POTENTIAL OF ORGANIC RANKINE CYCLE BASED HEAT RECOVERY SYSTEMS FOR POWER GENERATION

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DECLARATION

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ABSTRACT

Due to intense fuel dependency on energy production in the world, cost of energy has a greater bearing on the prices of fossil fuels. Most of the countries in the world are suffering due to this and Sri Lanka is no exception. It is in this context promotion of optimize the usage of thermal power generation, is so vital to the country. Even though fossil fuel base power generation plays a greater role as a source of primary energy in the country, major portion wasted to environment. WHR systems have been already introduced, but most of them are not performing effectively and efficiently. On other hand, novel systems and technologies required to investigate, to recovery most of the wasted heat of thermal plant while increasing the system efficiency and reducing the fuel cost. Conceptual thermodynamic cycles such as Trilateral Flash cycle, Organic Rankine cycle, Kalian cycle and Gaswami cycle, can be successfully incorporate for WHR applications. Hence, purpose of this research was to assess the amount of waste heat generated by thermal plants in the country while discussing the possible technologies that can be introduce for heat recovery. Further, discuss about selection of most suitable option and carryout thermo-economic analysis as a case study.

Fluid selection and system optimisation based on heat source temperature are two most critical aspect of Organic Rankine Cycle. Eleven fluids were investigated to optimize the work output by varying the evaporator temperature and varying the expander pressure ratio with theoretical model. In evaporator analysis, Heptane, Pentane and Decane shows favourable results in terms of work outputs while, in terms of efficiency, Decane and Heptane are better. Further it is recommended to use fluid Pentane, when source temperatures of WHR lies between 45 – 190 ^{o}C, while fluid Heptane is recommended when source temperature between 190 – 260 ^{o}C. Fluid Decane is recommended when temperature between 260 – 340 ^{o}C. Respective monographs were developed where one point on the graph can denote approximate work output, efficiency, pressure, temperature, etc. Based on expander analysis, Decane, Heptane and Toluene fluids have shown higher work outputs while, in terms of efficiency, Decane is better. In expander selection, when inlet/outlet pressure ratios are less than 10, fluid Decane is recommended. Further, when ratios are in between 10 – 13 and 13 – 20, fluid Heptane and fluid Toluene are recommended respectively. Refer to these 03 fluids, monographs were developed accordingly.

Refer to optimum working regions of temperature analysis; fluids were selected for economic evaluation. Waste heat recovery opportunities were selected from existing thermal plants for the case study and electric outputs were obtained for each plant, based upon selected fluids from theoretical model. Then maximum work out of each opportunity was selected for further economic evaluation under 07 different scenarios. Possible future economic situations of the country were predicted under those scenarios and carryout NPV calculations for each, to evaluate the investment feasibility. Scenario 2, 3 and 7 are the most possible situations of the country in future and for those conditions, WH opportunities at Supugaskanda, Lakvijaya, Keravalapitiya and Kelanithissa are most feasible to recover waste heat with ORC system.

ACKNOWLEDGEMENTS

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Most difficult part of this exercise was the collection of data related to waste heat energy of each thermal power plant in the country. My special thanks in this regard should go to my fellow engineers who have supported in data collection in various thermal power plants.

The biggest challenge I faced during this research was to find time for this and meeting the targets. I am sure if not for my wife, Nadeeka De Silva, my new born son Dihas and my mother I.D. Priyangani, I could not have completed this research on time. Hence, my gratitude should go to my beloved family members for the sacrifices they made during this research.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

Chapter 07

LIST OF ABBREVIATIONS

CEB	– Ceylon Electricity Board
NRE	– Non-Renewable Energy
WHR	– Waste Heat Recovery
IPP	– Independent Power Producers
HSFO	– High Sulfur Fuel Oil
LSFO	– Low Sulfur Fuel Oil
SPS	– Sapugaskanda Power Station
KPS	– Kelanithissa Power Station
GT	– Gas Turbine
ST	– Steam Turbine
CCPP	– Combined Cycle Power Plant
CCGT	– Combined Cycle Gas Turbine
LPT	– Low Pressure Turbine
HPT	– High Pressure Turbine
IPT	– Intermediate Pressure Turbine
CHP	– Combined Heat & Power
ORC	– Organic Rankine Cycle
TFC	– Trilateral Flash Cycle
NPV	– Net Positive Value

$\dot{W}_{exp}$	- Expander work
$\dot{m}$	- Fluid mass flow rate
h	- Specific enthalpy
η_{is}	- Expander isentropic efficiency
$\dot{W}_{exp.is}$	- Expander isentropic work
Q_{evap}	- Evaporator heat energy addition
$\dot{W}_{pump.is}$	- Pump isentropic work
P	- Pressure
ρ_{liquid}	- Fluid density
$\dot{W}_{pump}$	- Pump work
η_{pump}	- Pump efficiency
$\eta_{pump.is}$	- Pump isentropic efficiency
η_{cycle}	- Cycle Efficiency
$\dot{W}_{in}$	- Work input
$\dot{W}_{out}$	- Work output
$Q_{evap.\ max}$	- Maximum available heat energy for evaporator
η_{evap}	- Evaporator efficiency
P1/P2	- Expander pressure ratio between inlet and outlet

LIST OF APPENDICES

1.0 INTRODUCTION

Energy plays a vital role in our day to day activities. Especially, energy by electricity has become an essential need for our life, but, Sri Lanka's electricity sector has been ailing for the last two decades due to its excessive dependency on fossil fuels and lack of diversity in energy sources in the energy supply mix. Out of the available energy supplies, coal and petroleum supply the base demand while hydropower injects its limited energy to meet the rest of the demand. Figure 1.1 shows the share of each major resource in the annual electricity generation mix of Sri Lanka in the years 2012 and 2013 [1].

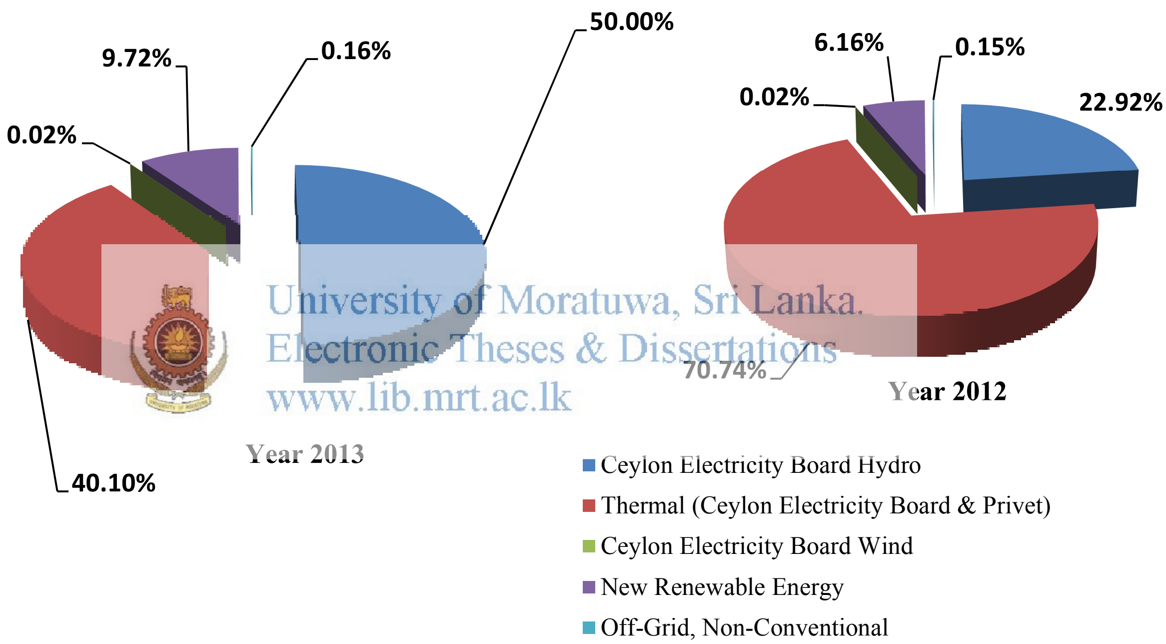

Figure 1.1: Electricity energy generation mix of Sri Lanka in the years 2012 and 2013 [1]

The total amount of electricity generated during 2012 was 11,878.8 GWh out of which 70.74% was from thermal power plants (both oil and coal); while 23.0% was from major hydro and the balance 6.2% was from non-renewable energy. In the year 2013, total electricity generated was 12,005.5 GWh out of which 50.0% was from major hydro plants and 40.1% from thermal plants. The NRE generation reached 9.7% in 2013.

Country has received very good rainfall in 2013 compared to previous years, recording highest hydro electricity generation in history. Enhanced diversity of fossil fuel resources used in power generation managed to reduce the imported oil volumes by more than 20%.

This clearly shows the domination of thermal base electricity generation in the power generation mix of the country. Government spent around 5 billion rupees to import petroleum products annually, which is equal to 1/3 of country's GDP. Petroleum being an external resource, their price is fully governed by external factors over which Sri Lanka has hardly any control. During the last several years, surging petroleum prices had adverse repercussions on the electricity industry and made the utilities run into losses. Hence, energy conservation will give a huge hand to improve the economy at this critical situation. Priority should be given in energy conservation is to maximize utilization of energy in combustion fuel. Maximum utilization of thermal power reduces the unit cost and directly reduces the fuel consumption. Overall efficiency of the thermal plants recorded in 2013 is 32% [1]; it means 68% of fuel energy is wasted during power generation. From an economic point of view, if overall efficiency of thermal plants in the country increased by 1%, Rs. 15 Million can be earned additionally by the sale of generated electricity units. Major sources of energy wasted are exhaust gas & cooling system. Proper operation & Waste Heat Recovery (WHR) systems are critical factors of a thermal plant that increase the overall performances. As thermal power generation is very expensive, optimum performance of heat recovery systems is a must. Most of the heat recovery systems in existing thermal plants in the country are outdated as technology vise and, as system vise; they are not operating up to the standard. Hence, there is a wide gap between existing WHR systems and modern innovations to be examined at research level. Further, it was hard to find any research done in Sri Lanka, to evaluate waste heat as qualitatively and quantitatively to match with modern WHR systems.

Aim and Objectives

Main focus of this research is to identify main waste heat sources of selected major thermal power plants in the country and assess the amount qualitatively and quantitatively. Further, new concepts and alternative technologies have been introduced to the world, which provide many solutions for the shortcomings in the existing systems. Hence, this work expects to identify existing waste heat opportunities in the plants and modern solutions to recovery of waste energy. Further, this research study will help for future rehabilitations and newer plants, to improve the overall performance.

Research aim and objectives can be scrutinized as follows;

Aim

To investigate the waste heat recovery potential of thermal power plants in Sri Lanka. Further, carryout techno-economic feasibility analysis for selected waste heat opportunities, and evaluate their viability of implementation.

Objectives

- To investigate waste heat opportunities of different thermal power plants in Sri Lanka.
- To analyses waste heat recovery methods and technologies.
- To evaluate the viability of implementation, technically and economically with appropriate recovery technologies.

2.0 THERMAL POWER GENERATION

Few decades back, the entire electricity requirement of Sri Lanka was met by hydro based power generation. But in the last 02 decades, demand for electricity had rapidly increased and hydro capacity couldn't match the required demand. Hence, thermal power generation has been introduced to the country during the latter part of the 20th century and from that point onwards, electricity generation has being shifting more towards thermal based power generation.

2.1 Present Status

At present, around 60% [4] of annual electricity demand of Sri Lanka is supplied by thermal based power generation and this average figure tends to vary with annual rain fall. Figure 2.1 given below shows the transformation of electricity generation from Hydro to Thermal power over the last few decades [2].

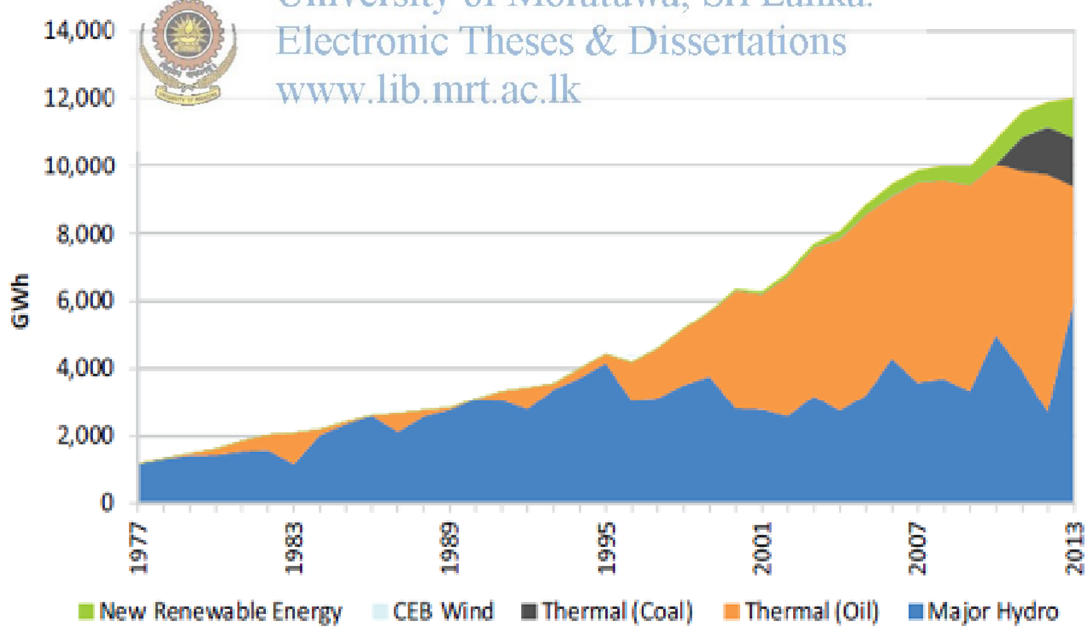

Figure 2.1: Hydro/Thermal/Non-conventional energy share in the National Grid [2]

Thermal power generation has been continuously increased after year 2000, while hydro power stagnated over the period, as depicted in the above figure. The transfer from an oil base to a coal base in thermal power generation began with the commissioning of the 1st coal power plant in 2011.

Considered from another perspective it is obvious that, increase in thermal power means, high unit cost & increase in fossil fuel imports. Frequent fluctuations due to political instability in the world & rapid increase of fossil fuel prices in the world market during last 02 decades have caused a crisis in Sri Lanka's electricity sector. Presently, fossil fuel import is a heavy burden on the national income. Figure 2.2 shows the graph of total imports Vs petroleum imports and Figure 2.3 shows total exports Vs petroleum imports to Sri Lanka during 2008 to 2013.

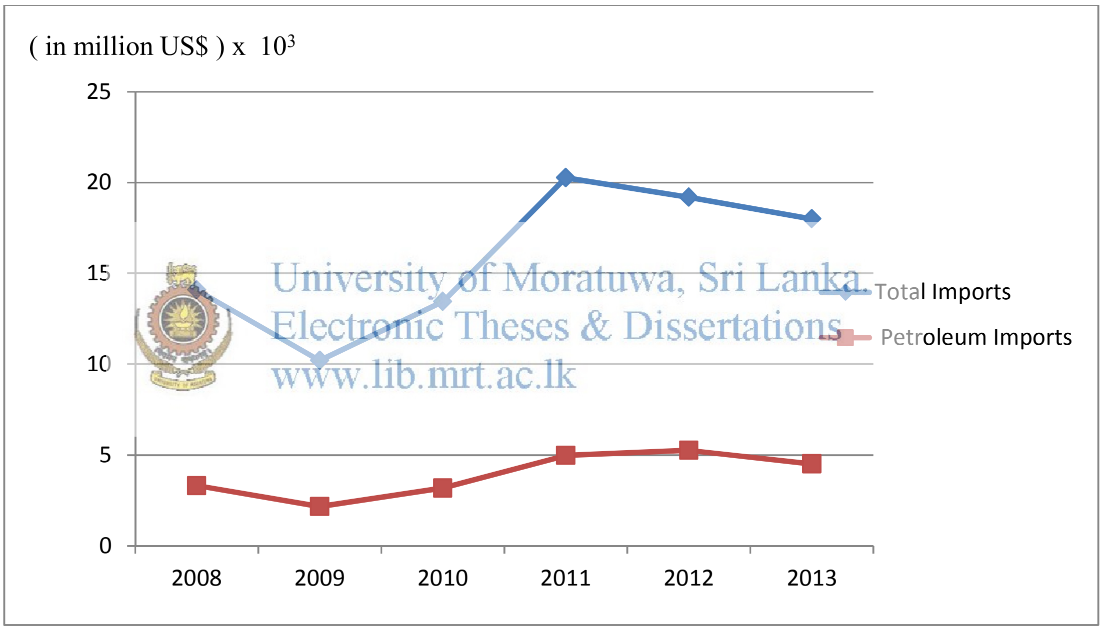

Figure 2.2: Total imports Vs petroleum imports over the last 05 years [1]

As per the figure 2.2, petroleum import is equal to ¼ of total imports to the country in 2013. According to figure 2.3, government spent over 5 billion rupees to import oil annually, which is equal to 1/3 of countries gross domestic production. Hence, energy conservation will give huge hand to be sustained at this critical situation.

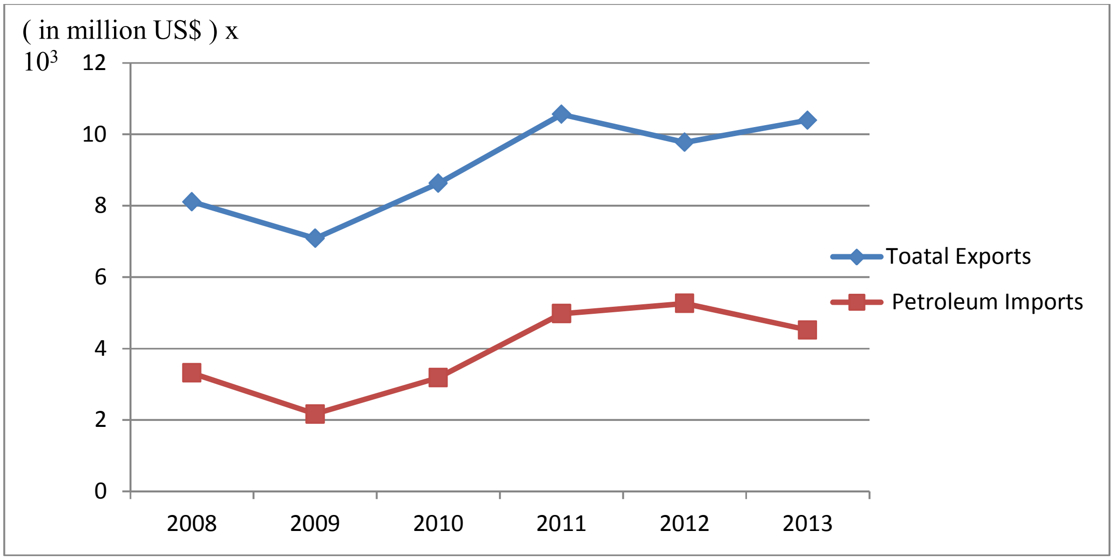

Figure 2.3: Total exports Vs petroleum imports over the last 05 years [1]

Even though, the electricity sector heavily depends on petroleum and hydropower sources, both these sources are highly unreliable as there is very little control over them. Petroleum is an external resource and their prices are fully governed by external factors over which Sri Lanka has hardly any control.

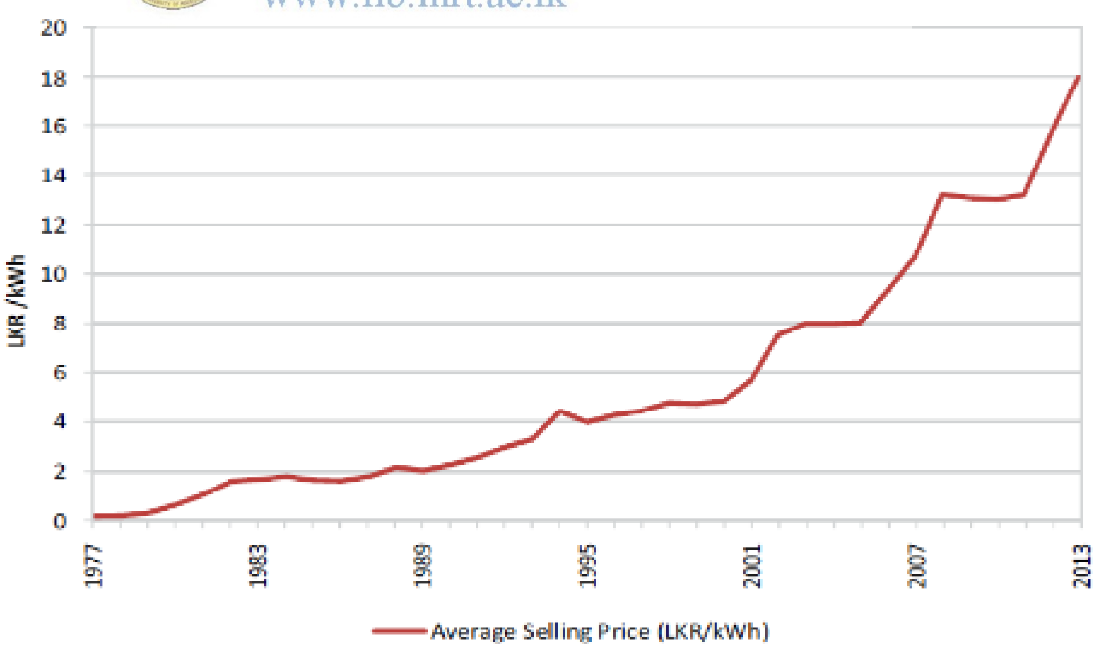

Figure 2.4: Graph of Average Selling Price of Electricity [2]

Figure 2.4 shows how average electricity price has increased over the last few decades. During last several years, surging petroleum prices had adverse repercussions on the electricity industry and making the utilities run into losses. Although hydropower is an economic and renewable source, its performance depends on rainfall received in the catchment areas of the hydro reservoirs. Every time the country had a dry spell, the CEB had to face great difficulties in meeting the electricity demands of the country.

2.2 Thermal Power Plants in Sri Lanka

Some of the thermal power plants owned by Ceylon Electricity Board and others are operated by Independent Power Producers (IPP) of the country. Following paragraphs discuss about the ownership of existing plants, their capacities and electricity generation.

Table 2.1: CEB owned thermal power plants and respective generations 2013 [2] [4]

Name of the Power Station	Technology Type	Fuel Type	Capacity (MW)	Gross Generation (GWh)	Share in Generation %
CEB					
Kelanithissa Power Station	GT Stage 2	Auto Diesel	115	16.6	0.3
	GT Stage 3	Auto Diesel	100	1.0	-
Sapugaskanda Power Station	Diesel Engine	Auto Diesel	80	6.1	0.1
		HSFO 380 cst		175.9	3.6
Sapugaskanda Extension Plant	Diesel Engine	Auto Diesel	80	7.0	0.1
		HSFO 380 cst		383.9	8.0
Small Generators	Diesel Engine	Auto Diesel	8	0.3	-
Kelanithissa Power Station	Combined Cycle Plant	Auto Diesel	165	221.7	4.6
		Naphtha		388.5	8.1
Uthuru Janani	Diesel Engine	Auto Diesel	24	13.8	0.3
		HSFO 180 cst		111.3	2.3
Lakvijaya Power Station	Steam	Auto Diesel	300	4.0	0.1
		Coal		1465.4	30.4
Total				**4795.8**	**57.9%**

Year 2013 was generally considered as a wet year and out of the total generation for the year; CEB thermal plants have produced 4,795.8 GWh which is equal to a contribution of 57.9% of the total thermal power generation.

Table 2.2: IPP owned thermal power plants and respective generations 2013 [2][4]

Name of the Power Station	Technology Type	Fuel Type	Capacity (MW)	Gross Generation (GWh)	Share in Generation %
Lakdhanavi	Diesel Engine	HSFO 180 cst	22.5	Agreements are terminated	
Asia Power	Diesel Engine	HSFO 380 cst	51	161.4	3.3
Colombo Power	Diesel Engine	HSFO 180 cst	60	331.8	6.9
Ace Power Matara	Diesel Engine	HSFO 180 cst	20	Agreements are terminated	
Ace Power Horana	Diesel Engine	HSFO 180 cst	20	Agreements are terminated	
AES - Kelanithissa	Combined Cycle	Auto Diesel	110	156.0	3.2
Heladanavi	Diesel Engine	HSFO 180 cst	100	476.4	9.9
Ace Power Ambilipitiya	Diesel Engine	HSFO 180 cst	100	413.8	8.6
Aggreko	Diesel Engine	Auto Diesel	15	Agreements are terminated	
Yugadhanavi Kerawalapitiya	Combined Cycle	LSFO 180 cst	270	460.2	9.5
Northern Power	Diesel Engine	HSFO 180 cst	20	24.4	0.5
Total				**2023.9**	**42%**

Thermal plants of Independent Power Producers (IPP's) have produced 2,023.9 GWh which is equal to a 42% contribution of the total thermal power generation for the year.

2.3 Power Generation and Efficiencies of Thermal Plants

Following tables 2.3 and 2.4, elaborate energy input, output of Sapugaskanda Power Station (SPS) and Kelanithissa Power Station (KPS), comparing the overall efficiency of each unit for a month in year 2013.

Table 2.3: Generation and Efficiencies of SPS [4]

Engine No:	Used Diesel Qty (m^3)	Used Heavy Fuel Oil Qty (m^3)	Total Input Energy	Generation (MWh)	Energy Output	Overall Efficiency
Scheme A 20 MW each						
E 01	20.5	1562.5	65,136,555	6240	22,464,000	34.49%
E 02	28.6	2567.5	106,843,461	10506	37,821,600	35.40%
E 03	22.9	2311.1	96,068,032	9470	34,092,000	35.49%
E 04	7.2	757.8	31,488,833	3047	10,969,200	34.84%
Scheme B 10 MW each						
E 01	6.946	1486.06	61,483,709	6318.746	22,747,486	37.00%
E 02	8.33	1344.93	55,720,732	6140.325	22,105,170	39.67%
E 03	5.126	1371.32	56,688,851	6302.475	22,688,910	40.02%
E 04	6.539	711.16	29,542,833	3330.225	11,988,810	40.58%
E 05	3.692	1012.50	41,852,113	4651.35	16,744,860	40.01%
E 06	2.469	1382.26	57,040,750	6356.775	22,884,390	40.12%
E 07	0.641	1395.99	57,538,724	6447.225	23,210,010	40.34%
E 08	5.107	1343.20	55,529,684	6233.775	22,441,590	40.41%

Note: Following values were used for above calculations

Calorific value of Heavy Fuel Oil taken as 41.2 MJ/l
Calorific value of Diesel taken as 42.7 MJ/kg
Density of Heavy Fuel Oil taken as 870 kg/m^3
Density of Diesel taken as 930 kg/m^3

Overall efficiency of scheme A around 35% and scheme B remains around 40%.

Table 2.4: Generation Efficiencies of KPS

Se. No:	Plant	Capacity	Fuel	Efficiency
01	GT Frame 02	20 MW	Diesel	19.9%
02	GT Frame 03	20 MW	Diesel	20.81%
03	GT Frame 04	20 MW	Diesel	21.39%
04	GT Frame 05	20 MW	Diesel	21.28%
05	GT Frame 07	115 MW	Diesel	28%
06	Combined Cycle			
	GT	105 MW	Naptha & Diesel	29.73%
	ST	60 MW	Naptha & Diesel	44.14%

Note: Following values were used for above calculations,

Calorific value of Naphtha taken as 44.938 MJ/kg

Density of Naphtha taken as 650 kg/m^3

Table 2.4; elaborates efficiency calculations of KPS, where data is based on energy input and output for the year 2013. Efficiencies of Frame 5 gas turbine (20 MW) varies between 19 % - 22%, while Frame 07 gas turbine remains 28%. In combined cycle power plant (CCPP), efficiencies are separately calculated and it doesn't show the overall figure.

Table 2.5, 2.6 and 2.7 shows the **heat rate** figures and plant **efficiencies** of selected thermal power plants in the country at different loads. Heat rate is a common measurement of system efficiency a thermal power plant which can be defined as "the energy input to a system, typically in Btu/hr or kcal/hr, divided by the electricity generated, in kW."

Table 2.5: Combined Cycle Power Plant- Kelanithissa [3]

CCPP		**Capacity : 165 MW**
Loading (MW)	**Heat Rate (kcal/kWh)**	**Efficiency %**
120 (40%)	2173	39.44%
130 (50%)	2081	41.18%
140 (100%)	2072	41.36%
152.7 (100%)	2020	42.43 %

Table 2.6: AES Kelanithissa Plant [3]

Combined Cycle Gas Turbine Plant		**Capacity : 163 MW**
Loading (MW)	**Heat Rate (kcal/kWh)**	**Efficiency %**
101 (65%)	3038.36	28%
118 (75%)	2739.81	31%
136 (85%)	2502.18	34%
157 (100%)	2027.57	42%

Table 2.7: West Coast Power Plant [3]

Combined Cycle Gas Turbine Plant		Capacity : 270 MW
Loading (MW)	**Heat Rate (kcal/kWh)**	**Efficiency %**
108 (40%)	2745.70	31%
135 (50%)	2483.94	35%
270 (100%)	2083.46	41%

Based on the above information, generations efficiencies of different types of thermal plants in the Sri Lankan context can be summarized as follows;

- Combined thermal plant efficiency range - 40 – 43 % at full load
- Gas turbine plant efficiency range - 19 – 28 % at full load
- IC engine plant efficiency range - 34 – 41 % at full load

The table 2.8 below summarizes the details of exhaust gas temperature and flue gas mass flow rate of thermal power plants owned by Ceylon Electricity Board.

Table 2.8: Summary of exhaust gas temperatures and volumes of CEB thermal plants

Plant Type	Plant Capacity (MW)	Exhaust Gas Temperature (^oC)	Pressure	Load Pattern	Flue Gas mass flow rate (tons/hr)
Sapugaskanda Plant *- IC Engines*					
SPS- A	20	250-300	Atmospheric	100%	140-160
SPS- B	10	250-300	Atmospheric	100%	70-80
Kelanithissa Plant					
Combined Cycle	165	105-110	Atmospheric	100%	400
Gas Turbines	20	440-470	Atmospheric	100%	14400
Coal Power Plant					
Lakvijaya Plant	300	80-90	Atmospheric	100%	1000

Based on above summary, Sapugaskanda and Kelanithissa GT plants have high energy potential in exhaust in terms of temperature. On the other hand, Lakvijaya and both Kelanithissa plants have high energy potential when considering the exhaust mass flow rate. Qualitative and quantitative evaluations for thermal plant exhausts in the country have been done in the forthcoming chapter.

2.4 Thermodynamic Cycles

Different thermodynamic cycles are used for various configurations in thermal power generation. In order to optimize the utilization of thermal energy and to reduce waste heat, different thermodynamic cycles are combined in cascade pattern for combined cycle power plants.

These combinations are considered as the best waste heat recovery solution for certain thermodynamic cycles such as Joule cycle (gas turbine), which is commonly coupled with Rankine cycle (steam turbine). Certain technologies have been well developed over the last few decades while some are still at the initial stage. Combination cycles are introduced where first cycle is considered as topping cycle while following cycle is considered as bottoming cycle. This combination of topping or bottoming is determined based on cycle operation temperatures. Figure 2.5 shows the operating temperature ranges of different thermodynamic cycles [5].

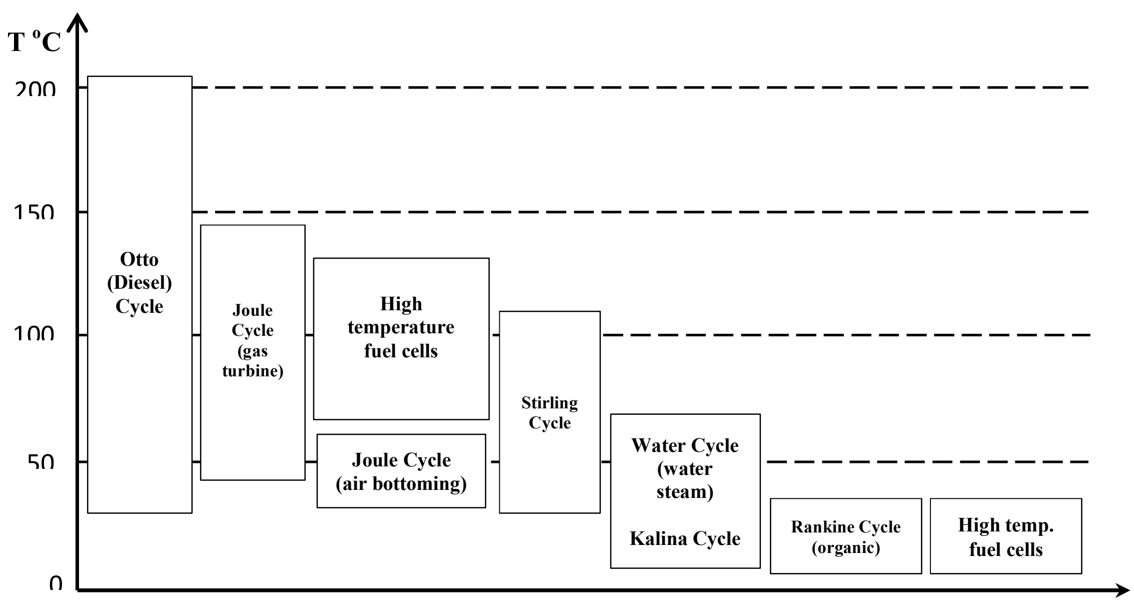

Figure 2.5: Thermodynamic cycles, according to their operating temperature range [5]

According to the above diagram, some of the thermodynamic cycles reject heat at high temperature while some reject heat at low temperatures. On the other hand, shortcomings of topping cycle can often be compensating with bottoming cycle. Combining high temperature cycle with those of medium or low grade temperature provides the most effective way in approaching Carnot efficiency, and thus better utilization of fuel exergy. However, the possibilities for combination may be limited by various factors, such as the status of development, power output, fuel requirements, or part load characteristics.

High temperature cycles [5], will be good candidates for topping cycle while medium or low temperature cycles [5], are ideal for bottoming cycles. This means the shortcoming of one cycle may become a benefit when it is combined with another cycle. However, fuel cell technology for mass scale power generation is still at a developing stage. Even though, Kalina cycle shows some potential, very few plants are operated worldwide at a commercial level. Recently, Organic Rankine cycle has been commercially used as bottoming cycle than conventional steam Rankine cycle, due to proven technology and availability of reputed manufacturers.

Table 2.9: Thermodynamic cycle combination matrix [5]

		Topping Cycle			
	Thermo Cycles	**Rankine**	**Otto/Diesel**	**Joule**	**Fuel Cell**
Bottoming Cycle	**Rankine**	●	●	●	●
	Kalina	●	●	●	●
	Joule		●	●	●
	Otto/Diesel			●	●
	Stirling	●	●	●	●
	Fuel Cell				●
	Heat Pump	●	●	●	●

When thermodynamic cycles are put into a matrix based on their temperatures, a number of combinations can be identified as shown in Table 2.9. Rankine cycle is suitable for both topping and bottoming, just as the Stirling cycle. The Joule cycle (Gas Turbine) along with Otto/Diesel cycle can be better applied as topping cycles. Kalina cycle can be applied as bottoming, with cycles such as Otto, Joule, high temperature fuel cell and Rankine cycles.

2.5 Plant Configurations

According to different topping and bottoming arrangement of thermodynamic cycles, thermal power plants can have various configurations. Based on those configurations, combined power plants are identified by different names. Most common configurations are classified in table 2.10.

Table 2.10: Common configurations of thermodynamic cycles

Thermo Cycle	Configuration	Description
Brayton Cycle (Gas Turbine)	- Open cycle operation	Efficiency < 30%
	- Cascade with Rankine cycle	Combined cycle
Rankine Cycle (Steam Turbine)	- Open cycle operation	Efficiency < 40%
Brayton and Rankine Cycle (GT and ST)	Combined cycle power plant (CCPP) - Single Pressure (SP) - Double Pressure (DP) - Triple Pressure (TP)	Overall efficiency < 55% CCPP/SP- GT/ST CCPP/DP- GT/LPT/HPT CCPP/DP- GT/LPT/IPT/HPT
	Combined cycle co-generation plant (CCCP)	Efficiency < 50% GT-ST and exhaust is used for separate process.
	Combined cycle gas turbine plant (CCGT)	Efficiency < 40% Horizontal Single shaft GT-ST configuration
Otto/Diesel Cycle	- IC engine coupled with generator - With Turbo charges	Efficiency < 40% Efficiency < 40-42% Air & oil pre-heating with exhaust gas.

Out of the above configurations, Kelanithissa plant has open cycle gas turbines and one combine cycle power plant. Kerawalapitiya plant consists of combined cycle gas turbine while Sapugaskanda plant consists of diesel cycle IC engines with turbo chargers.

3.0 WASTE HEAT RECOVERY

In power generation more than 50% of fuel energy is emitted as waste heat to environment. Lots of concepts and technologies have been invented and currently used to recover wasted heat. The amount of energy recovered depends on many factors, including waste heat temperature, quantity, accessibility, quality/cleanness, corrosiveness and intend use. These factors often determine the viability of recovering the WH as emission free energy source which affect for greater plant efficiencies and minimize the operation cost.

3.1 Waste Heat Definitions and Classifications

Waste heat can be defined as the thermal energy generated during a certain process, but is dumped into the environment without utilizing [6].

Waste heat recovery (WHR) can be defined as capturing, converting and utilizing the WH to do a useful work [6]. Process of WHR can be classified based on the type of use;

- Waste heat to heating - Utilizing the WH for heating purpose in the process to reduce the heating cost.
- Waste heat to cooling and refrigeration - Utilizing the WH for cooling purposes by means of absorption systems to reduce the cooling costs.
- Waste heat to power - Utilizing the WH for electricity generation by means of steam turbine, organic Rankine cycle and other technologies to reduce the electricity costs.

WHR systems in Thermal Power Plants

Thermal power plants produce large amounts of WH due to operating nature of thermodynamic cycles. Approximately 2 MW is discharged in the form of WH when producing each MW of electricity generated. Common practice for WH handling without recovering, involved heat rejection to lakes, streams or use of cooling towers, which are well established methods that offer reliable operation of the system. However energy lost as WH cannot be fully recovered and recoverable amount will depend on the;

- Quality of waste heat
- Quantity of waste heat

Quality of waste heat

Depending on the type of process, WH is rejected at various temperatures from very low values to very high values. Usually higher the temperatures, higher the waste heat quality which can be recovered more cost effectively.

Quantity of waste heat

It is essential to know in any heat recovery situation, the amount of heat recoverable and how it can be used. On the other hand quantity refers to the amount of available heat to be recovered.

Bottoming cycle, topping cycle and combined cycle are the commonly used methods for heat recovery purpose. A thermodynamic cycle which generates electricity from waste heat is called a bottoming cycle. In combined cycle these cycles are combined for electricity generation by connecting 02 heat engines in series [6].

3.2 WH Classification Based on Temperature

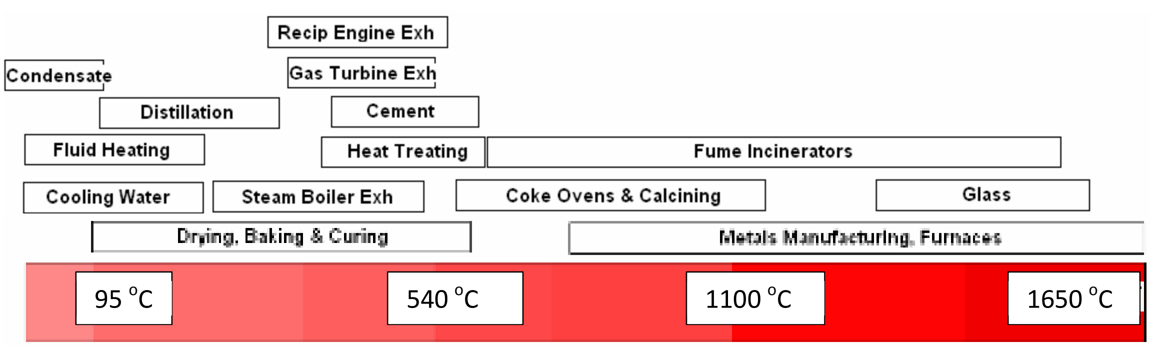

Figure 3.1: Waste heat source classification based on temperature [7]

Many of the industries require large quantities of thermal energy, much of which is eventually exhausted to the environment, either to the atmosphere or water. Recovering this waste heat represents the largest opportunity for reducing industrial energy consumption in the world. Since the majority of waste heat sources have temperatures less than 600 ºC, it is especially important that we implement technologies suitable for recovering those waste heat opportunities [7]. Figure 3.1 shows the temperature ranges of common industrial waste heat sources.

The old rule of thumb that industrial heat recovery is cost effective only for temperatures of at least 540 ^{o}C is not true today with increasing energy prices, technological development by equipment manufacturers and decreasing equipment costs. However, economic feasibility of investing in WHR system can be determined approximately through basic level by calculating associated simple payback period. Here, if simple payback is less than year 1 to 5, then a project is recognized as viable for investment. Further economical evaluations can be very much site specific & complicated, hence qualified specialist familiar with these systems can ensure proper calculation of benefits of the system.

Table 3.1 shows, a classification of industrial waste heat based on their source temperature and characteristics. According to the waste heat source, qualitative and quantitative factor will vary; hence, adoptable recovery method shall vary accordingly. According to Table 3.1, high temperature waste heat sources are the furnaces from metal industry where the temperatures are likely to have around 1000 ^{o}C or above. Exhaust heat temperatures of thermal power generation usually below 600^{o}C belong to medium and low grade heat based on the classification. The medium grade exhaust heat power plants are Gas Turbines and Reciprocating engines. Further, waste heat from steam plants and combine power plants are considered to be low grade waste heat [8].

When temperature reduces, quality of the heat source reduces, so heat recovery will be more difficult and less economical. Hence, novel technologies and systems are required to harness energy from low grade sources which are listed in the last column of Table 3.1.

Table 3.1: Classification of Waste Heat by temperature [8]

Temperature Classification	**Waste Heat Source**	**Characteristics**	**Commercial WH to Power Technologies**
High (> 650 ^{o}C)	• Furnaces – Steel electric arc – Steel heating – Basic oxygen – Aluminium reverberator – Copper reverberator – Nickel refining – Copper refining – Glass melting • Iron cupolas • Coke ovens • Fume incinerators • Hydrogen plants	• High quality heat • High heat transfer • High power generation efficiencies • Chemical and mechanical contaminants	• Waste heat boilers and steam turbines
Medium (260 – 650 ^{o}C)	• Prime mover exhaust streams – Gas turbine – Reciprocating engine • Heat treating furnaces • Ovens – Drying – Baking – Curing • Cement kilns	• Medium power generation efficiencies • Chemical and mechanical contaminants (some streams such as cement kilns)	• Waste heat boilers and steam turbines (>260 ^{o}C) • Organic Rankine cycle (<425 ^{o}C) • Kalina cycle (<540 ^{o}C) • Absorption Cycle
Low (< 260 ^{o}C)	• Boilers •Steam condensate • Ethylene furnaces • Cooling Water of; – Furnace doors – Annealing furnaces – Air compressors – IC engines – Refrigeration condensers – Glass melting • Low temperature ovens • Hot process liquids or solids	• Energy contained in numerous small sources • Low power generation efficiencies • Recovery of combustion streams limited due to acid concentration if temperatures reduced below 120 ^{o}C	• Organic Rankine cycle (>150 ^{o}C gaseous streams, >65 ^{o}C liquid streams) • Kalina cycle (>95 ^{o}C)

.

3.3 Waste Heat Recovery Classification

Heat recovery options can be broadly classified into three strategies:

- Recycling energy back into the process
- Recovering energy for other on-site uses
- Using it to generate electricity in combined heat and power systems

Recycling the waste heat energy back into the process is mostly done by passive recovery methods while combined heat and power is produced by active recovery methods. Figure 3.2 shows the basic classifications of waste heat recovery systems for thermal power plants.

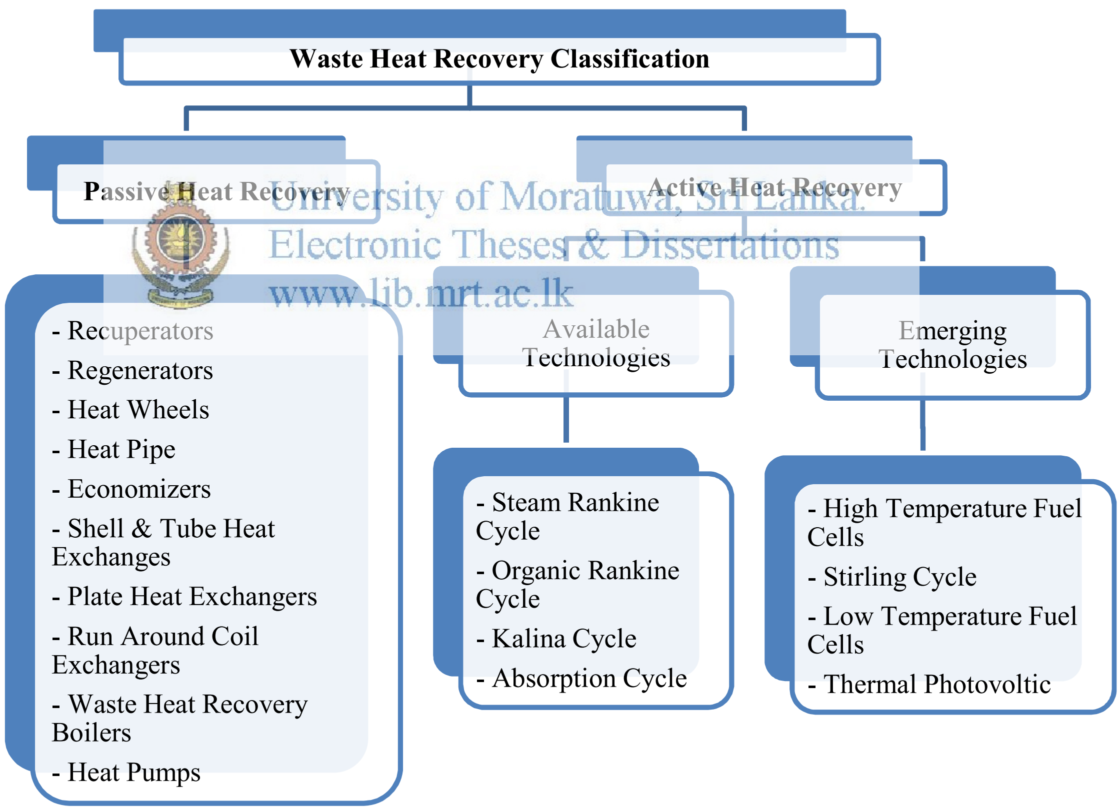

Figure 3.2: Waste heat recovery method classification

Passive heat recovery makes use of heat exchangers of various types to transfer heat from a higher temperature source to a lower temperature stream. Passive heat recovery technologies do not require significant mechanical or electrical input for their operation, except for auxiliary equipment such as pumps or fans. Active heat recovery technologies on the other hand require the input of energy to “upgrade” the waste heat to a higher temperature or to electricity. These technologies include industrial heat pumps and combined heat and power systems.

