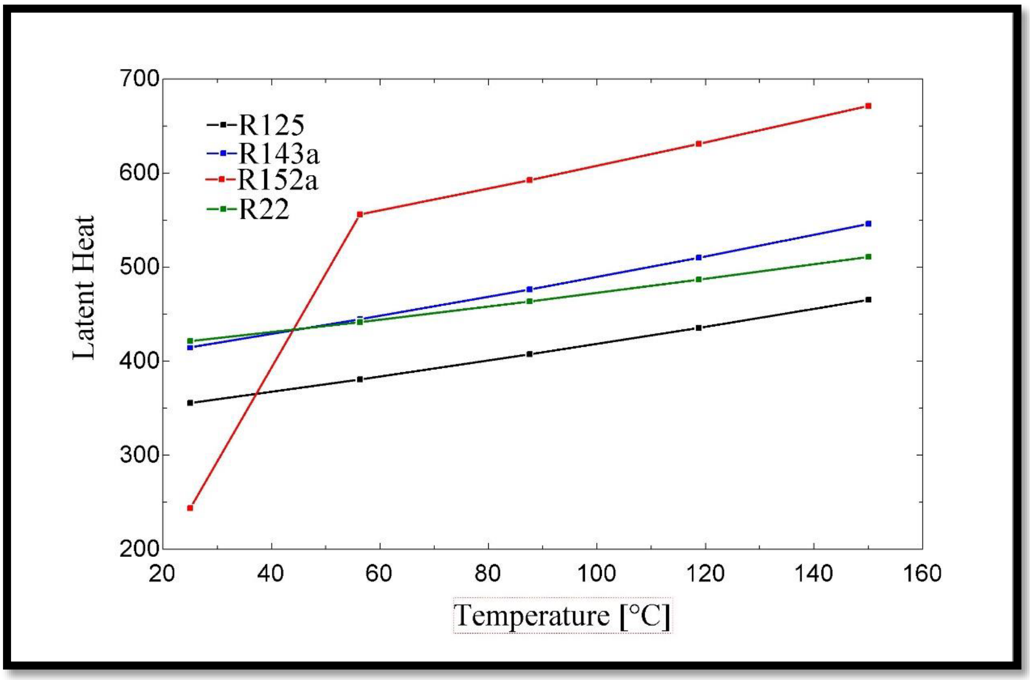

Figure 4.2d: Latent Heat of Vaporization for Wet Fluids

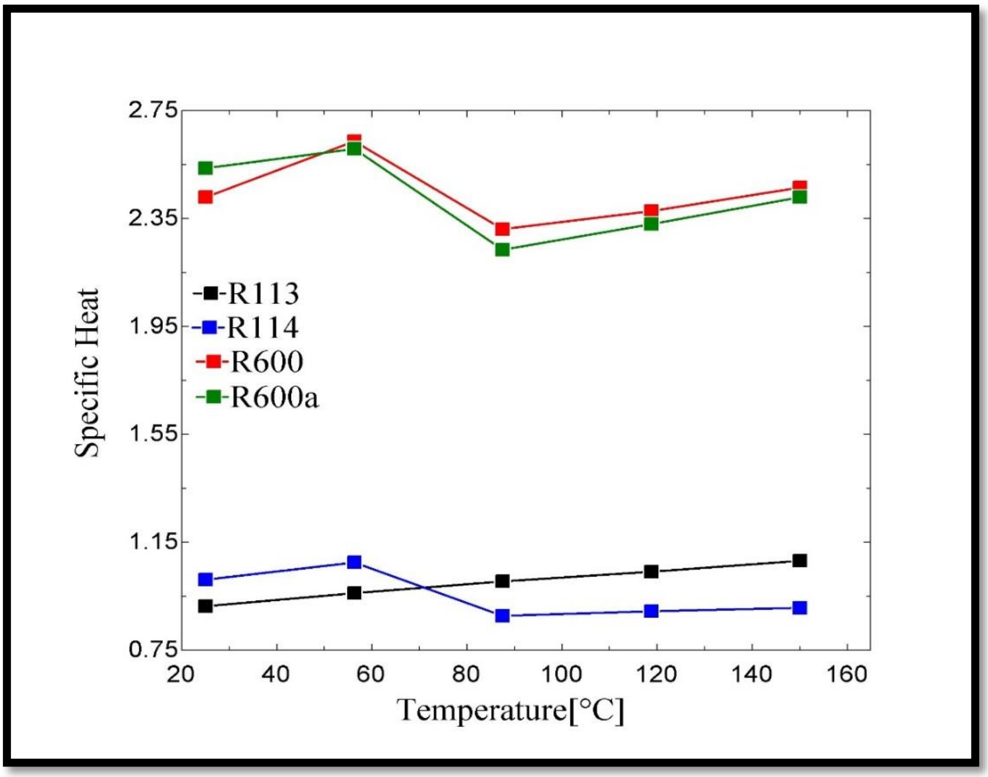

Figure 4.3a: Specific Heat of Dry Fluids

Figure 4.3a, 4.3b, 4.3c and 4.3d shows the thermophysical properties of dry fluids. Those figures shows the plots on specific heat capacity, thermal conductivity, specific volume and latent heat at turbine outlet temperature range 25˚C to 150˚C.