The recoverability of waste energy is largely determined by source temperature and WHR systems are manufactured to operate at their appropriate temperature regimes. Other considerations are the flow rate, its availability over the course of the day and year, and the fouling characteristics of the exhaust. Further, the following classification on WHR technologies is based on WH temperature.

Heat Recovery Technology Classification [9][10]

- Passive heat recovery: Temperatures greater than 95 ºC
- Industrial closed-cycle mechanical heat pumps: Temperatures less than 95 ºC
- Absorption chillers and heat pumps: Temperatures between 95 ºC and 200 ºC
- Organic Rankine Cycle, Combined Heat & Power (CHP): Typically 150 ºC to 400 ºC
- Kalina Cycle, CHP: 120 ºC to 540 ºC

Passive heat recovery systems have been in the industry for long time and most of them are well known and developed. Most of the active heat recovery systems are recently developed with modern technologies and some of them are still at an emerging state. In the power sector, passive systems are heavily utilized to increase the overall performance in plants, and now it is focusing on modern technologies where active heat recovery systems are introduced to incorporate for further optimization of energy utilization. This research study is mainly focused on active heat recovery technologies that can be combined with waste heat of thermal power plants of Sri Lanka to increase system efficiencies.

3.4 Low Grade Heat Recovery Cycles

Low temperature heat recovery systems are less economical due to high cost and less effectiveness. Passive systems are used under limited conditions. Hence, active heat recovery systems have been researched and developed.

Recovering the thermal energy from low grade energy sources and producing electricity is not profitable with conventional steam Rankine cycles. Hence, many low grade heat recovery cycles have been developed. The following thermodynamic cycles have been introduced for low temperature heat recovery [11];

- Organic Rankine Cycle
- Kalina Cycle
- Goswami Cycle
- Trilateral Flash Cycle

These cycles offer low equipment cost, high effectiveness and higher profit by using other working fluids than pure water. Following paragraphs will briefly discuss about each cycle, and their unique features.

3.4.1 Organic Rankine Cycle (ORC)

ORC has the same working principle and components similar to steam Rankine cycle. The main differences in ORC's are the working fluids and heat source temperatures. On the other hand ORC can extract energy from lower heat source temperature and produce electricity than traditional Rankine cycle [12].

There are three types of ORC systems depending on the four thermodynamic processes of heat addition, expansion, heat rejection and compression.

- Subcritical ORC

  Here, four thermodynamic processes in the cycle occur at pressures lower than the critical pressures of working fluid.

- Trans – Critical ORC

  Here, heat rejection process occurs at a pressure lower than the critical pressure and heat addition occurs at a pressure higher than critical pressure. The other processes such as compression and expansion occur between the two pressure levels.
- Super – Critical ORC

  Here, four thermodynamic processes in the cycle occur at pressures higher than the critical pressures of working fluid.

Lot of attention has been paid on ORC in recent years in particular due to the fact that depletion of fossil fuels and global warming has increased the interest on low grade energy recovery.

ORC have several advantages over conventional steam cycle such as;

- Less heat is required to evaporate the organic fluid as the evaporation is taken place at lower pressures and temperatures.
- Often, the expansion process ends at vapour region, superheating is not desirable.
- The risk of blade erosion due to vapour condensation is avoided.
- Pressure ratio is smaller as temperature difference between evaporator & condenser is small. Hence related cost is less.
- As smaller pressure ratios, simple single stage expander turbines can be used.

ORC have wide variety of applications which depends upon the working fluid where heat can be extracted from waste heat of thermal power plants, biomass combustion, and geothermal, solar and industrial waste heat.

3.4.2 Kalina Cycle

Kalina cycle introduced in 1984 by Alexander I. Kalina, can be successfully used to convert low grade heat into electrical power [13]. The system comprises additional components such as recuperator, separator and absorber compared to conventional Rankine cycle. Mixture of two working fluids called as binary fluid is used for this cycle. Water and ammonia mixture is the most commonly used fluid. Reason for using a binary fluid is to reduce the thermodynamic irreversibility in the process and increase cycle efficiency. Thermodynamic irreversibility will reduce as ratio of two components in the working fluid varies at different location. Further, when irreversibility reduces, overall thermodynamic efficiency will be increased. Non-isothermal boiling will take place in the boiler as the working fluid has the ability to shift the mixture composition during heat absorption. As a result fluid will have good thermal match [14].

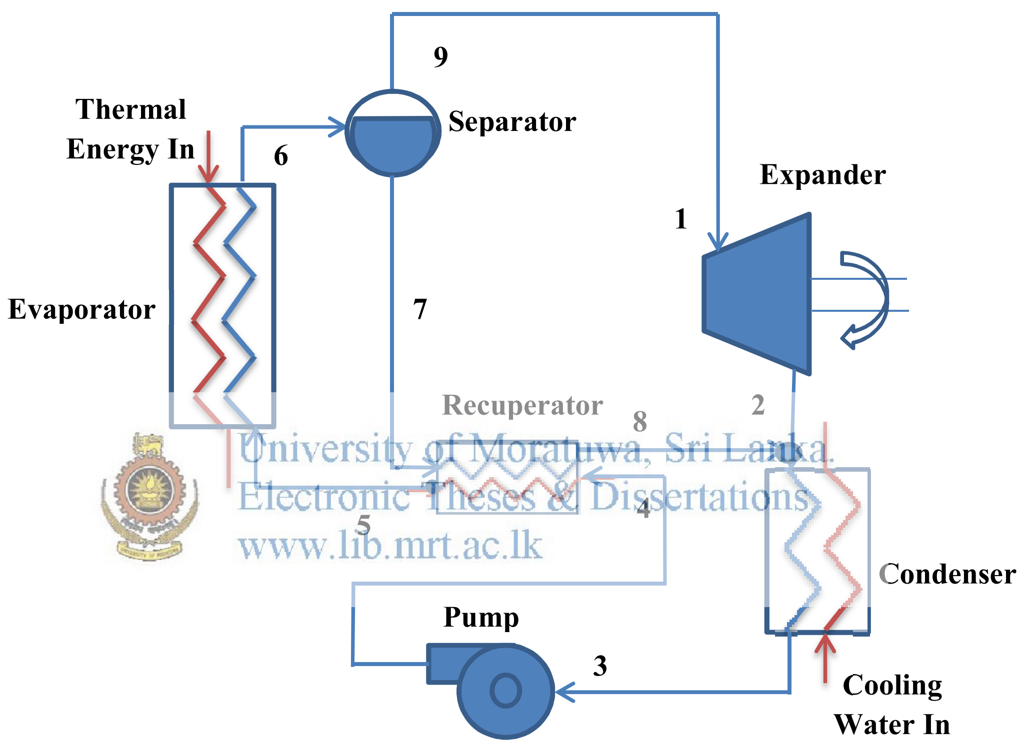

Figure 3.3: Basic configuration of Kalina cycle [13]

The ammonia-water mixture is heated in the evaporator (5-6). Working fluid is separated into ammonia-rich vapour mixture (9) and weak liquid mixture (7) in the separator. Ammonia-rich vapour pass through turbine (1-2) and expand generating electricity. Weak liquid mixture passes through recuperator (7-8) transferring considerable amount of thermal energy to concentrated fluid pumped to evaporator (4-5). Weak fluid coming out from recuperator (8) mixed with working fluid coming out from turbine (2) before entering the condenser. Condenser is cooled by external system and saturated condensate fluid coming out from condenser and fed to pump (3) for cycle circulation [13][15][16].

At the beginning, many scientists including the inventor have shown the theoretical advantages of Kalina cycle over Rankine cycle where thermal efficiency reported 10 to 60% compared to steam plant [17][18]. Comparison of Kalina cycle with Rankine cycle on WHR applications, favourable results for Kalina cycle in terms on power production, but cost is high compared to ORC, as the cycle pressure increases, surface requirement also increases for evaporator [19].

Very small numbers of plants are operated commercially in power generation based on Kalina cycle principle in the world and table 3.2 shows details of some plants.

Table 3.2: Kalina cycle case-studies [16]

Se. No.	Name	Country	Commissioned	Output (MW)	Heat Source
1	Canoga Park	USA	1992	6.5	Nuclear waste heat
2	Fukuoka	Japan	1998	4	Waste incineration
3	Sumitomo Metals	Japan	1999	3.5	Waste heat
4	Husavik	Iceland	2000	2	Geothermal
5	Fuji oil	Japan	2005	3.9	Waste heat
6	Bruschal	Germany	2009	0.6	Geothermal
7	Unterhaching	Germany	2009	3.5	Geothermal
8	Shanghai Expo	China	2010	0.05	Solar hot water
9	Quingshui	Taiwan	2011	0.05	Geothermal

3.4.3 Goswami Cycle [11]

A novel thermodynamic cycle called Goswami cycle was proposed by Dr. Yogi Goswami in 1998 [10], which uses binary mixture to produce electricity and cooling effect simultaneously. The principle of the cycle is combination of Rankine cycle and absorption cooling cycle. System is running on binary mixture fluid, and most commonly used water with ammonia mixture. Unique advantages of Goswami cycle can pointed out as follows;

- Generate power and cooling effect at the same time.
- Flexibility of varying the power generation based on requirement while cooling effect varies and vice versa.
- Better utilization of energy sources when both power and cooling is required.
- Efficient conversion of moderate temperature heat sources to power.

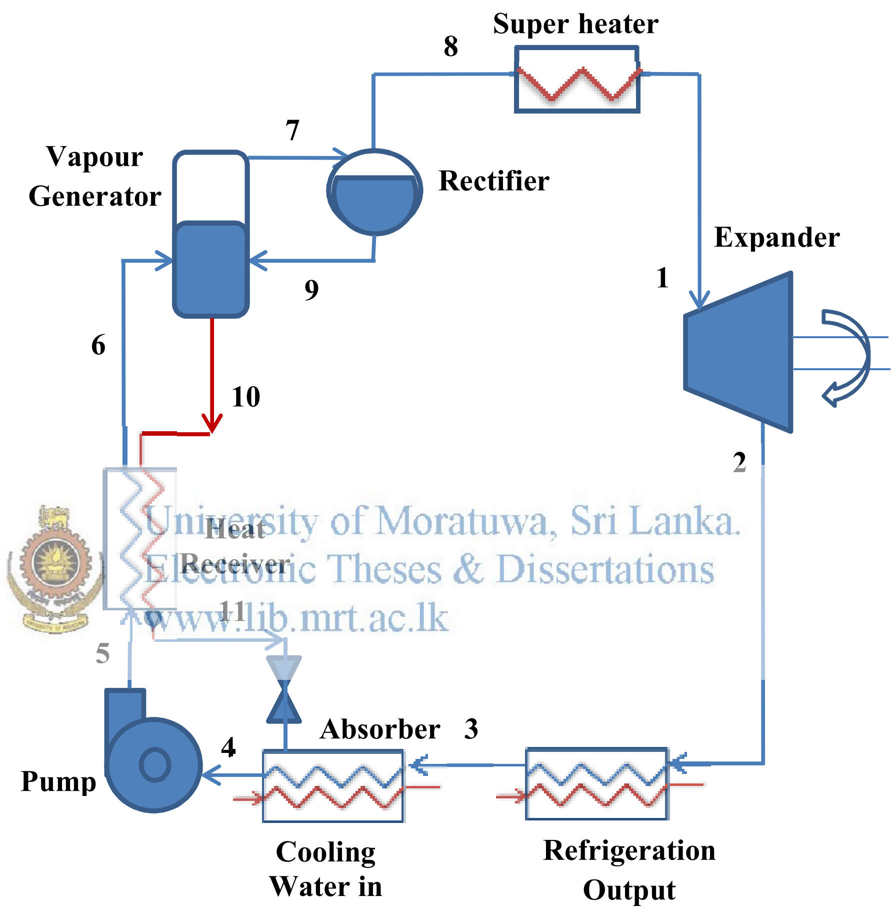

Figure 3.4: Basic configuration of Goswami cycle

Binary fluid mixture is pumped to high pressure (4-5) and pre-heated from lean solution returning from the vapour generator (5-6). Then mixture is sent to the vapour generator where ammonia vapour is generated (6-7) and passes to rectifier. The rectifier is used to purify the vapour (7-8) by condensing the water if needed (9). Then vapour is sent to super heater (8-1) and to turbine/expander (1-2). Since the working fluid is condensed by absorption (2-3), this can be expanded to temperature other than ambient. This will provide cooling effect in addition to power generation. Remaining lean solution from boiler/vapour generator is throttled and fed into the absorber (11).

The conceptual cycle is still at research stage and no commercial application is reported in the world. However, an experimental setup has been established in the research park of the University of South Florida.

3.4.4 Trilateral Flash Cycle

Trilateral Flash Cycle (TFC) is a system where thermodynamic expansion starts from the saturated liquid rather than the saturated, superheated or supercritical vapour phase. The expansion process will undergo saturated liquid to liquid-vapour (two phase) region in the expander. System potential power recovery could be 14 – 85% more than from ORC or flash steam cycle provided that the two-phase expansion process is efficient [23].

In TFC system, working fluid is heated up to its boiling point only and then expands it as a two-phase vapour through expander. Even though TFC system is theoretically efficient, but developing of efficient expander for two-phase flash has been the main drawback for practical implementation. However extensive research and development are in progress in the world. The layout components are shown in figure 3.4 which are more similar to Rankine cycle.

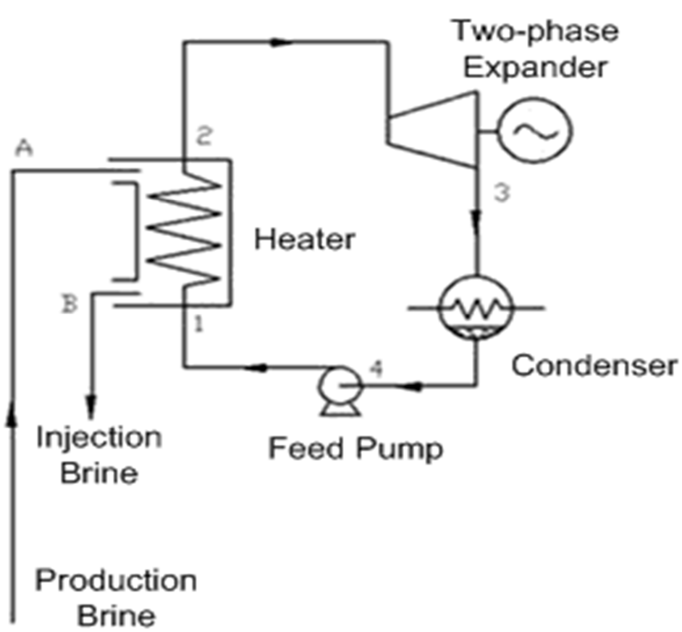

Figure 3.5: Layout of Trilateral Flash cycle [23]

The most promising feature of TFC is that there is a perfect thermal match between heat source and heat recovery fluid. This can be clearly understood by referring the A-B line and 1-2 line in figure 3.6. A-B line denotes the temperature reduction in heat source while 1-2 line denotes the temperature increment in heat recovery. Significantly, both these line are almost parallel in TFC.

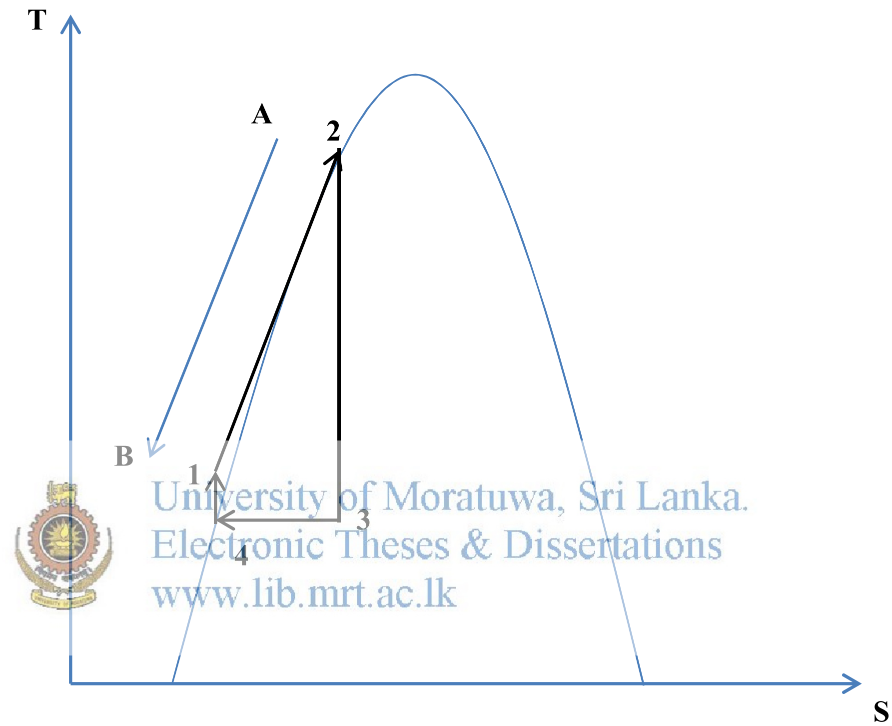

Figure 3.6: T-S diagram of Trilateral Flash cycle

Other significant feature of TFC is the very high reversibility. The working fluid is heated under pressure to a temperature above boiling point. The expansion phase in the expander starts from saturated liquid state and flashes to the condenser pressure. Further, cycle has most perfect temperature match compared to other cycles. Hence, TFC is considered as a high reversible process.

Even though, TFC has some distinguishing features; the concept has been considered for over 30 years. The main failure is to find expander that can operate under two phase working fluid while maintaining high adiabatic efficiency.

3.5 Selection of Thermodynamic Cycle

All the thermodynamic cycles discussed above, have desirable characteristics and drawbacks in terms of adopting for waste heat recovery. Further, this research scope is confined to an investigation of the possibilities for converting waste energy to power on existing thermal power plants. Hence, it is required to select best concept, among previously discussed systems for further analysis technically and economically.

First of all, further consideration of Trilateral Flash cycle is not worth due to the absence of efficient two-phase expander up to now [20]. Goswami cycle is still in the research state; hence commercial viability in the industry is yet to be tested in the future [21]. Hence, further discussion on TFC and Goswami cycle will be discontinued from here on. The concept of ORC and Kalina cycle vie with one another for supremacy in performance. Commercially, ORC is a more established concept worldwide and researches are continuing for higher efficiencies. Further, lots of plants are in under operation and the number of plants continues to expand in future. On the other hand, concept of kalian cycle has now started to get popular. At an early stage of Kalina concept, researches showed positive signs in terms of efficiencies compared to ORC. But, recent investigations haven't shown positive results, even though the potential is there [22].

Table 3.3 has compared the TFC, ORC and Kalina cycle under same conditions for better analysis. Here ORC has been evaluated for four organic fluids. According to the following table, ORC have shown better results than the Kalina cycle [24].

Table 3.3: Comparison of TFC, ORC and Kalina cycle [24]

Cycle (Fluid)	η (%)	P (kW)	Q (kW)
ORC (R141b)	10	13	132
ORC (R123)	9	17	179
ORC (R245ca)	9	18	189
ORC (R21)	9	18	198
Kalina (NH3-H2O)	3	13	373
TFC ((NH3-H2O)	8	38	477

Thermodynamic performance of the Kalina cycle and ORC in the case of heat recovery has been evaluated for Wartsila 20V32 8.9 MW diesel engine for exhaust gas where there is a mass flow of 35 kg/s at 346 ºC. Almost equal cycle efficiencies were obtained 19.7% and 21.5% for Kalina cycle and ORC respectively [25]. But, Kalina system operated at very high pressures and required very high rotational speeds for turbine compared to ORC. Further, higher ratio gear box is required when connecting to generator for Kalina cycle. Hence, in cost comparison, ORC is better.

Based on above information's and factors, Organic Rankine cycle was selected for further study in this research.

4.0 ORGANIC RANKINE CYCLE

Organic Rankine cycle (ORC) has become a field of intense research and appears as a promising technology for conversion of low grade heat into electricity. It is exactly the steam Rankine cycle, except that the working fluid in the system will be refrigerants or hydrocarbons. The operating costs of the ORC system is strictly linked to the thermodynamic properties of the working fluid.

4.1 Properties of Working Fluid

Selection of working fluid is the most critical factor of ORC for efficient and economic operation. Desirable features/properties for ideal working fluid are as follows;

- Thermal efficiency - Should be high as possible.
- Condensing pressure - Should be higher than the atmospheric pressure to avoid leakage issues.
- Specific volume and density - Low specific volume and higher density.
  Higher fluid density will cause lower the specific volume and low volumetric flow rate. On the other hand, low fluid density will have high specific volume and large volumetric flow rate which requires bigger components (cost will be increased). Additionally, pressure drop also increases with specific volume in heat exchanger and need more power for the pump. Hence, low volumetric flow is desirable to achieve smaller component and more compact machines.
- Fluid cost - Low cost.
- Saturated vapour line - Positive or infinite slope is desirable.
  For dry and isentropic fluids, saturated vapour line will be positive and infinite respectively. During expansion, formation of droplets will not occur for dry and isentropic fluids which are desirable for expander life time.

- Specific heat capacity - High specific heat capacity is desirable.
  High heat capacity leads to recover energy effectively while decreasing the fluid mass flow rate.
- Enthalpy variation - Large enthalpy variation.

Higher enthalpy variation during the expansion leads high work output.

- Toxicity - Low toxicity for safety.
- ODP & GWP - Desirable to have low ODP and low GWP

  When considering the environment aspect low Ozone Depletion Potential and low Global Warming Potential is ideal for the fluid.
- Chemical & thermal status - Desirable to have chemically and thermally stable fluid.

4.2 Fluid Classification Based on T-S diagram

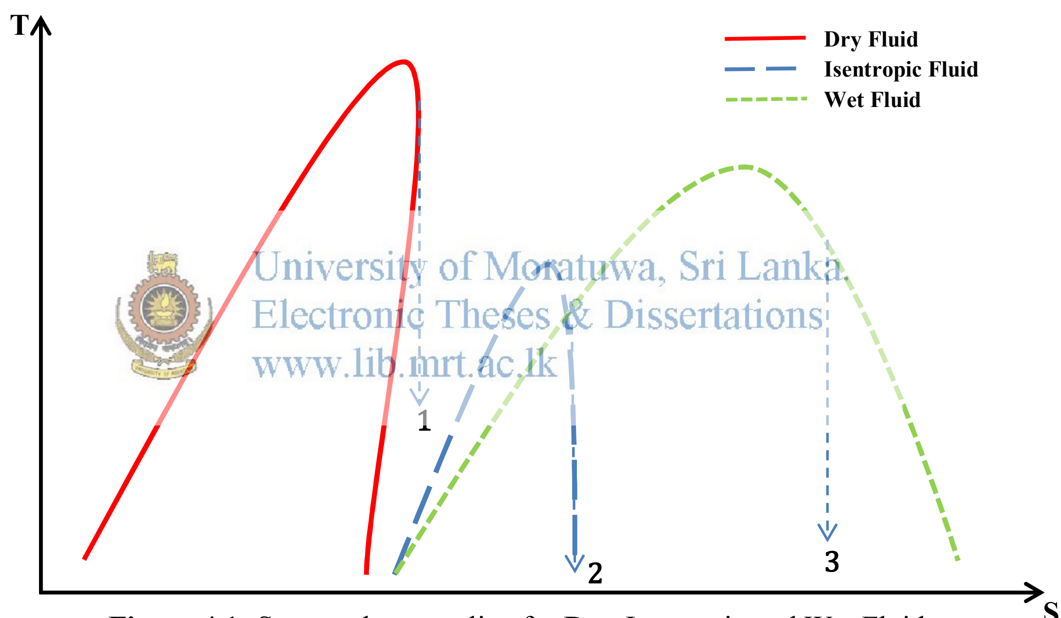

Figure 4.1: Saturated vapour line for Dry, Isentropic and Wet Fluids

According to saturated vapour line, organic fluids can be classified as dry, wet & isentropic (please refer figure 4.1). This classification is based on the slope of the saturated vapour line. When saturated vapour line slope is positive, droplet formation in the expansion is avoided and in negative slope, droplet will be formed in expansion. Based on this, fluids are classified as dry and wet respectively.

Position 1 in the figure denotes the fluid state during the dry fluid expansion, which located on superheated region. On the other hand, position 2 denotes fluid state during wet fluid expansion, which located on liquid-vapour region. Further, the fluids having infinite slope is considered as isentropic fluid as the entropy remains unchanged during the expansion. Consider the blue dash line in the figure, during expansion; the fluid will come to position 2 which located on saturated line.

Formation of droplets by wet fluids during the expansion process cause serious damages to blades of the expander. Hence, superheating is required to avoid the droplet formation. Usage of dry or isentropic fluid will eliminate this problem.

In the selection of working fluid, it is very important to consider the slope of saturated vapor line in the T-S diagram. According to slope of the saturated line, working fluids can be categorized as follows;

- Wet fluids - Fluids that have the negative slope in saturated line, commonly considered as wet fluids.

(eg. Heavy water, Ethanol, Methanol, R21, Sulfur dioxide, etc.)

- Dry fluids - Fluids that have the positive slope in saturated line, commonly considered as dry fluids.

(eg. Toluene, Decane, Nonane, Octane, Heptane, Cyclohexane, Hexane, R113, etc.)

- Isentropic fluids - Fluids that have infinite slope in saturated line are considered as wet fluids.

(eg. R142b, R11, R141b, Cis-butane, Acetone, etc.)

Influence on overheating

Overheating or superheating is used in conventional steam Rankine cycle in order to improve the vapour quality during the operation when it leaves the expander. This ensures that condensation will not occur in the fluid before leaving the expander which cause serious issues for expander. On the other hand, low vapour quality leads to drop formation in the final stages of the expansion process.

Overheating an ORC increases the thermal efficiency at a very low slop, but more significantly decreases the efficiency of second law of thermodynamics. Further, overheating increases the cycle pressure which increases the investment cost of the system. Hence, superheated cycles are not recommended unless to gain more power at the expense of losing efficiency [26]. *Because of that, for all the fluids, feasibility of saturated Rankine cycle was investigated in this research*.

Critical pressure

Concerning a critical pressure of a particular fluid, a small change in temperature causes large change in pressure difference. This large change in pressure difference near to critical point may cause instability in the system. Because of this, cycles are developed in such a way that considerable pressure difference required is maintained from the fluid's critical point. Hence, during the analysis, 4 bar pressure difference from fluid's critical point were maintained.

4.3 ORC Configuration

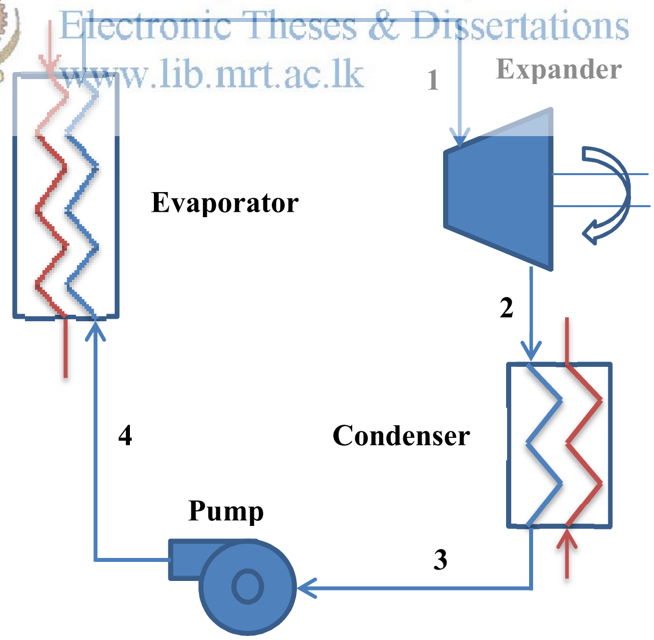

Figure 4.2: Basic configuration of ORC system

Figure 4.2, shows a general representation of the actual saturated basic Organic Rankine cycle configuration, consisting of expander, condenser, evaporator (heat recovery unit) and working fluid pump. Heat from different waste heat sources is pumped into the evaporator. Through the evaporator, an organic fluid is circulated and certain amount of heat in the heat source is transferred to organic fluid. This organic fluid comes out as saturated vapour phase from evaporator. This saturated vapour is fed into the expander and drives it to generate electricity while reducing the pressure and temperature. The low temperature and low pressure organic fluid is cooled to liquid phase when passing through the condenser. The liquid pump sends the organic liquid to evaporator for heat absorption. In this way, above process is repeated in the cycle.

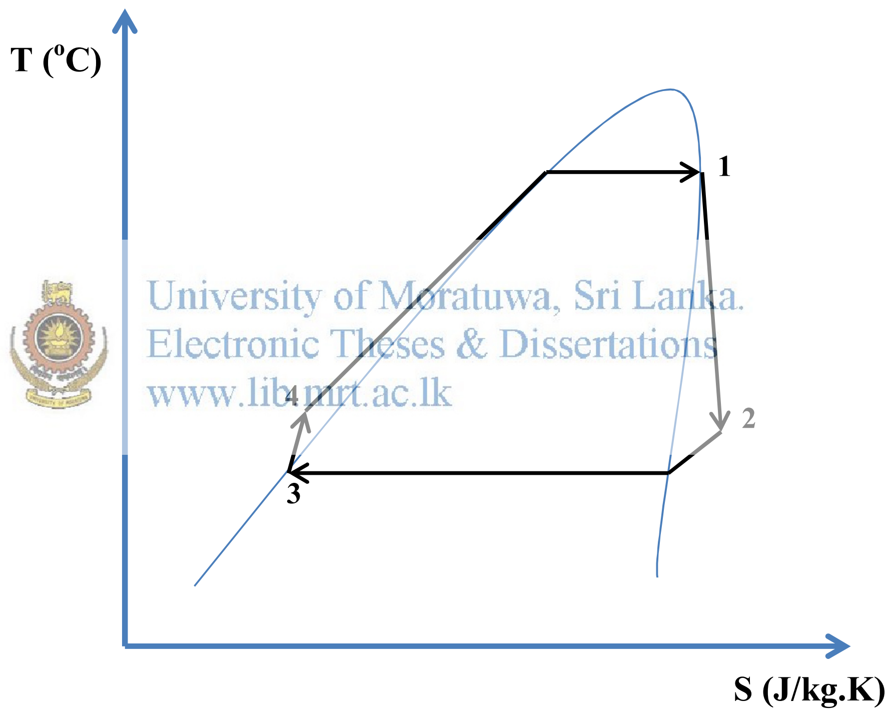

Figure 4.3: Basic T-s diagram for ORC system

ORC heat recovery plant converts heat into electrical power through four thermodynamic processes shown in Figure 4.3. In process 1 – 2, organic fluid passes through the expander to generate mechanical power. Ideally this process is an isentropic process, but in actual case isentropic efficiency (η_{is}) is not equal to 100%. Exhaust organic vapour fed into the condenser where it cooled to liquid in isobaric process 2 – 3. In process 3 – 4, condensed liquid is pressurized and sent to the evaporator. In the evaporator, heat absorption takes place from the heat source denoted as process 4 – 1. During the heat absorption, liquid fluid is transferred to saturated vapour phase in the outlet.

4.4 System Modelling

In this model following assumptions are made;

1. This research focuses only in the performance of ORC under steady stable conditions.
2. Isentropic efficiency for expansion process 1 – 2 is assumed as 75 % [27][28][29]
   (η_{is} = 0.75). Note that expander efficiencies vary between 70-85% in practice.
3. For simplification, pressure drop across pipeline, heat exchanger and condenser assumed as zero for all operating conditions.
4. It is assumed that the mechanical efficiency of working pump, expander and generator is 75%, 96% and 98% respectively. It's further assumed that feed pump work is isentropic.
5. For the calculation and comparison purposes, mass flow rate of the organic fluid is taken as 0.5 kg/s for all fluids at all conditions. Because, volume rate is high for these organic fluids and boiling temperature is low. Further, high volume rate will reduce the effectiveness of the heat recovery unit. Hence, above mass flow rate was taken for theoretical ORC modelling and actual mass flow rates were calculated for different selected scenarios.

Modelling equations

Expander;

$$\dot{W}_{exp} = \dot{m} \times (h1 - h2) \times \boldsymbol{\eta}_{is} \qquad \text{-------------} \quad 4.1$$

$$\boldsymbol{\eta}_{is} = {\dot{W}_{exp}}/{\dot{W}_{exp.is}} \times 100\% \qquad \text{-------------} \quad 4.2$$

Evaporator;

$$\dot{Q}_{evap} = \dot{m} \times (h1 - h4) \qquad \text{-------------} \quad 4.3$$

Working pump;

$$\dot{W}_{pump.is} = (P4 - P3) \times \dot{m}/\boldsymbol{\rho}_{liquid} \qquad \text{-------------} \quad 4.4$$

Equation 4.4 denotes standard power equation for pump, which uses the flow rate, density and pressure difference or rise in pressure. This equation denotes the ideal power requirement for pump.

Nevertheless, actual power requirement of a pump can be defined in terms of the pump efficiency;

$$\boldsymbol{\eta}_{pump} = \dot{W}_{pump.is}/\dot{W}_{pump} \times 100\% \qquad \text{-------------} \quad 4.5$$

Practically, pump efficiency is less than 100%, as the part of energy that goes to raise the temperature of outlet fluid. Hence, the above equations can be combined for actual pump requirement;

$$\dot{W}_{pump.is} = (P4 - P3) \times \dot{m}/(\boldsymbol{\rho}_{liquid}) \times (\boldsymbol{\eta}_{pump.is}) \qquad \text{--------} \quad 4.6$$

Based on above assumptions & calculation simplicity, pump total work can be defined as;

$$\dot{W}_{pump} = \dot{m} \times \mathcal{V}_{pump} \times (P4 - P3) \times 100/\boldsymbol{\eta}_{pump.is} \qquad \text{--------} \quad 4.7$$

Efficiency;

Cycle efficiency can be calculated from total energy input (Heat energy absorbed & pump work) and work output. Work out will be expander work and work input to the system will be energy absorbed by evaporator plus energy consumed by the pump.

$$\dot{W}_{in} = \dot{Q}_{evap} + \dot{W}_{pump} \qquad \text{-------------} \quad 4.8$$

$$\boldsymbol{\eta}_{cycle} = \frac{\dot{W}_{out}}{\dot{W}_{in}} = \frac{\dot{W}_{exp}}{\dot{Q}_{evap} + \dot{W}_{pump}} \qquad \text{-------------} \qquad 4.9$$

Heat recovery efficiency;

Amount of heat recovered is depends on the efficiency of the evaporator. If evaporator is 100% efficient, the organic fluid will reach mean average temperature of the heat source.

$$\boldsymbol{\eta}_{evap} = \frac{\dot{Q}_{evap}}{\dot{Q}_{evap.max}} = \frac{\dot{m} \times \bar{C}_{p,f}\,(\boldsymbol{T}_{f,evap\ out} - \boldsymbol{T}_{f,evap\ in})}{\dot{m} \times \bar{C}_{p,f}\,(\boldsymbol{T}_{avg.\ source} - \boldsymbol{T}_{f,evap\ in})} \qquad \text{----} \qquad 4.10$$

5.0 RESULTS AND DISCUSSION

This chapter the fluid selection from both organic and refrigerant categories is elaborated. Those fluids were evaluated under modelled system by varying the evaporator temperature and expander pressure ratios. Work input and outputs were graphically reviewed and the summary was tabulated.

5.1 Selected Fluids

In the fluid selection, certain aspects mentioned in the previous chapter were considered. Accordingly, eleven commonly used fluids were selected [27], half from organic fluids and other half from refrigerant fluids which are mentioned below;

- Organic Fluids - Toluene, Benzene, Decane, Pentane, Heptane
- Refrigerant Fluids - R245fa, R123, R113, R245ca, R134a, R114

Fluids were selected from low boiling points to high boiling points to match the different grades of waste heat sources. Because, waste heat of thermal plants may vary from low grade heat to medium grade heat, suitable fluid has to be selected in heat recovery at the evaporator or waste heat boiler. Further, the thermal conductivity, molecular mass, environment safety, atmospheric life time, ozone depletion potential (ODP), global warming potential (GWP) and fluid viscosity are the key factors to be considered when selecting suitable working fluids. Environmental and safety factors should be considered in the selection as some of the organic fluids were globally banned due to their toxicity and unhealthy repercussions on environment.

Other most critical factor was the fluid classification as wet, dry and isentropic fluid, based on expansion. Dry or isentropic fluids are ideal to avoid droplet formation in expansion. Further, this research was carried out on thermodynamic properties of saturated vapour [30], as the overheating is not recommended for ORC [25]. Table 5.1 shows the physical, environmental and expiation details of selected fluids.

Table 5.1: Physical, Safety and Environmental Data of selected fluids

Se. No.	Fluid	Physical Data			Safety Data	Environmental Data			Expansion
		NBP (^oC)	Tc (^oC)	Pc (^oC)	ASHRAE Safety Group	Atm Life Time	ODP	GWP (100 yrs)	
1	**R 245fa**	14.9	154.1	36.4	B1	7.7	0	1050	Dry
2	**R 123**	27.82	183.68	36.62	B1	1.3	0.01	77	Dry
3	**Toluene**	110.6	318.6	41.26					Dry
4	**R 113**	47.58	214.06	33.92	A1	85	0.85	6130	Dry
5	**Benzene**	80.08	288.9	48.94					Dry
6	**R 245ca**	25.13	174.42	39.25		6.5	0	726	Dry
7	**Decane**	174.12	344.55	21.03					Dry
8	**R 134a**	-26.07	101.06	40.59	A1	14	0	1430	Dry
9	**R 114**	3.586	145.68	32.57	A1	190	0.58	9180	Dry
10	**Pentane**	36.06	196.6	33.7	A3	0.009	0	20	
11	**Heptane**	98.38	266.98	27.36					

Note: Tc – Temperature at critical point
Pc – Pressure at critical point
NBP – Normal boiling point
ODP – Ozone depletion potential
GWP – Global warming potential

According to physical properties of fluid Decane in table 5.1, it has the highest normal boiling temperature of 174.12 ^oC, highest critical point temperature of 344.55 ^oC and lowest critical point pressure of 21.03 bar. Refer to ASHRAE standard 34, the letter A denotes the lower toxicity and letter B denote higher toxicity in safety data of above table. The number 1.2 and 3 refer to flame propagation, number 1 means no flame propagation, number 2 means lower flame flammability and number 3 means higher flammability. Further, environment data of table 5.1 divided into atmospheric life time in years, ozone depletion potential and global warming potential for each fluid. Also fluid classification based on expansion state mentioned in last column.

5.2 Fluid Analysis

Basic mathematical modelling equations of Organic Rankine cycles were mentioned in the previous chapter. Based on that, eleven numbers of organic fluids were analysed in this research. The fluid analysis has been performed according to the following steps;

- Varying the evaporator temperature

Evaporator temperature denotes the exhaust temperature of the heat source. Hence, evaluating the temperature variation will represent the ORC performance under different source temperatures. Further analysis will easily provide maximum electrical and efficiency ranges of each fluid with optimum evaporator temperatures. These factors are critical when designing ORC for heat recovery. Also this analysis provides reference for selection of fluids based on heat source temperature.

- Varying the pressure ratio of the expander

When considering the expander performance, inlet and outlet pressures are key factors. For optimization of ORC, it is required to find best performing inlet and outlet pressure ratio. The analysis will provide optimum pressure ratios for each working fluid which can be compared with others. If the pressure ratios are high, component cost and pump energy consumption will be increased.

Note that, temperature analysis would provide guideline for selection of suitable fluid for ORC system, based on temperature. Further provide approximate power output and turbine in/out temperatures. On the other hand, pressure ratio would provide guide for turbine selection at component designing stage after selecting the suitable fluid.

Based on the analysis maximum efficiencies, optimum temperatures, optimum pressure ratios and work outputs were tabulated.

5.2.1 Analysis on Evaporator Temperature Variation

In this analysis the following were considered,

- Organic fluids were selected in such a way that boiling points were varied from low value to higher value.
- Basic criteria for condenser temperature to maintain it at least 15 ^{o}C higher than atmospheric temperature, which is 45 ^{o}C. Some of the fluids, liquid phase started at higher than 45 ^{o}C, hence, condenser temperature was adjusted accordingly to meet the minimum liquid phase temperature.
- Evaporator temperature was varied starting from condenser temperature to temperature just below the critical point of the fluid.
- Condenser pressure was maintained higher than atmospheric level to avoid air mixing in a possible leakage.
- The output of the evaporator was maintained as saturated vapour and also the input to expander. The output of the expander was maintained at low temperature saturated vapour which condensate in further cooling at condenser.
- Pump efficiency was taken as 0.75, expander isentropic efficiency was taken as 0.75, expander mechanical efficiency was taken as 0.96 and generator efficiency 0.98 for the calculation.
- Graphs such as work in, work output and efficiency were drawn for each fluid.

Based on above conditions, work input, work output and efficiencies graphs were plotted for each fluid (see appendix B). Then, combining the work output and efficiencies of all fluids, 02 separate graphs were drawn for the convenience of comparison with the temperature variation in evaporator. According to the analysis, fluid Decane has shown highest expander output and highest efficiency, related graphs are shown in Figure 5.1, 5.2 and 5.3.