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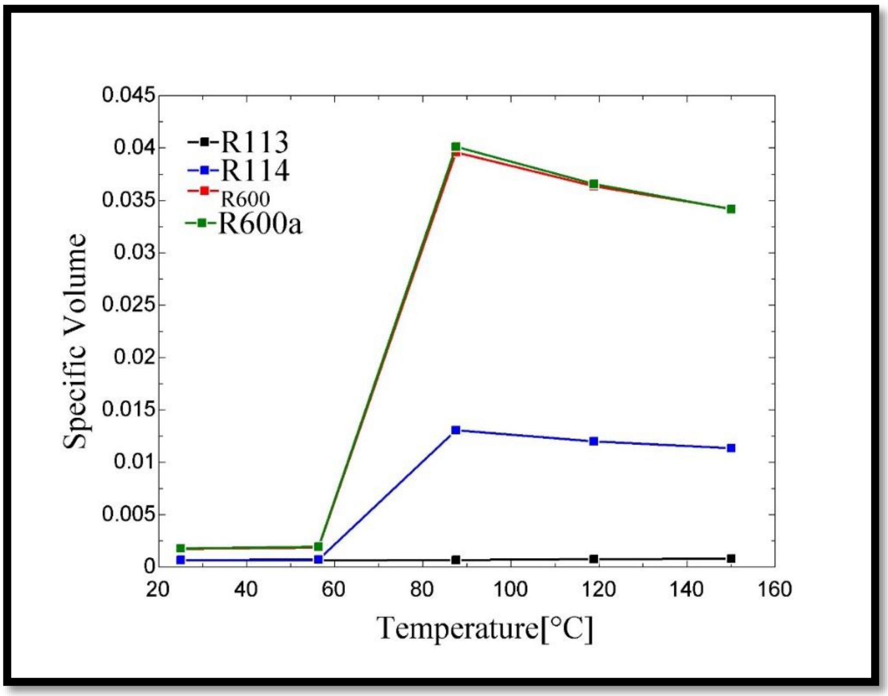

Figure 4.3b: Specific Volume of Dry Fluids

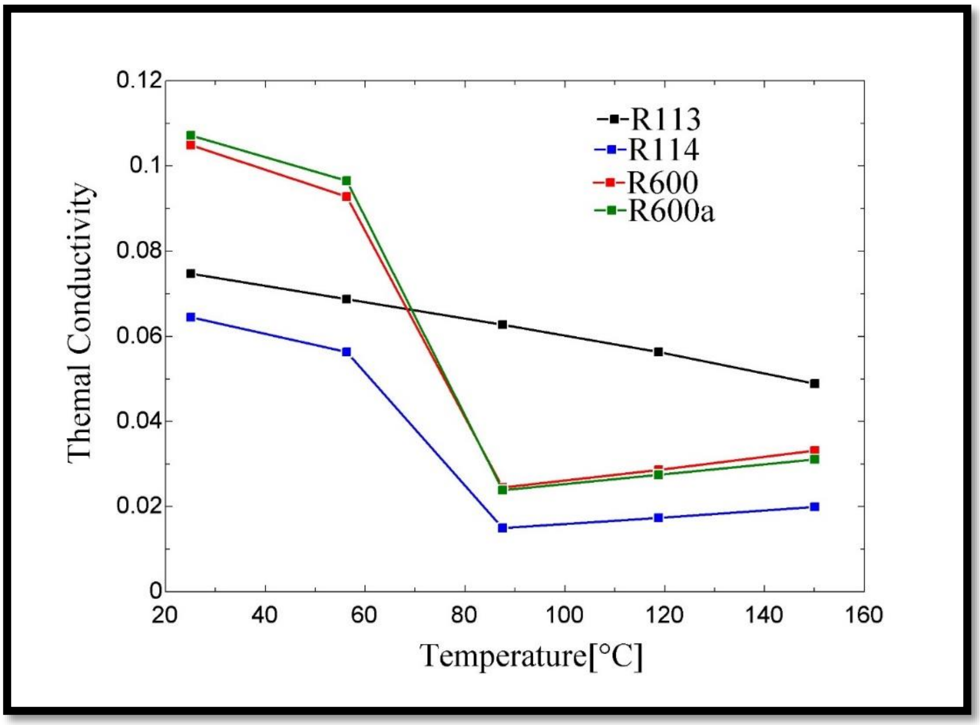

Figure 4.3c: Thermal Conductivity of Dry Fluids

The Figure 4.3b shows the specific volume of dry fluids, explains that R600 and R600a has the highest specific volume giving it higher work of pump theoretically. Besides that, the Figure 4.3c shows the thermal conductivity of dry fluids.

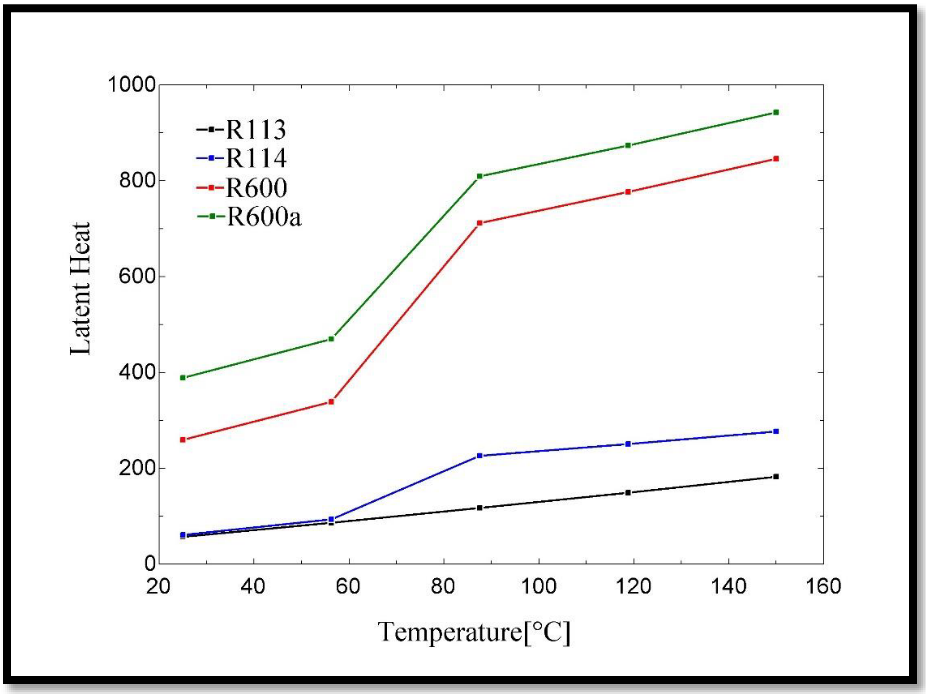

Figure 4.3d: Latent Heat of Vaporization for Dry Fluids

From the Figure 4.3d, R600a exhibits the highest latent heat of vaporization, giving highest work net among all the fluids in the group. Besides, the parallel pattern reading of R600a and R600 gives more or less approximated values of work net.

4.2 Discussion of Result

Work Pump

From the previous Figure 2.1, under Chapter 2 Literature Review, the T-s graph of an Organic Rankine Cycle shows the four point, at where it numerically presents the inlet and outlet of subsystems in the cycle. Identification of work of pump eases by applying equation, (E-8), under Chapter 3 Methodology and getting the actual work of pump. Work of pump is an important parameter to concern when comes to evaluating back work ratio and performance of the system. Table 4.1a, 4.1b and 4.1c shows the work of pump between the turbine outlet temperature range of 25°C to 150°C, for dry, wet and isentropic fluids. The values of work pump for working fluids on each group tabulated from the Engineering Equation Solver software. From the Table 4.1a, it shows that R600 has higher work of pump and R113 has lowest work of pump from dry fluid group. Besides that, Table 4.1b reveals that, R152a has the highest work of pump among wet fluids and specifically R125 has the lowest work of pump, in fact, of all the group of working fluids giving it value of 0.12kJ/kg. Furthermore, Table 4.1c shows that R141b has the highest work of pump and R134a has the lowest work of pump.

Table 4.1a: Work of Pump for Dry Fluids

Dry Fluids' Work Pump (kJ/kg)			
R113	**R114**	**R600**	**R600a**
1.40	1.02	1.91	1.85

Table 4.1b: Work of Pump for Wet Fluids

Wet Fluids' Work Pump (kJ/kg)			
R125	**R143a**	**R152a**	**R22**
0.12	0.30	1.18	0.44

Table 4.1c: Work of Pump for Isentropic Fluids

Isentropic Fluids Work Pump (kJ/kg)			
R123	**R134a**	**R141b**	**R290**
1.04	0.81	1.58	1.30

Work Net

The difference between work or turbine and work of pump yields another important parameter for selecting best working fluid, which is Work Net. Table 4.2a, 4.2b and 4.2c below shows the work net of each group working fluid at turbine outlet temperature range of 60°C to 150°C. The zero work net simply represents neglected negative work net or total work done is zero.

Table 4.2a: Work Net of Dry Fluids

Dry Fluids' Work Net (kJ/kg)				
Temperature (°C)	**R113**	**R114**	**R600**	**R600a**
60	0	0	0	0.42
82.5	0	0.17	1.65	4.34
105	0	10.02	31.20	31.79
127.5	0	11.37	35.10	35.40
150	0.64	12.577	38.58	38.71

Table 4.2b: Work Net of Wet Fluids

Wet Fluids' Work Net (kJ/kg)				
Temperature (°C)	**R125**	**R143a**	**R152a**	**R22**
60	1.02	1.04	0.4	0.35
82.5	14.49	19.90	2.155	19.05
105	16.78	23.28	27.44	22.21
127.5	18.85	26.31	31.64	25.06
150	20.81	29.14	35.44	27.75

Table 4.2c: Work Net of Isentropic Fluids

Isentropic Fluids' Work Net (kJ/kg)				
Temperature (°C)	**R123**	**R134a**	**R141b**	**R290**
60	0	0	0	1.296
82.5	0	14.91	0	36.98
105	0	18.15	0	43.42
127.5	0.87	20.83	0	49.19
150	13.58	23.27	16.88	54.61

From the Table 4.2c, R290 has highest work net, of 54.61kJ/kg. Table 4.2b shows that R152a has highest work net among all the fluids in the group giving it a value of 35.44kJ/kg. Observing the dry fluids, all the fluids shows a better work net parameter after the temperature 82.5°C, except for R113, whose work net relatively very small compared to all other working fluids in three groups. This is because the work of pump is higher before that temperature. In the dry fluids group R600a exhibits highest work net giving it a value of 38.71kJ/kg. Again looking into the value of work net, R600 and R600a gives almost similar result.