Analysis of fluid Decane

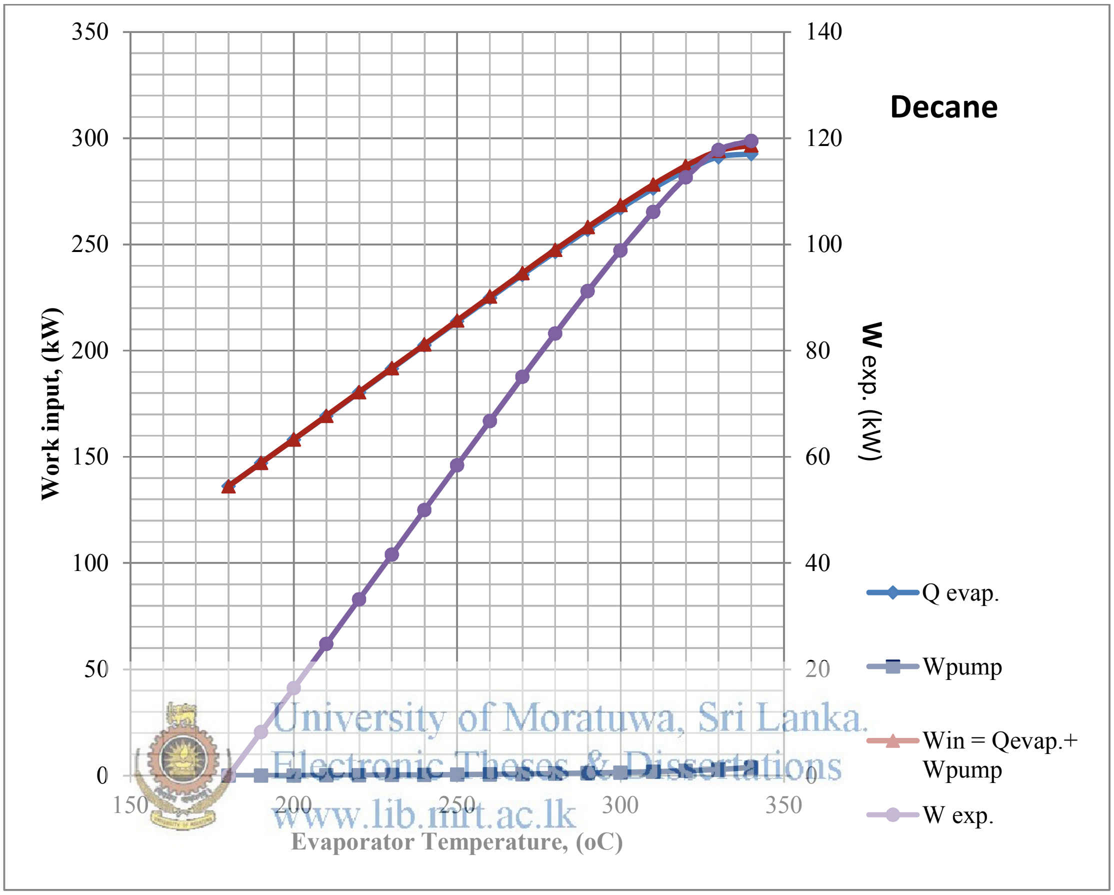

Figure 5.1: Input and expander work variation with different evaporator temperatures for Decane

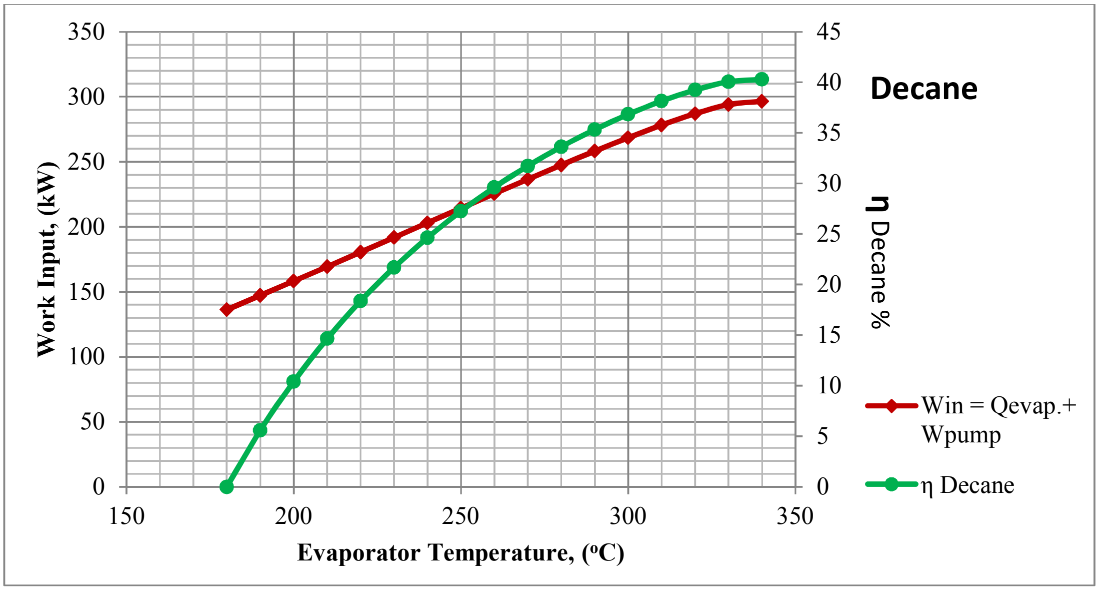

Figure 5.2: Work input and efficiency variation with different evaporator temperatures for Decane

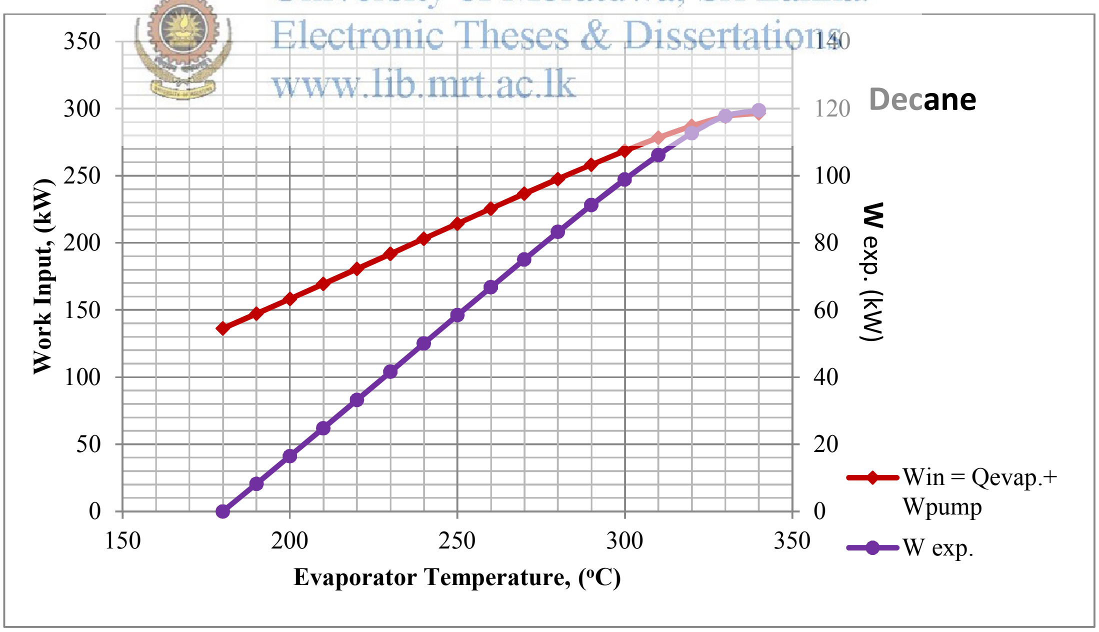

Figure 5.3: Input and expander work variation with different evaporator temperatures for Decane

Accordingly, expander output curves and efficiency curve of each fluid were plotted in combined graph of figure 5.4 and 5.6 respectively. Further, temperature axis of both figures varies between 45 ^{o}C – 340 ^{o}C range.

Combined analysis on work outputs

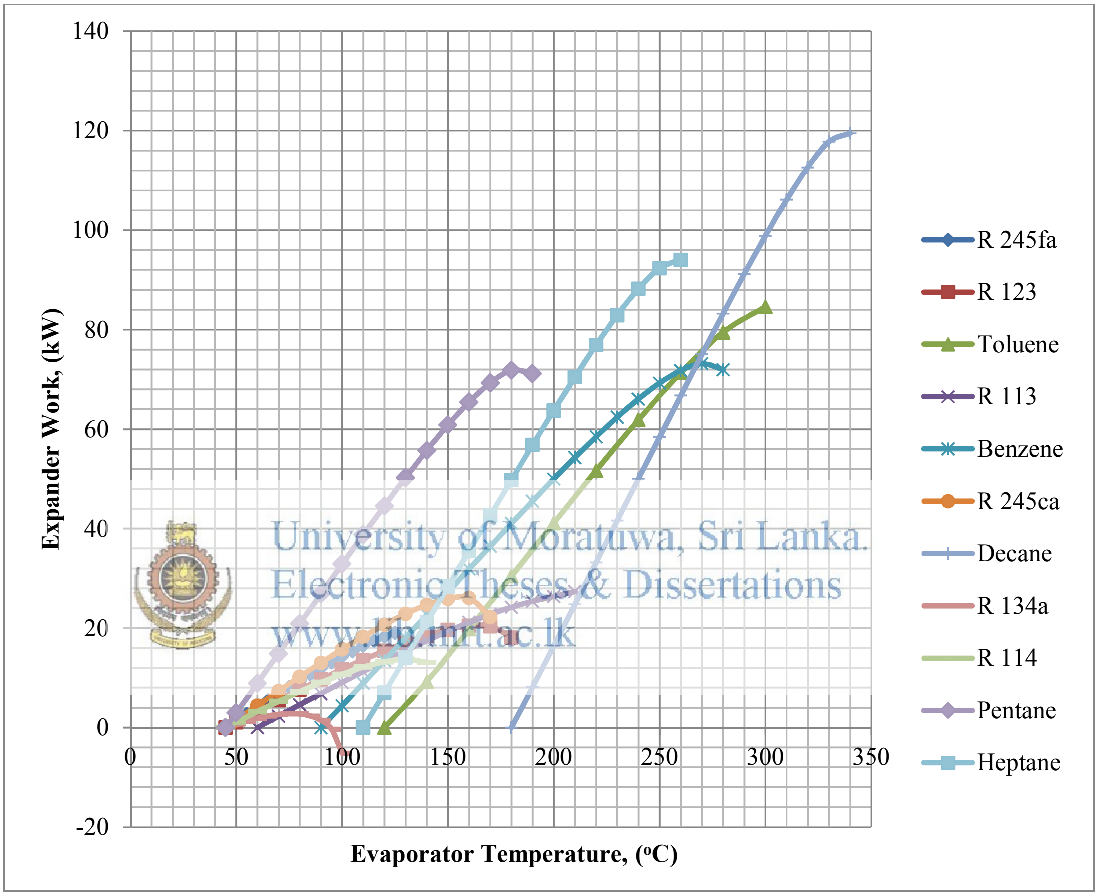

Figure 5.4: Work output variation with different evaporator temperatures

Following key points can be identified from the Figure 5.4;

- Generally work output increases with temperature increment. But fluid R245ca, R123 and R134a have shown a drop before coming to their flashing points.
- Further, organic fluids have shown good linear relationship with temperature and given higher output compared to refrigerant fluids.
- Maximum work outputs were equal or less than 120 kW for all fluids, where fluid mass flow rate was maintained at 0.5 kg/s. For refrigerant fluids, work

outputs were less than 30 kW and operating temperatures were less than 200 ^{o}C except R113.

- Generally, refrigerant fluids have shown the ability to operate at low temperatures (>120^{o}C), while organic fluids have shown the ability to operate at higher temperatures (<120^{o}C).
- However organic fluid Pentane has shown the ability to operate in both, low and high temperatures. Significantly, at low temperatures (>120 ^{o}C), work outputs were higher than refrigerant fluids.

Figure 5.5 shows a bar chart for maximum work output recorded for each fluid and respective output values were mentioned in kilowatt accordingly. In the chart, first six fluids from left hand side denote refrigerant fluids while next five denote organic fluids.

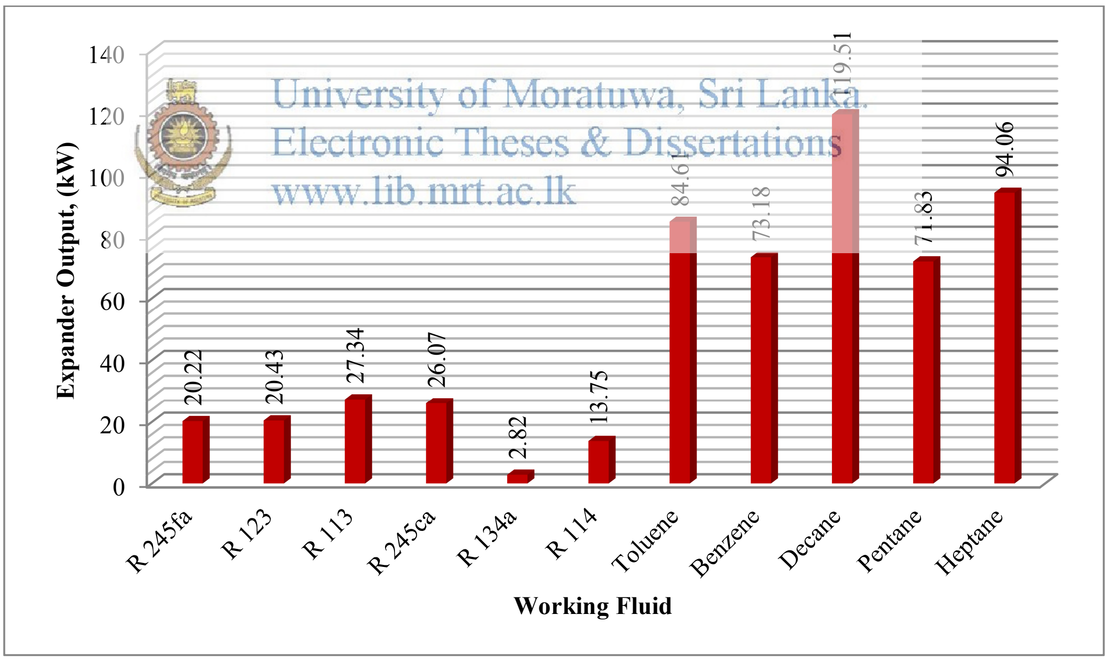

Figure 5.5: Graphical view of maximum possible work outputs of each fluid with temperature

According to Figure 5.5, Decane has the highest work output of 119.5 kW when the evaporator at 340^{o}C and R134a has the lowest output of 2.8 kW when evaporator temperature is around 75 ^{o}C.

Among refrigerant fluids, R113 has the highest work output of 27.3 kW when the evaporator temperature at 210 °C.

Combined analysis on efficiencies

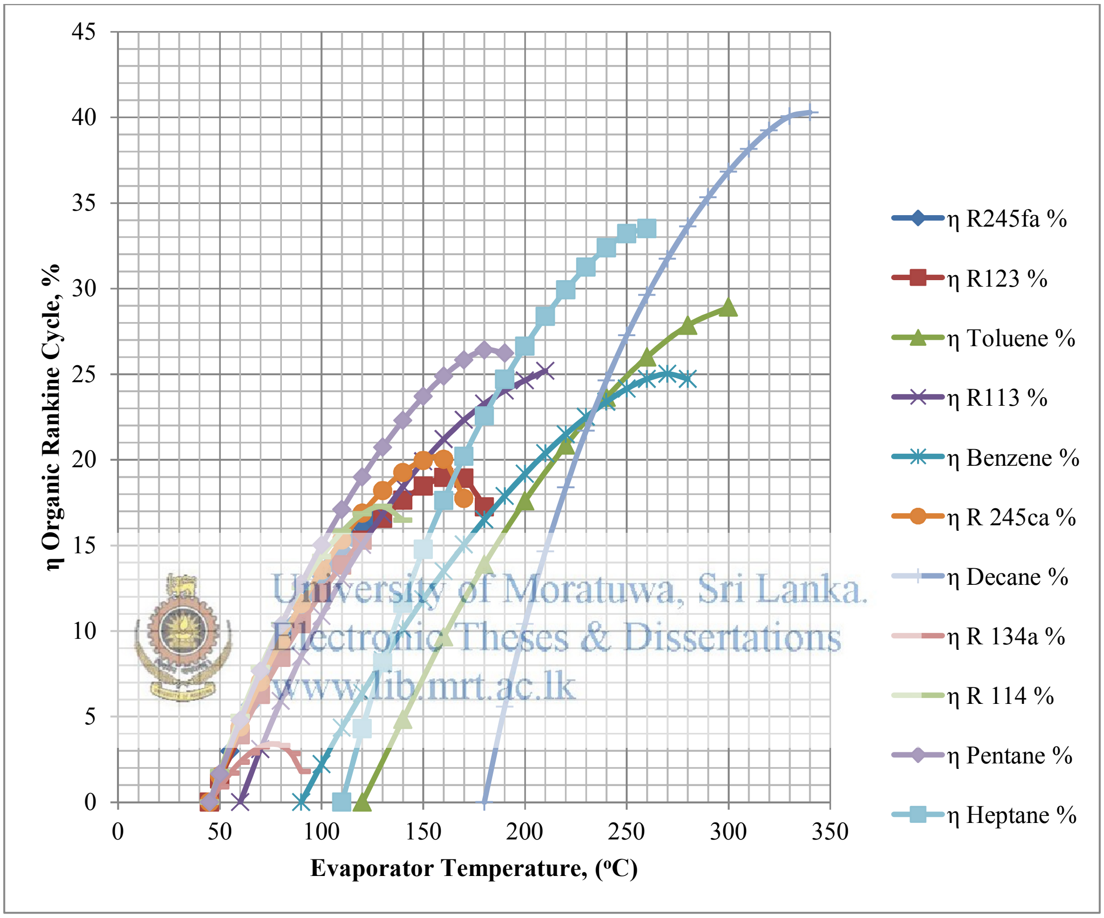

Figure 5.6: Cycle efficiency variation with different evaporator temperatures

Figure 5.6 highlights following key points;

- Generally, efficiency increases with evaporator temperature increment. Almost all the fluids have good linear relationship between efficiency and temperature, except R134a.
- Maximum efficiency was around 40% for all fluids, and for refrigerant fluids, maximum efficiency was around 25%. Further, for most of the refrigerant, cycle efficiencies were hovering between 15 – 25%, except R134a and most of the organic fluids, cycle efficiencies were hovering between 25 – 40%.

- Most of the refrigerant fluids have shown their maximum efficiency between the temperatures of 130 - 170 ^{o}C. On the other hand, all the organic fluids have shown their maximum efficiencies at higher temperature than 170 ^{o}C. Significantly, Pentane has shown higher efficiencies than all refrigerants between the temperatures of 130 - 170 ^{o}C.
- Organic fluids such as Heptane, Toluene & Decane have shifted the condenser temperature as higher boiling temperatures due to greater molecular mass.
- Most of the fluids were capable of operating temperature between 100 - 200 ^{o}C except R 134a, where cycle efficiencies vary within 10 - 25%.

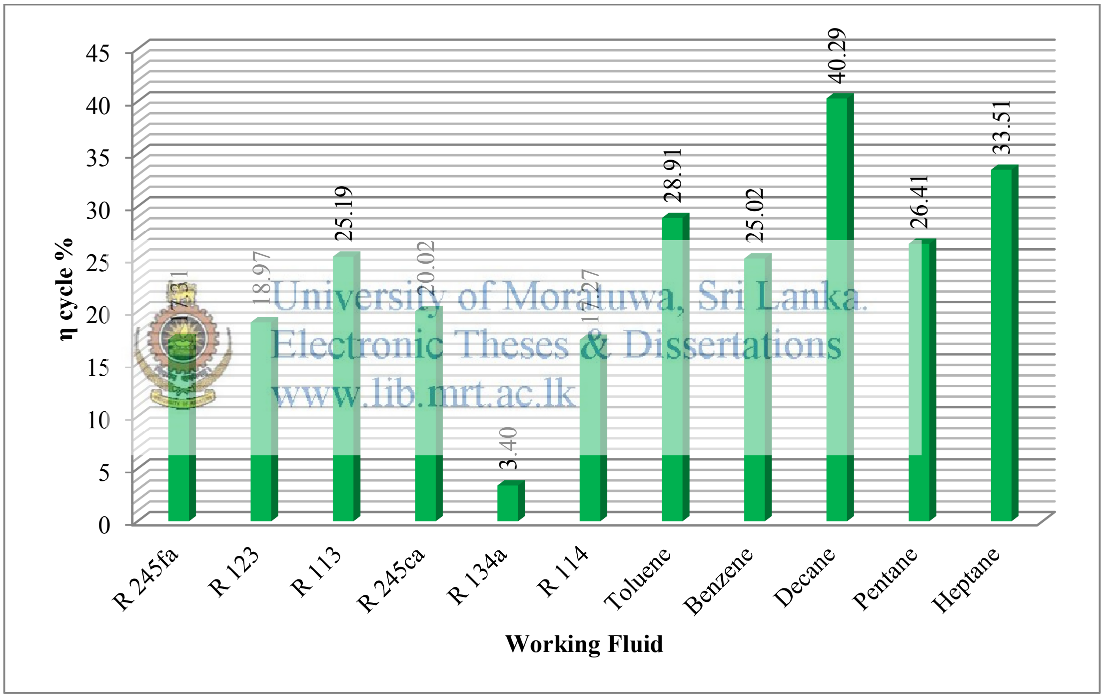

Figure 5.7: Graphical view of maximum possible efficiencies of each fluid with temperature

Figure 5.7 shows a bar chart for maximum efficiency recorded for each fluid and respective efficiency values were mentioned accordingly. In the chart, first six fluids from left hand side denote refrigerant fluids while next five denote organic fluids.

Accordingly; highest cycle efficiency of 40.2% was given by Decane when evaporator temperature at 340 ^{o}C and lowest efficiency of 3.4% was given by R134a when evaporator temperature at 75 °C. Respective to refrigerants, highest efficiency was 25.2% shown by R113.

Except R134a, all other fluids are capable of achieving 15% cycle efficiency at their respective evaporator temperatures.

Table 5.2: Details of maximum work output point for each fluid

Working Fluid	Expander Output		Evaporator		Work Input (kW)	Condenser	
	Max. Work (kW)	η %	Temperature (°C)	Pressure (bar)		Temperature (°C)	Pressure (bar)
R 245fa	20.22	17.31	140	30.87	116.71	45	2.96
R 123	20.43	18.97	160	24.90	107.74	45	1.83
Toluene	84.61	28.91	300	32.76	292.62	120	1.31
R 113	27.34	25.19	210	30.31	108.50	60	1.51
Benzene	73.18	25.02	270	38.67	292.48	90	1.36
R 245ca	26.07	20.02	160	30.56	130.19	45	2.07
Decane	119.51	40.29	340	19.75	296.60	180	1.17
R 134a	2.82	3.40	75	23.64	82.75	45	11.60
R 114	13.75	17.27	130	24.87	79.65	45	3.93
Pentane	71.83	26.41	180	26.10	271.94	45	1.37
Heptane	94.06	33.50	260	24.79	280.72	110	1.41

Table 5.2 shows the maximum work outputs of each fluid in related to respective temperatures and pressures. Accordingly, fluid Decane has highest recorded work output of 119.51 kW, cycle efficiency of 40.29% when evaporator temperature is at 340 °C and at that point, condenser at 180 °C and pressure 1.17 bar.

Further, analysis results have been organized and tabulated in such a way that, the maximum work outputs and efficiencies regions of each fluid are summarized in table 5.3. The work output region is obtained based on efficiency region which can be defined as highest efficiency of particular curve minus 4, and the work output values of either side of the curve. The temperature range is obtained according to work output values. For an instant, fluid Benzene has cycle efficiency ≥ 22 %, when evaporator temperature is between 230 – 280 °C and work output vary with in 62 – 74 kW.

Table 5.3: Details of maximum output and efficiency ranges of each fluid

Working Fluid	Max. Efficiency Range, η %	Work Output Range (kW)	Evaporator		Condenser	
			Temperature Range (^{o}C)	Pressure Range (bar)	Temperature (^{o}C)	Pressur e (bar)
R 134a	η ≥ 2.8 %	2 - 3	65 - 85	18 - 30	45	11.60
R 245fa	η ≥ 14 %	16 - 21	110 - 150	15 - 34	45	2.96
R 114	η ≥ 14 %	10 - 14	100 - 140	14 - 30	45	3.93
R 123	η ≥ 16 %	17 - 21	130 - 170	14 - 30	45	1.83
R 245ca	η ≥ 17 %	20 - 27	120 - 160	14 - 37	45	2.07
R 113	η ≥ 22 %	22 - 28	170 - 210	17 - 31	60	1.51
Benzene	η ≥ 22 %	62 - 74	230 - 280	22 - 44	90	1.36
Pentane	η ≥ 23 %	60 - 72	150 - 190	15 - 31	45	1.37
Toluene	η ≥ 26 %	71 - 85	260 - 300	19 - 33	120	1.31
Heptane	η ≥ 30 %	76 - 95	220-260	13 - 25	110	1.41
Decane	η ≥ 37 %	98 - 120	300 - 340	11 - 20	180	1.17

Refer to the observed values in Table 5.3, regarding maximum work output ranges and respective evaporator temperature ranges were plotted in above Figures 5.8 and 5.9 for all working fluids. Accordingly, Figure 5.8 represents the range of maximum work output for each fluid. The maximum work output range was defined as highest efficiency of particular curve minus four, and the work output values were taken with respective temperature values. Figure 5.9 represent the respective temperature region for maximum work output. Further, respective range sizes of work outputs and evaporator temperatures were mentioned in Figures of 5.8 and 5.9 for each fluid.

Here, fluid Heptane is having a maximum work output region of 60 to 72 kW and respective evaporator temperature range varies from 220 to 260 ^{o}C.

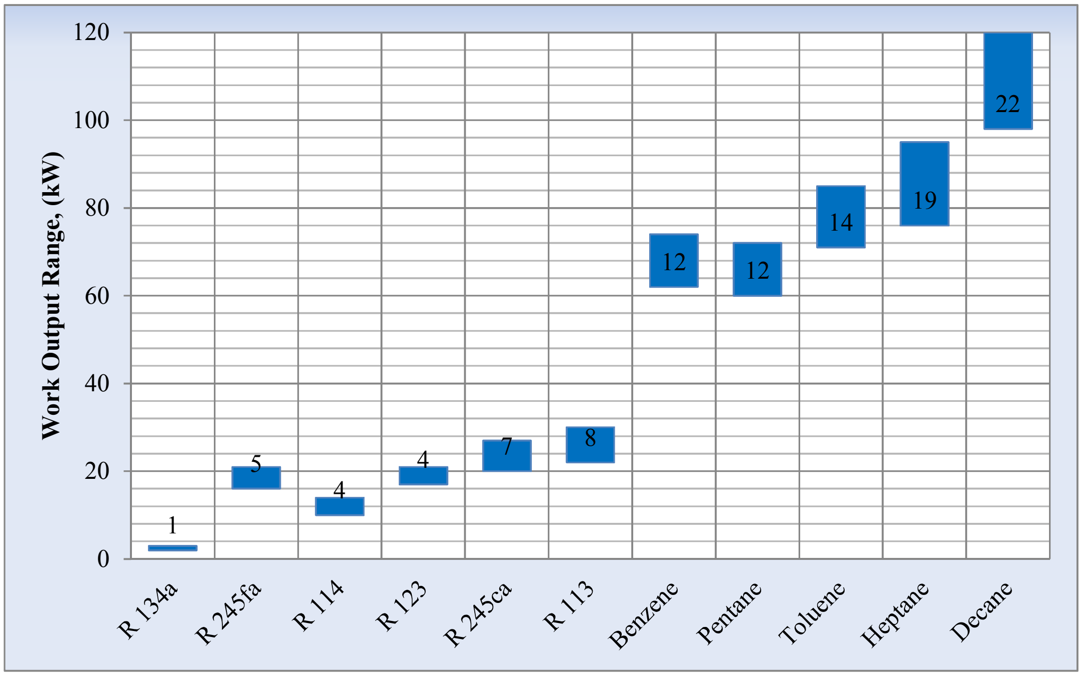

Figure 5.8: Maximum work output region for each fluid

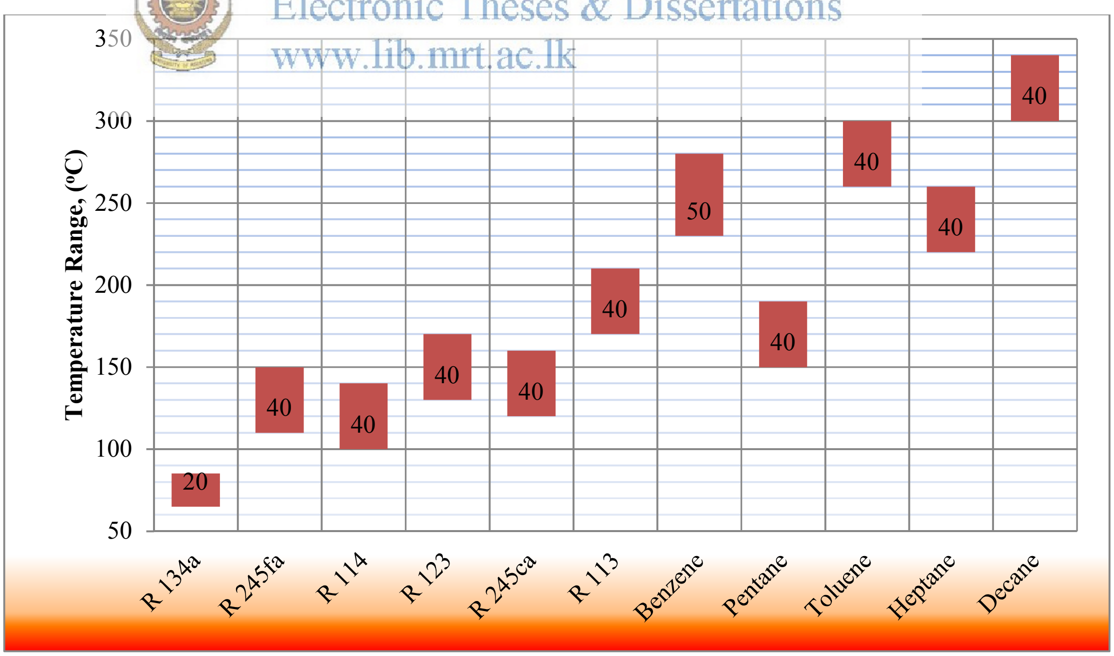

Figure 5.9: Evaporator temperature range for maximum work output region

5.2.2 Key Findings in Temperature Analysis

- According to Table 5.3 and Figure 5.8; refrigerant fluids have lesser work output range (less than 26 kW) compared to organic fluids which range from 60–120 kW. Based on Figure 5.8; refrigerants have narrow maximum work output than organic fluids.
- According to Figure 5.9; refrigerant operating temperature ranges were less than 210^{o}C. This means, the refrigerants are more suitable for cycles that evaporator temperature is below 200^{o}C. The fluids other than refrigerants, shows their maximum outputs at higher temperatures (>200^{o}C), which means they are more suitable for cycles that evaporator temperature is higher than 200^{o}C.
- As mentioned previously, organic fluids work outputs lay in between 60-120 kW as Figure 5.8 shows. Also their respective evaporator temperatures varied 200–350^{o}C, except for Pentane fluid, where temperature varied from 150–190^{o}C. This also suggests that, these fluids are preferable to be used at temperature higher than 200^{o}C.
- R 134a shows lowest output while Decane shows highest output. Hence, R134a is not recommended.
- Figure 5.8 shows, Benzene & Pentane equal in work outputs and efficiencies at maximum region, but according to 5.9, evaporator operates at two different temperatures. Fluid Benzene operates at temperature higher than 200^{o}C while Pentane operates at temperatures between 150–190^{o}C.
- According to Figure 5.4; fluid Pentane shown far better work output than any other fluid at temperatures below 190^{o}C. In temperature between 190-260^{o}C, Fluid Heptane has shown higher work output than any fluid. Temperature higher than 260^{o}C, fluid Decane shown better performance.
- Refer to Figure 5.6; fluid Pentane shown better cycle efficiencies for temperatures below 190 ^{o}C, Heptane shown higher efficiencies temperature between 190-260^{o}C and fluid Decane shown better efficiencies when temperature higher than 260 ^{o}C.

Surface graphs were plotted for fluid Decane, shown in Figure 5.10 and 5.11, for cycle efficiency and power output with variation of pressure and temperature. These plots are based on data of evaporator temperature variation.

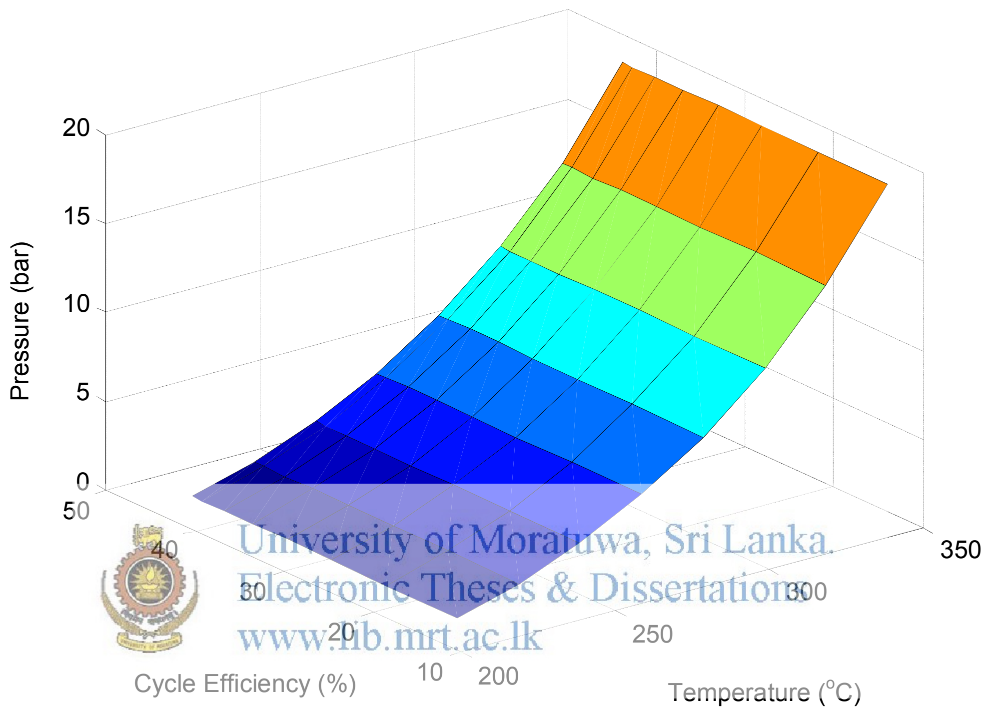

Figure 5.10: Efficiency variation of pressure and temperature of fluid Decane

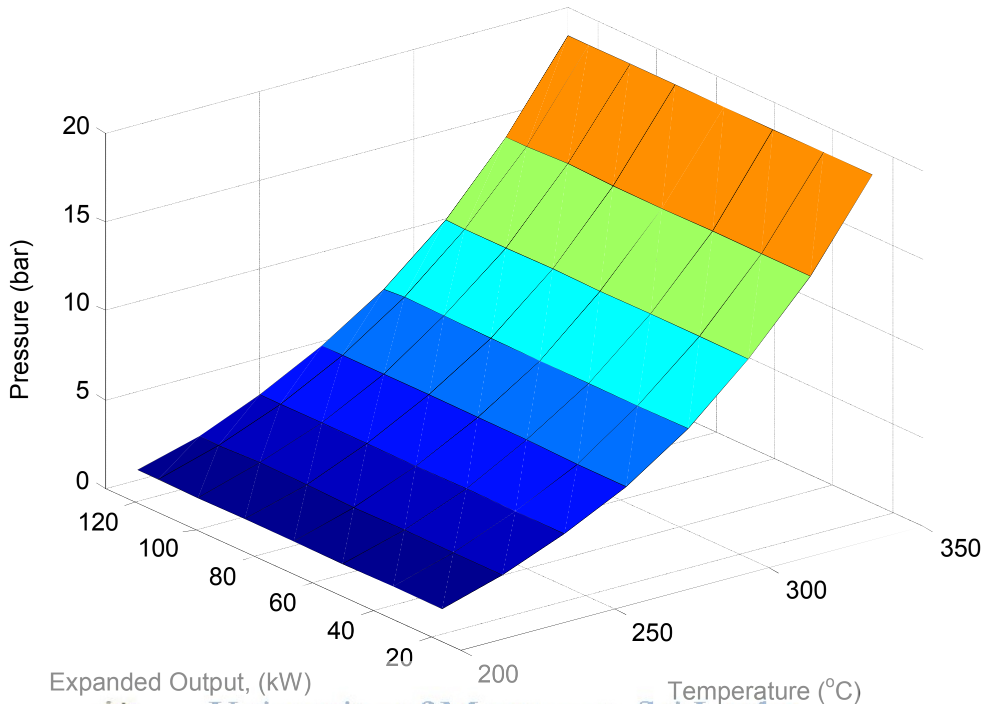

Figure 5.11: Expander output variation with pressure and temperature of fluid Decane

5.2.3 Analysis of Pressure Ratio Variation on Expander

Practically this analysis shows the influence of expander pressure ration variation for cycle performance. Further, analysis will provide basic criteria on optimum pressure ratios which will be important for expander selection. For an instant, output curve of particular fluid will have optimum region, and it is required to select smallest pressure ratio for expander selection which provide same optimum output. This is important because, small increment in system pressure cause large addition in cost component for the system.

In this analysis followings were considered;

- Basic Rankine cycle, inlet side pressure to expander maintained at higher value while outlet end at lower value. When pressure difference increases either side of the expander, work out put tends to increase. Hence, this pressure difference vs. work output in the expander was analysed during this research.
- Expander receive high pressure saturated vapour from evaporator and after expansion, exit low pressure saturated vapour.
- During the analysis, only dry and isentropic fluids were investigated to eliminate droplet formation in the expansion.
- Pressure ratio of high side and low side of the expander was increases from minimum value to maximum and stopped before fluid's critical point reaches.
- Low pressure side pressure was maintained higher than atmospheric pressure.
- Pump efficiency was taken as 0.75, expander isentropic efficiency was taken as 0.75, expander mechanical efficiency was taken as 0.96 and generator efficiency 0.98 for the calculation.
- Graphs such as work in, work output and efficiency were drawn for each fluid.

Based on above conditions, analysis was done and graphs were drawn for work outputs and efficiencies with reference to pressure ratio variation for each fluid (see appendix C). Then, combining the work output and efficiencies of all fluids, 02 separate graphs were drawn for the convenience of comparison with the pressure ratio variation in evaporator.

According to the analysis, fluid Decane has shown highest expander output and highest efficiency, related graphs are shown in figure 5.12, 5.13 and 5.14.

Analysis of fluid Decane

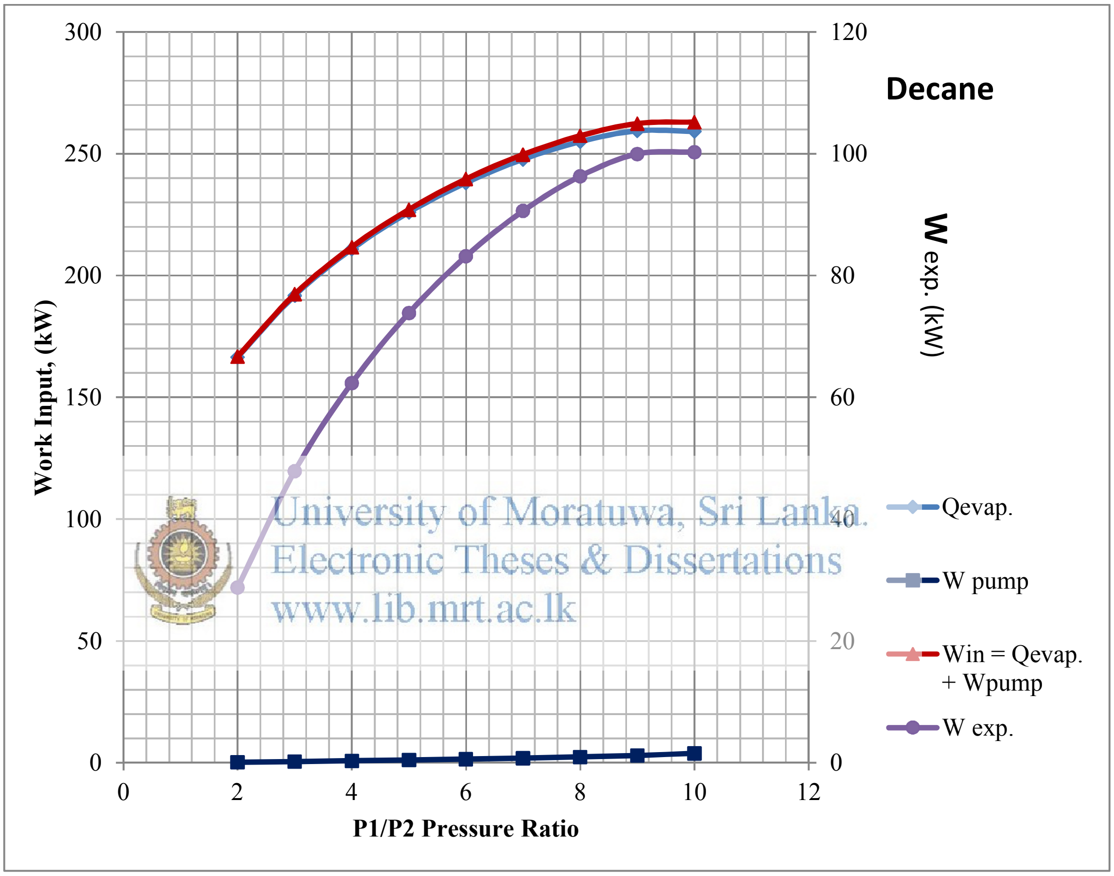

Figure 5.12: Input and expander work variation with different expander pressure ratios for Decane

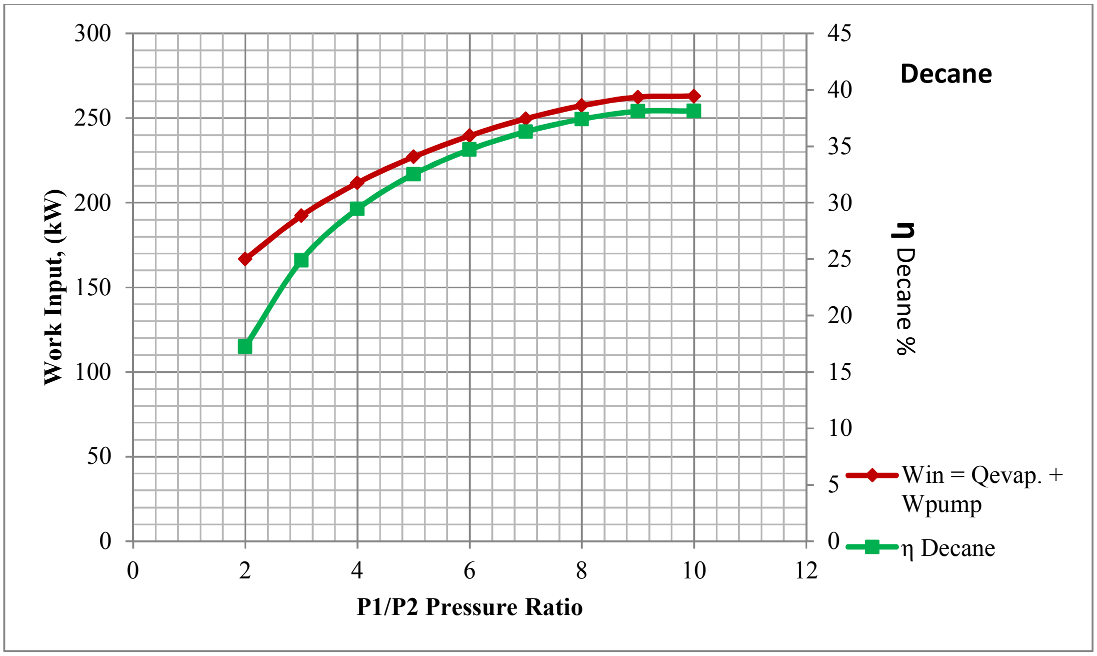

Figure 5.13: Work in and eff. variation with different expander pressure ratios for Decane

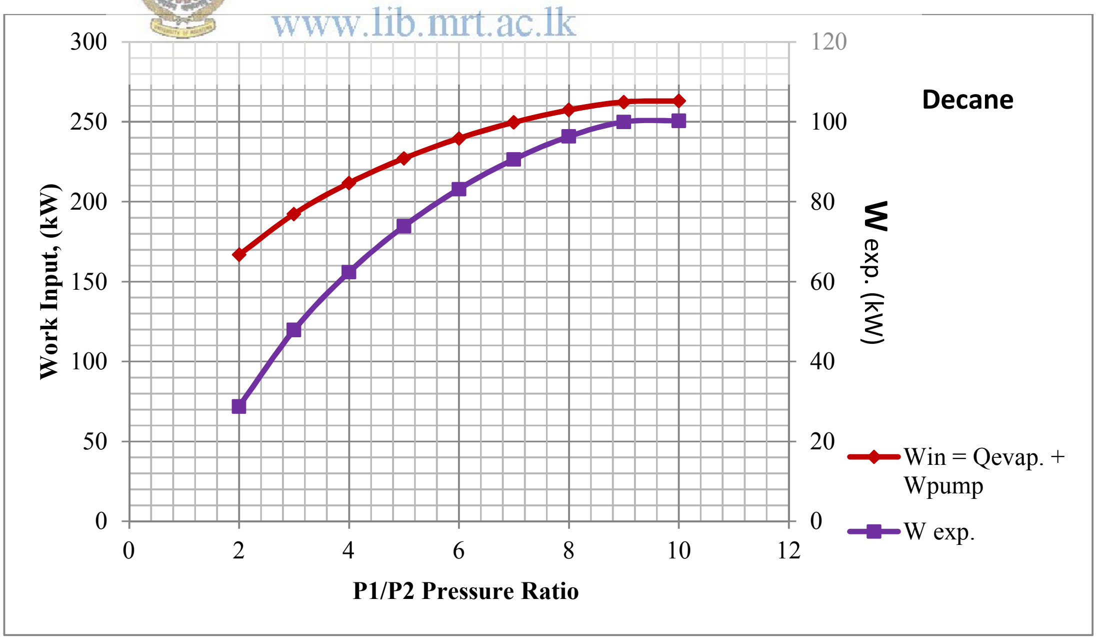

Figure 5.14: Input and expander work variation with different expander pressure ratios for Decane

Combined graph for all fluids for expander outputs and efficiencies are plotted in the figure 5.15 and 5.17 respectively. Observed pressure ratio variation for the plot was 2 - 25 range.

Combined analysis on work outputs

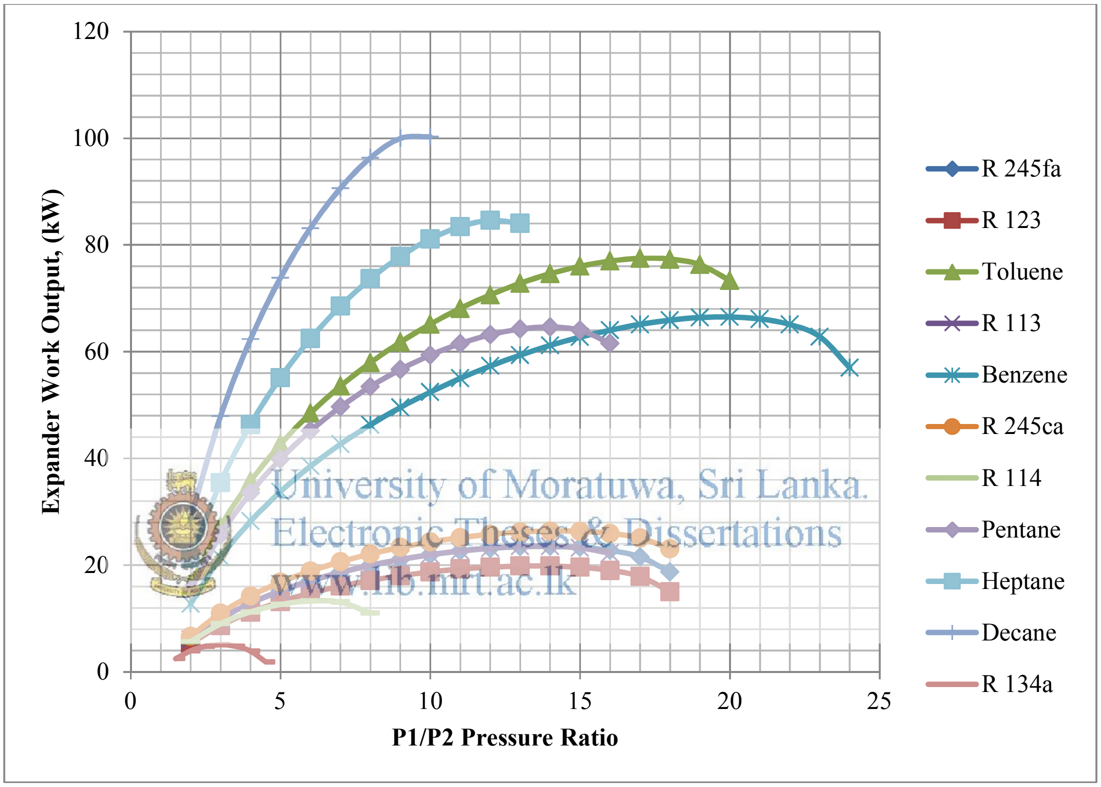

Figure 5.15: Expander work output variation with different pressure ratios

Following points were highlighted from figure 5.15;

- Generally all the fluids have positive relationship between pressure ratio and work output. Rate of raising the work output against pressure increment is generally two times higher in organic fluids than refrigerant fluids.
- Comparatively refrigerant fluids provide low work outputs and organic fluids provide high outputs. In terms of pressures, most of the refrigerants have approximately same output with flat peak in between pressure ratio 10 – 15. For organic fluids, maximum peaks have given at different pressure ratios.