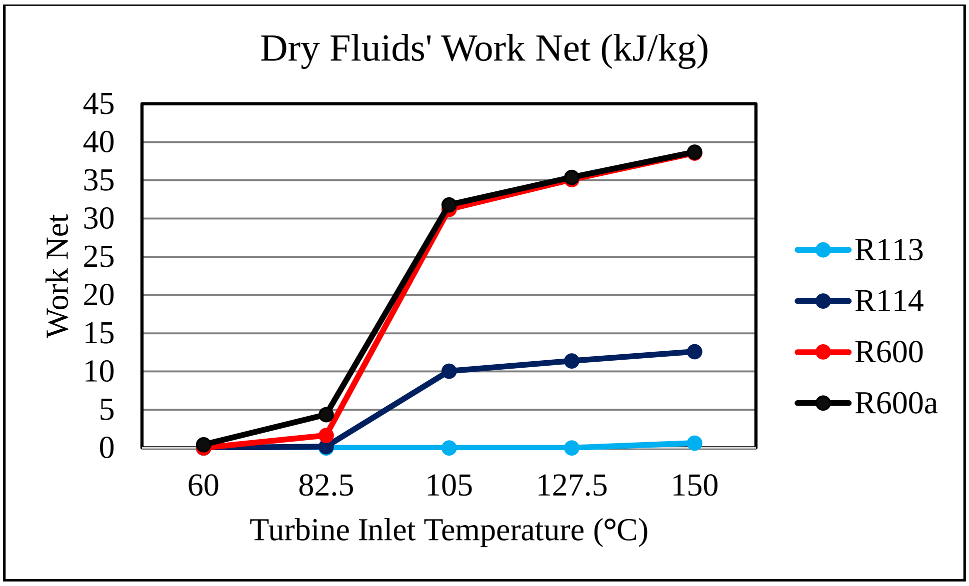

Figure 4.4a: Work Net of Dry Fluids

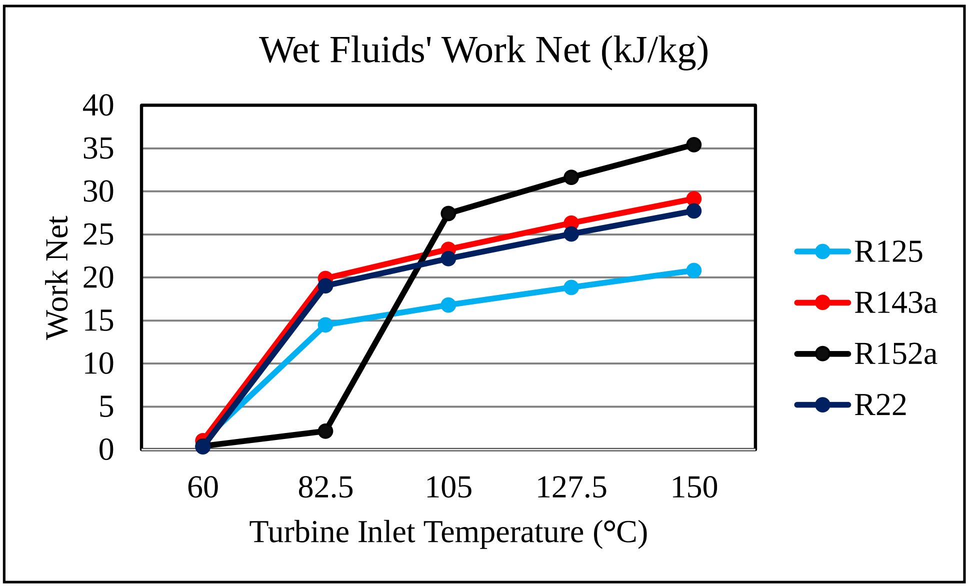

Figure 4.4b: Work Net of Wet Fluids

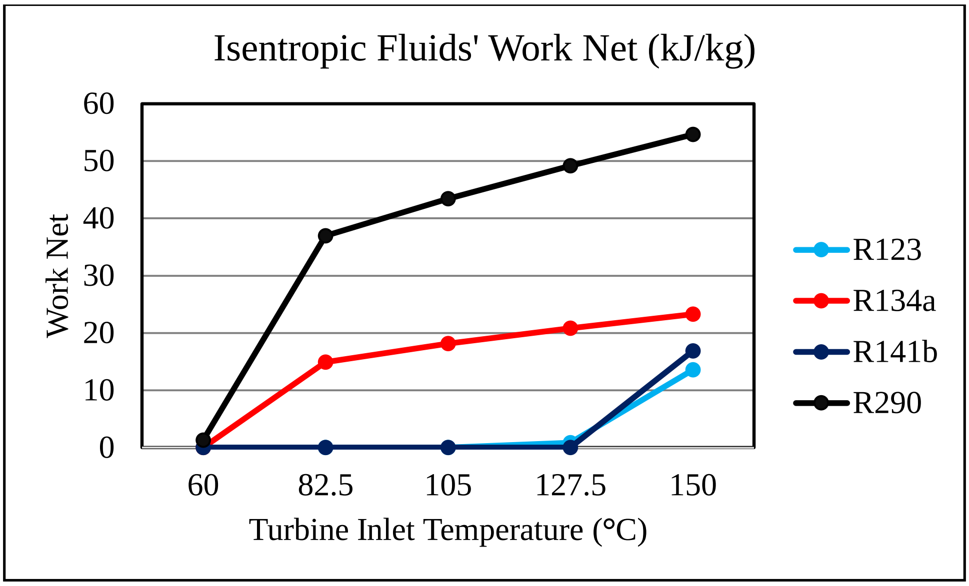

Figure 4.4c: Work Net of Isentropic Fluids

Figures 4.4a, 4.4b and 4.4c above shows the graph of work net for each working fluid group which are, dry fluids, wet fluid and isentropic fluid. At the temperature range of 80°C to 100°C in dry fluids, the R600a shows better work net at lower temperature from 80°C compared to R600. Above all, R290 as usual exhibits the highest work net. In wet fluid group, the fluid R152a shows better work net compared to other fluids in that group.

Back Work Ratio

The ratio of work of turbine to work of pump, as mentioned in the equation, (E-6), under Chapter 3 Methodology, is known as the back work ratio. The higher value of back work ratio is preferable, because it indicates larger work of turbine and smaller pump work. Table 4.3a, 4.3b and 4.3c show the back work ratio of dry, wet and isentropic working fluids.

Table 4.3a: Back Work Ratio of Dry Fluids

Dry Fluids' Back Work Ratio				
Temperature (°C)	**R113**	**R114**	**R600**	**R600a**
60.00	0.00	0.17	0.76	1.23
82.50	0.37	1.17	1.86	3.35
105.00	0.40	10.80	17.32	18.18
127.50	0.57	12.12	19.36	20.14
150.00	1.46	13.29	21.18	21.92

Table 4.3b: Back Work Ratio of Wet Fluids

Wet Fluids' Back Work Ratio				