- Maximum work outputs were equal or less than 100 kW for all fluids, where fluid mass flow rate was maintained at 0.5 kg/s. For refrigerant fluids, work outputs were less than 25 kW and operating pressure ratios were less than 18. However, R134a ability operates at pressure ratios less than 5.
- Most of the fluids operate at their maximum output in between pressure ratios 10 to 16.

Figure 5.16 shows a bar chart for maximum work output recorded for each fluid and respective efficiency values were mentioned accordingly. In the chart, first six fluids from left hand side denote refrigerant fluids while next five denote organic fluids.

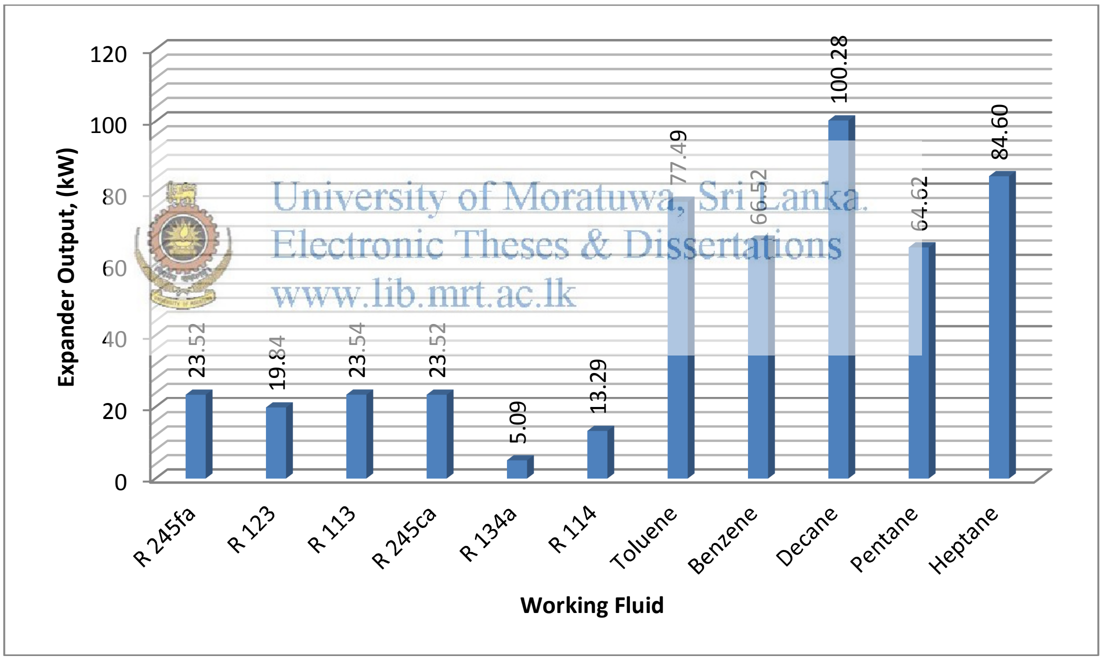

Figure 5.16: Graphical view of work outputs of each fluid with pressure

Most highlighting point in Figure 5.16 is that, Decane is having the highest work output of 100.27 kW, among all the fluids while expander operates at pressure ratio of 10. Lowest work output of 5.09 kW is having by R134a fluid where operates at pressure ratio of 3. Among refrigerant fluids, R113 and R245ca have the highest work output of 23.5 kW where expander pressure ratios are 23 and 20 respectively.

Combined analysis on efficiencies

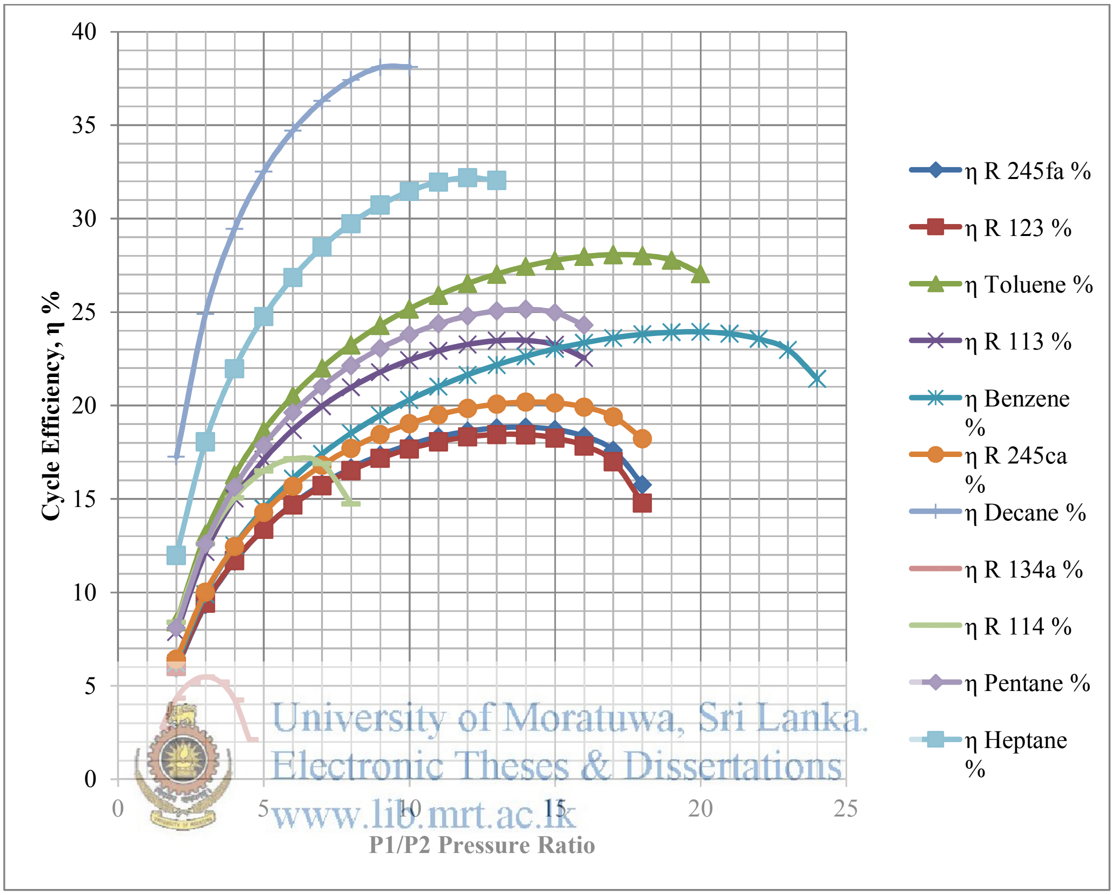

Figure 5.17: Cycle efficiency variation with different expander pressure ratios

Following points were highlighted from Figure 5.17;

- Generally fluid efficiencies were varied in between 17 – 38% except fluid R134a. Most of the fluids have their maximum efficiencies in between pressure ratios of 9 – 16. All the efficiency curves have flat trajectory.
- Fluid Decane has shown significantly higher efficiency from the beginning. Curve efficiency starts from 17%, where all other fluids recorded less than 8% except Heptane at the beginning. Further curve comes to its peak when pressure ratio at 10 which is comparatively low pressure.
- Even though fluid Benzene has the highest pressure ratios, in terms of efficiency fluid Pentane, Toluene, Heptane and Decane have higher values.

- In low pressure ratios as well as high ratios, organic fluids have shown better efficiencies than refrigerant fluids. R134a fluid has shown poor performance out of all fluids at all pressures.

Figure 5.18 shows a bar chart for maximum efficiency recorded for each fluid and respective efficiency values were mentioned accordingly. In the chart, first six fluids from left hand side denote refrigerant fluids while next five denote organic fluids.

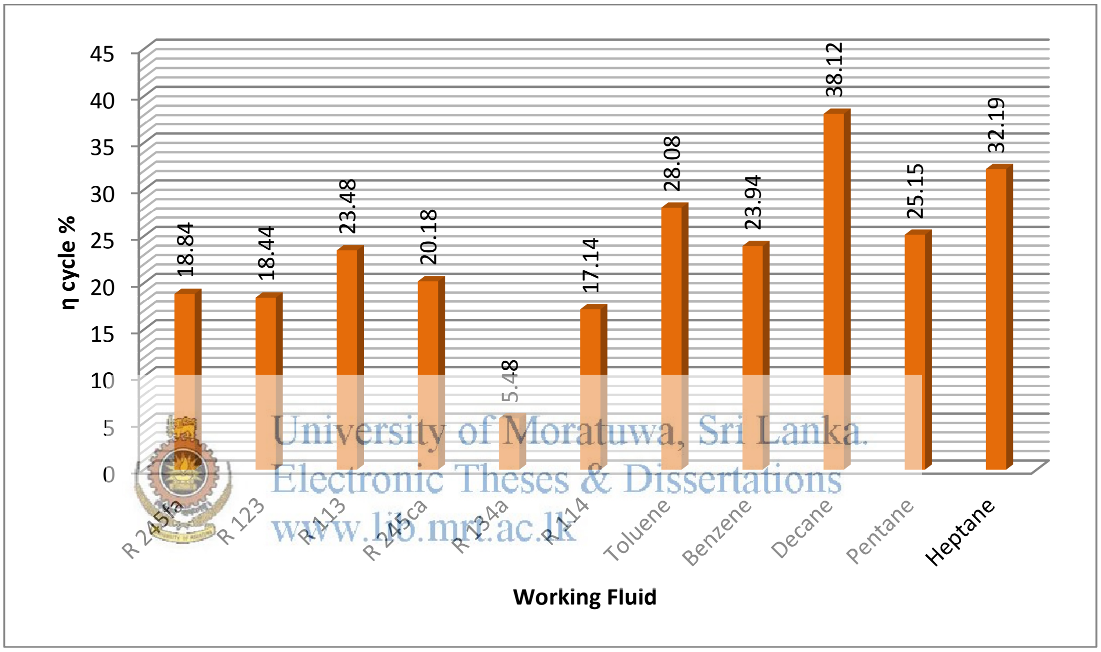

Figure 5.18: Graphical view of efficiencies of each fluid with pressure

Refer to 5.18; highest cycle efficiency of 38.12% was given by Decane when expander pressure at 20 bar and lowest efficiency of 5.47% was given by R134a when expander pressure at 24 bar. Respective to refrigerants, highest efficiency was 23.48% shown by R113. Except R134a, all other fluids achieved more than 17% cycle efficiency at their respective pressures.

Table 5.4 shows the maximum expander work with respective pressures and temperatures of cycle configuration. Accordingly, fluid Decane has highest recorded work output of 100.27kW, cycle efficiency of 38.12% when expander pressure ratio is 10 and at that point, condenser temperature is at 202.95 ^{o}C.

Table 5.4: Details on maximum work output point in each fluid

Working Fluid	Expander Output		Evaporator			Work Input (kW)	Condenser	
	Max. Work (kW)	η %	Pressure (bar)	P/P Ratio	Temp. (^{o}C)		Pressure (bar)	Temp. (^{o}C)
R 245fa	23.52	18.84	28	14	139.67	124.83	2	33.21
R 123	19.84	18.44	28	14	167.58	107.59	2	48.05
Toluene	77.49	28.08	34	17	302.97	275.98	2	136.41
R 113	23.54	23.48	28	14	201.12	100.24	2	70.08
Benzene	66.52	23.94	40	20	272.8	277.88	2	105.02
R 245ca	26.40	20.18	28	14	155.04	130.78	2	44.28
Decane	100.27	38.12	20	10	340.93	263.03	2	202.95
R 134a	5.09	5.48	24	3	75.70	92.96	8	31.34
R 114	13.29	17.14	24	6	128.22	77.52	4	46.91
Pentane	64.62	25.15	28	14	184.53	256.97	2	57.71
Heptane	84.60	32.19	24	12	257.71	262.79	2	123.4

Further, optimum regions in terms of work and efficiencies have been tabulated for each fluid in convenient manner in Table 5.5. The work output region is defined based on efficiency region as mentioned in previous analysis. Based on work output region, pressure ratio range is obtained. For an instant, fluid Benzene has cycle efficiency ≥ 21 %, when expander pressure range is between 22 - 48 bar and work output vary with in 55 – 67 kW. Here, condenser pressure is at 2 bar and temperature is at 105.02 ^{o}C.

Table 5.5: Details of maximum output range and efficiency ranges of each fluid

Working Fluid	Max. Efficiency Range, η %	Work Output Range (kW)	Expander	Evaporator	Condenser	
			Pressure Range (bar)	Temperature Range (^{o}C)	Temp. (^{o}C)	Pressure (bar)
R 134a	η ≥ 4 %	3 -5.5	16 - 32	58 - 90	31.34	8
R 114	η ≥ 15 %	11 - 14	16 - 28	106 - 138	46.91	4
R 123	η ≥ 16 %	17 - 20	16 - 34	130 - 180	48.05	2
R 245fa	η ≥ 16 %	20 - 24	16 - 34	115 - 150	33.21	2
R 245ca	η ≥ 17 %	22 - 27	16 - 36	125 - 170	44.28	2
R 113	η ≥ 21 %	21 - 24	18 - 32	170 - 210	76.08	2
Benzene	η ≥ 21 %	55 - 67	22 - 48	230 - 287	105.02	2
Pentane	η ≥ 22 %	53 - 65	16 - 32	150 - 193	57.71	2
Toluene	η ≥ 25 %	65 - 78	20 - 40	260 - 316	136.41	2
Heptane	η ≥ 29 %	73 - 85	16 - 26	230 - 263	123.4	2
Decane	η ≥ 35 %	90 - 101	14 - 20	315 - 340	202.95	2

Refer to the values of Table 5.5 regarding maximum work output ranges and respective expander pressure ranges were plotted in above Figures 5.19 and 5.20 for all working fluids. Accordingly, Figure 5.19 represents the range of maximum work output for each fluid. The maximum work output range has defined as highest efficiency of particular curve minus four, and the work output values were taken with respective pressure ratios. Figure 5.20 represent the respective expander pressure region for maximum work output. Further, respective range sizes of work outputs and expander pressures were mentioned in Figures 5.19 and 5.20 for each fluid.

According to Figure 5.19 and 5.20, fluid Toluene is having a maximum work output region of 64 to 80 kW and respective expander pressure range varies from 20 to 40 bar. Here, maximum work range size is 13 and expander pressure range size is 20.

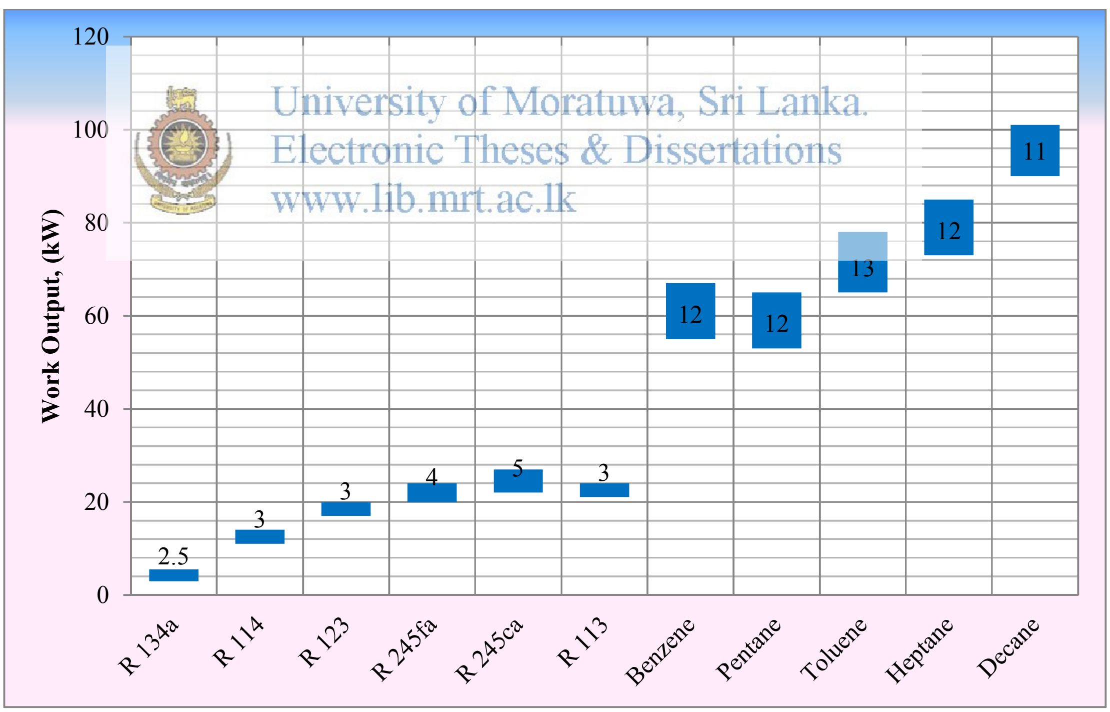

Figure 5.19: Maximum work output range with expander pressure variation

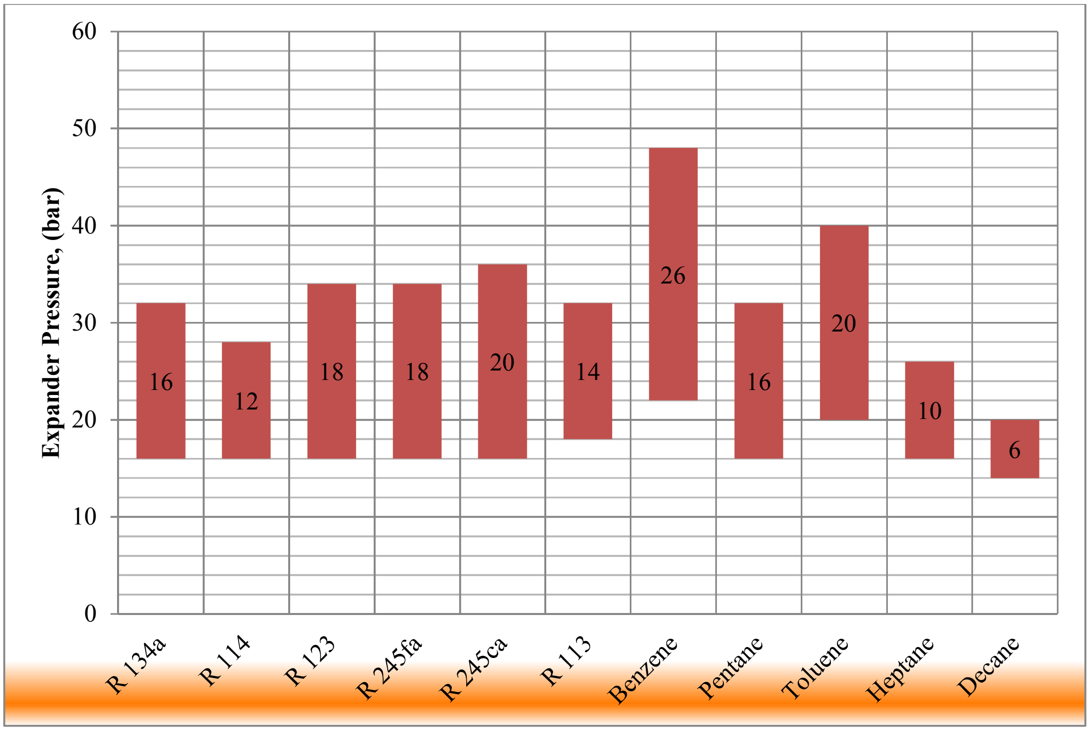

Figure 5.20: Expander pressure range for maximum work output region

5.2.4 Key Findings in Pressure Ratio Analysis

- Refer to Table 5.5; refrigerant fluids have lesser work output compared to other fluids (less than 28 kW). According to Figure 5.19; refrigerants having narrow work output regions compared to organic fluids. Further, organic fluids have higher work output ranging from 52 – 100 kW.
- Based on Figure 5.20; all most all refrigerants reach their maximum output when expander pressure region vary from 20 – 30 bar. For organic fluids, expander pressure varies at vast region from 15 – 45 bar.
- With refer to Figure 5.19 and 5.20; Benzene has the largest expander pressure region while maintaining the maximum output. Decane has the lowest pressure ratio region while maintaining the maximum output.
- Fluid R134a is not recommended due to poor overall performance.
- Refer to Figure 5.15; fluid Decane shown the far better performance than other fluids at pressure ration below 10. Between pressure ratios 10 – 13, fluid Heptane shown higher performance. Further pressure ratios 13 – 20, fluid Toluene has shown the higher work output. In terms of cycle efficiencies in Figure 5.17; identical results have repeated in respective pressure ratios for same fluids.

Further, surface graphs were plotted for fluid Decane, shown in Figure 5.21 and 5.22 for cycle efficiency and power output with variation of pressure and temperature. These plots are based on data of expander pressure ratio variation.

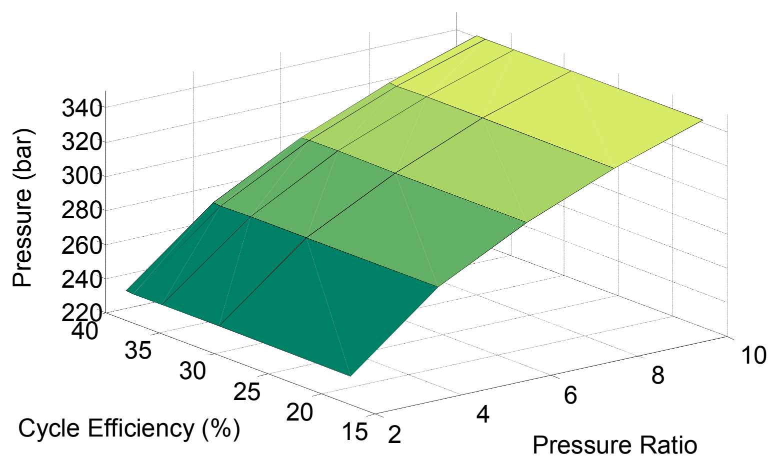

Figure 5.21: Efficiency variation with pressure and temperature of fluid Decane

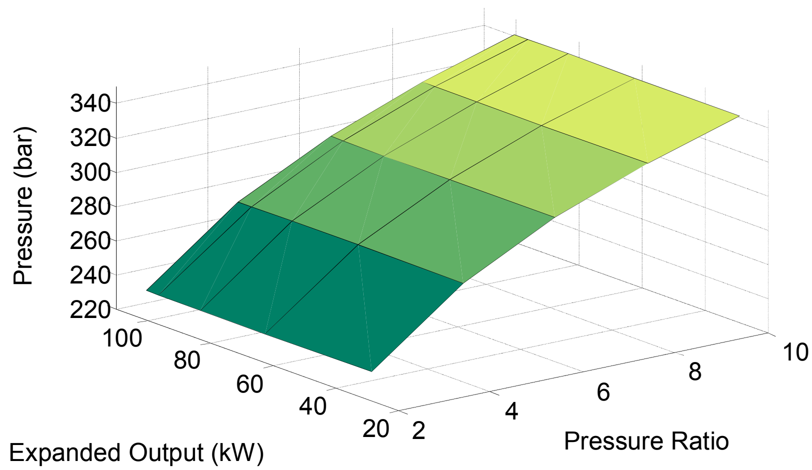

Figure 5.22: Expander output variation with pressure and temperature of fluid Decane

5.3 Development of Monographs

Based on theoretical evaluation of temperature and pressure ratio analysis, best performing fluids were selected upon their highest work output in between regions. Then inlet pressures/temperatures were varied to obtain theoretical work output at different points. These points were marked on same plot against the work output at Y axis and equal pressure/temperature lines were drawn. Further, system efficiencies of each point were obtained and equal efficiency lines were drawn. In the calculations of above points, unit mass flow rate (1 kg) has been considered.

One point on the developed monograph will denote pressure, temperature, work output and efficiency.

5.3.1 Temperature Based Monographs

In temperature analysis, evaporator temperature has been varied from 45 ^{o}C to 340 ^{o}C, evaluating eleven different fluids. Then theoretical outputs were plotted on combined graph of Figure 5.4. In that graph, 03 distinct fluids have provided far better performances than others, at different temperature regions. When evaporator temperature varied from 45 – 190 ^{o}C, fluid Pentane has shown higher work output than any other fluids. Similarly, evaporator temperature varied from 190 - 260 ^{o}C, fluid Heptane has shown the highest work. Further, temperature between 260 - 340 ^{o}C, fluid Decane has shown the highest work. Significantly, these fluids have shown predominantly higher work outputs along their respective regions.

Accordingly, 03 monographs for fluid Pentane, Heptane and Decane have been developed with temperature variation and they are shown in Figure 5.23, Figure 5.24 and Figure 5.25. In the calculations assumed that, expander out will be at saturated state and no energy loss up to evaporator inlet.

Finally, these monographs would provide guideline for industrial users as well as designers in selecting ORC fluids. Further, this will provide approximate guide about system in terms of possible power generation and system parameters such as pressures, temperatures, efficiencies, etc.

5.3.2 Pressure Ratio Based Monographs

In pressure ratio analysis, expander inlet pressure/outlet pressure ratio was varied from 1 to 25, while maintaining constant outlet pressure at expander. For all fluids theoretical outputs were plotted on combined graph of Figure 5.15. In that graph, 03 distinct fluids have shown far better performances than others, in different pressure ratio regions. When pressure ratio was less than 10, fluid Decane has shown far better performances, while fluid Heptane has higher work outputs where pressure ratio in between 10 to 13. Further, fluid Toluene has the highest performances when pressure ratio 13 to 20.

Accordingly, 03 monographs for fluid Decane, Heptane and Toluene have been developed in terms of pressure ratio Vs work output, and they are shown in Figure 5.26, Figure 5.27 and Figure 5.28. These monographs will provide guideline for design engineers when selecting the expanders during detail design stage of Organic Rankine Cycle for waste heat recovery. Further in the calculations assumed that, expander out will be at saturated state and no energy loss up to evaporator inlet.

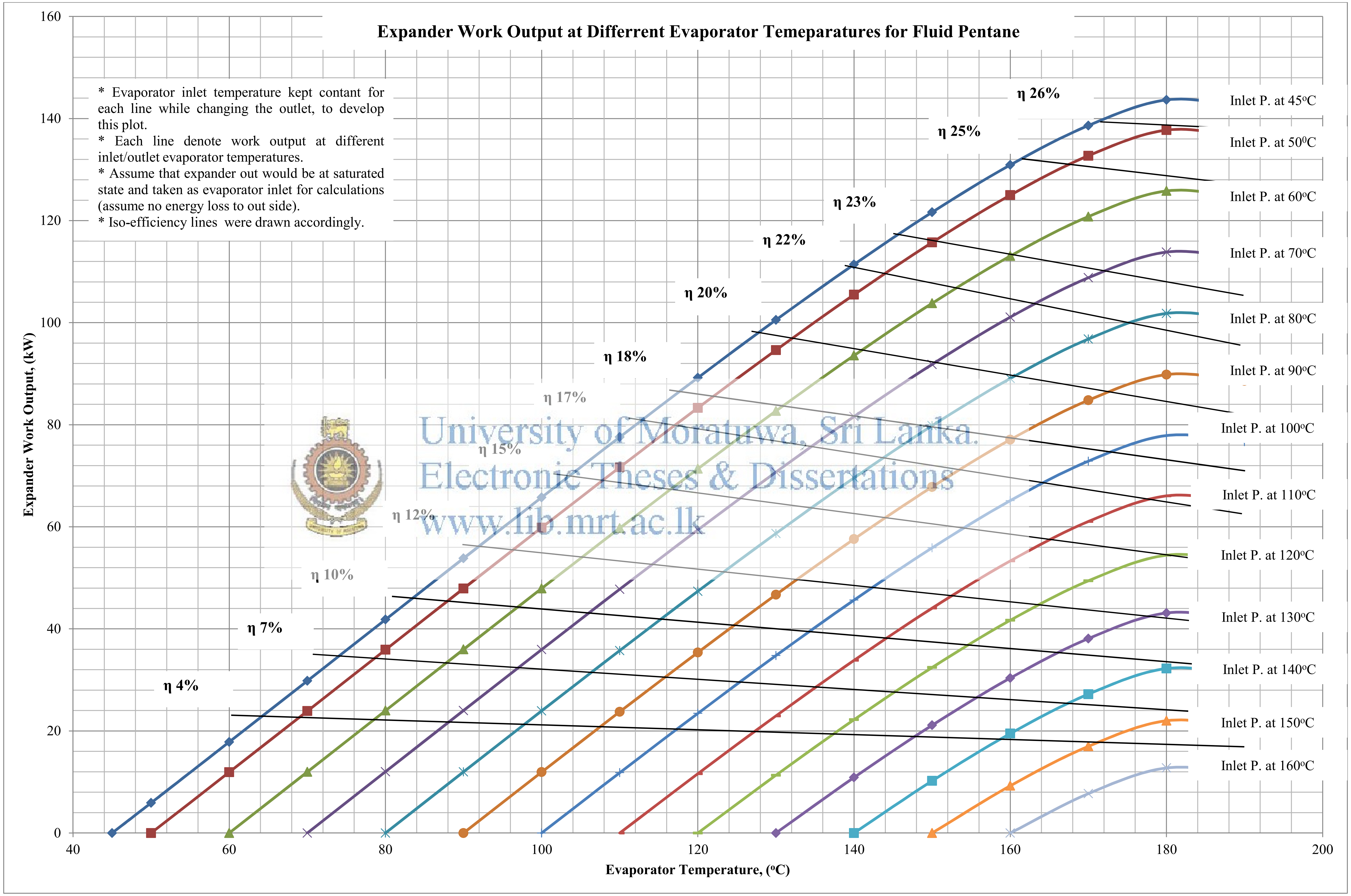

Figure 5.23: Different temperature curves with iso-efficiency lines against work output for fluid Pentane

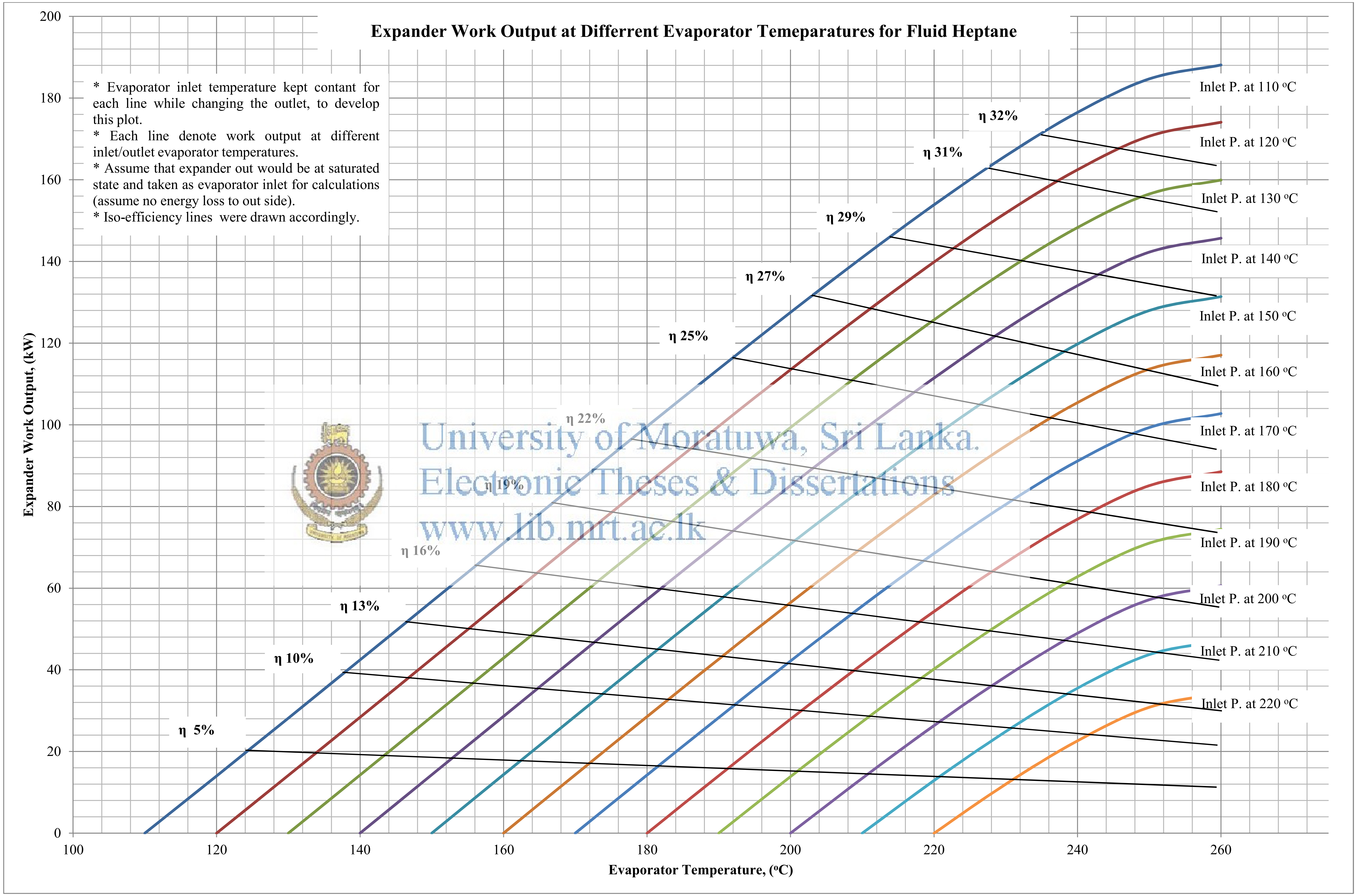

Figure 5.24: Different temperature curves with iso-efficiency lines against work output for fluid Heptane

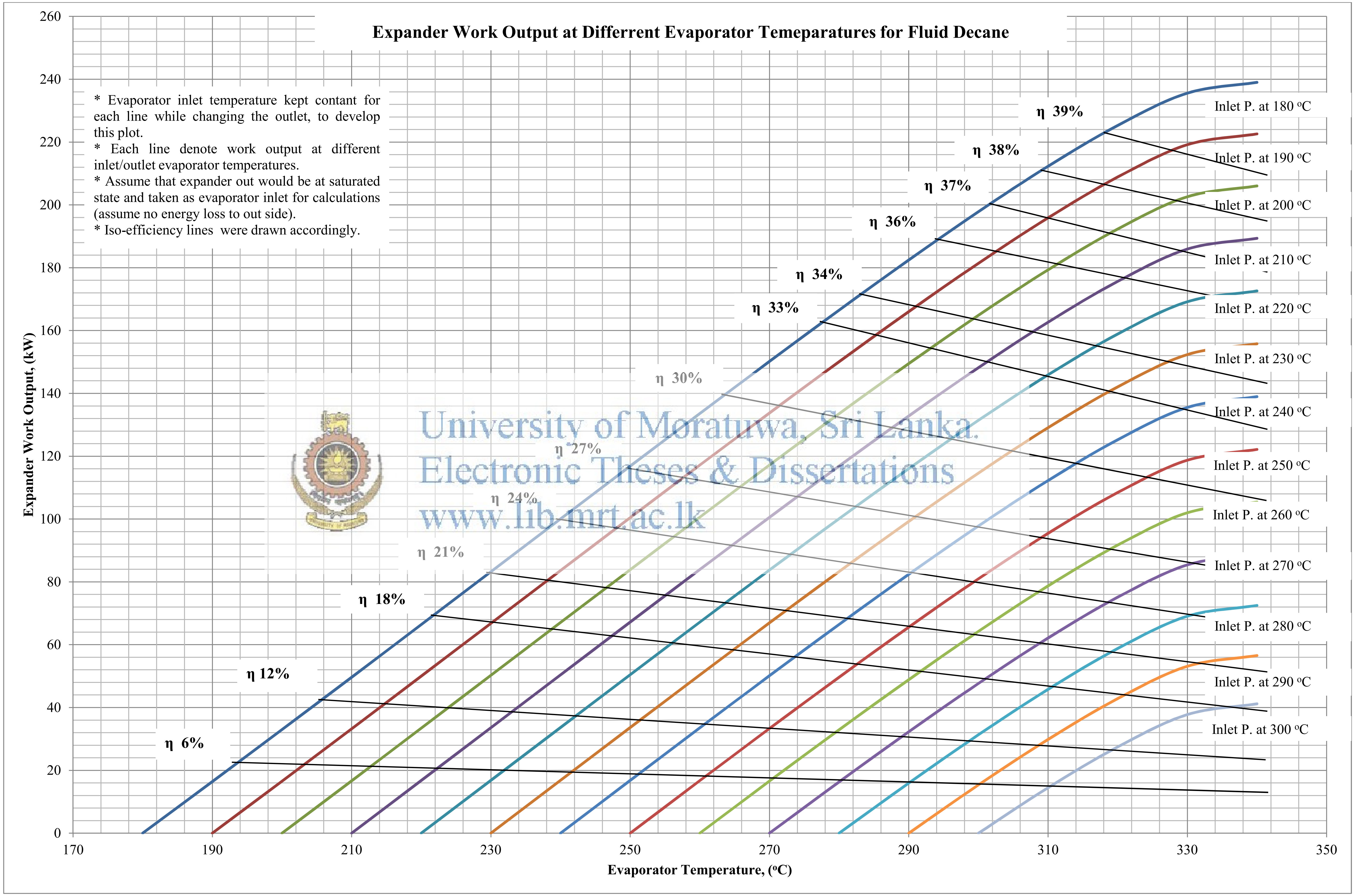

Figure 5.25: Different temperature curves with iso-efficiency lines against work output for fluid Decane

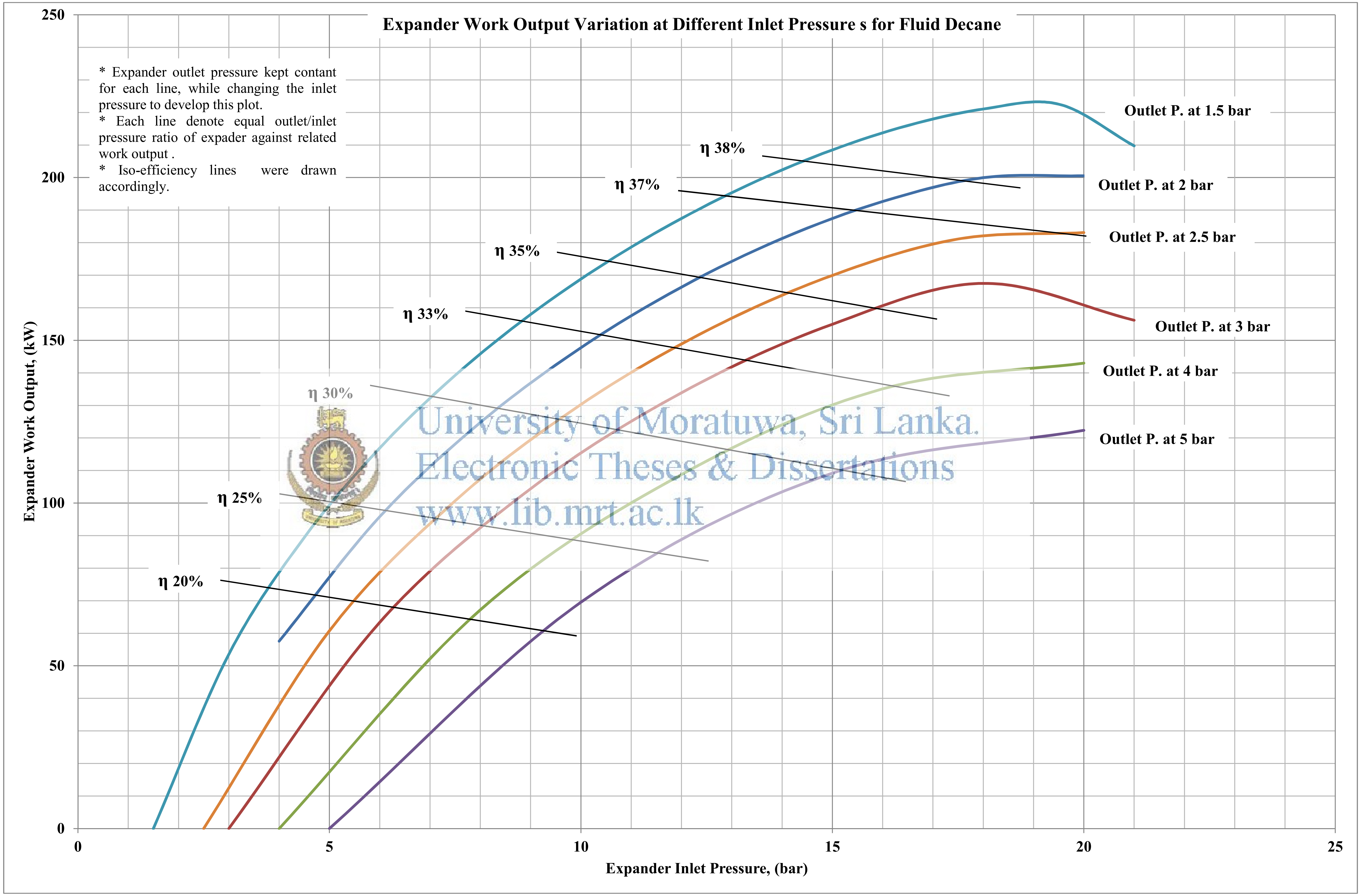

Figure 5.26: Different pressure ratio curves for expander with iso-efficiency lines against work output for fluid Decane

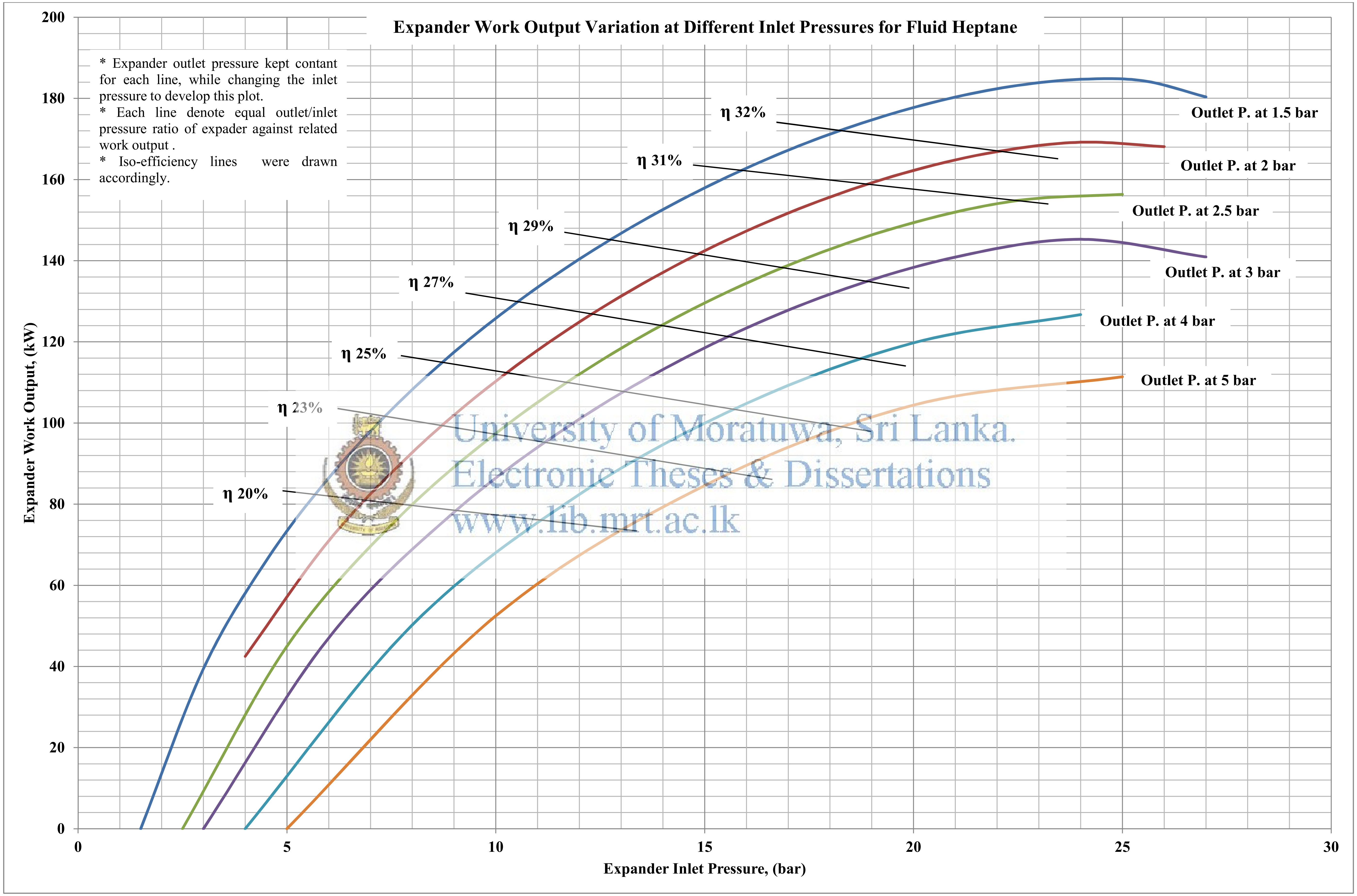

Figure 5.27: Different pressure ratio curves for expander with iso-efficiency lines against work output for fluid Heptane

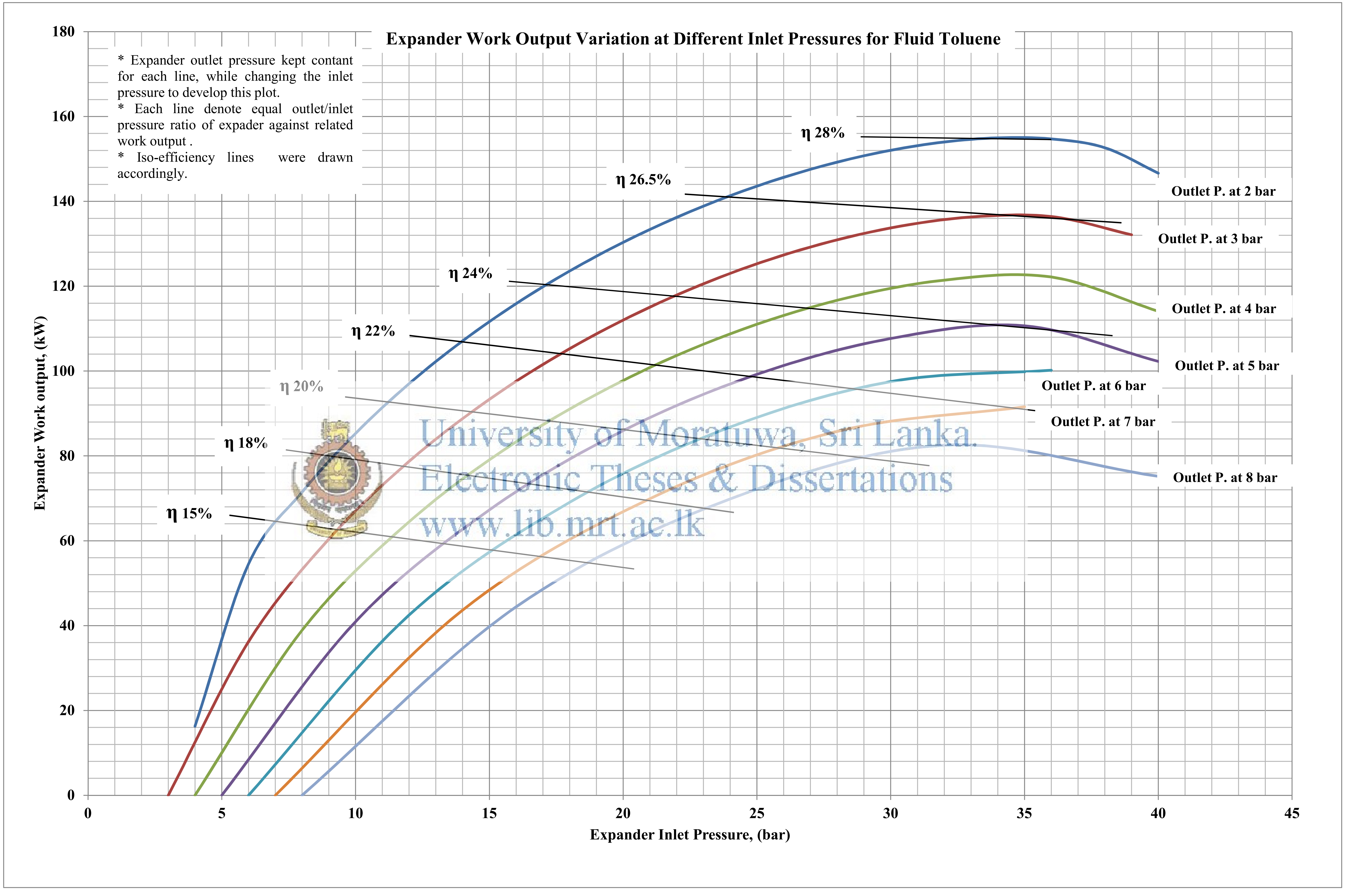

Figure 5.27: Different pressure ratio curves for expander with iso-efficiency lines against work output for fluid Toluene

6.0 CASE STUDY

During this research, waste heat opportunities were investigated in thermal power plants of the country. In the investigation, feasible waste heat sources were identified with their locations and tabulated in Appendix-A. Out of them, selected opportunities were tabulated and calculated the work output based on modelled system in Appendix-D. Please refer to mentioned appendixes; most feasible opportunities were selected for further studies in this chapter.