Temperature (°C)	**R125**	**R143a**	**R152a**	**R22**
60.00	9.52	4.46	0.30	1.78
82.50	121.03	66.70	2.83	43.35
105.00	139.98	77.85	24.25	50.38
127.50	157.12	87.85	27.81	56.71
150.00	173.34	97.19	31.03	62.69

Table 4.3c: Back Work Ratio of Isentropic Fluids

Isentropic Fluids' Back Work Ratio				
Temperature (°C)	**R123**	**R134a**	**R141b**	**R290**
60.00	0.50	0.80	0.36	1.23
82.50	0.53	19.36	0.39	29.27
105.00	0.57	23.35	0.43	34.20
127.50	1.93	26.65	1.01	38.61
150.00	14.09	29.67	11.79	42.75

From the tables so far from work of pump, work net and back work ratio above, the working fluid that has the highest reading in work net and having back work ratio of 42.74, is R290. From the Table 4.3b, noted that the highest back work ratio is R125, value of 173.34, which is very large compared to the R152a which is 31.03. This can be explain from the work of pump in previous Table 4.1b, under Work of Pump, where R125 has the smallest work of pump, 0.12kJ/kg, among all the working fluids in three groups. Besides, R152a, having the value of 1.18kJ/kg, is the highest work of pump in the wet group. Thus the highest the work of pump, the lower the back work ratio. In Table 4.3c, R600a shows higher back work ratio compared to R600 in the dry group. The Figures 4.5a, 4.5b and 4.5c below show the graph of back work ratio of each group working fluids under the turbine outlet temperature range of 25°C to 150°C.