6.1 Selection of Waste Heat Opportunities

In the previous chapter, ORC configuration has been analysed on variable evaporator temperatures and variable expander pressure ratios. Based on that analysis, suitable organic fluids were chosen for selected waste heat opportunities as per Table 5.3. Reason for selecting the temperature variation analysis is that, the ORC system has to be modelled based on evaporator temperature. According to the table mentioned above, fluids were recommended in Table 6.1 for case study analysis.

Table 6.1: Waste heat of thermal plants and recommended fluids

Thermal Plant	Waste Heat Opportunity	Temperature Range (^oC)	Recommended Fluids
Sapugaskanda Power Station	Exhaust gas	430 - 180	R123, R113, Pentane
Lakvijaya Coal Plant	Exhaust gas	150 - 90	R134a, R114, R245fa, R245ca
	Boiler continuous blow down	275 - 100	R114, R245fa, R245ca, R123
Jaffna Power Plant	Exhaust gas	420 - 240	Heptane, Benzene, Toluene
Keravalapitiya Power Plant	Exhaust gas	500 - 160	R245ca, R123, R113, Pentane
Kelanithissa GT Plant	Exhaust gas	470-180	R123, R113, Pentane

Temperature ranges given in the above table were obtained in the following manner. The first value of the range denotes the average heat source temperature while the second value denotes the possible reduced figure or minimum maintained figure of heat source. Maximum utilization of waste heat, evaporator should maintain the minimum value of heat source. Hence, organic fluids which performed at higher outputs in previous analysis within the minimum value of heat source for each location were selected. Further, in heat exchangers, pinch point has to maintain 10 ^{o}C to 15 ^{o}C temperature difference with heat source to have better heat transfer. Considering all these factors, suitable organic fluids were recommended for performance evaluation.

6.2 Performance Evaluation of Selected Fluids

Each waste heat opportunity was evaluated by modelled system with recommended fluids. Main calculations were done Appendix-D and summarized details are mentioned in this chapter.

For the calculations, following assumptions were made:

- Isentropic efficiency (η iso) of the expander was taken as 0.75.
- For electrical output calculation overall efficiency (ηo) of the expander and generator was taken as 0.9 (η mech. x η elec. = η overall).
- Boiler blow down calculation in Lakvijaya plant, compressed water exit 100 ^{o}C at 2.5MPa and organic fluid leaving the evaporator will have 100 ^{o}C at 2.85 bar.
- For the calculation, it was assumed that GT's of Keravalapitiya plant will run as open cycle.
- Efficiency of the heat exchanger at waste heat recovery was taken as 0.20.

Summary of the work output calculations of each recommended fluids are tabulated in Table 6.2, 6.3, 6.4, 6.5, 6.6 and 6.7. Out of these tables, exhaust heat recovery of Sapugaskanda power station is presented in Table 6.2. Possible energy recovery potential is calculated under 03 fluids R123, R113 and Pentane. Fluid R113 has given the highest electric output of 458.83 kW at 10.02 kg/s mass flow rate, when exhaust flue gas temperature reduced from 430 ^{o}C to 180 ^{o}C. Similarly other tables also presented the possible energy recovery by selected fluids.

Table 6.2: Sapugaskanda plant exhausts heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^{o}C)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
R123	10799.13	430-180	10.06	410.81	0.90	369.73
R113	10799.13	430-180	10.01	509.81	0.90	458.83
Pentane	10799.13	430-180	4.07	396.46	0.90	356.82

Table 6.3: Lakvijaya plant exhausts heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^{o}C)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
R134a	17916.67	150-90	21.73	118.83	0.90	106.95
R114	17916.67	150-90	25.77	373.12	0.90	335.81
R245fa	17916.67	150-90	17.65	331.06	0.90	297.95
R245ca	17916.67	150-90	16.51	337.60	0.90	303.84

Table 6.4: Lakvijaya plant blow down heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^{o}C)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
R114	1723.45	275-100	2.32	49.83	0.90	44.85
R245fa	1723.45	275-100	1.59	45.84	0.90	41.26
R245ca	1723.45	275-100	1.49	46.79	0.90	42.11
R123	1723.45	275-100	1.80	42.30	0.90	38.07

Table 6.5: Jaffna plant exhausts heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^{o}C)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
Heptane	384.44	420-240	0.11	27.48	0.90	24.73
Benzene	384.44	420-240	0.14	16.55	0.90	14.90
Toluene	384.44	420-240	0.12	21.40	0.90	19.26

Table 6.6: Keravalapitiya plant exhausts heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^oC)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
R245ca	163838.43	500-160	127.41	6444.70	0.90	5800.23
R123	163838.43	500-160	155.17	5932.42	0.90	5339.18
R113	163838.43	500-160	158.87	6965.81	0.90	6269.23
Pentane	163838.43	500-160	64.81	7552.95	0.90	6797.65

Table 6.7: Kelanithissa GT plant exhausts heat recovery

Working Fluid	Exhaust Energy at Stack (kW)	Exhaust Temp. (^oC)	Fluid Mass Flow Rate (kg/s)	Expander Work (kW)	η_m X η_e = η_o	Electric Output (kW)
R123	32790.30	470-180	30.56	1247.37	0.90	1122.63
R113	32790.30	470-180	30.41	1547.98	0.90	1393.19
Pentane	32790.30	470-180	12.38	1203.82	0.90	1083.44

Graphical representations of energy output under each opportunity with respective fluids were plotted in Figure 6.1 and 6.2. The amount of recovery in Keravalapitiya power station was comparatively very large as it considered open cycle operation. Hence, Figure 6.1 won't represent the clear situation of energy recovery in Jaffna and Lakvijaya plant. Figure 6.2 was drawn separately removing the Kelavarapitiya and Kelanithissa plant details to provide better representation of other thermal plants.

According to Figure 6.2, Sapugakanda power plant has the highest potential energy recovery and amount the fluids; R113 has the best recorded performance of 458.83 kW.

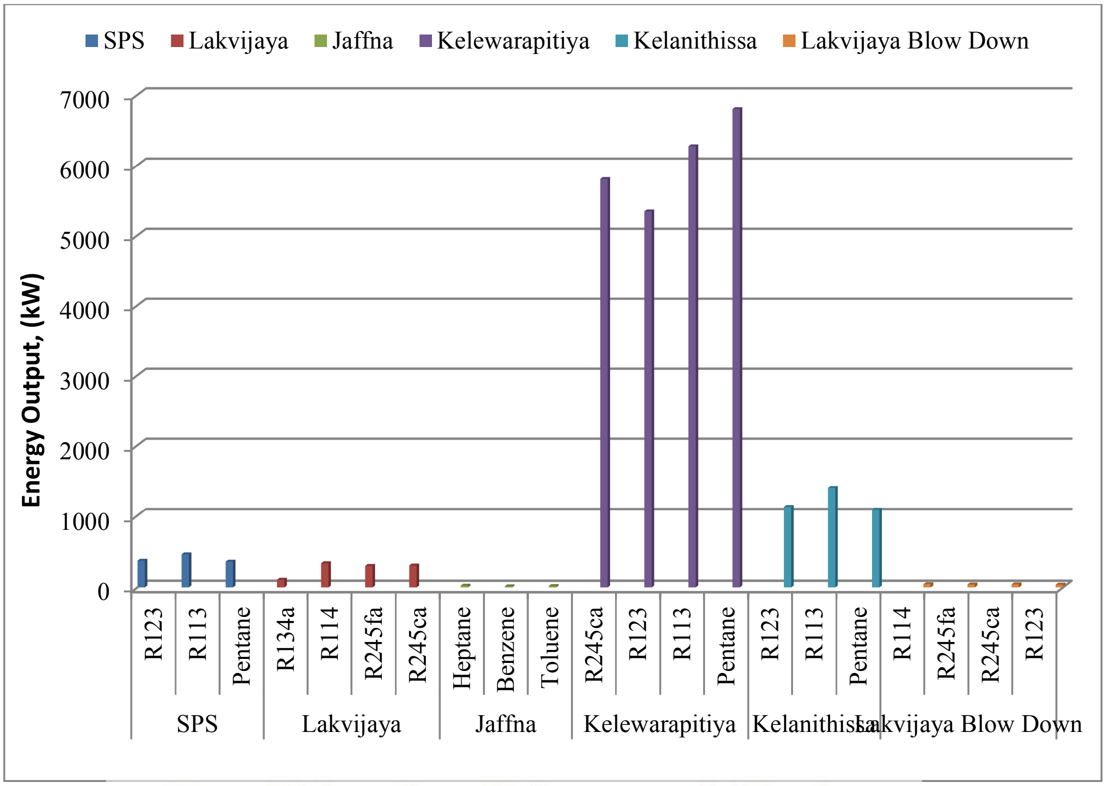

Figure 6.1: Energy recovery at each opportunity by different fluids

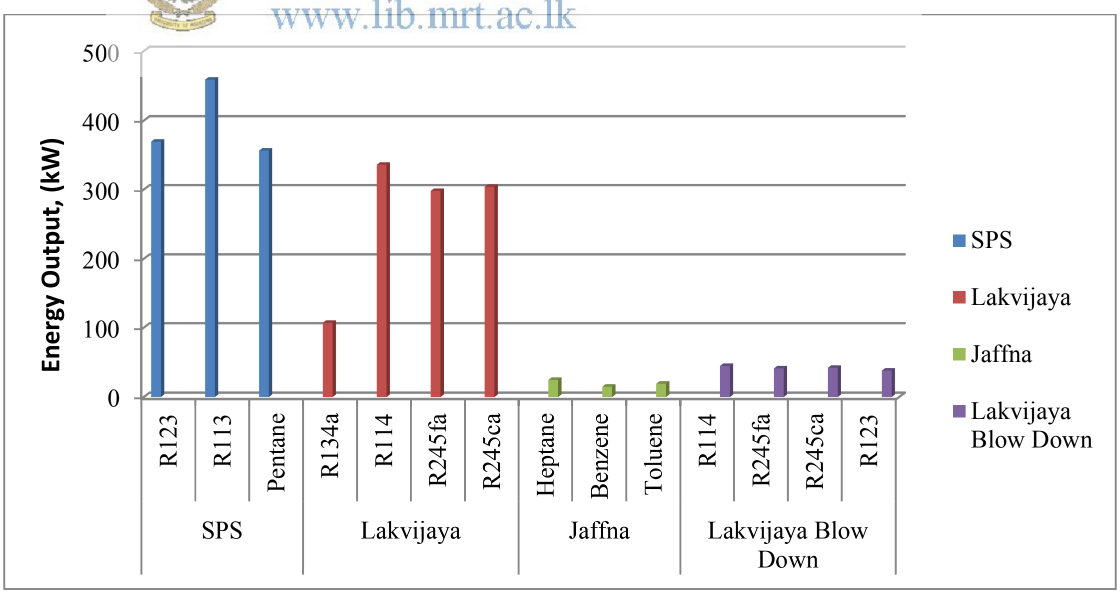

Figure 6.2: Energy recovery at each opportunity by different fluids, except Keravalapitiya and Kelanithissa plants

Based on above calculations, expander work and possible electrical outputs relating to each recommended fluid were given. Based on that, fluid which has the highest electrical output for each waste heat opportunity is summarized in Table 6.8.

Table 6.8: Maximum electrical output and related fluid for each opportunity

Thermal Plant	Waste Heat Resource	Maximum Electrical Output (kW)	Related Fluid for Max. Output
Sapugaskanda Power Station	Exhaust gas	458.83	R113
Lakvijaya Coal Plant	Exhaust gas	335.81	R114
	Boiler continuous blow down	44.85	R114
Jaffna Power Plant	Exhaust gas	24.73	Heptane
Keravalapitiya Power Plant	Exhaust gas	6797.65	Pentane
Kelanithissa GT Plant	Exhaust gas	1393.19	R113

Mass flow rate of refrigerant fluids are much higher than organic fluids due to less specific heat capacity. This will increase the power consumption of the feed pump considerably. But for the Jaffna power plant, mass flow rate is very low due to high heat capacities of organic fluids. This is identified as a limitation due to practical difficulty to find suitable expander for very low mass flow rates. According to 6.8, it can be concluded that, both refrigerant and organic (R113, R114, Heptane & Pentane) fluids are capable of extracting energy from WH successfully.

7.0 ECONOMIC ANALYSIS

Economic feasibility of selected WH opportunities are analyzed in this chapter. Plant capacities were decided according to available source and investment costs have been calculated. Total investments with payback were compared considering seven possible scenarios governed by external factors of the economy for better prediction. Method of net positive value (NPV) was adopted for the project investment evaluation.

7.1 Investment Cost on ORC

Capital investment of a project includes, costs associated with all the plant and equipment, lands, buildings, utility services, project management, design and consultancy, construction and installation works and many other approvals. Related to ORC plants, the system comes as a compacted unit to fix on the site with suitable alterations. Hence, cost components such as land, building and approvals would be minimized. Yet, cost involving plant and equipment, consultancy, may have much higher cost due to sophisticated machinery and some of the patent rights.

As the technology and the concept is novel, not many suppliers are available around the world. Due to competition, those suppliers won't directly provide the information about their plant costs. Most common application for ORC is exhaust of furnaces in steel, cement, glass and ceramic industries, due to high quality waste heat source which replaces conventional Rankine cycle. Further, investment costs of ORC systems based on different sources vary from USD 1,800 to 3,000 per kW depending on the plant capacities, technologies and configurations.

Based on some worldwide reputed manufacturers of ORC systems, investment cost were derived for each opportunity. Those manufacturers have given some basic guidelines for cost estimation with required source temperatures and cost per kW. The cost of the organic fluid is small compared to component cost, hence working fluid do not have considerable impact. Further, manufacturers have done the cost estimation including the fluid cost.

Based on manufacturer's criteria [31], following cost factors were defined and investment costs of each opportunity are summarized in table 7.1.

- Plants below 250kW - USD 2500 per kW
- Plants above 250kW - USD 2200 per kW
- Plants above 1MW - USD 1800 - 2000 per kW

Note: Calculations, cost for plants above 1MW has been considered as USD 1800/kW.
1 USD equals to Rs. 130.00

Table 7.1: Reputed ORC manufacturers and their plant details [31]

Manufacturer	**Product Line**	**Size Range**	**Minimum Temperature (^oC)**	**Estimated Nominal Cost per kW**
Turboden	Heat Recovery Units	400kW – 5MW	260	NA
Tri-O-Gen	Tri-O-Gen ORC	60kW – 165kW	350	NA
Energetix	Kingston	1kW	NA	NA
Infinity	Infinity Turbine ORC Power	10kW – 280kW	80	$ 2260/kW
Ormat	Ormat Energy Converter	250kW – 20MW	90	$ 1800-2000/kW
United Technologies	Pure Cyle	280kW	74	$ 2857/kW
Electratherm - Gas - Water	Green Machine	50kW	204 88	$ 2530/kW
Caluetix	Clean Cycle	100kW – 150kW	121	NA
Cryostar	Lo-C	1MW – 15MW	100	NA
Barber Nichols	Waste Heat Recovery Systems	500kW – 2MW	115	NA

7.2 Net Present Value (NPV)

NPV can be defined as "the difference between the present value of cash inflows and the present value of cash outflows". Here, cash inflows are the cash generated from the investment and cash outflows are the expenditures including investment. Hence, net cash flow is the difference of cash in and out flow. **NPV** is used in capital budgeting to analyze the profitability of an investment or project [32][33].

NPV relationship between net cash flow and the investment can be determined using the following equation;

$$NPV = \sum_{i=1}^{n} (B - C)i * Ai$$

Where;

NPV - Net present value

B - Cash inflow or benefit

C - Cash outflow or investment

A - Discount rate

The discount rate can be explained as the interest rate used in discounted cash flow (DCF) analysis to determine the present value of future cash flows. The discount rate in DCF analysis takes into account not just the time value of money, but also the risk or uncertainty of future cash flows. Hence, greater the uncertainty of future cash flows, the higher the discount rate [32][33].

The discount rate can be calculated as follows;

$$A = \frac{1}{(1+i)^p}$$

Where;

i - Interest rate

p - Period or years

Approximate capital investment for selected waste heat sources were estimated based on their potential output calculated in the previous chapter. Further, suitable organic fluid also was selected by referring to the previous calculations. Following table 7.2 shows the estimates for the possible capital investment for selected heat recovery opportunities;

Table 7.2: Estimated capital investment for heat recovery opportunities

Waste Heat Recovery Opportunity	Organic Fluid	Expected Electric Output (kW)	Estimated Plant Capacity (kW)	Investment Cost (USD) per kW	Total Investment Rs.
Sapugaskanda Power Station	R113	458.83	460	2200	131,560,000.00
Lakvijaya Coal Plant	R114	335.81	340	2200	97,240,000.00
Jaffna Diesel Plant	Heptane	24.73	25	2500	8,125,000.00
Keravalapitiya Plant	Pentane	6797.65	6800	1800	1,591,200,000.00
Kelanithissa GT (20MW) Plant	R113	1393.19	1400	1800	327,600,000.00
Lakvijaya Plant Boiler Blow Down	R114	44.85	45	2500	14,625,000.00

For the above table, plant capacities were decided based on expected power output for each WHR opportunity. Most of the country's thermal power plants would generate power, more than 60% during the year. Specially, Lakvijaya coal plant is expected to run more than 80% during the year to minimize power purchasing from private owned plants. For calculations, annual running hours were estimated as 60%, which means 5256 hrs per year. Further, it has been assumed that operation pattern would continue unabated for the next 05 years.

Fluctuation of electricity prices and inflation rate of the country, have made the financial analysis part more critical and complex. The impact of different economic situations arising out of unstable economic conditions referred to as scenarios, on the investment has been considered in evaluating the project. If so called scenarios are capable of predicting the future trends in the economy and if the investment was evaluated under those scenarios, most accurate picture on investment can be gained. Hence, considering possible future occurrences seven different scenarios are defined based on average selling price of electricity unit, future economic condition and bank interest. Investments are evaluated with reference to those scenarios, to estimate the financial viability of project implementation.

Table 7.3 has summarized the defined scenarios and predicted bank interest rates with average unit selling prices.

Table 7.3: Different scenarios on which NPV calculations were done for project feasibility

Practical Situations	Average Unit Selling Price (Rs. Per kWh)	Bank Interest Rate %
Scenario 1	14.00	8%
Scenario 2	15.00	8%
Scenario 3	15.40	8%
Scenario 4	14.00	10%
Scenario 5	15.00	10%
Scenario 6	14.00	12%
Scenario 7	Refer table 7.4	

Table 7.4: Plant running hours and interest rate for scenario 7

Waste Heat Recovery Opportunity	Averaged Annual Running %	Exp. Running Hours per Year	Averaged Unit Selling Price Rs.	Interest Rate
Sapugaskanda Power Station	60 %	5256	15.00	10%
Lakvijaya Coal Plant	70 %	6132	15.00	10%
Jaffna Diesel Plant	60 %	5256	15.00	10%
Keravalapitiya Plant	60 %	5256	15.00	10%
Kelanithissa GT Plant	40 %	3504	15.00	10%
Lakvijaya Boiler Blow Down	70 %	6132	15.00	10%

Scenario 7 in table 7.4 represents the plant running hours based on actual situation for the last few years. For instance, Kelanithissa GT plants do not usually operate, unless low cost plants are not adequate to meet the system demand. On the other hand, Lakvijaya coal plant will run throughout the year except for maintenance requirement due to low cost power generation. Usually Lakvijaya plant will run more than 70%

annually, unless a significant breakdown occurs. Further, average actual unit selling price is Rs. 15.40, but for scenario 7, unit price was taken as Rs. 15.00, due to possible price reductions in the future. Current bank interest rates are low as 6 – 8 %, which is not a normal situation in the country. Hence, average interest rate for next five years has been taken as 10%. With all these variations, an effort has been made to match the economic evaluation of scenario 7, to actual situation as closely as possible.

Further, all overhead costs such as operation and maintenance, spare parts, labour, etc, have been estimated as 1% of the total investment for next 05 years.

7.3: NPV Results

Following calculations were done in the appendix to investigate the feasibility of ORC systems on selected heat sources;

- Based on different scenarios, expected turnovers were calculated.
- NPV calculations were carried out referring to the above expected turnovers and different interest rates.

Table 7.5 shows the values of net positive incomes over a period of 05 years under previously described scenarios.

Table 7.5: Net Positive Values for different WHR opportunities

WHR Opportunity	Net Cash Flow after 5 years (Rs.)						
	Scenario 1	Scenario 2	Scenario 3	Scenario 4	Scenario 5	Scenario 6	Scenario 7
Sapugaskanda Power Station	(2,008,469)	7,620,412	11,471,965	(8,560,277)	581,643	(14,595,765)	581,643
Lakvijaya Coal Plant	(2,462,527)	4,584,614	7,403,471	(7,255,707)	(564,961)	(11,671,157)	16,161,906
Jaffna Diesel Plant	(1,183,750)	(664,775)	(457,185)	(1,534,790)	(1,042,061)	(1,858,166)	(1,042,061)
Kelevarapitiya Plant	342,415,937	485,069,361	542,130,731	244,627,206	380,066,220	154,544,782	380,066,220
Kelanithissa GT Plant	68,637,554	97,874,530	109,569,321	48,598,638	76,357,013	30,138,902	(62,434,864)
Lakvijaya Blowdown	(2,031,999)	(1,090,789)	(714,306)	(2,668,865)	(1,775,255)	(3,255,542)	458,770

With reference to table 7.5, values mentioned in red colour within brackets are negative incomes and rests of the values are positive incomes. It was decided that investment shall recover the cost at least within 05 years for the project to be feasible. Based on that criterion, feasibility of each investment is summarized in table 7.6.

Table 7.6: Summary of feasibility of the investments

WHR Opportunity	Scenario 1	Scenario 2	Scenario 3	Scenario 4	Scenario 5	Scenario 6	Scenario 7
Sapugaskanda Power Station	x	√	√	x	√	x	√
Lakvijaya Coal Plant	x	√	√	x	x	x	√
Jaffna Diesel Plant	x	x	x	x	x	x	x
Kelevarapitiya Plant	√	√	√	√	√	√	√
Kelanithissa GT Plant	√	√	√	√	√	√	x
Lakvijaya Blowdown	x	x	x	x	x	x	√

According to the above summary, following key points were highlighted;

- For all scenarios, investment on Keravalapitiya plant is feasible.
- Except for scenario 7, investment on Kelanithissa plant is feasible.
- Under all circumstances, investment on Jaffna Diesel plant is not feasible.
- For scenarios 2, 3, 5 and 7, investment on Sapugaskanda power plant is feasible.
- Lakvijaya plant, investment is feasible under scenarios 2, 3 and 7.
- Further, investment on Lakvijaya boiler blow down is feasible only under scenario 7. Most importantly, conditions mentioned under scenario 7 are more realistic with actual situation for the projects.

Comment on selected scenarios;

- Scenario 7 is the most realistic from all other.
- Scenario 3 is the present situation in the country which has high possibility to alter in terms of interest rates.
- Scenario 4 and 6 are the worst conditions.

8.0 CONCLUSIONS

The selection of optimal working fluid for Organic Rankine cycle is the most critical aspect in designing of WHR system. There are many fluids to choose based on certain criteria, but, when some fluids show desirable properties thermodynamically, their environmental and safety aspects are not favorable. Hence, there is no ideal fluid that can achieve all the desirable criteria such as thermodynamic properties, environmental safety and personal safety, at the same time.

This thesis presents the ORC modeling and performance results for evaporator temperature variation with different working fluids. Further, performance when pressure ratio variation in expander also observed. The total evaluation was done on subcritical region for all fluids on waste heat recovery aspect. If trans-critical and supercritical ORC systems are considered, system components undergo high pressures and temperatures which automatically increase the overall cost. Considering the low grade heat transfer, minimum of 15 ^{o}C temperature difference was maintained with source and evaporator in the model analysis. Fluid flow rate and viscosity should be as low as possible to reduce the components size, pressure losses and work done by the pump.

From this thesis, following points can be abstracted as conclusions;

Theoretical Evaluation

1. Basically selection of optimum working fluid depends on temperature of waste heat source and evaporator. When consider the temperature analysis, whole region from 45 ^{o}C to 340 ^{o}C was be devided in to 03,
2. Results of temperature analysis shows that, considered temperature region from 45 - 340 ^{o}C, was be divided into 03, upon their work output. Region between 45 – 190 ^{o}C, fluid Pentane has shown higher work output than any other fluids while region of 190 - 260 ^{o}C, fluid Heptane has shown the highest work. Further, temperature between 260 - 340 ^{o}C, fluid Decane has shown the highest work. Significantly, these fluids have shown predominantly higher work outputs along their respective regions.
3. When waste heat source temperature lies between 45 – 190 ^{o}C, fluid Pentane based ORC system is recommended while, if source temperature lies between

190 - 260 ^{o}C, fluid Heptane based ORC system is recommended. If source temperature lies between 260 - 340 ^{o}C, fluid Decane based ORC system is recommended.

4. With referance to developed monographs for fluid Pentane, Heptane and Decane, approximate values for cycle parameters such as work output, efficiency, pressure, temperature, etc. were be obtained.
5. There is no ideal working fluid which gives significant performance than the others when using Organic Rankine Cycle for waste heat recovery.
6. In subcritical region, wet and isentropic fluids are recommended.
7. The selected working fluid should have better thermodynamic properties such as high heat capacity, large enthalpy variation and high thermal efficiency.
8. As the waste heat is free, more concern should be on energy recovery by the system rather than overall efficiency.
9. Pressure ratio analysis shows how inlet and outlet pressure variation affect on expander. Based on the analysis, the ratio region was be divided in the 03 upon work outputs. When pressure ratio was less than 10, fluid Decane has shown far better performances, while fluid Heptane has higher performance between the pressure ratios of 10 to 13. Further, fluid Toluene has the highest performances when pressure ratio in between 13 to 20.

10. Developed monographs for fluid Decane, Heptane and Toluene, based on pressure analysis will provide guide for expander selection in component designing stage.
11. The results shows, high pressure ratio, always do not gaurantee high work outputs.
12. Further, pressure ratio line peaks have more flat regions, hence it will be advisable to select starting points of the flat peaks for expander. Beacause, small increment of pressure would add large cost component to the system.
13. When comparing organic and refrigerant fluids, organic fluids give higher work output than refrigerants in both temperature analysis and pressure ratio analysis.
14. In both evaporator temperature and expander pressure ratio analysis, Decane has shown the highest overall efficiency and highest work output among the fluids. Hence, fluid Decane has given the best performance in this theoretical evaluation.

15. According to temperature variation analysis, refrigerant fluids cannot be used when evaporator temperature goes beyond 200 ^{o}C. Hence, for higher WHR sources, it is recommended to use organic fluids.
16. Result shows that fluid R 134a is not suitable for waste heat recovery applications due to poor working performances.

Case Study and Economic Evaluation

Case study and economic evaluation was done based on temperature analysis. Fluids were selected upon the parameters of their optimum working regions and energy generation was calculated for selected waste heat opportunities. Then economic analysis was done based on NPV method to evaluate the investment worthiness. Based on these evaluation, following point were highlighted;

17. Based on evaluation, feasible WHR options in scenario 7 are recommended. Even though, options such as Keravalapitiya and Kelanithissa are feasible in almost all scenarios, they are highly dependent on the number of working hours during the year. Further, Keravalapitiya plat has been considered as open cycle gas turbine for evaluation.
18. Feasible WHR opportunities are tabulated and, WHR opportunities feasible under scenario 2, 5 and 7 are recommended for further analysis individually, because, scenario 2 represent the current condition, scenario 5 represent the worst condition than present due to high interest rates and scenario 7 represent the most realistic condition with plant running hours.

Research Limitations

1. Number of fluids that were evaluated in the research was limited.
2. The fluid properties of most of the organic fluids were very difficult to find due to limited available resources. Further, environmental and safety data for many fluids are not available.

3. Efficiencies of heat exchangers, expanders, working fluid pumps and generators were assumed based on the real values. Additionally, external heat losses from the cycle were assumed as zero.
4. According to table 6.5 in chapter 6, calculated mass flow rates are seems to be small. Hence, finding a suitable expander is a practical limitation.

Future work

1. More number of fluids needs to be evaluated on waste heat recovery aspect with more practical studies.
2. Furthermore future work should include an investigation of the performances in Trans critical and Supercritical of ORC.
3. Working fluid prices, stability and availability are other issues which needed to be investigated as the fluid selection is a main concern in the system. Additionally, environmental and safety impact on organic fluids is yet to be studied further with more fluids.
4. Extensive investigations are required on different expanders with different cycle states such as subcritical, trans-critical and supercritical. Several expanders have been developed in the world and studies should continue to optimize them.
5. In is interesting to study fluid mixtures such as in Kalina Cycle concept, because most of the researches are based on pure working fluids.

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Appendix A

DETAILS ON WASTE HEAT IN THERMAL POWER PLANTS

Waste heat details of Sapugaskanda Power Station

Table A.1: Exhaust gas temperatures and approximate waste heat energy

	Avg. Exhaust Gas Temp. (°C)	**Stack Temp. to be maintained (°C)**	**Avg. Exhaust Gas Qty (kg/hr)**	**Specific Heat Capacity C_p (kJ/kgK)**	**Exhaust Energy at the Stack (kW)**
A 01	430	180	140160	1.109	10799.13
A 02	440	180	140160	1.109	11231.10
A 03	440	180	140160	1.109	11231.10
A 04	430	180	140160	1.109	10799.13
B 01	427	180	79200	1.109	6029.02
B 02	430	180	79200	1.109	6102.25
B 03	430	180	79200	1.109	6102.25
B 04	420	180	79200	1.109	5858.16
B 05	435	180	79200	1.109	6224.29
B 06	445	180	79200	1.109	6468.38
B 07	439	180	79200	1.109	6321.93
B 08	435	180	79200	1.109	6224.29

Table A.2: Average jacket water temperatures and approximate waste heat energy

<table>
<tr><th></th><th>Avg. Jacket Water Outlet Temp. (°C)</th><th>Avg. Jacket Water Inlet Temp. (°C)</th><th>Jacket Water Flow Rate (m^3/hr)</th><th>Specific Heat Capacity C_p (kJ/kgK)</th><th>Energy at Jacket Cooling Water (kW)</th></tr>
<tr><td>A 01</td><td rowspan="4">94</td><td rowspan="4">80</td><td rowspan="4">190</td><td rowspan="4">4.2</td><td rowspan="4">3103.33</td></tr>
<tr><td>A 02</td></tr>
<tr><td>A 03</td></tr>
<tr><td>A 04</td></tr>
<tr><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>B 01</td><td rowspan="8">83</td><td rowspan="8">75</td><td rowspan="8">135.6</td><td rowspan="8">4.2</td><td rowspan="8">1265.6</td></tr>
<tr><td>B 02</td></tr>
<tr><td>B 03</td></tr>
<tr><td>B 04</td></tr>
<tr><td>B 05</td></tr>
<tr><td>B 06</td></tr>
<tr><td>B 07</td></tr>
<tr><td>B 08</td></tr>
</table>

Table A.3: Average raw water temperatures and approximate waste heat energy

	Avg. Raw Water Outlet Temp. (^oC)	Avg. Raw Water Inlet Temp. (^oC)	Avg. Raw Water Temp. Diff. (^oC)	Raw Water Flow Rate (m^3/hr)	Specific Heat Capacity C_p (kJ/kgK)	Energy at Cooling Water (kW)
A 01	52.4	40	12.4	760	4.2	10994.67
A 02						
A 03						
A 04						
B 01	NA	NA	NA	NA	NA	NA
B 02						
B 03						
B 04						
B 05						
B 06						
B 07						
B 08						

Table A.4: Average charge air water temperatures and approximate waste heat energy

	Avg. Charge Air Cooling Water Outlet Temp. (^oC)	Avg. Charge Air Cooling Water Inlet Temp. (^oC)	Cooling Water Flow Rate (m^3/hr)	Specific Heat Capacity C_p (kJ/kgK)	Energy at Cooling Water (kW)
A 01	NA	NA	NA	NA	NA
A 02					
A 03					
A 04					
B 01	49	44	276.56	4.2	1613.27
B 02					
B 03					
B 04					
B 05					
B 06					
B 07					
B 08					

Waste heat details of Lakvijaya Coal Power Station

Table A.5: Exhaust gas temperatures and approximate waste heat energy

Plant	Avg. Exhaust Gas Temp. (^{O}C)	Stack Temp. to be maintained (^{o}C)	Avg. Exhaust Gas Qty (kg/hr)	Specific Heat Capacity C_p (kJ/kgK)	Exhaust Energy at the Stack (kW)
U 01	150	90	1,000,000	1.075	17916.67
U 02	150	90	1,000,000	1.075	17916.67
U 03	150	90	1,000,000	1.075	17916.67

Table A.6: Waste energy at continuous blow down from each unit

Plant	Blow Down Water Out Temp. (^{o}C) at 17.5Mpa	Assumed maintain Temp. (^{o}C) at 2.5MPa	Blow Down Flow Rate (kg/hr)	Enthaphy at T 275 ^{o}C, P 17.5MPa,h1 (kJ/kg)	Enthaphy at T 100 ^{o}C, P 2.5MPa,h2 (kJ/kg)	Energy at Blow Down Water (kW)
U 01	275	100	8700	1134	420.85	1723.45
U 02	275	100	8700	1134	420.85	1723.45
U 03	275	100	8700	1134	420.85	1723.45

Note: Following values were for above calculation,

Blow down details;

Feed water pressure for the drum 17.5 MPa at 275 ^{o}C.

Feed water rate for 300 MW, 870 tons/hr and continuous blow down rate is 1%.

Assumptions for calculation;

In heat recovery from blow down, pressurized water in the drum 17.5 MPa at 275 ^{o}C would be reduced to 2.5 MPa at 100 ^{o}C.

Waste heat details of Jaffna Power Station

Table A.7: Exhaust gas temperatures and approximate waste heat energy

	Avg. Exhaust Gas Temp. (^{O}C)	**Stack Temp. to be maintained (^{o}C)**	**Avg. Exhaust Gas Qty (kg/hr)**	**Specific Heat Capacity C_p (kJ/kgK)**	**Exhaust Energy at the Stack (kW)**
DG 01	413	240	6800	1.1307	369.49
DG 02	417	240	6800	1.1307	378.03
DG 03	420	240	6800	1.1307	384.44

Table A.8: Average jacket water temperatures of each generator

<table>
<tr><th></th><th>Avg. Jacket Water Outlet Temp. (^oC)</th><th>Avg. Jacket Water Inlet Temp. (^oC)</th><th>Jacket Water Flow Rate (m³/hr)</th><th>Specific Heat Capacity C_p (kJ/kgK)</th><th>Energy at Jacket Cooling Water (kW)</th></tr>
<tr><td>DG 01</td><td rowspan="3">96</td><td rowspan="3">40</td><td rowspan="3">NA</td><td rowspan="3">4.2</td><td rowspan="3">-</td></tr>
<tr><td>DG 02</td></tr>
<tr><td>DG 03</td></tr>
</table>

Note: Jacket water flow rates were not measured in the plant, also design references were not possible to find, hence waste energy could not calculated.