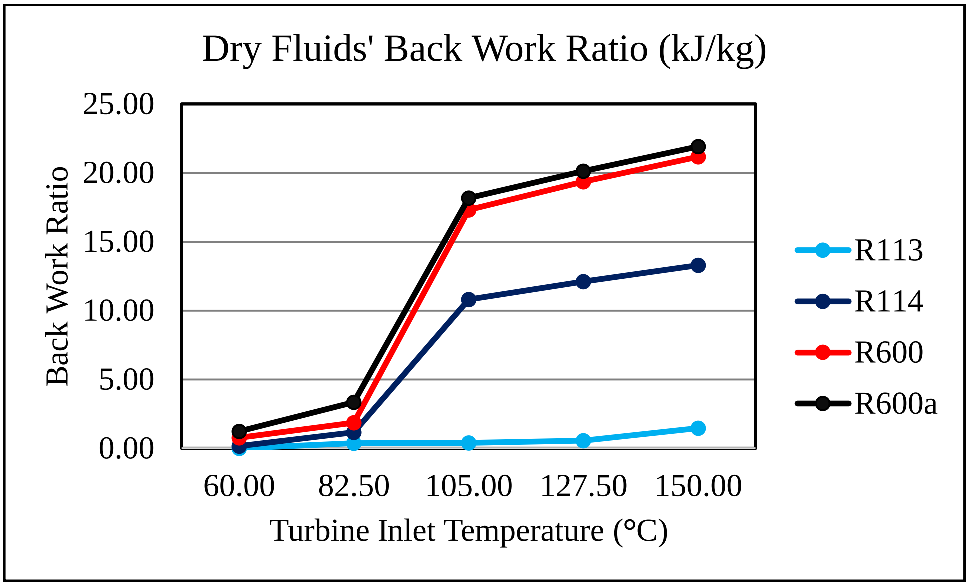

Figure 4.5a: Back Work Ratio of Dry Fluids

From the Figure 4.5a, the approach of back work ratio for R600a and R600 is similar but giving R600a the highest in dry fluid group. Besides that, the Figure 4.5b, reveals that R125 has the highest back work ratio in wet fluid group. Apart from that, Figure 4.5c, shows that R290 has the highest back work ratio among isentropic fluids.

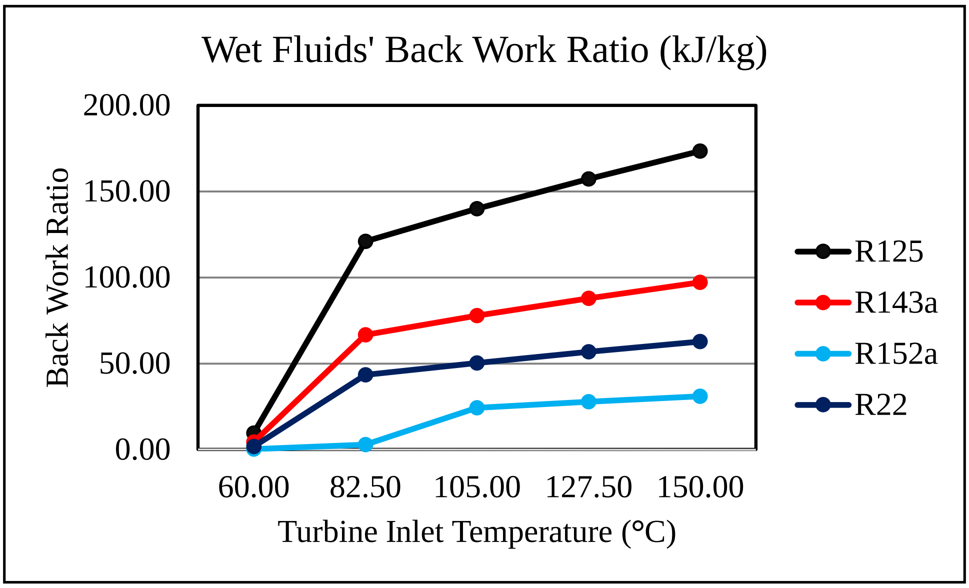

Figure 4.5b: Back Work Ratio of Wet Fluids

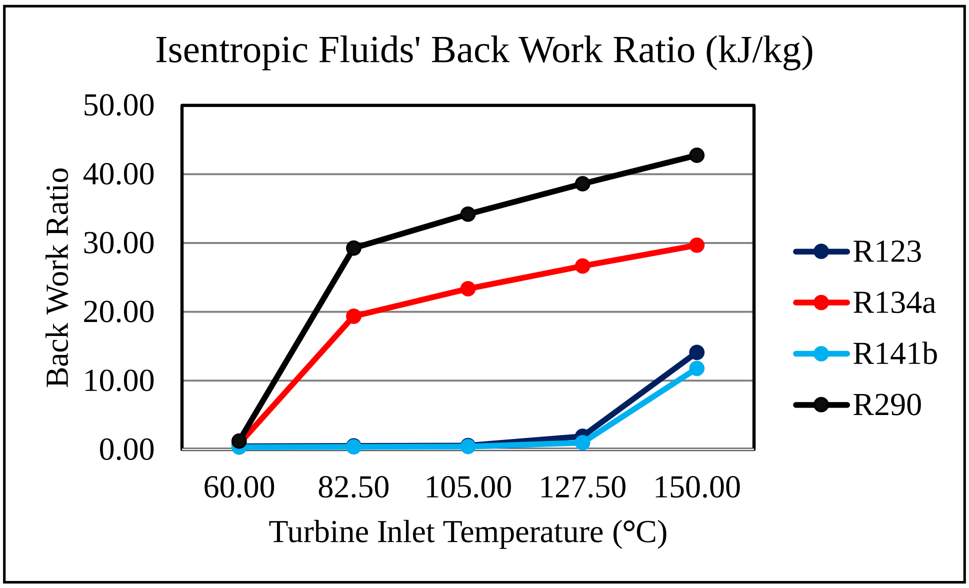

Figure 4.5c: Back Work Ratio of Isentropic Fluids

Efficiency

Efficiency is one of the important parameter, on selecting the right refrigerant. But sometimes the parameters like back work ratio can be given greater priority for selecting the right refrigerant. The Figure 4.6a, 4.6b and 4.6c below shows the efficiency graph of different working fluids at turbine outlet temperature range of 25°C to 150°C.