Waste heat details of Keravalapitiya Power Station

Table A.9: Exhaust gas temperatures at open cycle and approximate waste heat energy

	Avg. Exhaust Gas Temp. (^oC)	**Stack Temp. to be maintained (^oC)**	**Avg. Exhaust Gas Qty (kg/hr)**	**Specific Heat Capacity C_p (kJ/kgK)**	**Exhaust Energy at the Stack (kW)**
GT 01	505	160	1,512,000	1.1307	589818348
GT 02	510	160	1,512,000	1.1307	598366440
ST	NA				

Table A.10: Average close cooling water temperatures and waste energy

	Avg. Close Coolig Water Outlet Temp. (^oC)	**Avg. Close Cooling Water Inlet Temp. (^oC)**	**Close Cooling Water Flow Rate (m^3/hr)**	**Specific Heat Capacity C_p (kJ/kgK)**	**Energy at Close Cooling Water (kW)**
GT 01	48	35	720	4.2	39312.00
GT 02	48	35	720	4.2	39312.00
ST	NA				

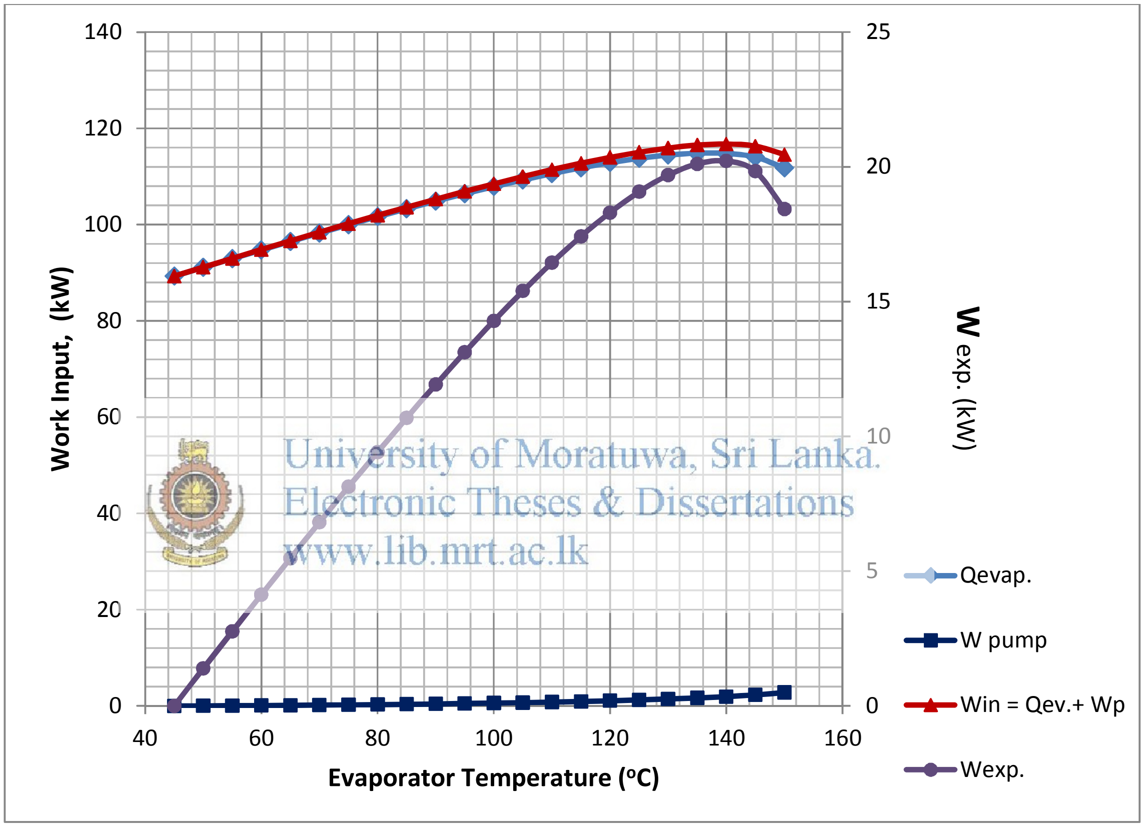
Table A.11: Average sea cooling water temperatures

	Sea Coolig Water Outlet Temp. (^oC)	**Sea Cooling Water Inlet Temp. (^oC)**	**Close Cooling Water Flow Rate (m^3/hr)**	**Specific Heat Capacity C_p (kJ/kgK)**	**Energy at Sea Cooling Water (kW)**
GT 01	38	32	1200	4.2	
GT 02	NA				
ST	38	32	20000	4.2	504000.00

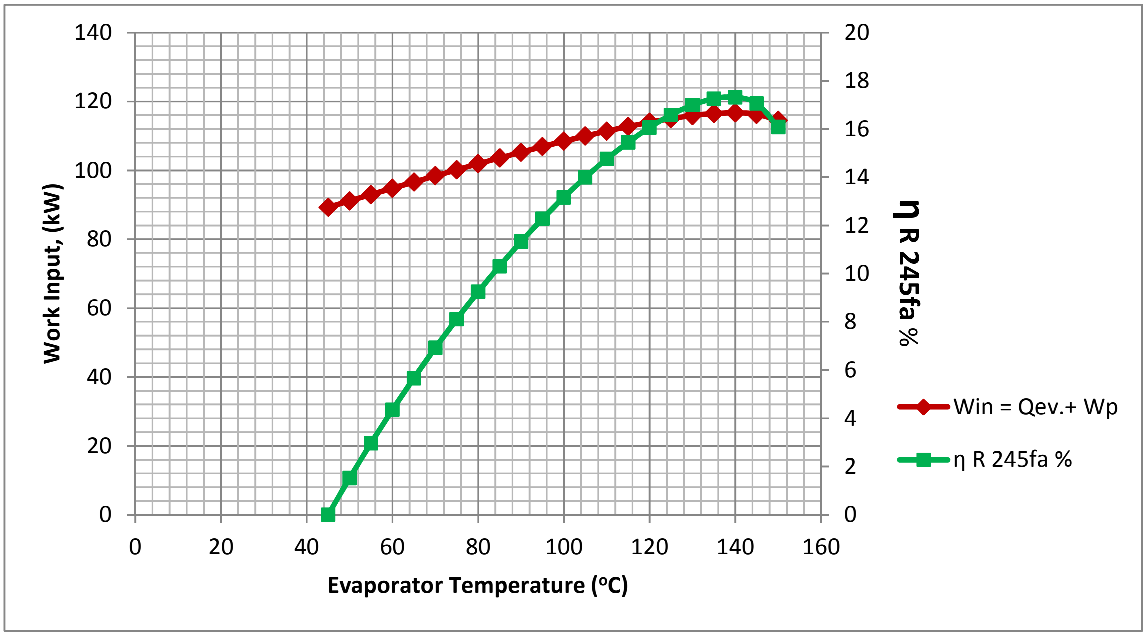
APPENDIX B

ORC PERFORMANCE ANALYSIS WITH EVAPORATOR TEMPERATURE VARIATION

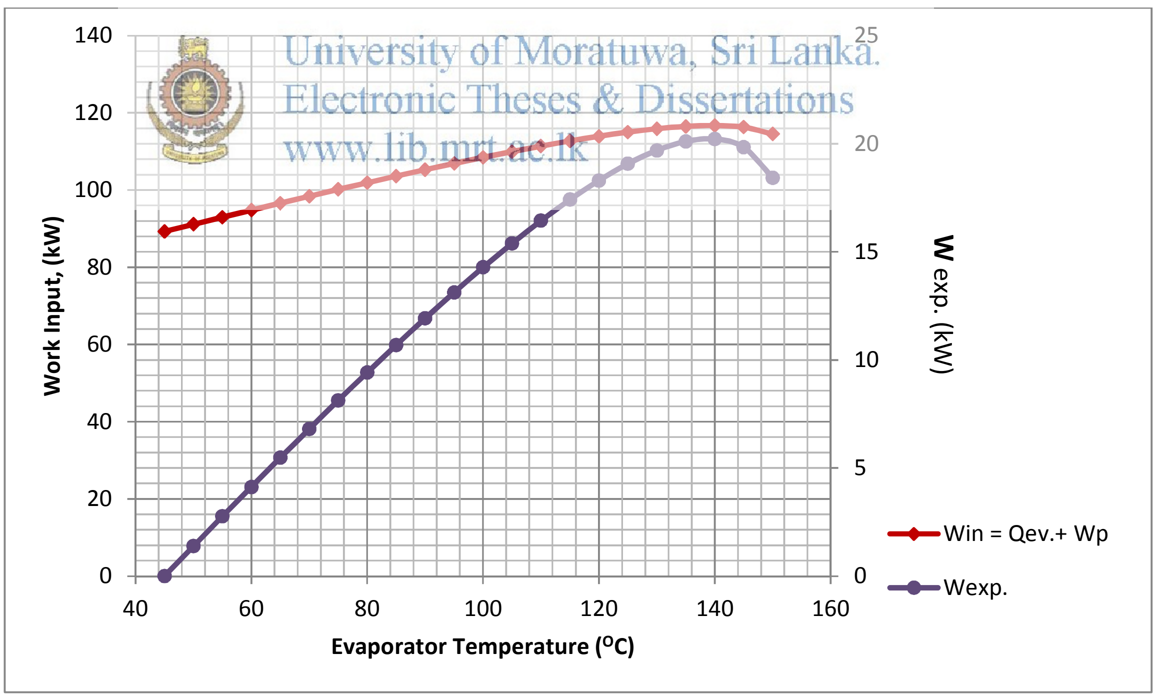
R245fa as working fluid



Graph B.1: Variation of Work component with different Evaporator temperatures for R245fa

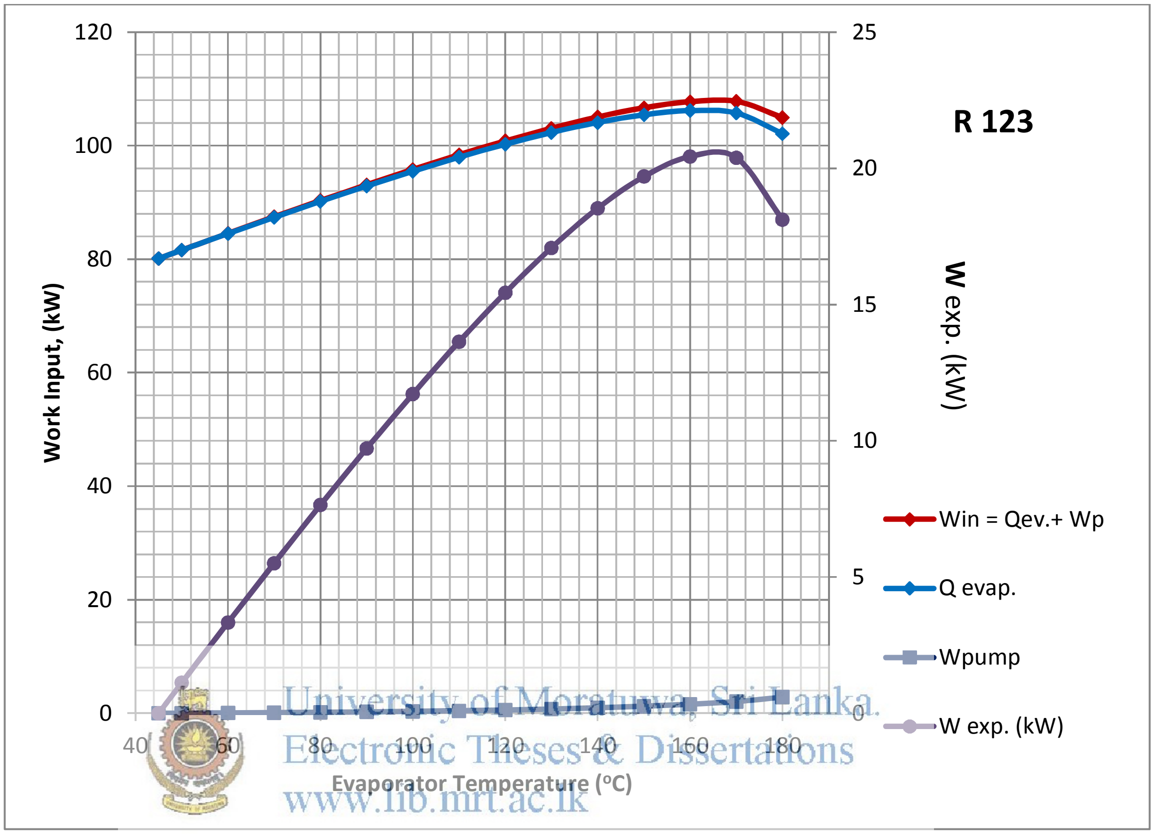


Graph B.2: Work Input and Efficiency variation with different Evaporator temperatures for R245fa

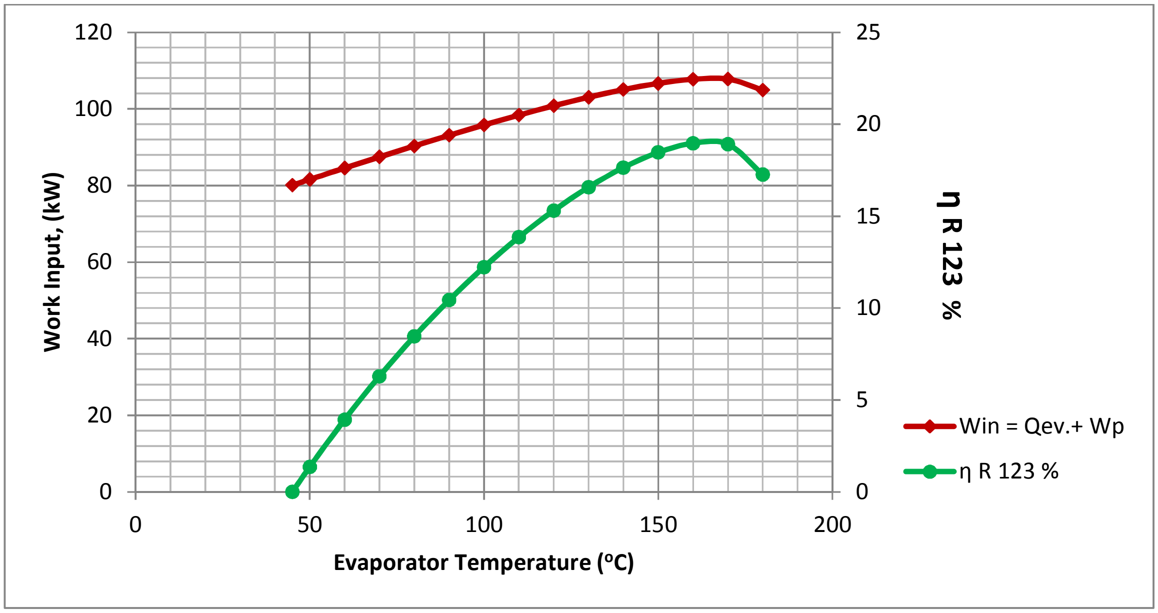


Graph B.3: Work Input and Work at Expander variation with different Evaporator temperatures for R245fa

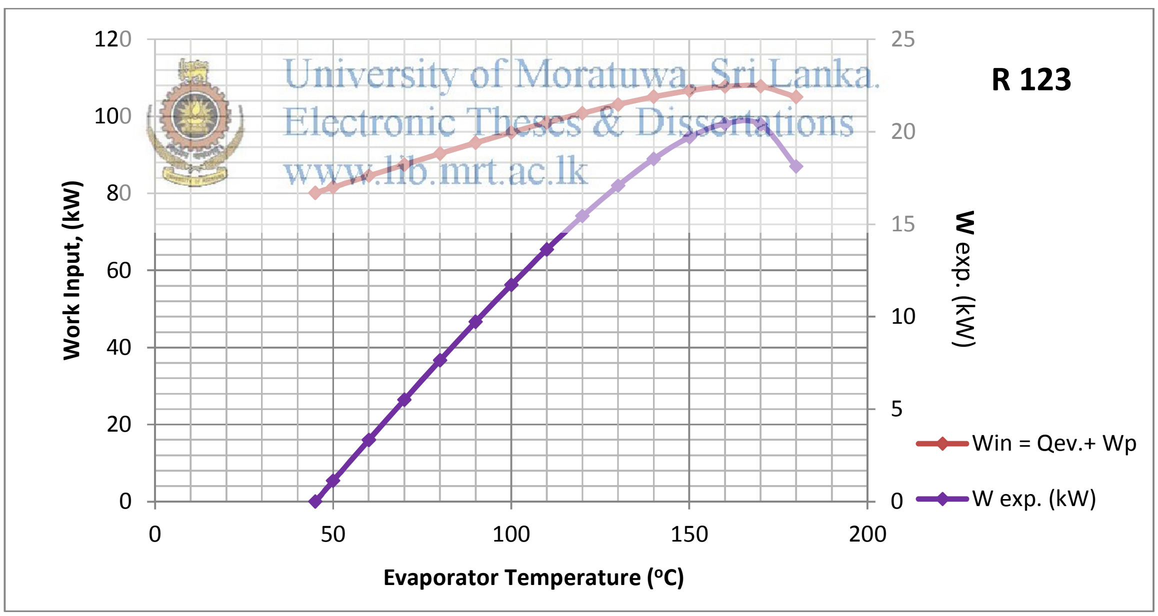
R123 as working fluid



Graph B.4: Variation of Work component in the cycle with different Evaporator temperatures for R123

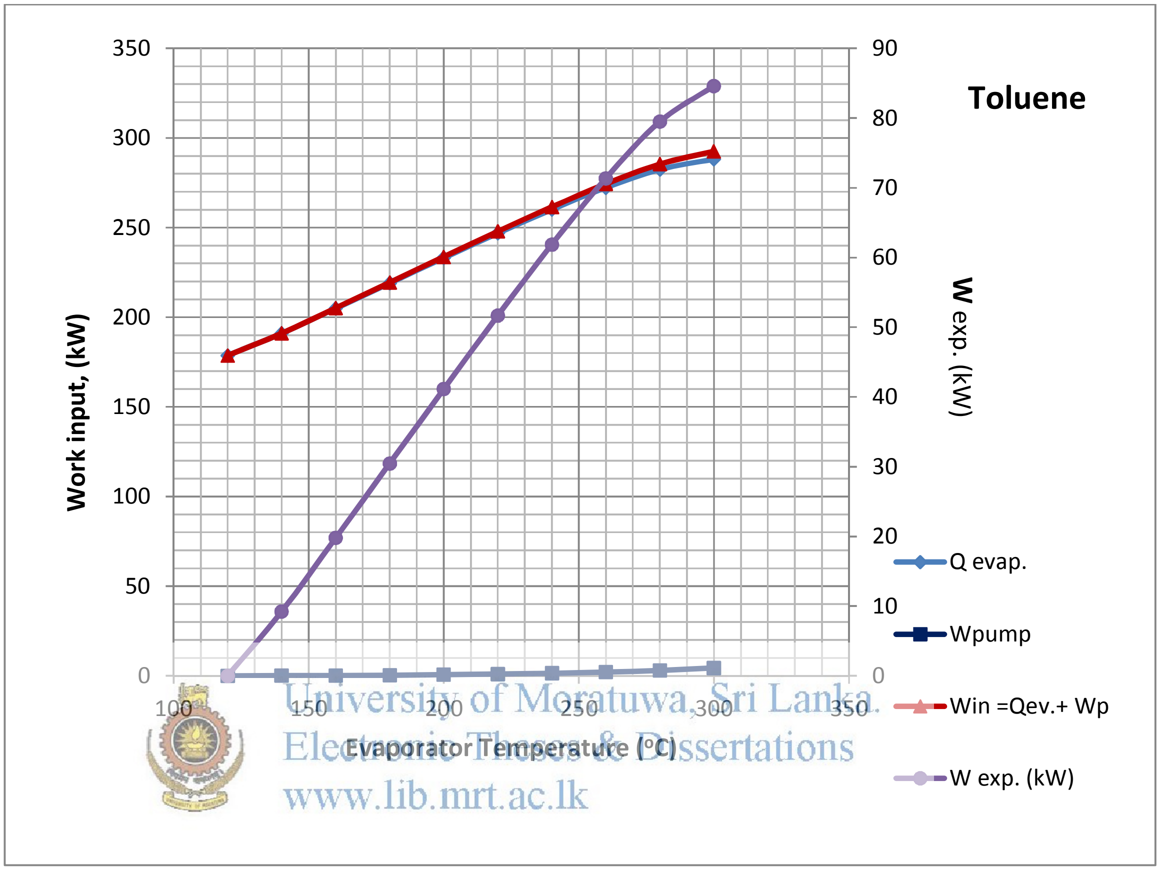


Graph B.5: Work Input and Efficiency variation with different Evaporator temperatures for R123

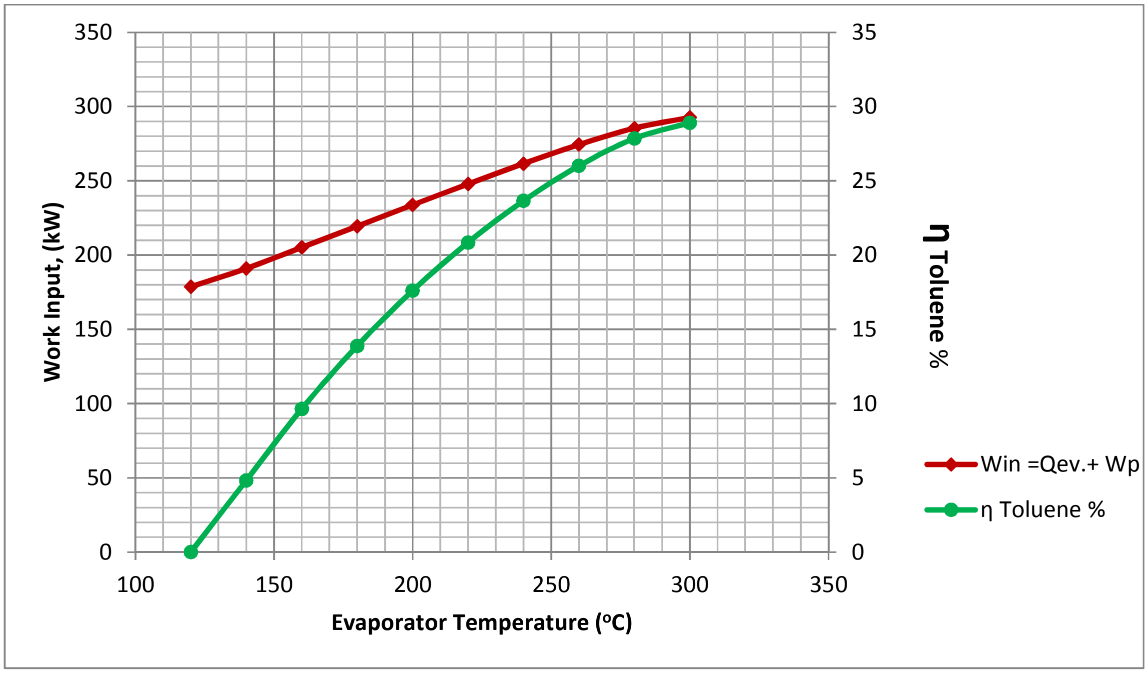


Graph B.6: Work Input and Work at Expander variation with different Evaporator temperatures for R123

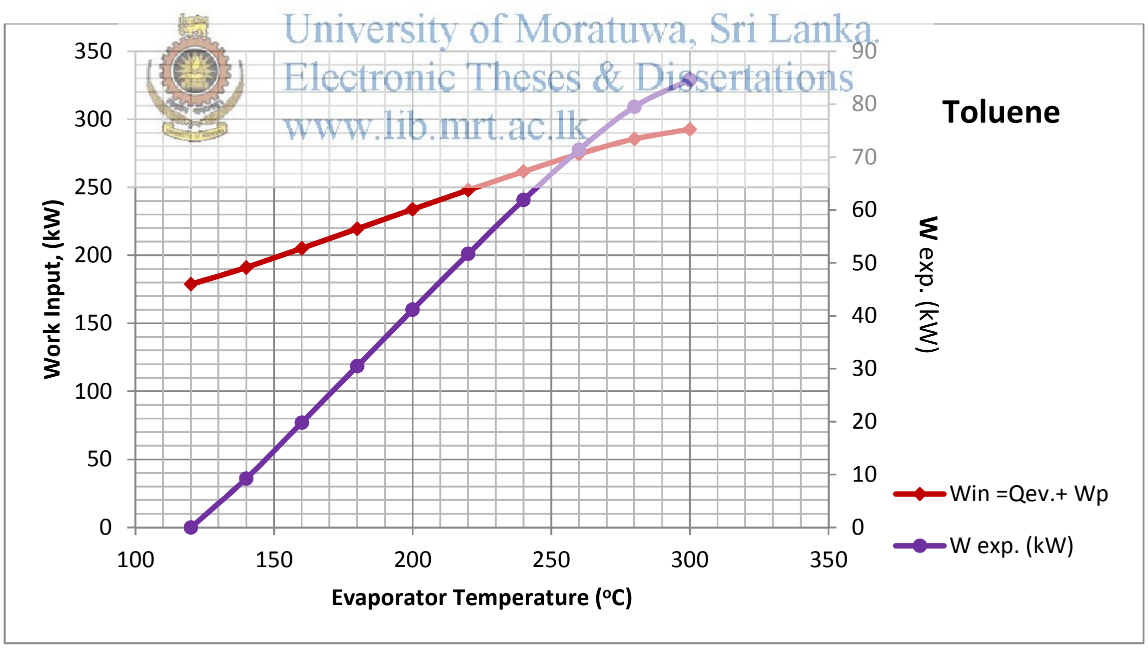
Toluene as working fluid



Graph B.7: Variation of Work component in the cycle with different Evaporator temperatures for Toluene

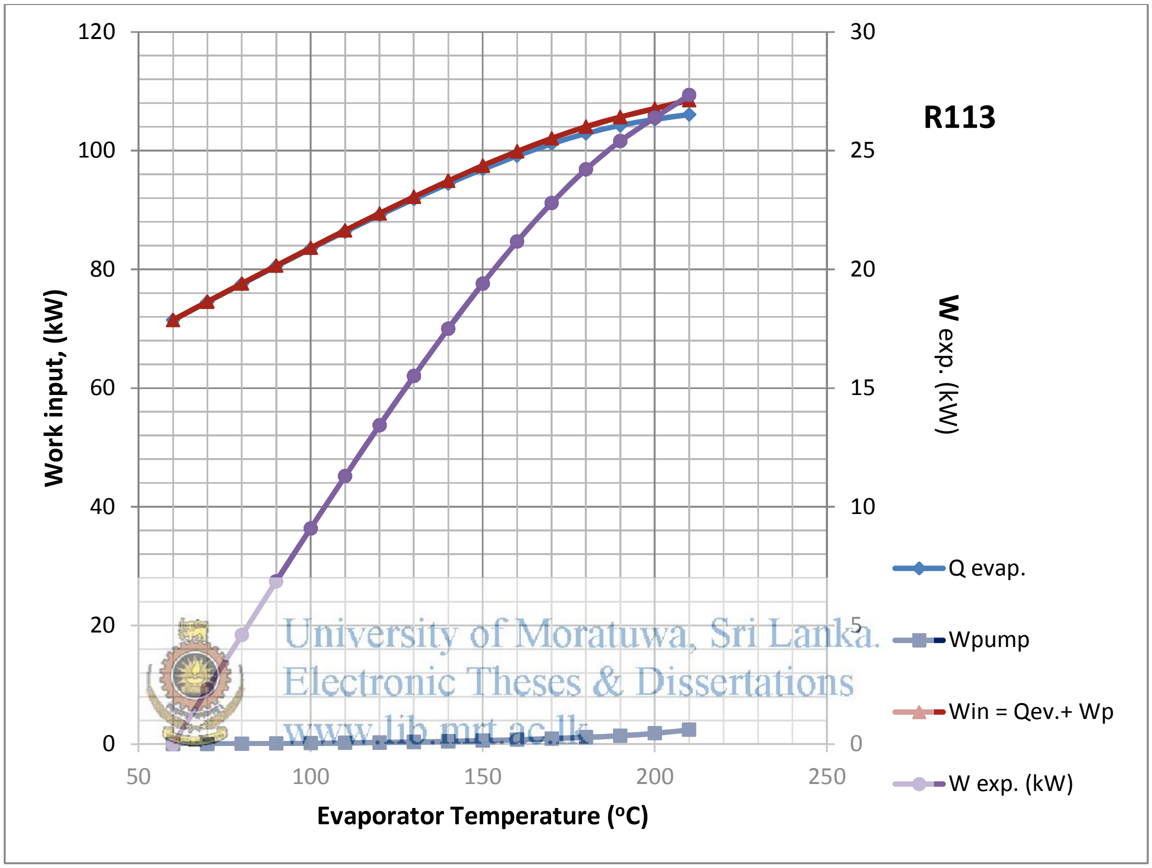


Graph B.8: Work Input and Efficiency variation with different Evaporator temperatures for Toluene

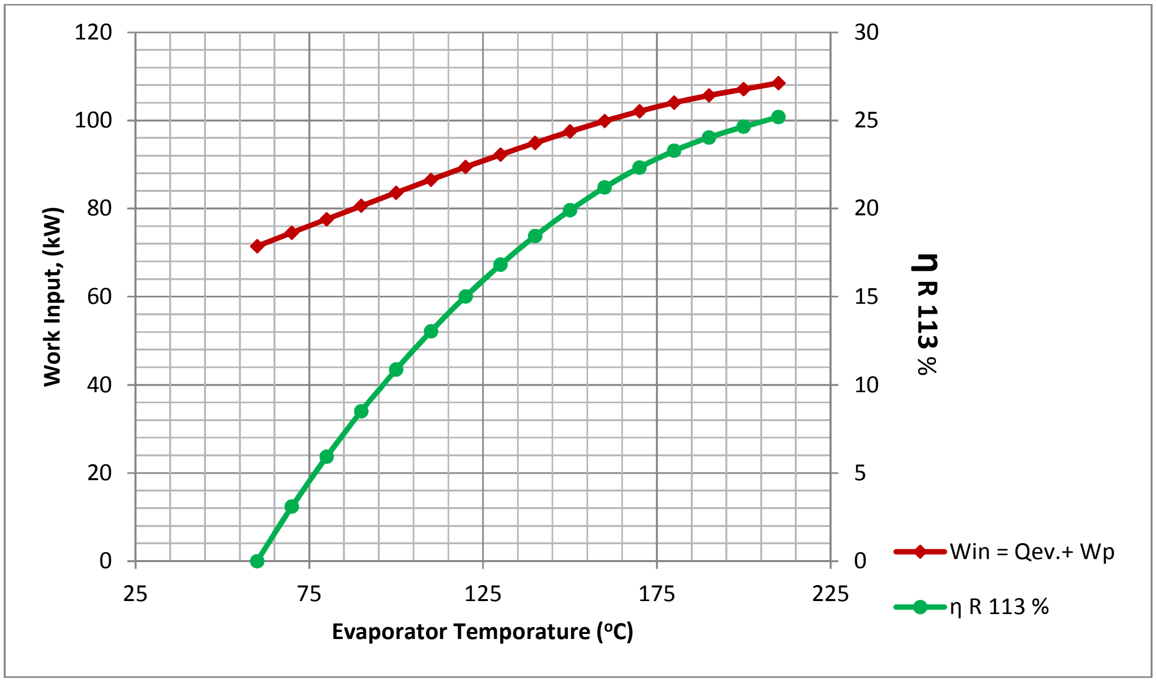


Graph B.9: Work Input and Work at Expander variation with different Evaporator temperatures for Toluene

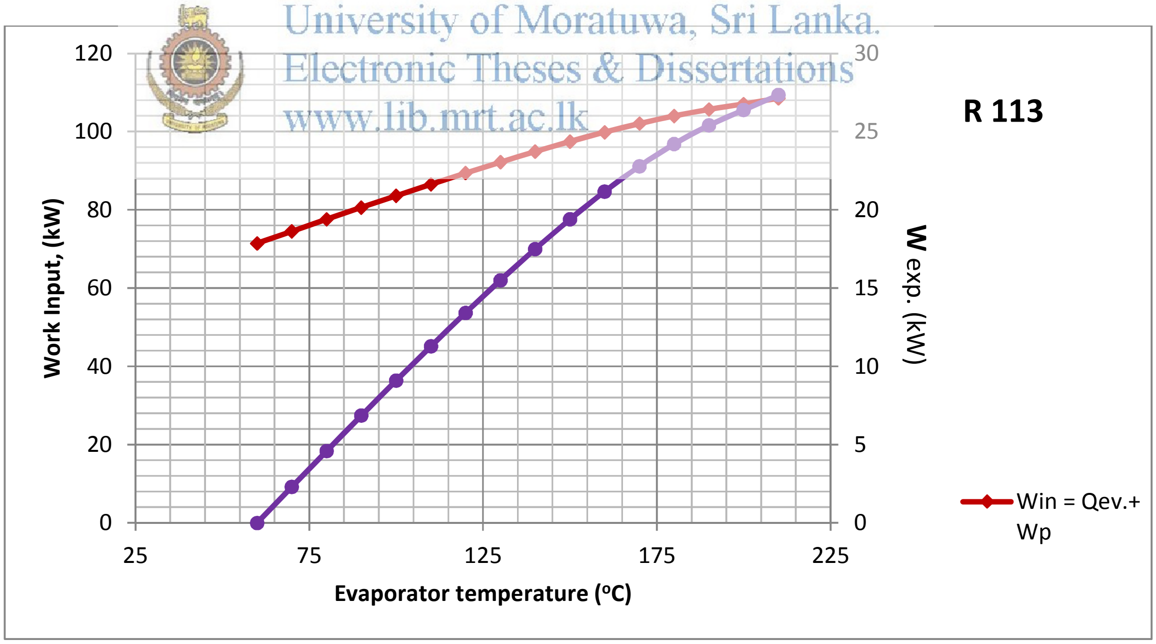
R113 refrigerant as working fluid



Graph B.10: Variation of Work component in the cycle with different Evaporator temperatures for R113

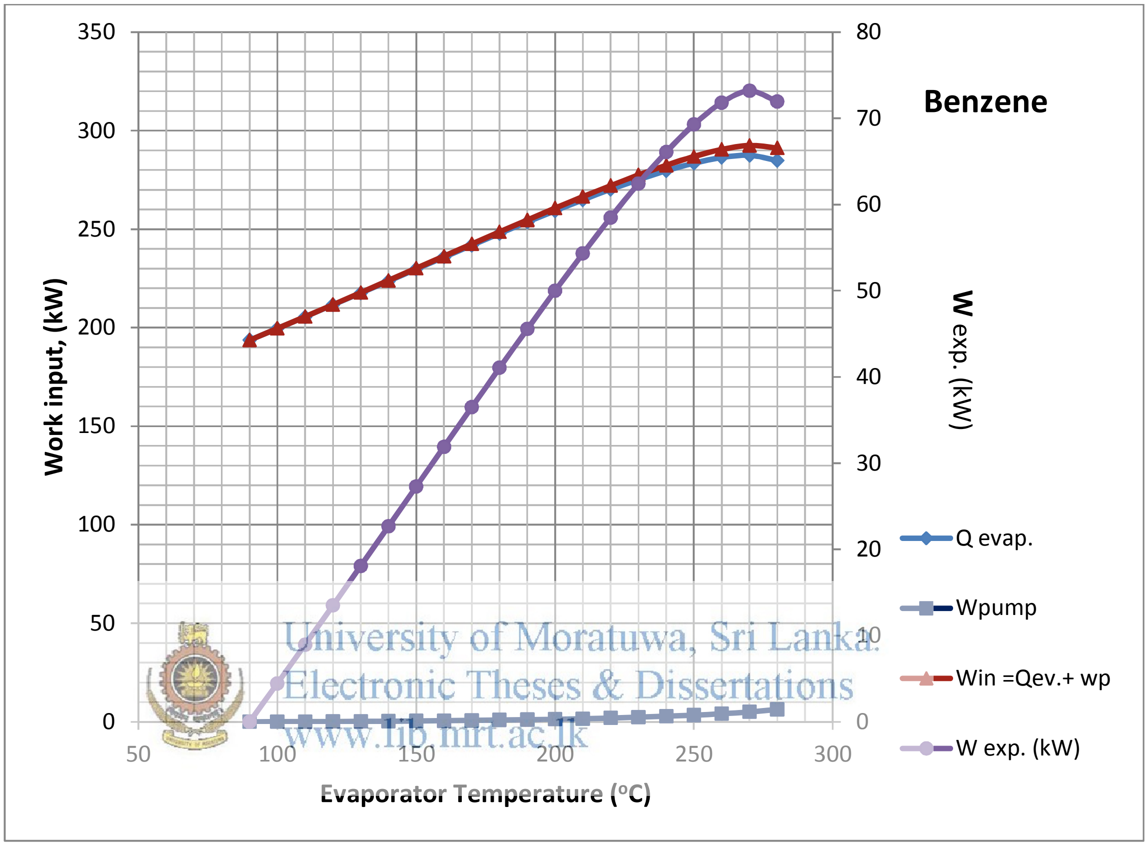


Graph B.11: Work Input and Efficiency variation with different Evaporator temperatures for R113

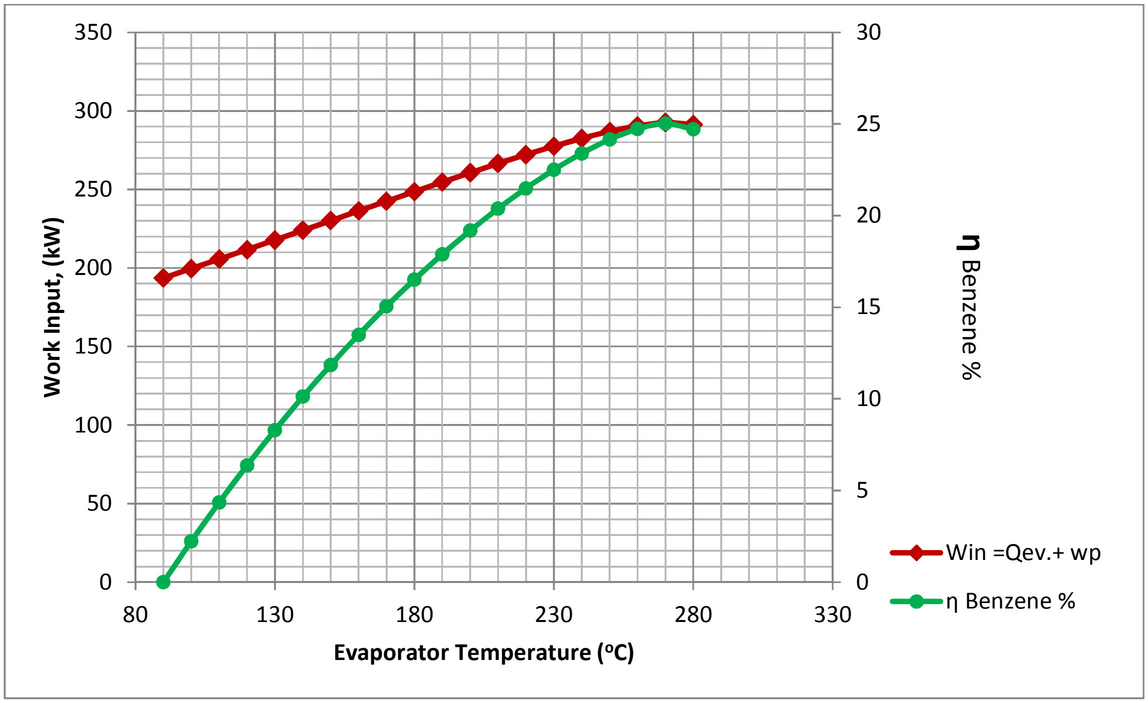


Graph B.12: Work Input and Work at Expander variation with different Evaporator temperatures for R113

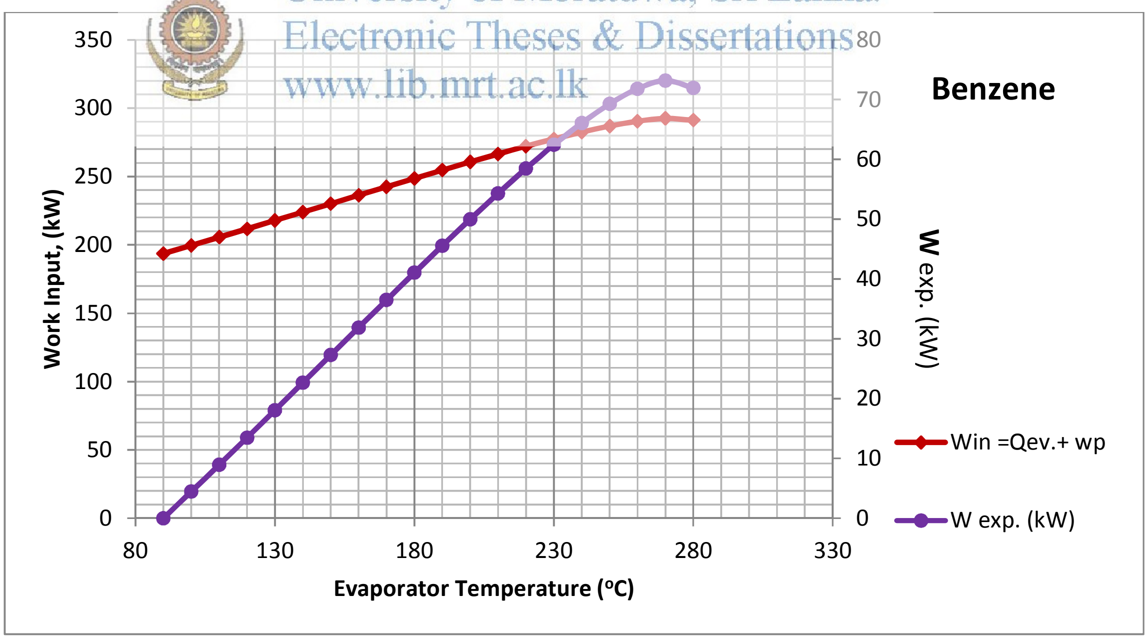
Benzene as working fluid



Graph B.13: Variation of Work component in the cycle with different Evaporator temperatures for Benzene

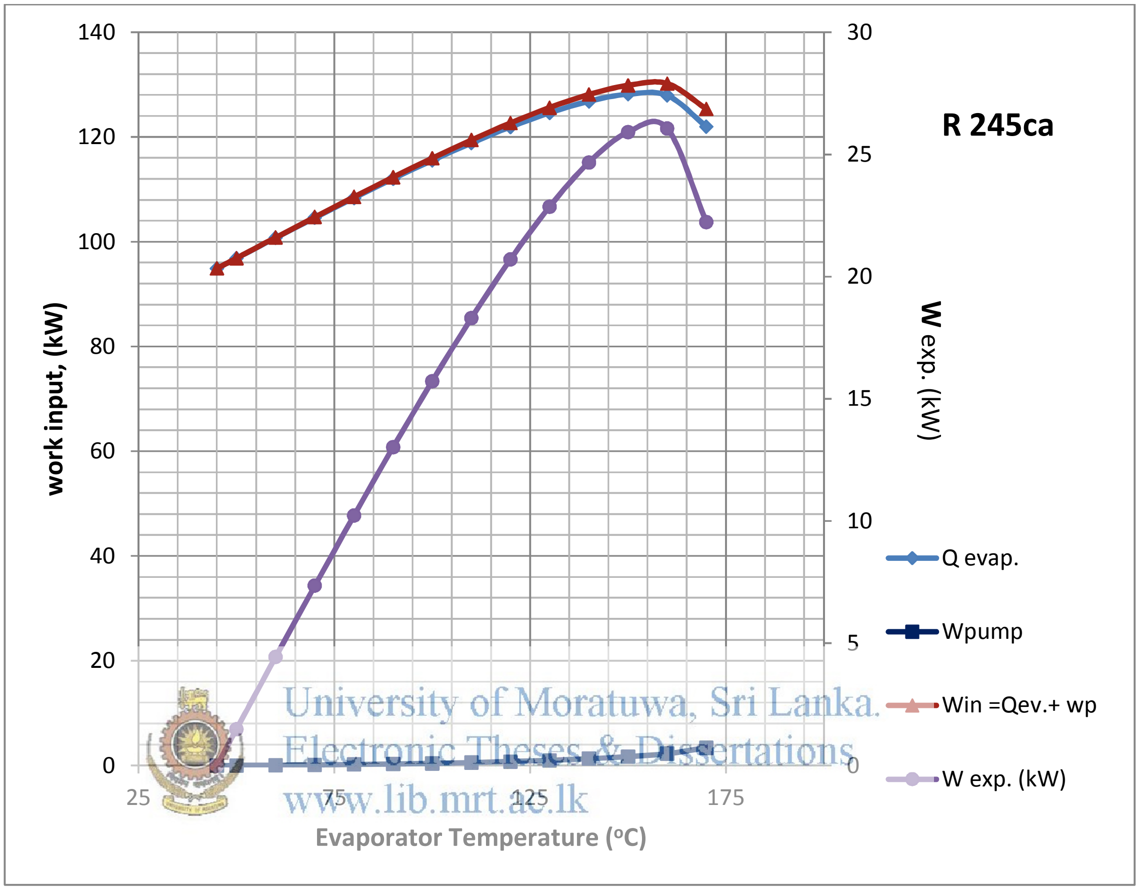


Graph B.14: Work Input and Efficiency variation with different Evaporator temperatures for Benzene

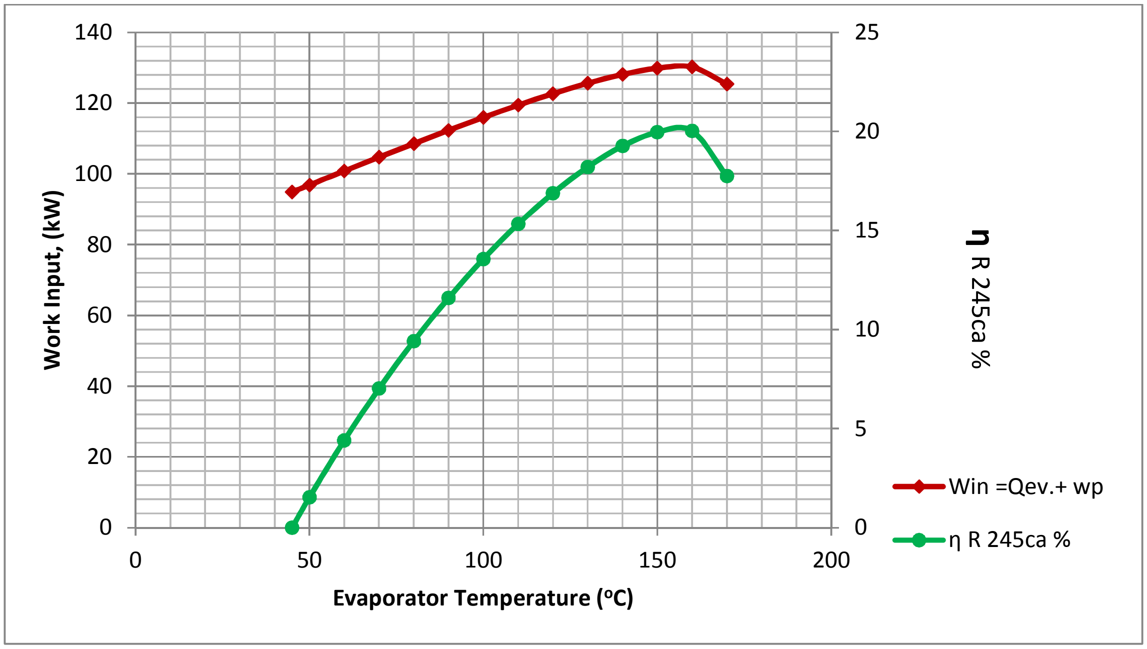


Graph B.15: Work Input and Work at Expander variation with different Evaporator temperatures for Benzene

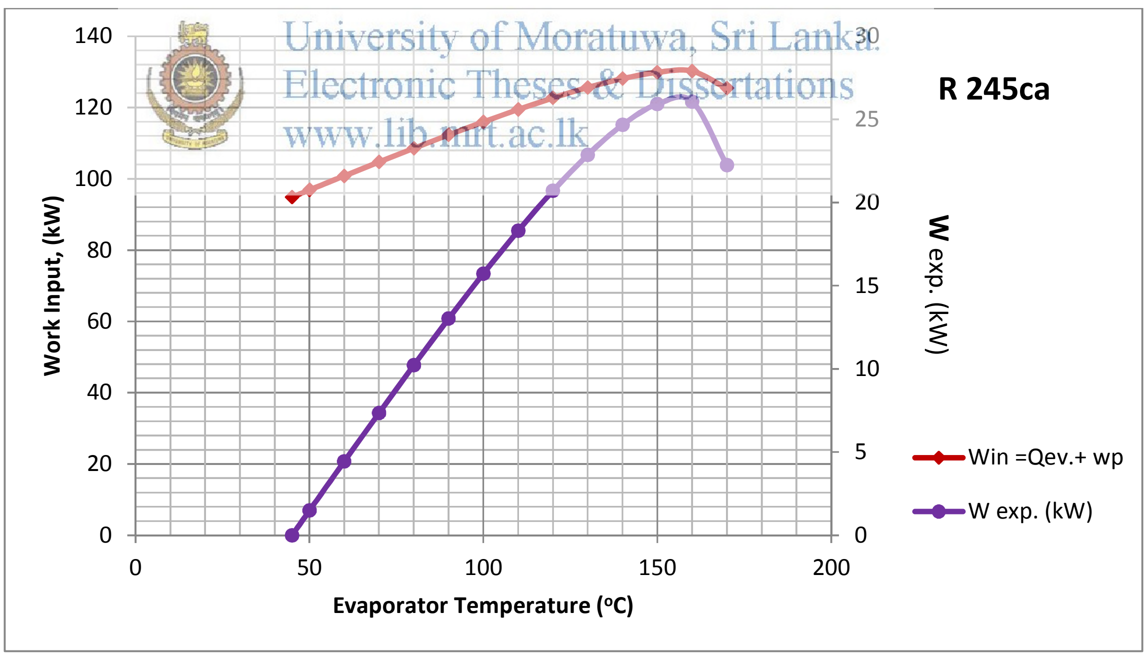
R245ca as working fluid



Graph B.16: Variation of Work component in the cycle with different Evaporator temperatures for R245ca

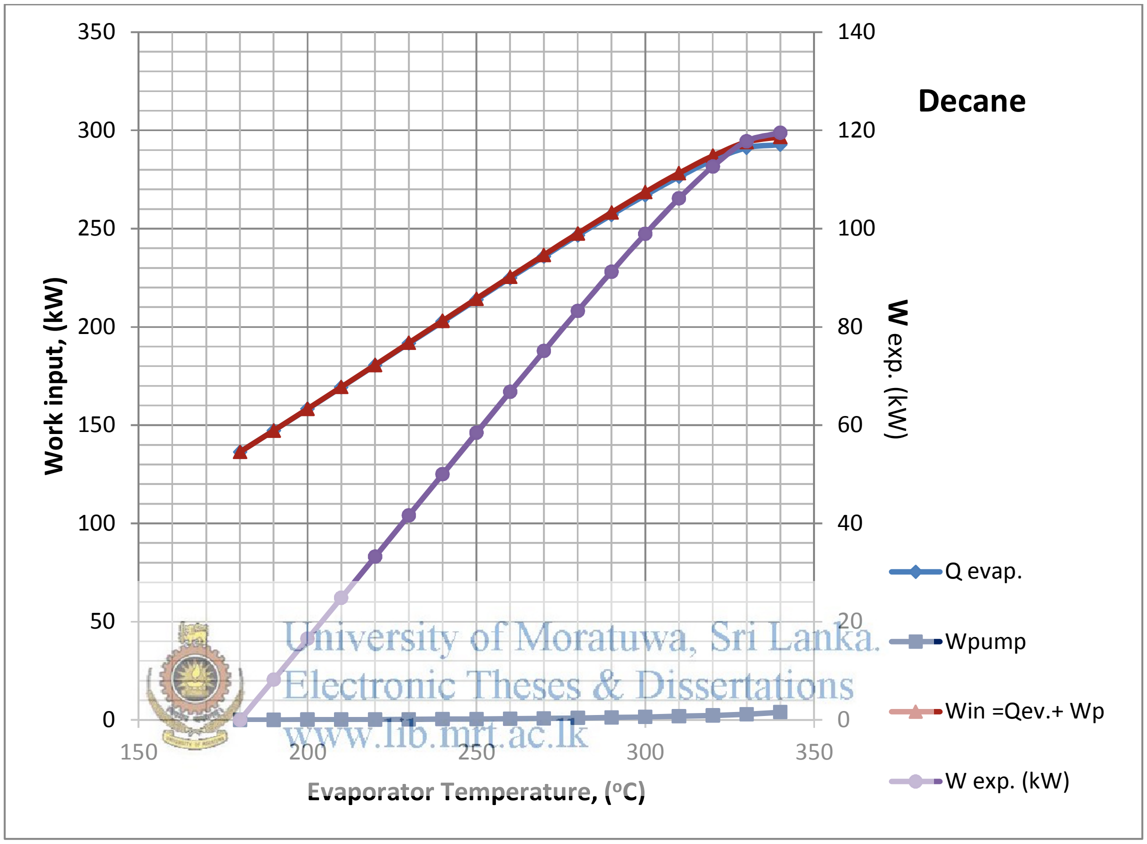


Graph B.17: Work Input and Efficiency variation with different Evaporator temperatures for R245ca

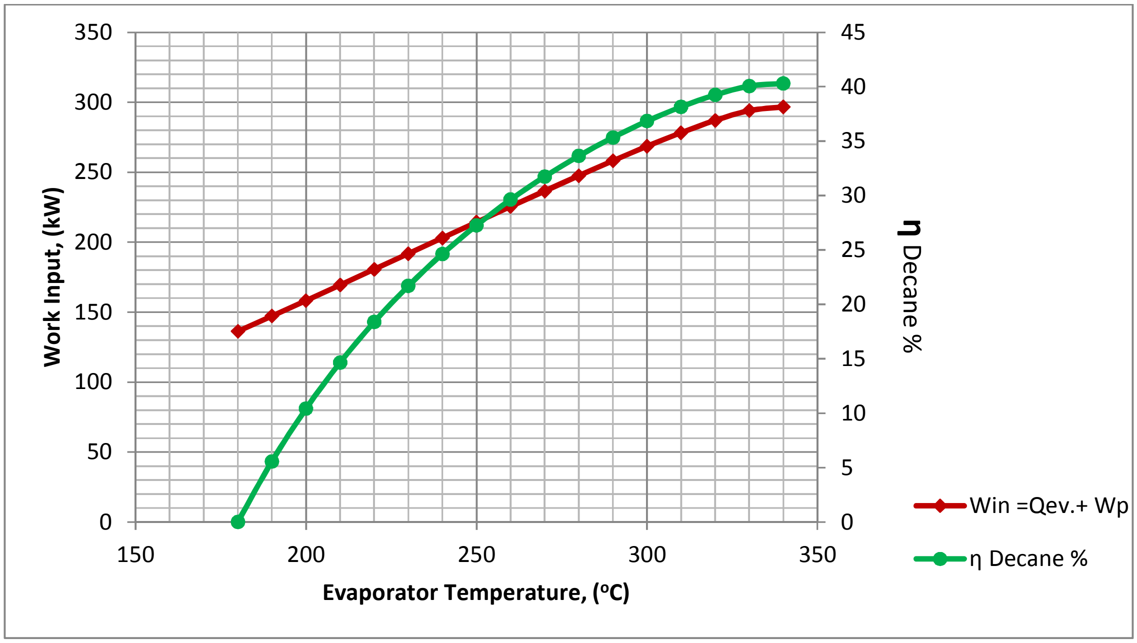


Graph B.18: Work Input and Work at Expander variation with different Evaporator temperatures for R245ca

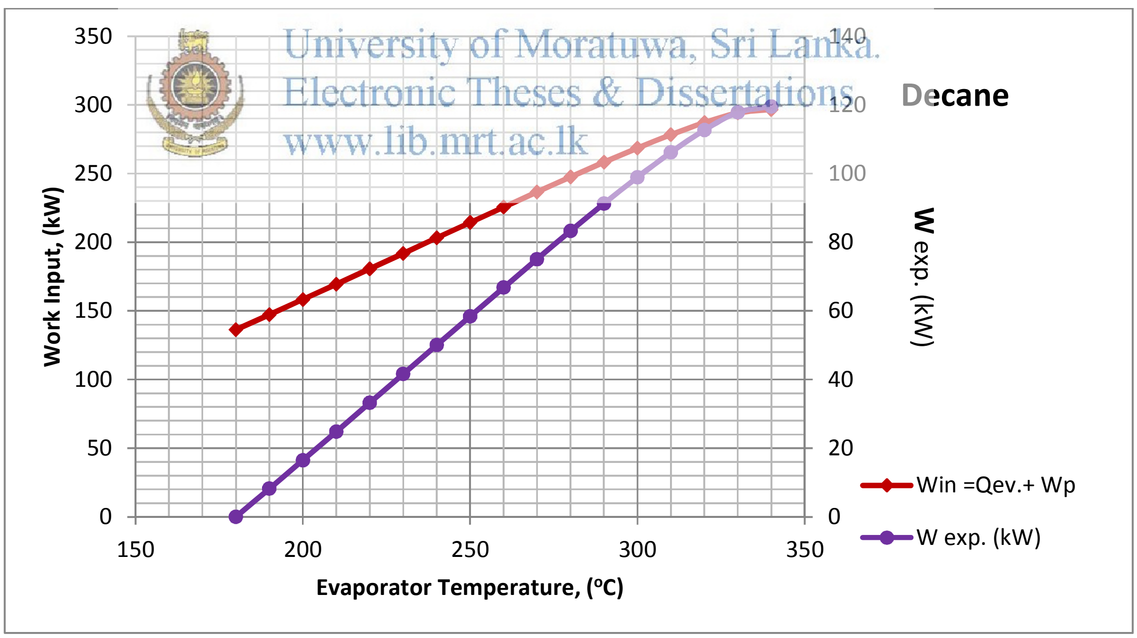
Decane as working fluid



Graph B.19: Variation of Work component in the cycle with different Evaporator temperatures for Decane

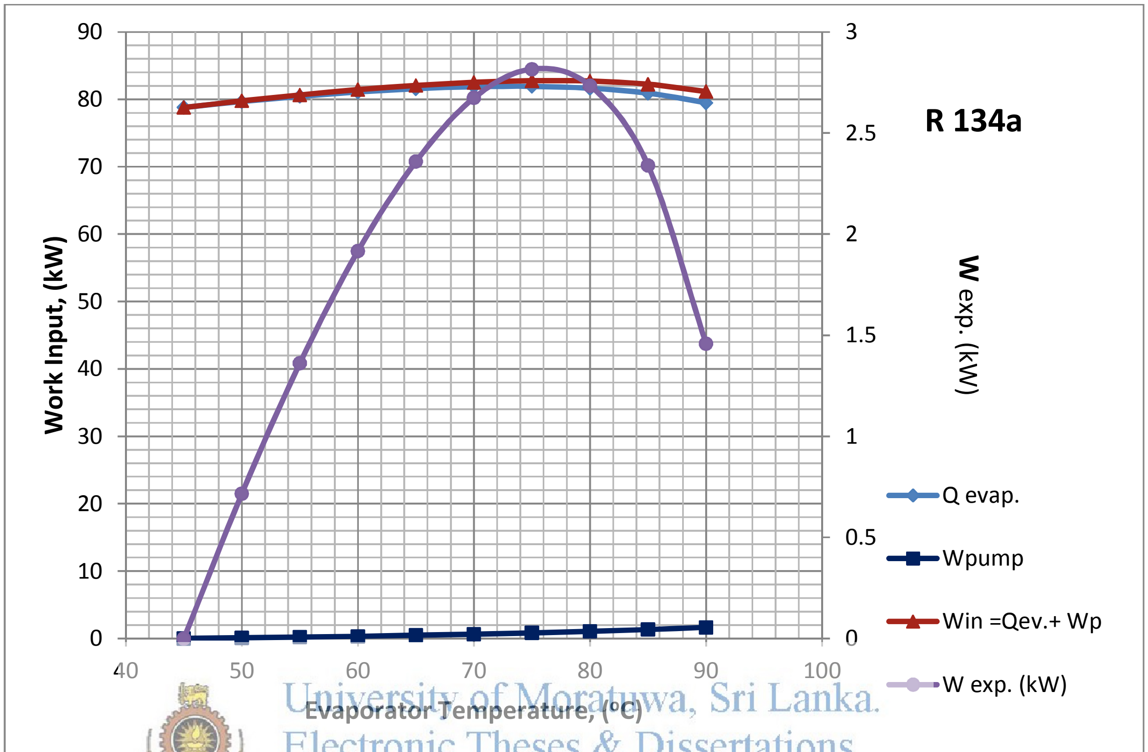


Graph B.20: Work Input and Efficiency variation with different Evaporator temperatures for Decane

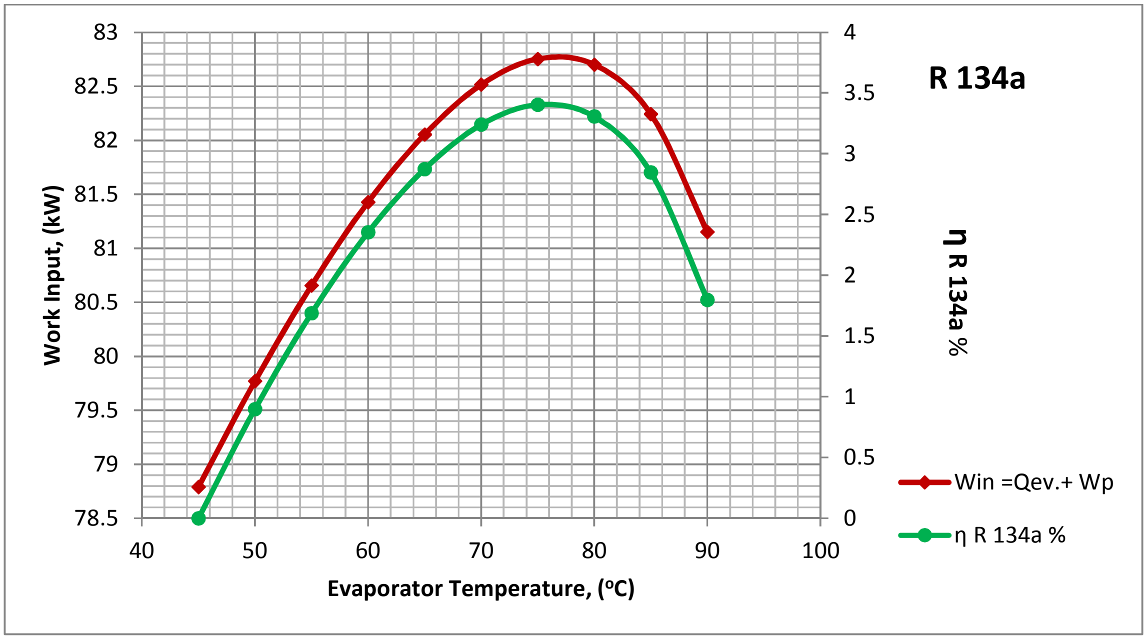


Graph B.21: Work Input and Work at Expander variation with different Evaporator temperatures for Decane

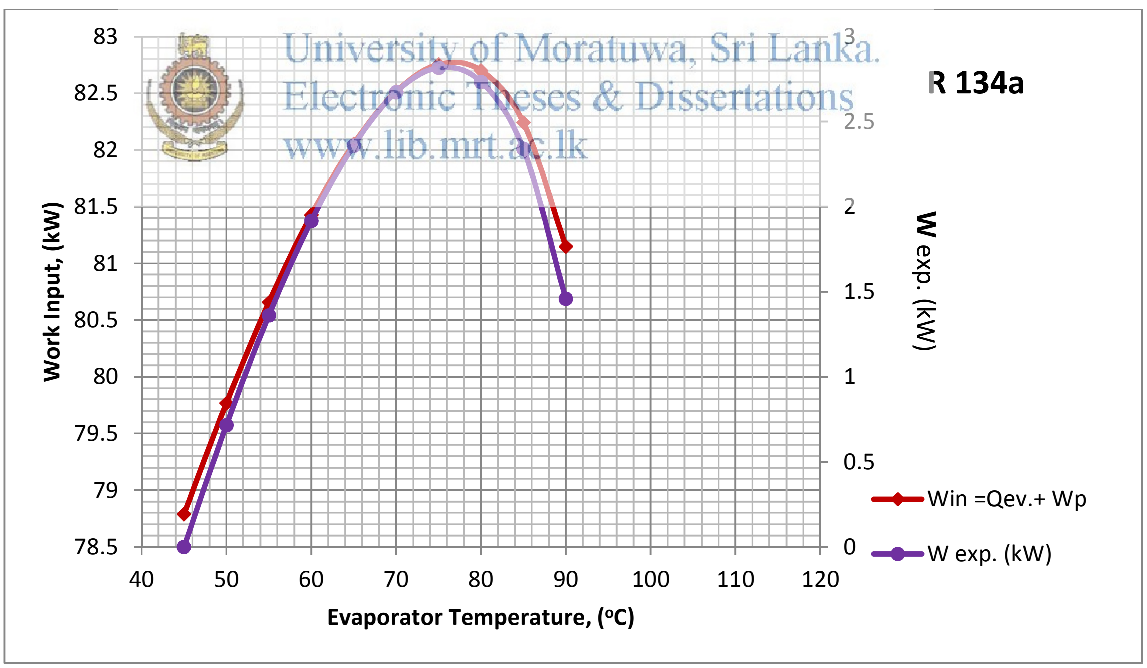
R134a as working fluid



Graph B.22: Variation of Work component in the cycle with different Evaporator temperatures for R134a

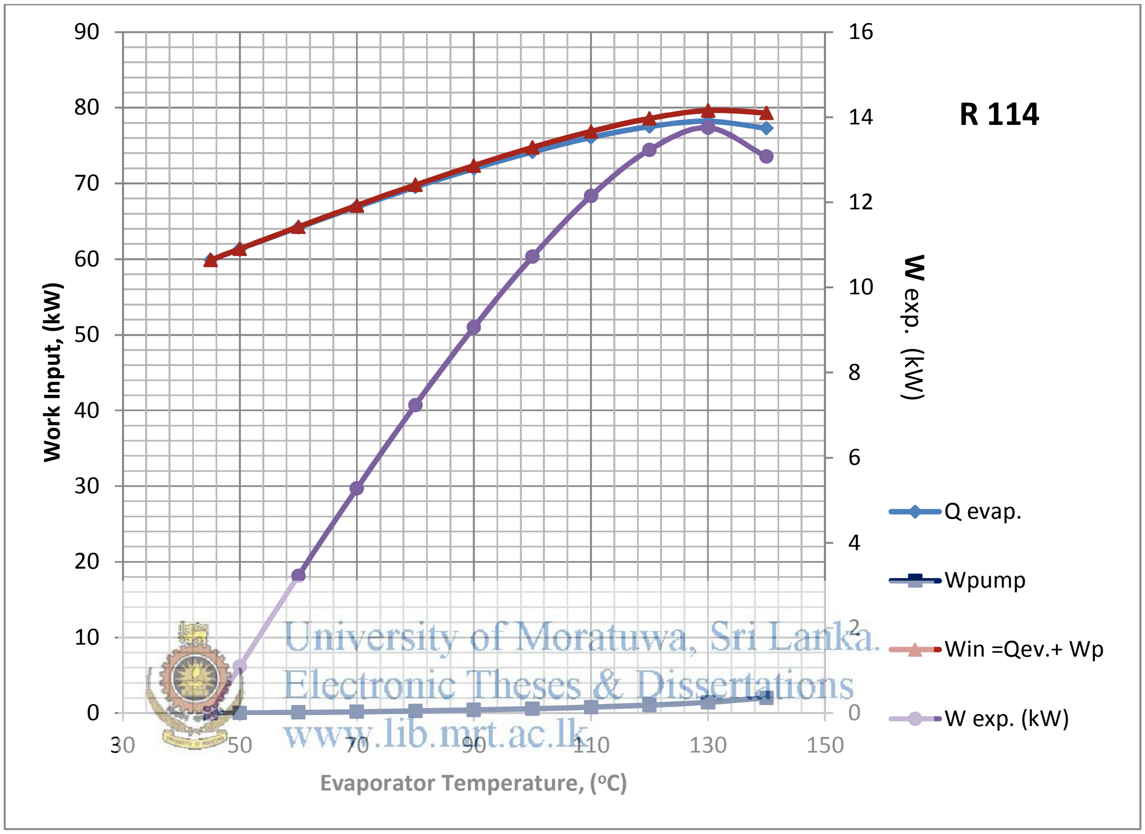


Graph B.23: Work Input and Efficiency variation with different Evaporator temperatures for R134a

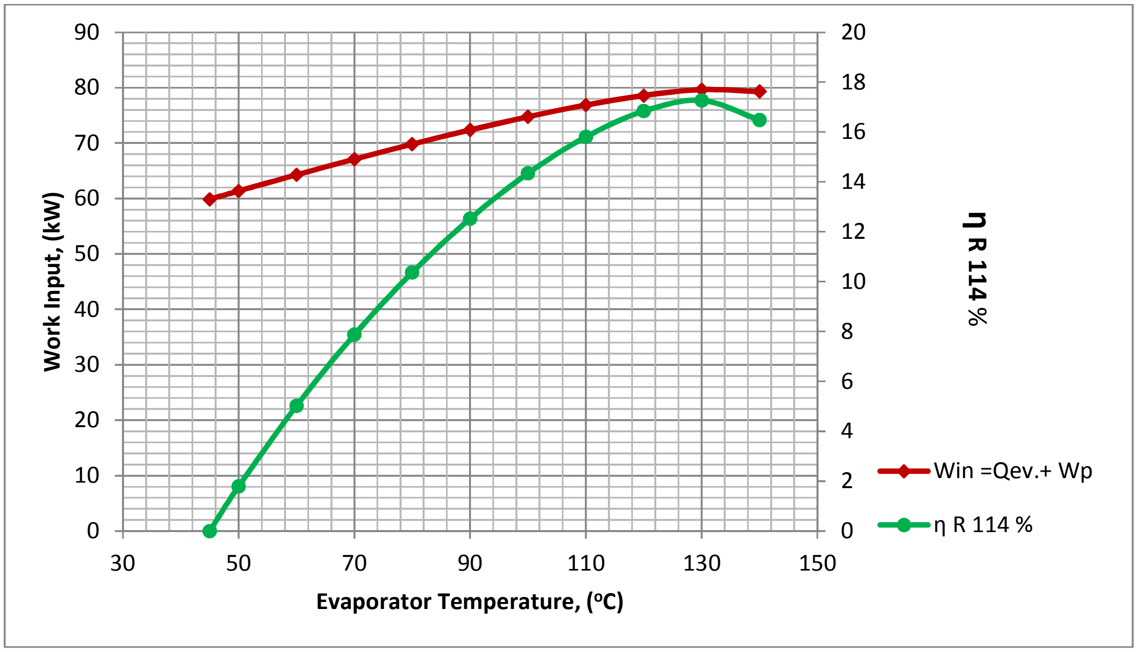


Graph B.24: Work Input and Work at Expander variation with different Evaporator temperatures for R134a

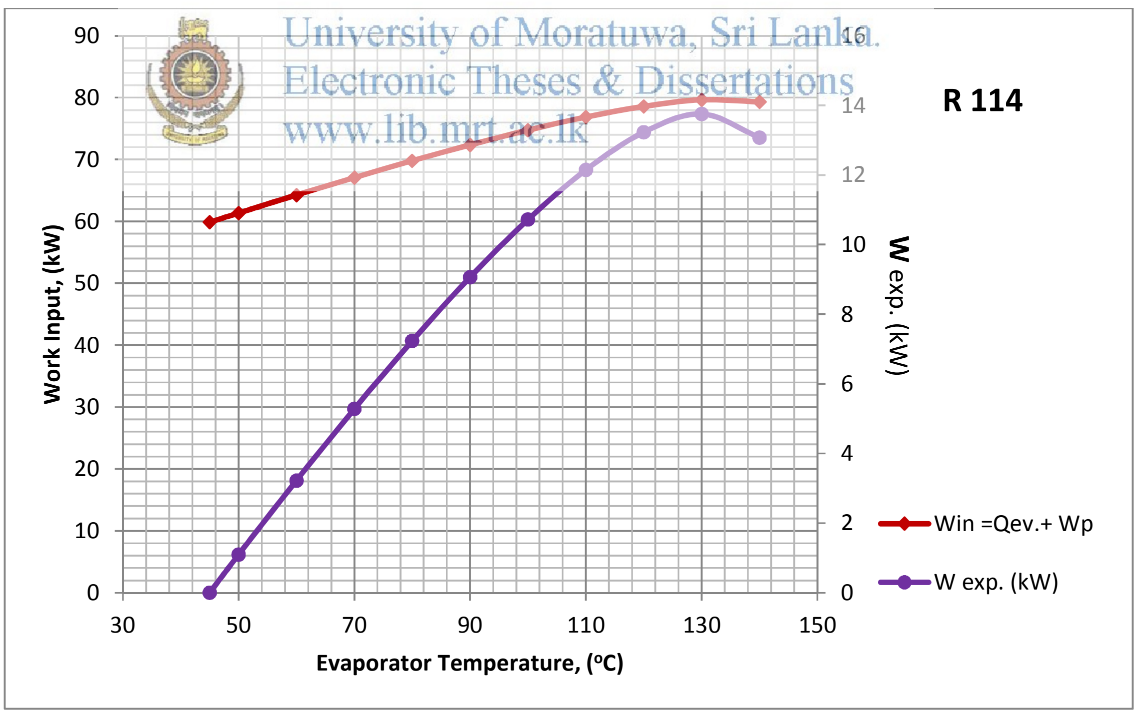
R114 as working fluid



Graph B.25: Variation of Work component in the cycle with different Evaporator temperatures for R114

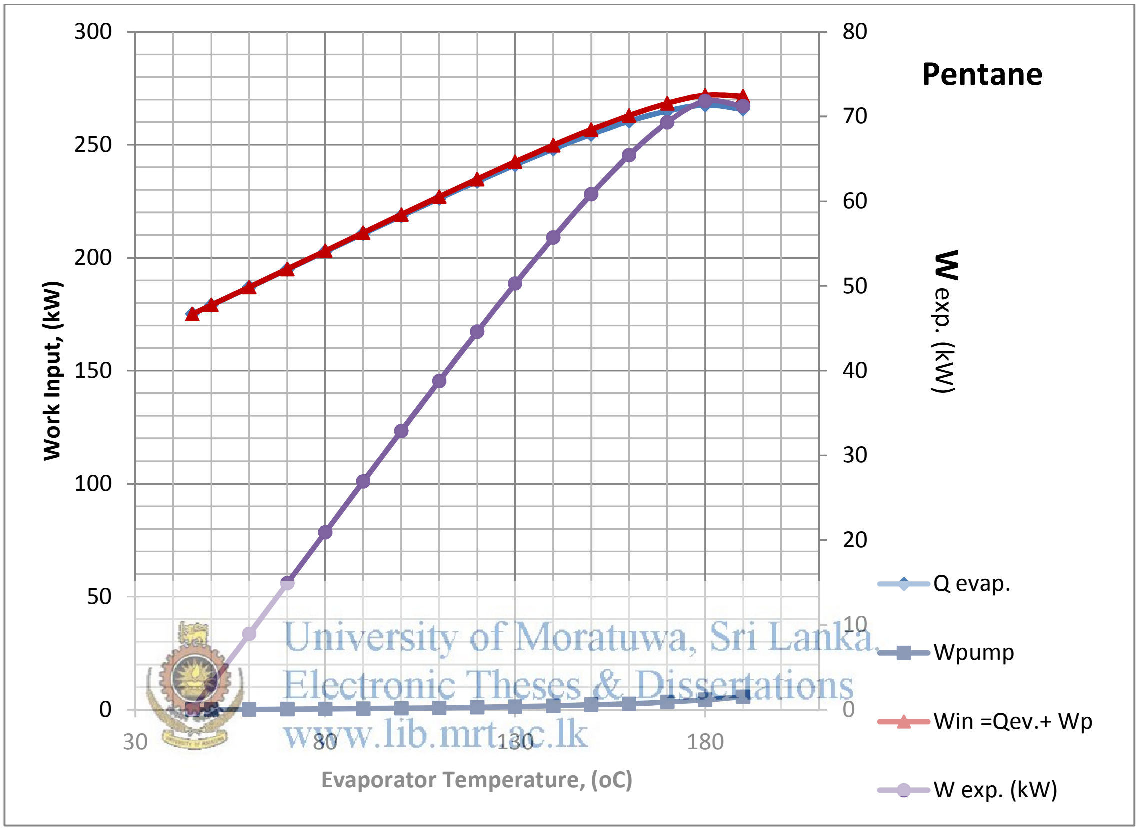


Graph B.26: Work Input and Efficiency variation with different Evaporator temperatures for R114

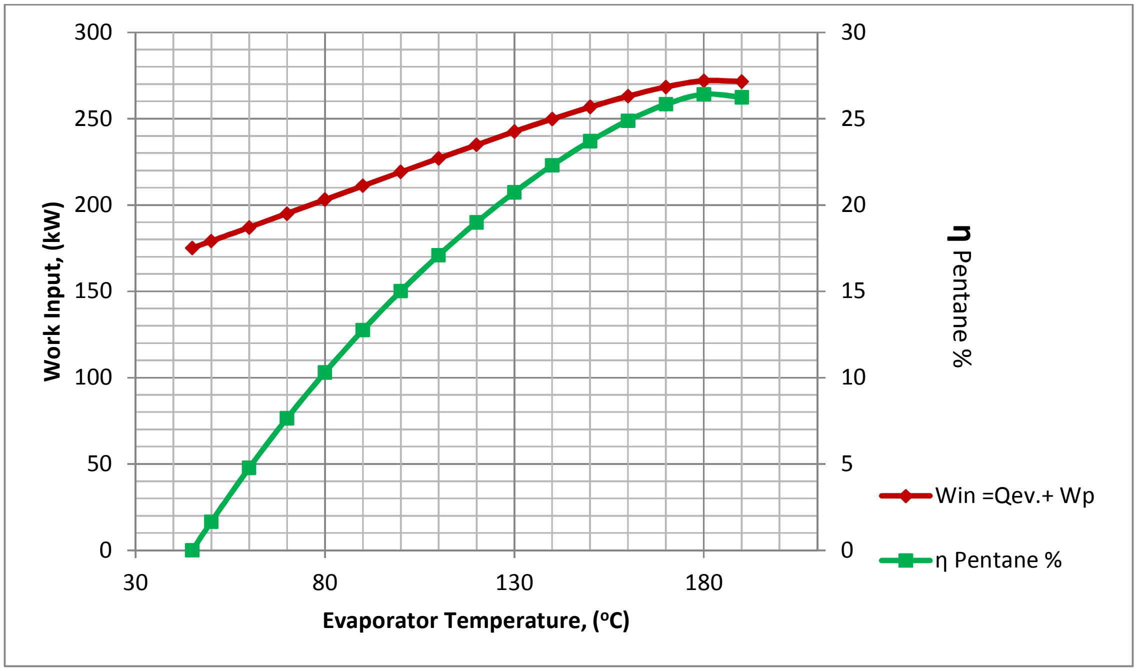


Graph B.27: Work Input and Work at Expander variation with different Evaporator temperatures for R114

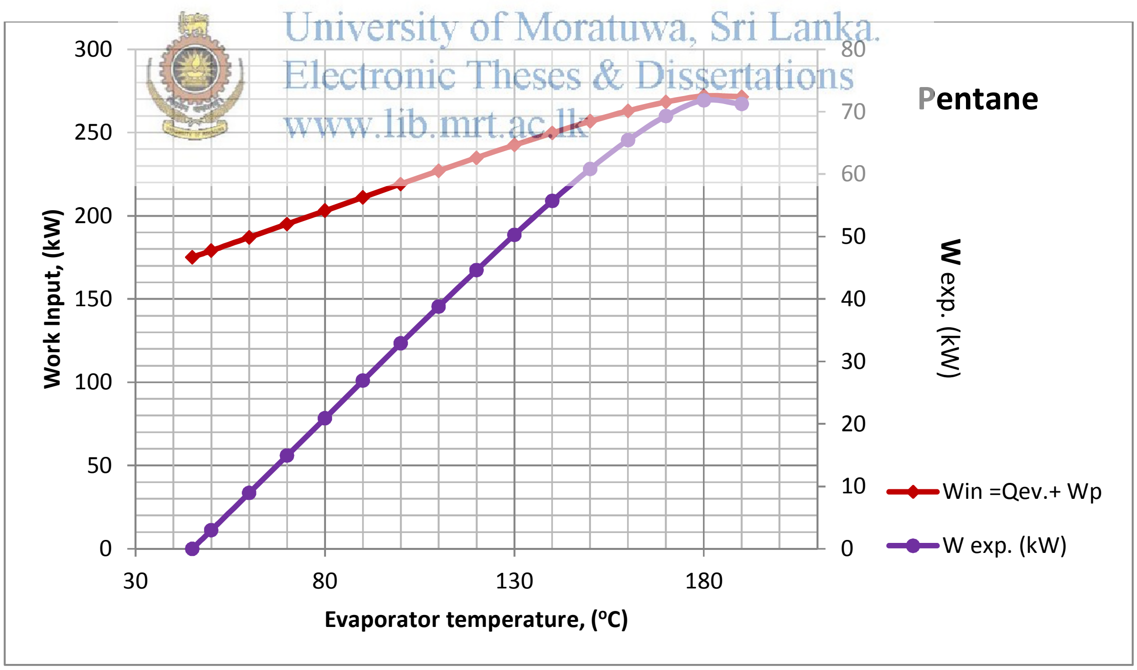
Pentane as working fluid



Graph B.28: Variation of Work component in the cycle with different Evaporator temperatures for Pentane

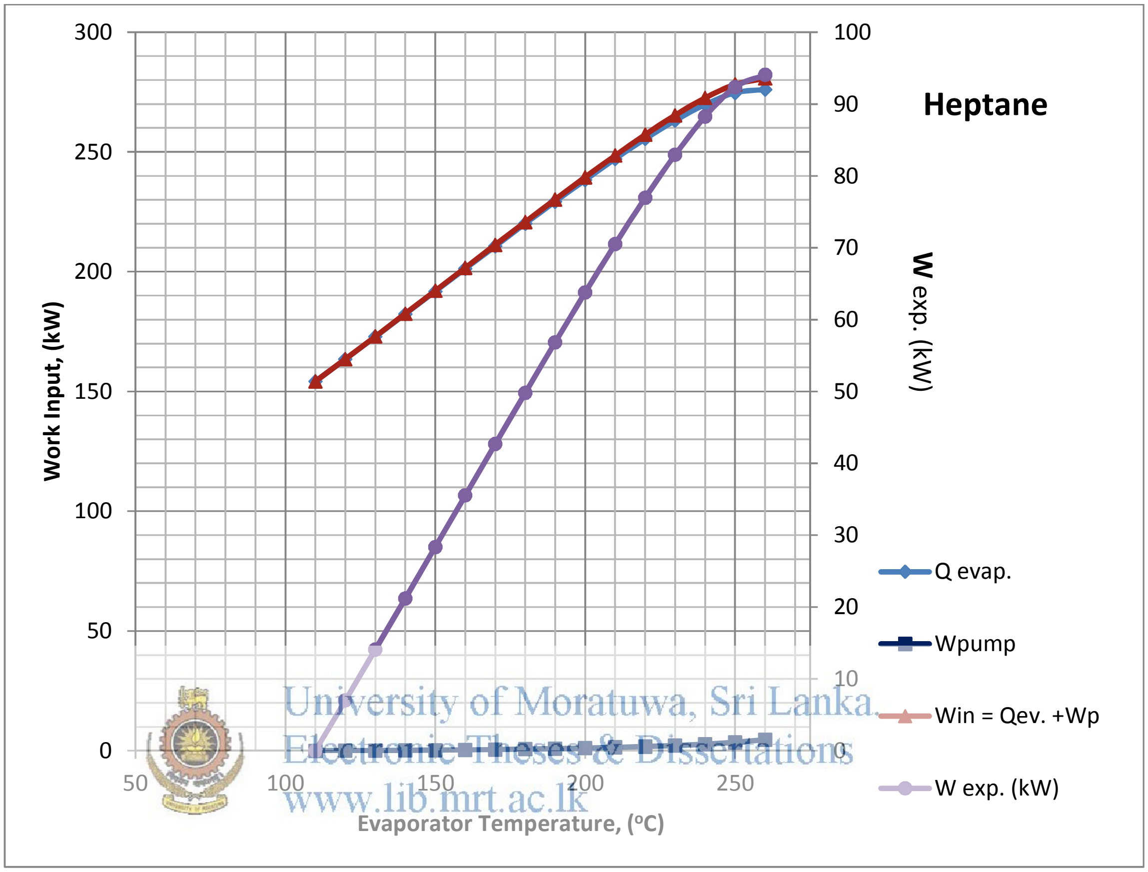


Graph B.29: Work Input and Efficiency variation with different Evaporator temperatures for Pentane

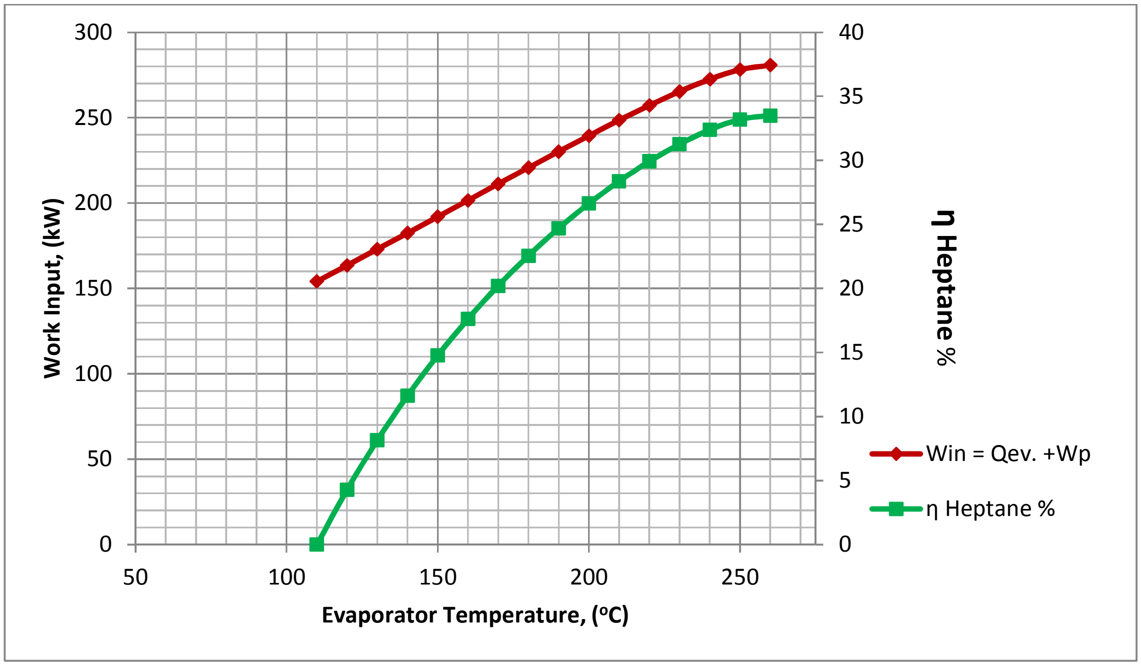


Graph B.30: Work Input and Work at Expander variation with different Evaporator temperatures for Pentane

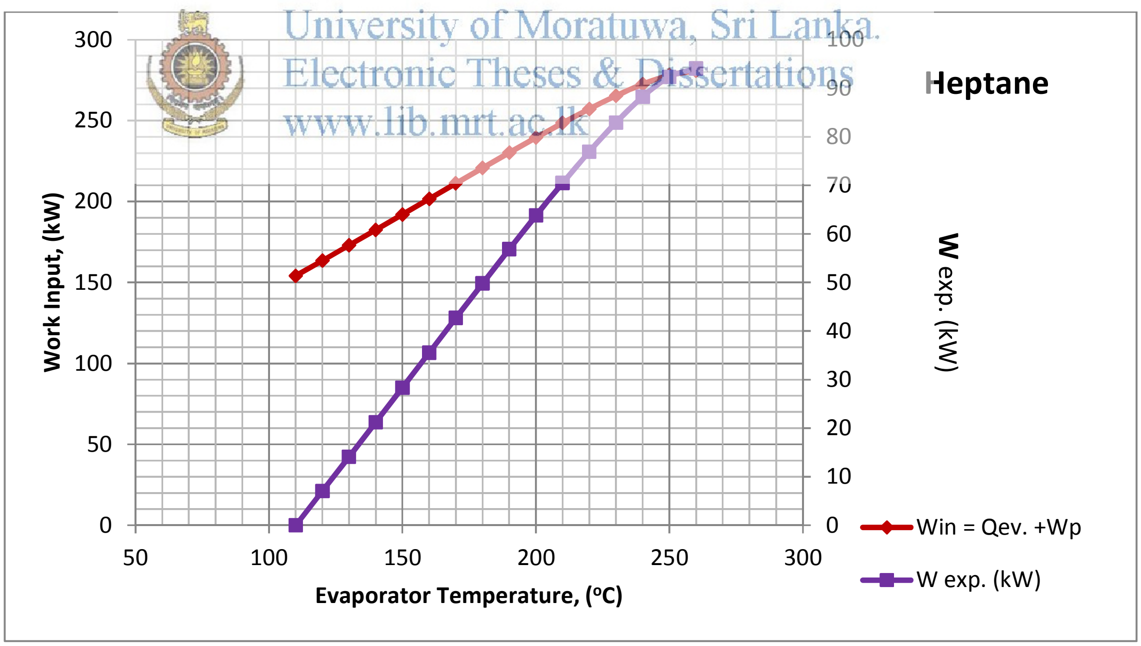
Heptane as working fluid



Graph B.31: Variation of Work component in the cycle with different Evaporator temperatures for Heptane



Graph B.32: Work Input and Efficiency variation with different Evaporator temperatures for Heptane

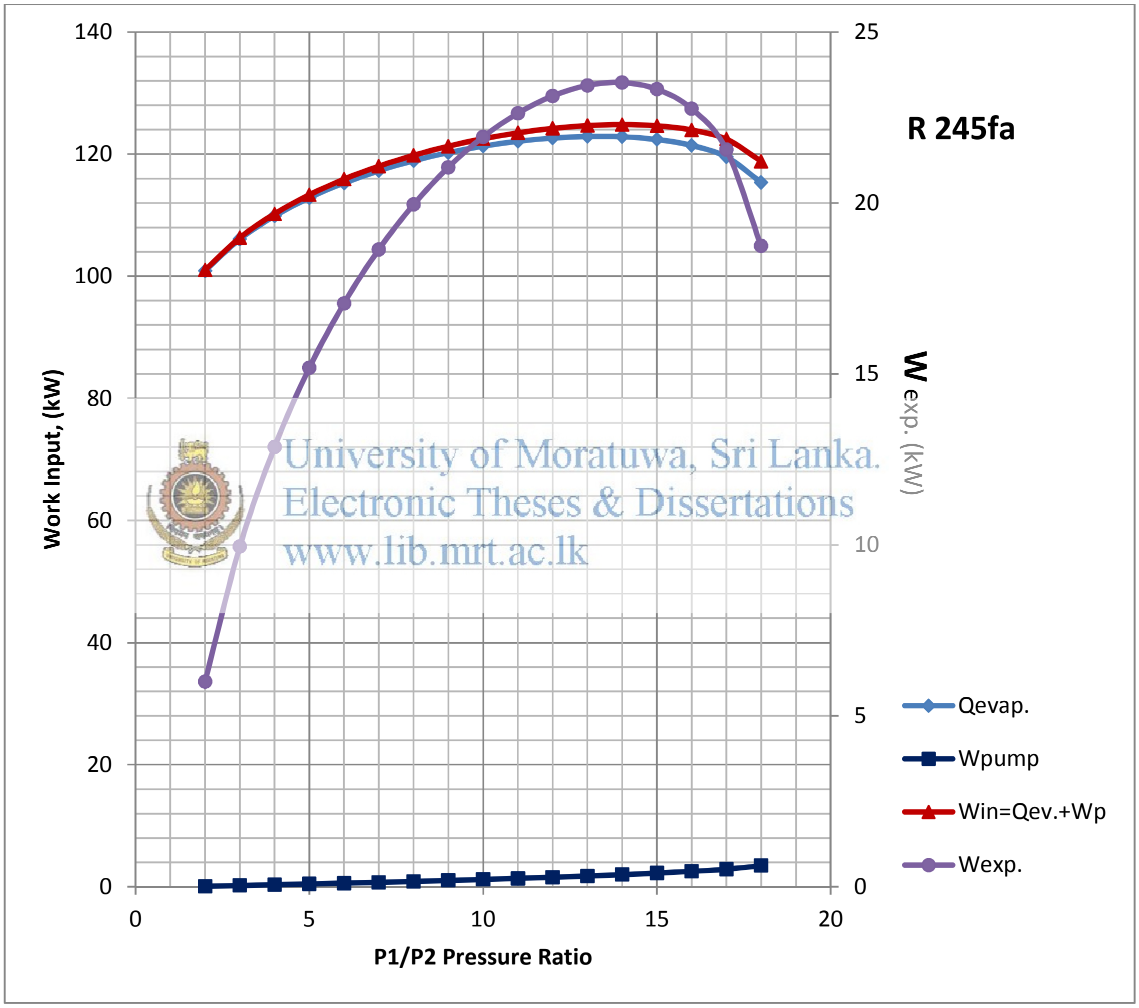


Graph B.33: Work Input and Work at Expander variation with different Evaporator temperatures for Heptane

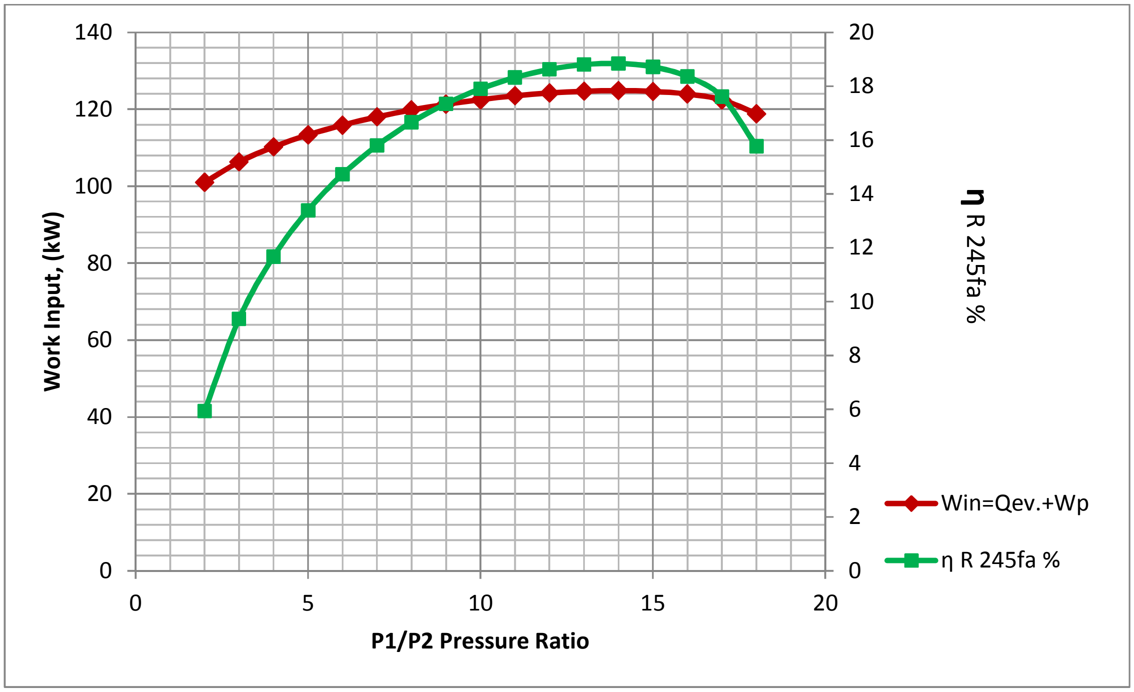
APPENDIX C

ORC PERFORMANCE ANALYSIS WITH EXPANDER PRESSURE RATIO VARIATION

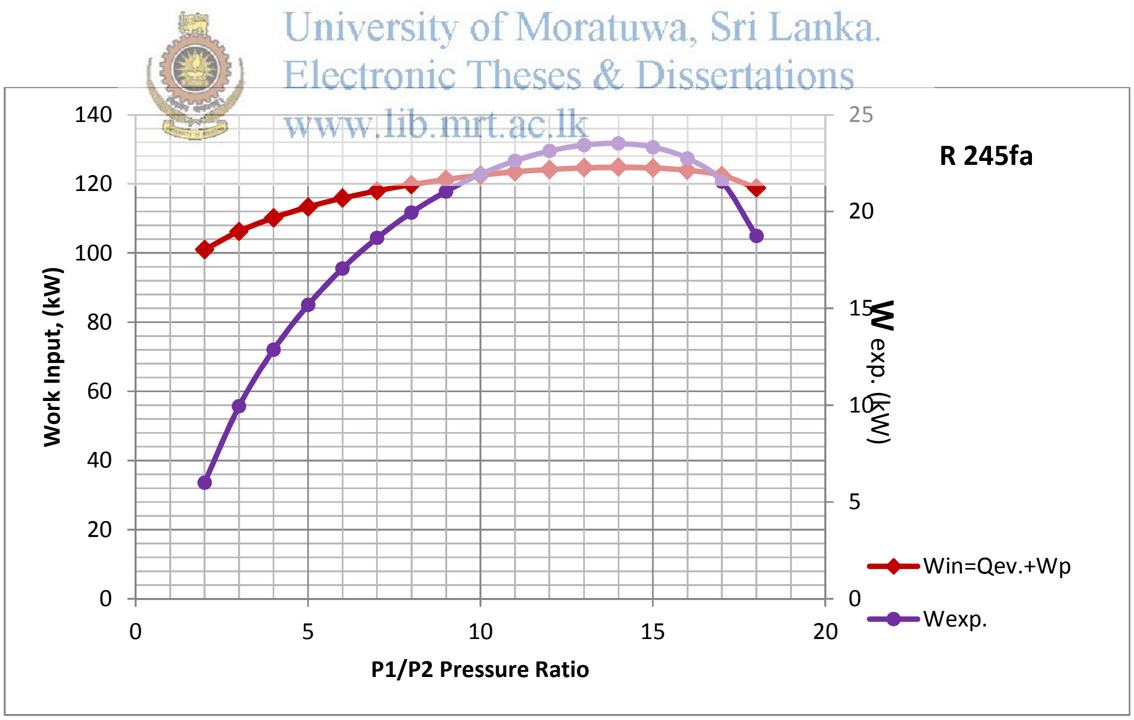
R245fa as working fluid



Graph C.34: Variation of Work component with different Expander pressure ratios for R245fa