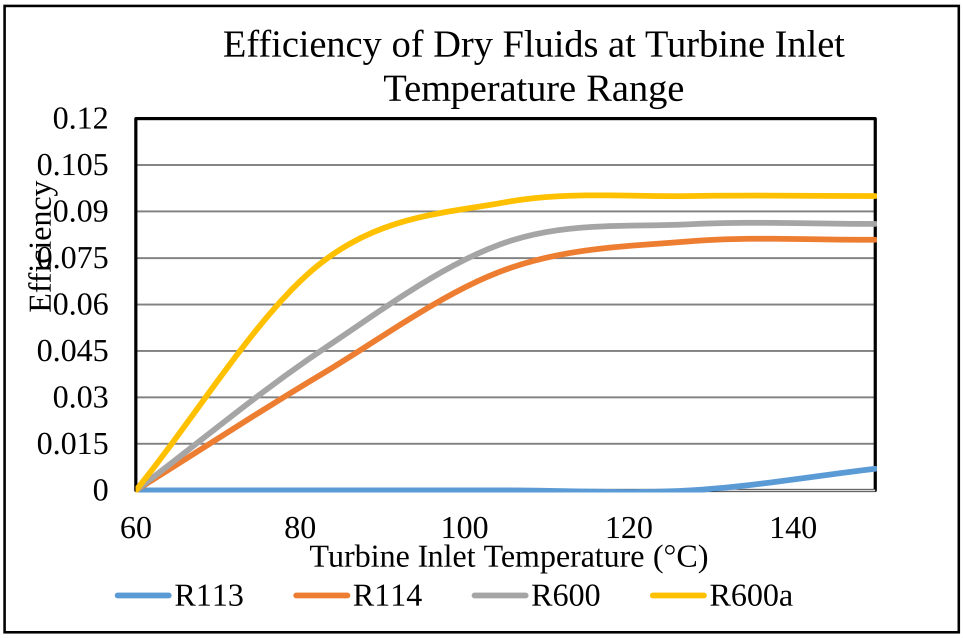

Figure 4.6a: Efficiency of Dry Fluids

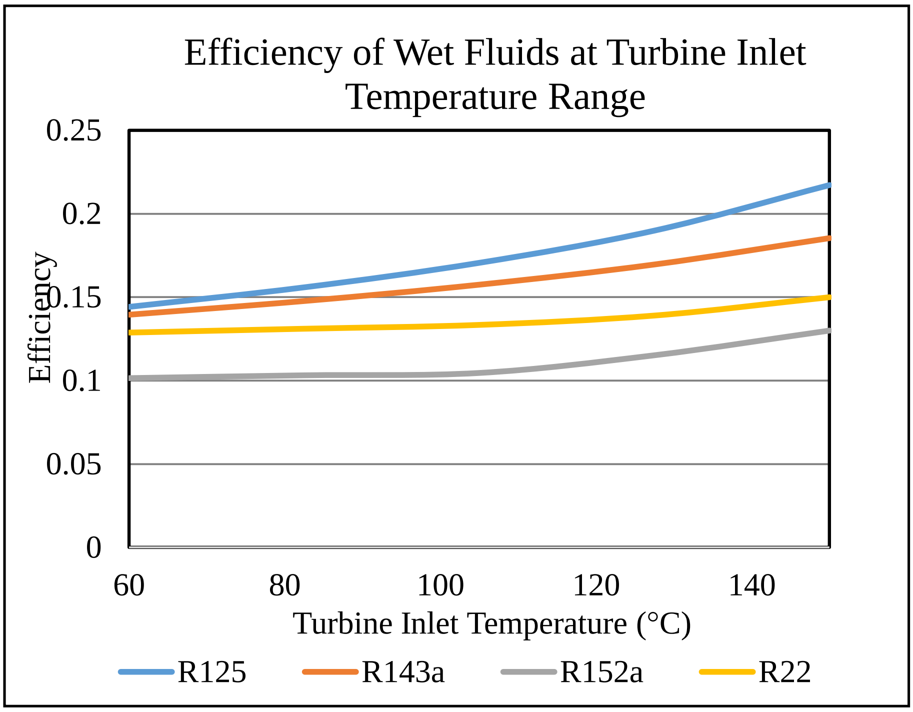

Figure 4.6b: Efficiency of Wet Fluids

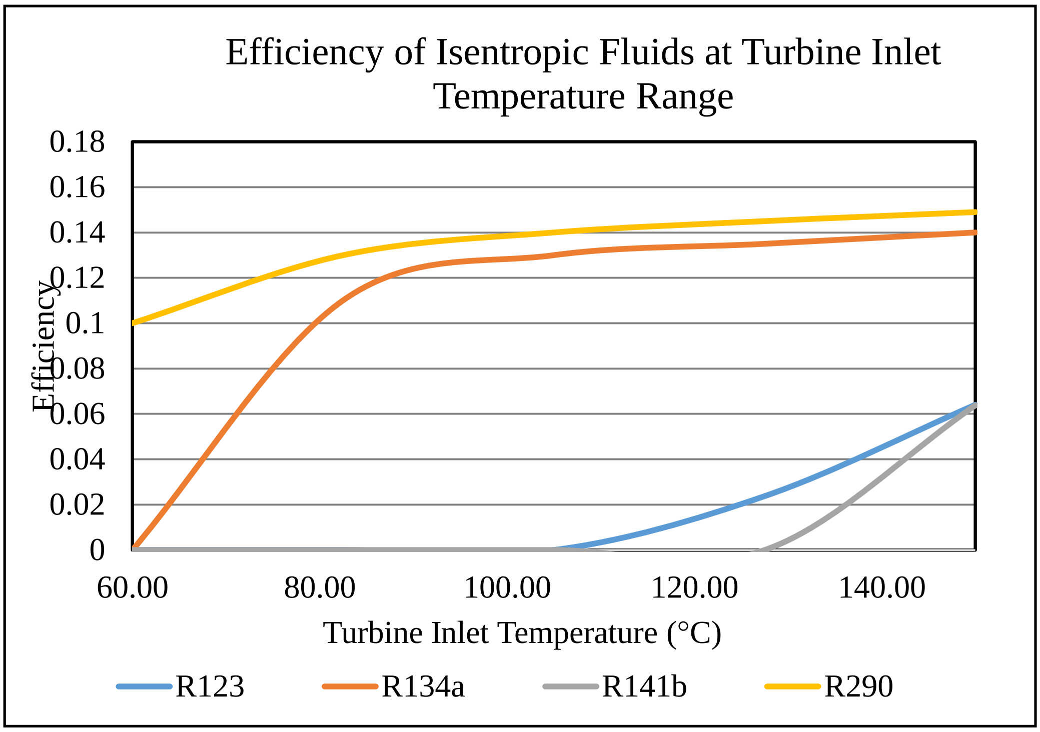

Figure 4.6c: Efficiency of Isentropic Fluids

From the graphs above, the overall efficiency is very small efficiency which is below 0.3, because usage of low grade heat sources that produces relatively lower work net compared to usage of high grade heat sources.

Thus from the graph, **the best working fluid of R290 from the isentropic group is chosen as the best working fluids under the given constraints applied in this research project**. This is because of the higher work net produced by the working fluid and its higher back work ratio among the isentropic group and better efficiency which is approximately 0.15. Even though the efficiency of R125 is the highest among all the group working fluids, consideration of overall best result in terms of efficiency, work net and back work ratio is needed. From the wet group R125 is chosen over the potential working fluid which is R152a. This is because of the back work ratio of R125 is very large and R152a is lowest in the group. It also shows that the work of pump for R125 is very low compared to R152a. Improvement in performance of turbine can be done better compared to pump. Thus under suitable and better constraints R125 could exhibits higher work net, thus giving the main idea that work of turbine could be increase in further research. Parameter of work net is concerned over efficiency when selecting R290 as the best fluid compared R125. R125 shows the highest efficiency up to 20% compared to R290 whose efficiency is 15% maximum. But R290 has better work net compared to R125’s which is 20.81kJ/kg. Consideration on selecting R290 as best fluid is higher, because usage of low grade heat sources relatively gives smaller work net when compared with work net produced by high grade heat sources which would go up to 70% efficiency. Efficiency parameter will be more prioritise if high grade heat sources is used. This is the reason why R290 chosen over R125. Whereas from the dry group, R600a is chosen as the best working fluid, as it shows better work net, back work ratio and efficiency compared to R600.

CONCLUSION

Organic Rankine Cycle is efficient cycle for energy production from low grade heat sources, like solar and biomass using organic working fluids. Purpose of organic working fluids, is justified briefly over the conventional one, water, as water possess higher boiling point than organic working fluids which is inefficient for Organic Rankine Cycle. The objective of this research project which is identifying the best working fluids for low grade heat sources, acting between the temperature of 60°C to 150°C, is met by the chosen suitable working fluid, R290, under the isentropic fluid group. It has the work net of 54.61kJ/kg making it the highest among all the group of fluids. The efficiency of the fluid is 15%, being the highest in isentropic group. Lastly, the back work ratio is also favourable giving the highest in the same group, thus R290 is the best fluid. From the wet fluid group, R125 is chosen over the potential fluid R152a due to higher back work ratio of the fluid among all the groups of working fluid giving up to 173.34 and efficiency among the same group of fluid giving up to 20%. The reason is because in term of overall selection of the fluid Work net was given more priority compared to efficiency due to usage of low grade heat sources. Low grade heat sources usually produces smaller efficiency for overall of the fluid, thus work net is focused more as that what we are aiming for under ORC. Among the dry fluids, R600a chosen as the best working fluids since it has higher work net of 38.71kJ/kg and highest efficiency among its group giving up to 10%. Recommended method of improving the performance of organic Rankine Cycle is through setting the temperature of turbine inlet and outlet range when the work net started to become positive. For example working fluids like R141b exhibits better performance after the temperature 120°C and R123 shows better performance after the turbine inlet temperature 100°C. Minimizing the work of pump and increase the efficiency of the system, are the benefits of internal heat exchanger, is highly recommended by connecting between before boiler and after turbine. The objective of this research project has met.

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