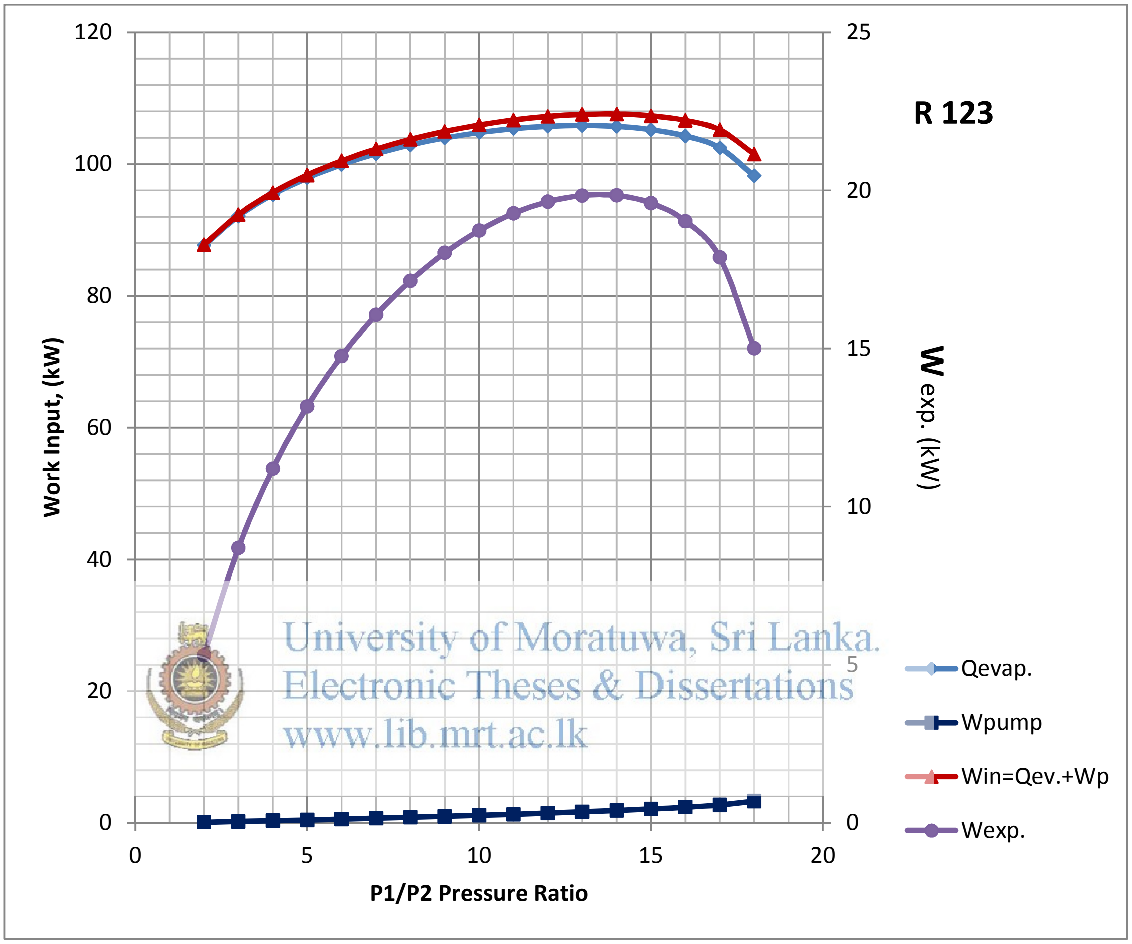


Graph C.35: Work Input and Efficiency variation with different Expander pressure ratios for R245fa

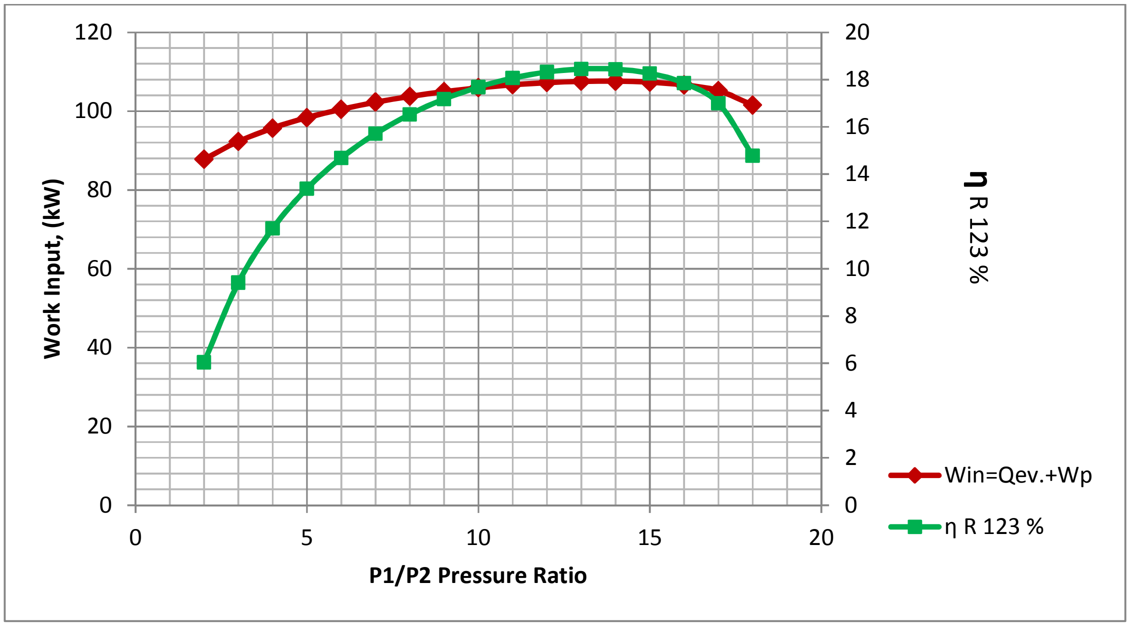


Graph C.36: Work Input and Work Output variation with different Expander pressure ratios for R245fa

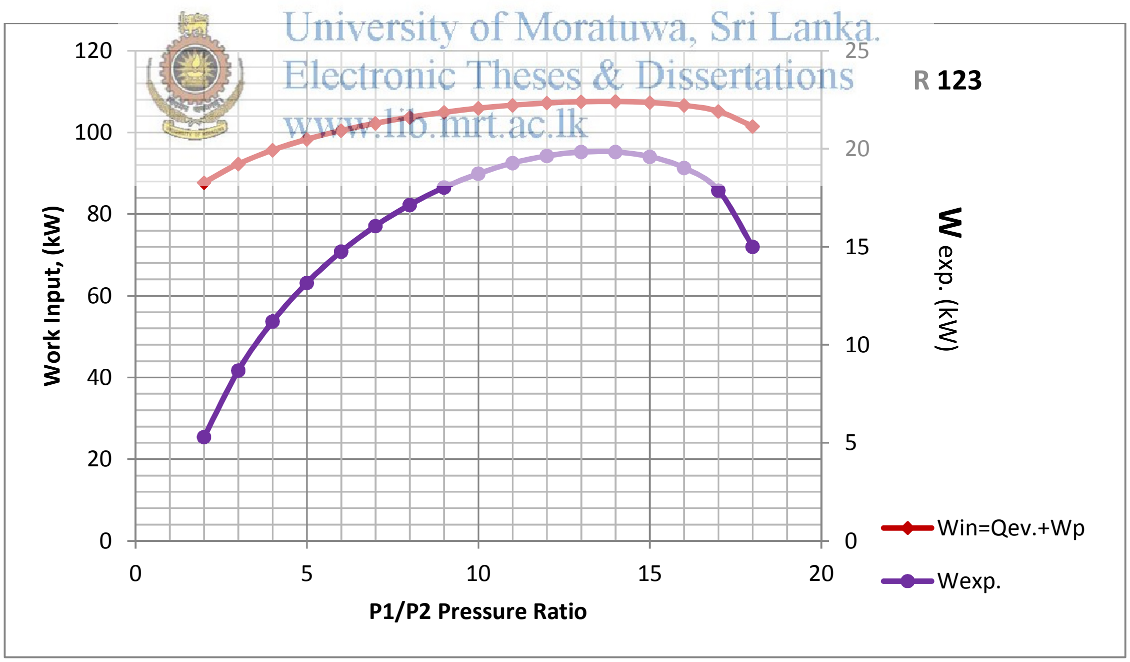
R123 as working fluid



Graph C.37: Variation of Work component with different Expander pressure ratios for R123

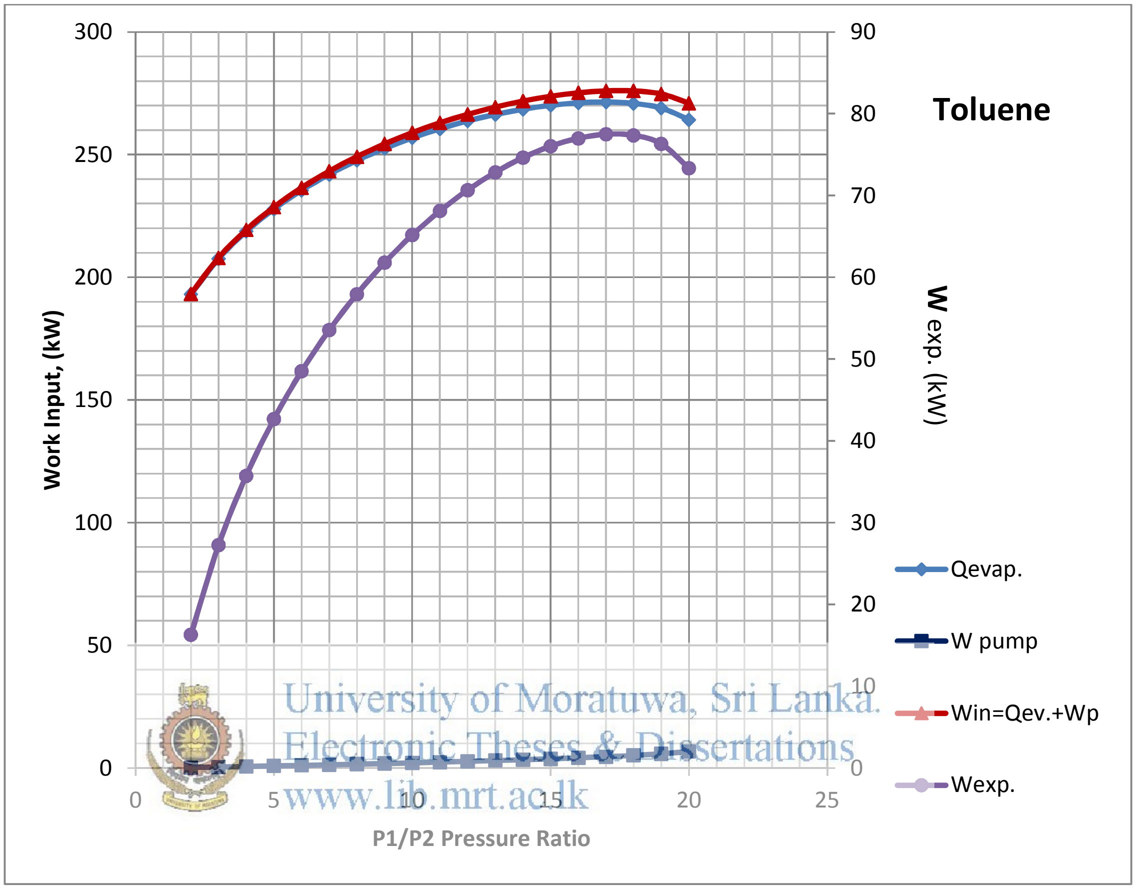


Graph C.38: Work Input and Efficiency variation with different Expander pressure ratios for R123

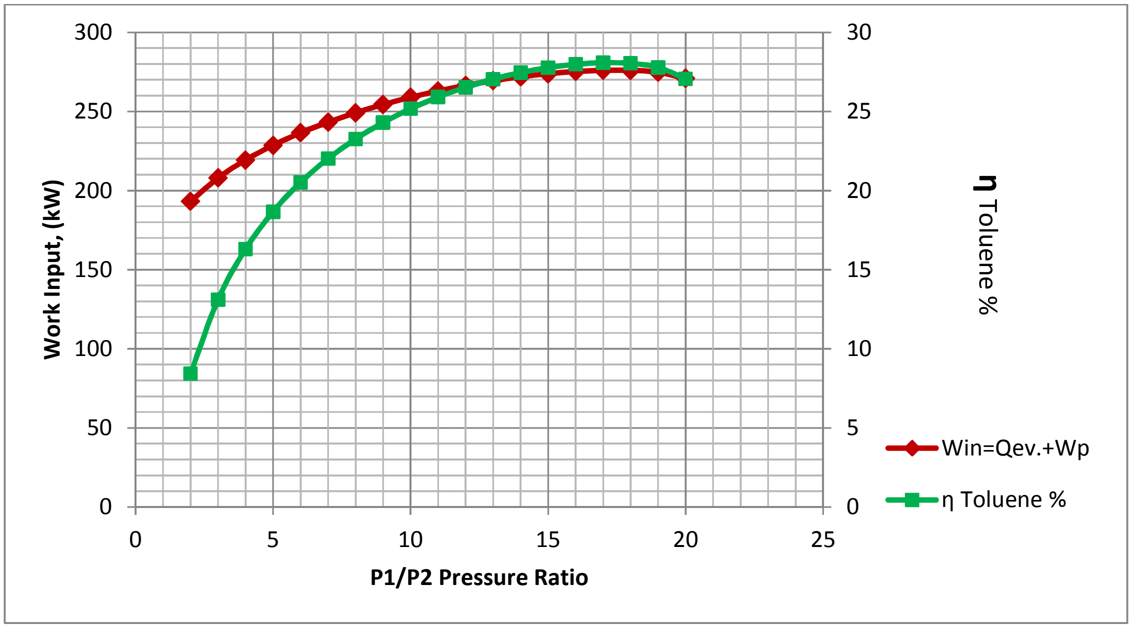


Graph C.39: Work Input and Work Output variation with different Expander pressure ratios for R123

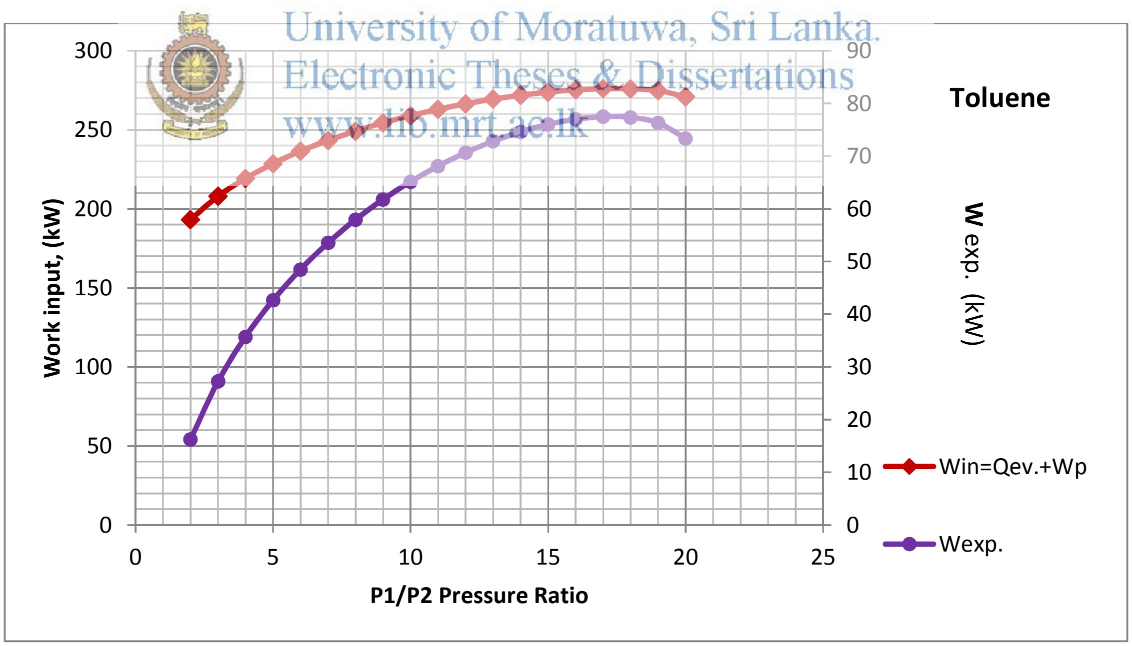
Toluene as working fluid



Graph C.40: Variation of Work component with different Expander pressure ratios for Toluene

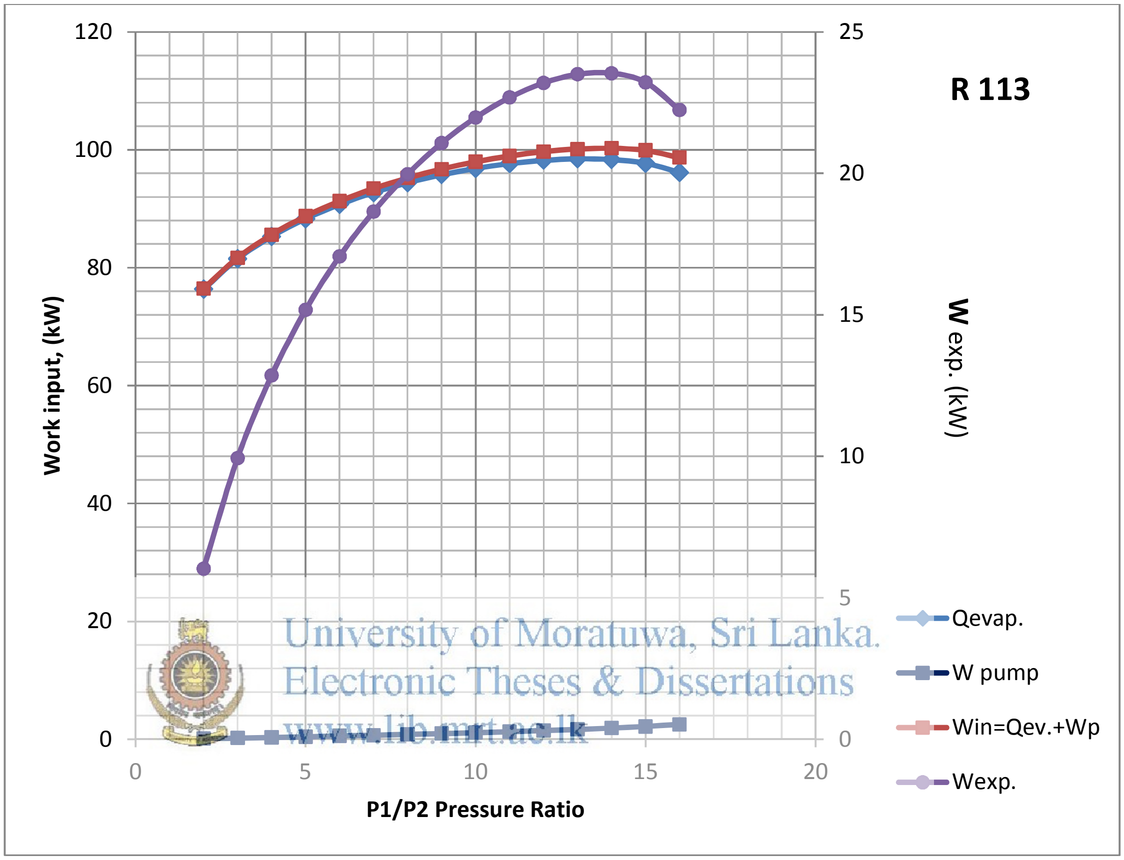


Graph C.41: Work Input and Efficiency variation with different Expander pressure ratios for Toluene

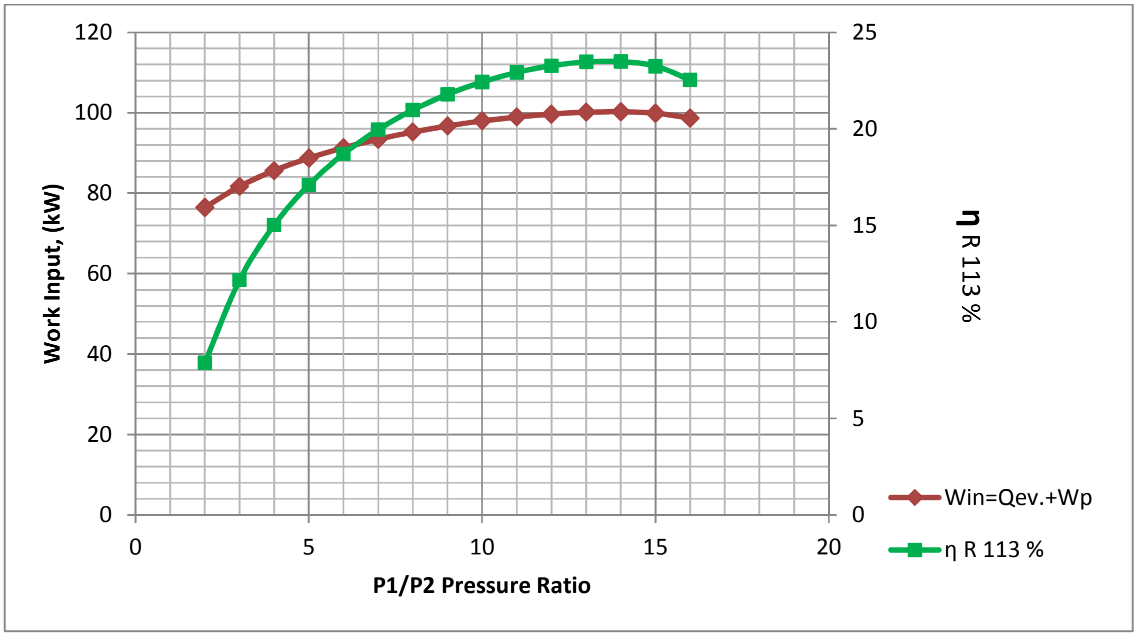


Graph C.42: Work Input and Work Output variation with different Expander pressure ratios for Toluene

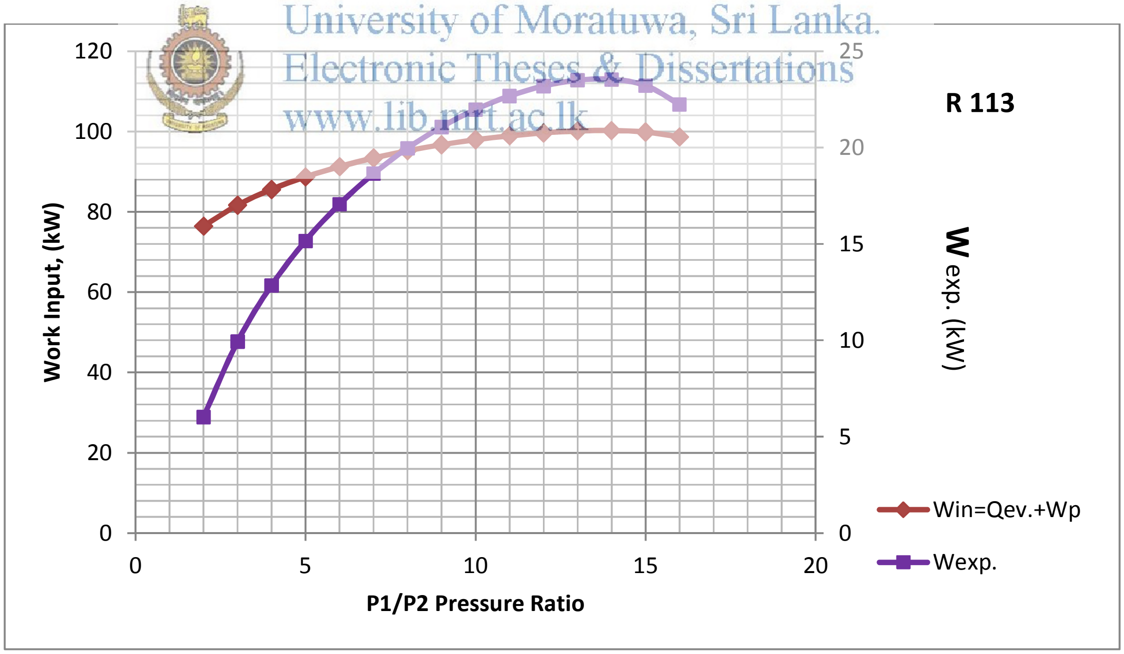
R113 as working fluid



Graph C.43: Variation of Work component with different Expander pressure ratios for R113

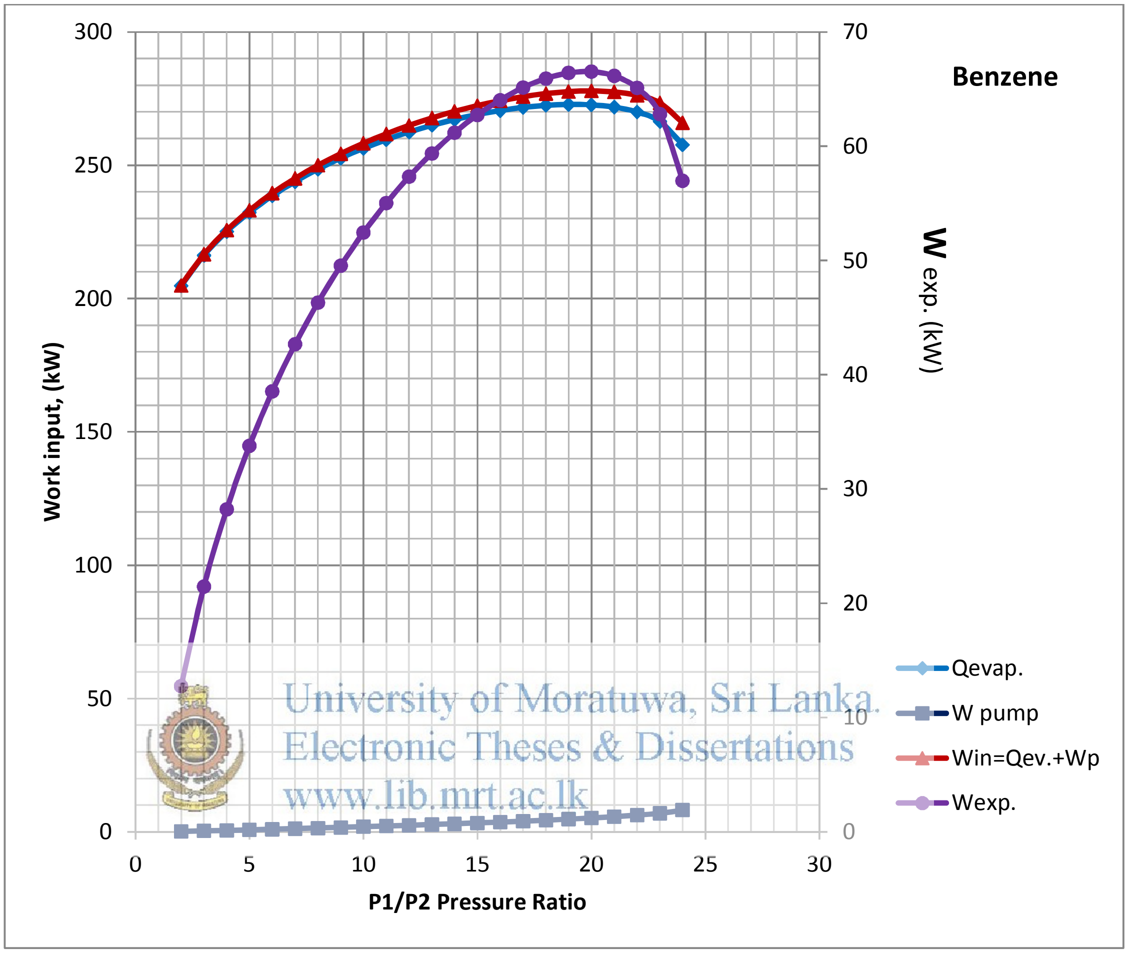


Graph C.44: Work Input and Efficiency variation with different Expander pressure ratios for R113

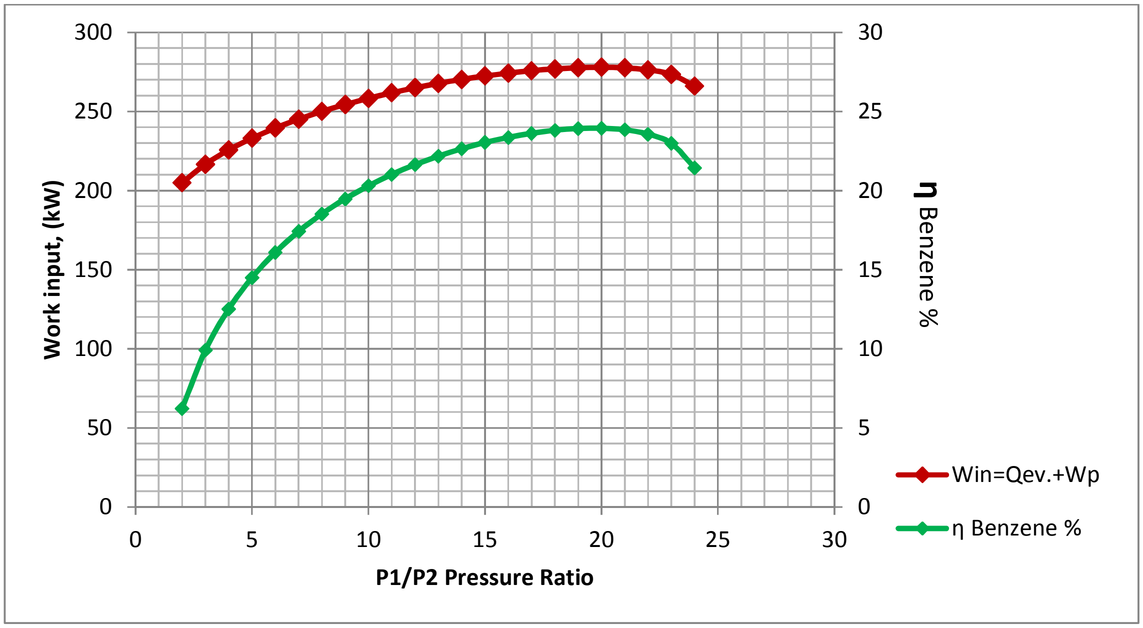


Graph C.45: Work Input and Work Output variation with different Expander pressure ratios for R113

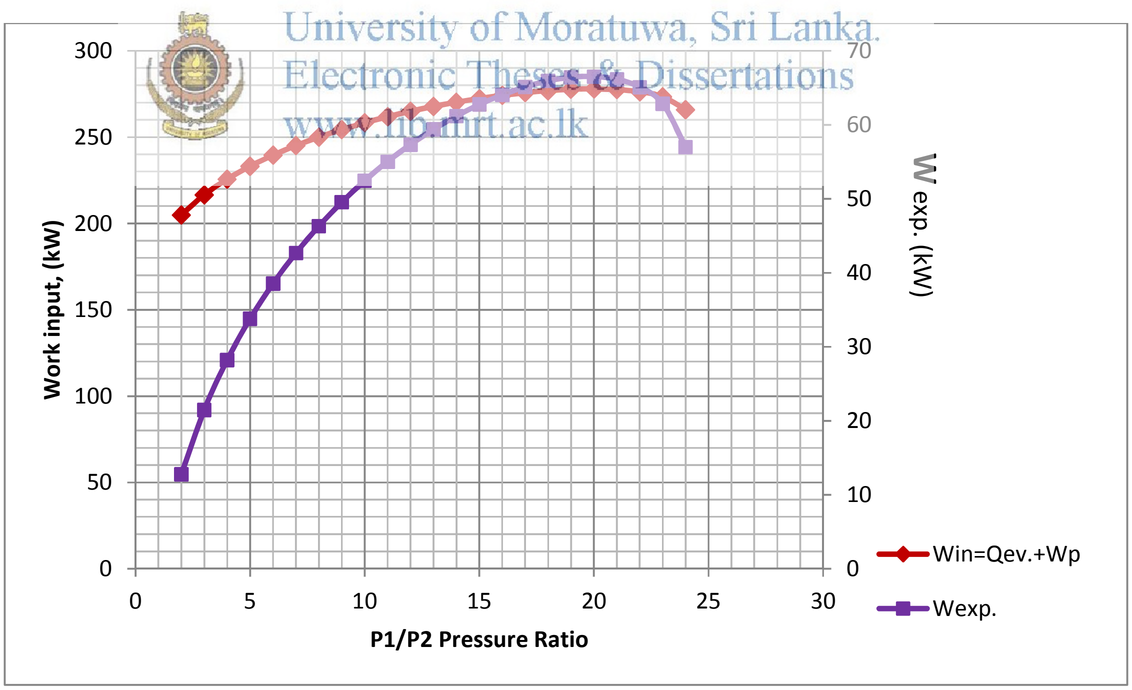
Benzene as working fluid



Graph C.46: Variation of Work component with different Expander pressure ratios for Benzene

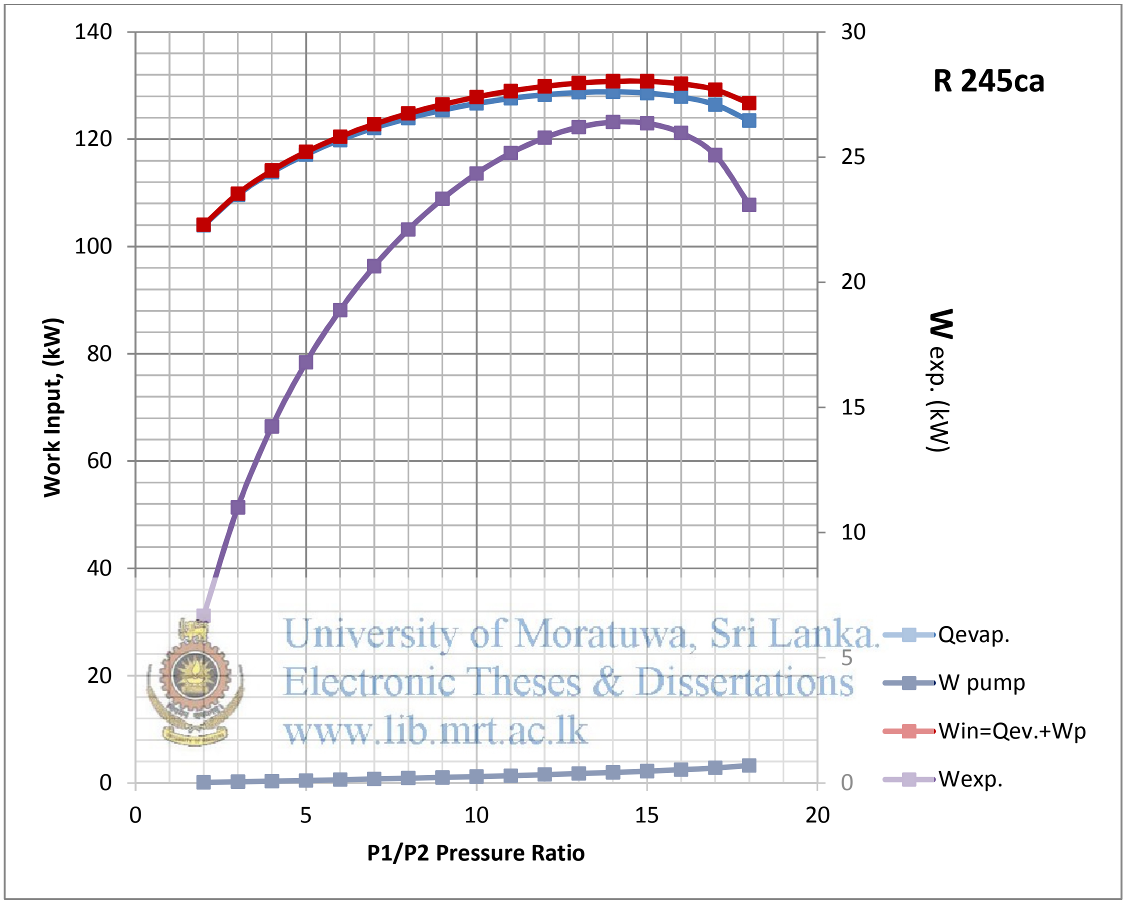


Graph C.47: Work Input and Efficiency variation with different Expander pressure ratios for Benzene

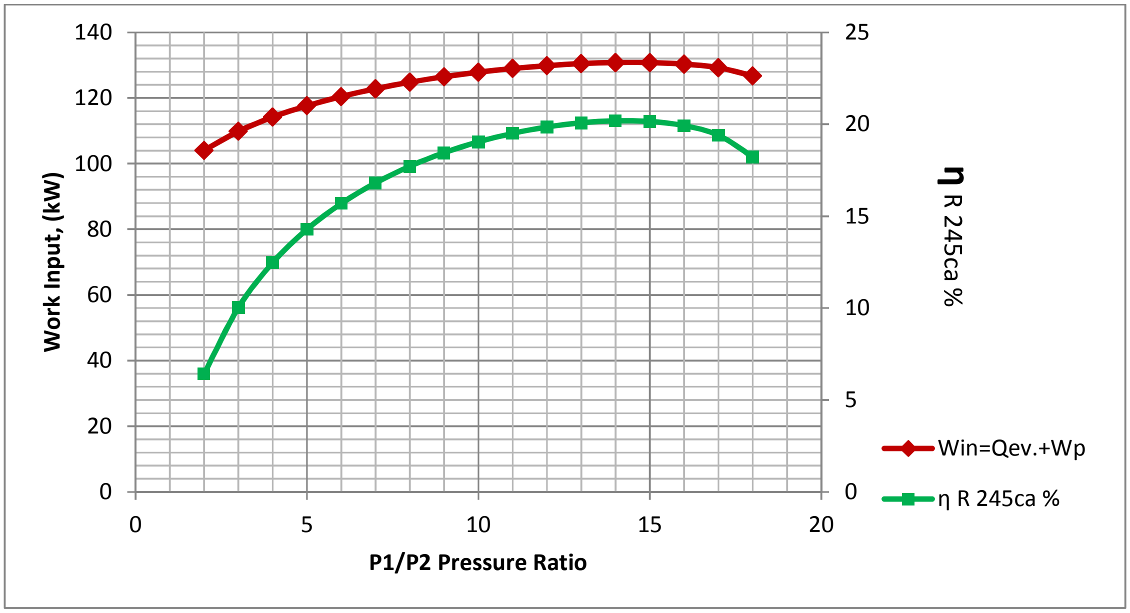


Graph C.48: Work Input and Work Output variation with different Expander pressure ratios for Benzene

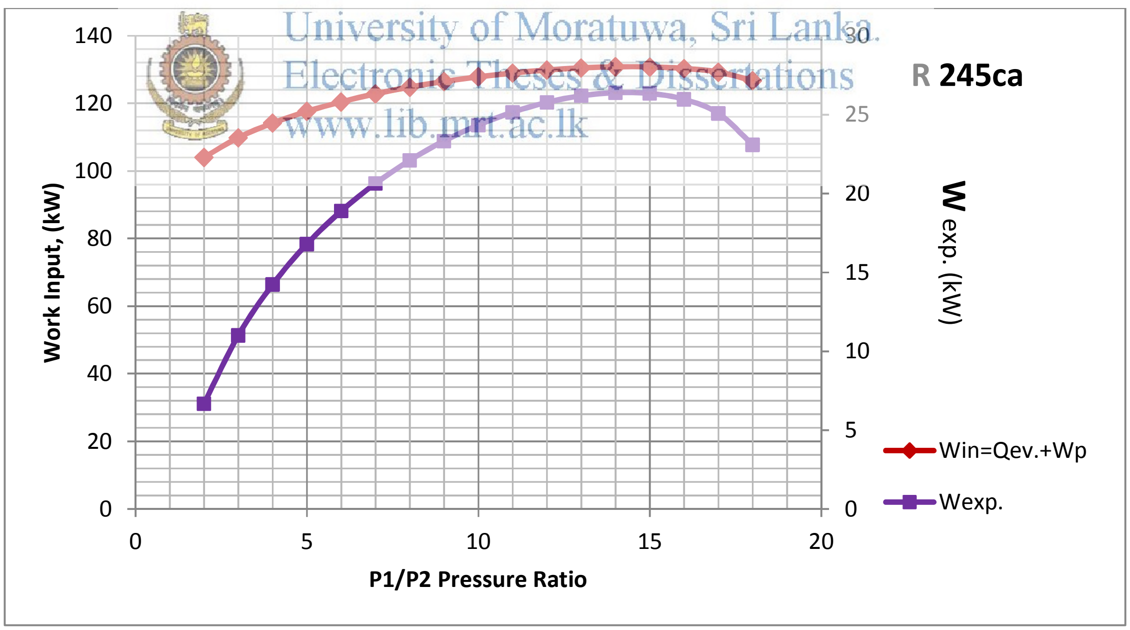
R245ca as working fluid



Graph C.49: Variation of Work component with different Expander pressure ratios for R245ca

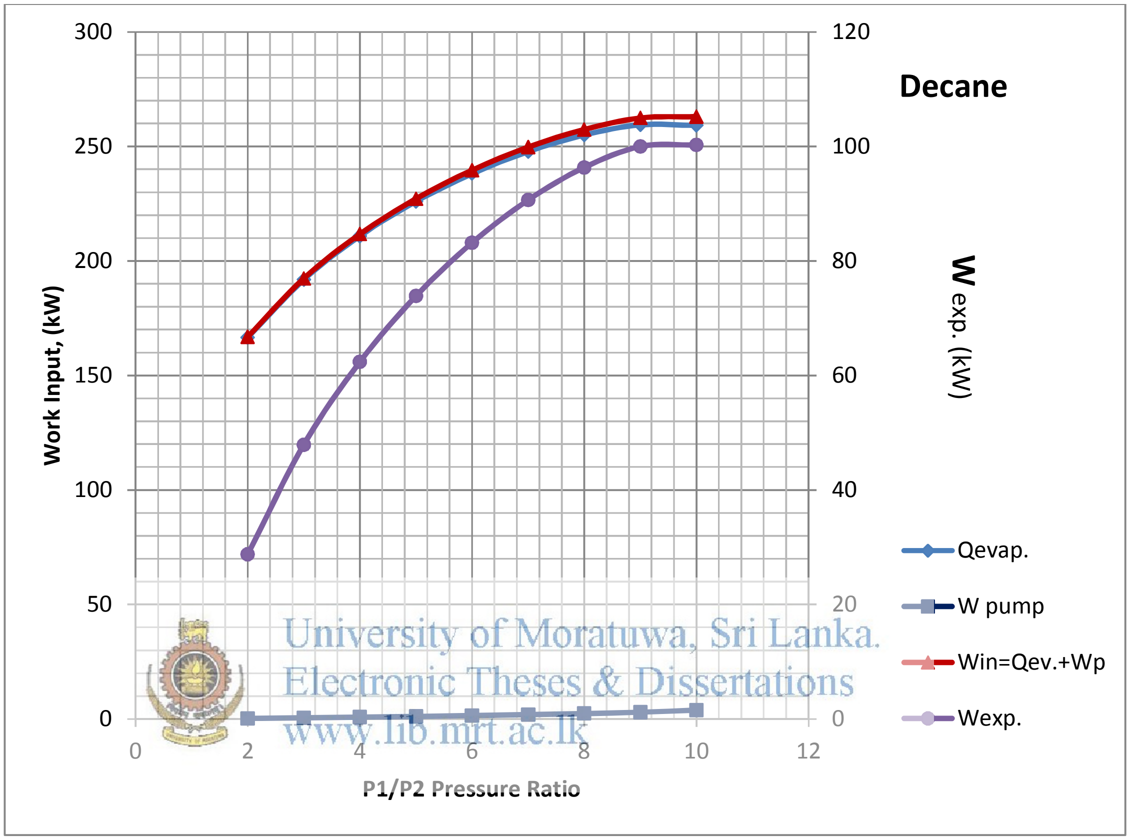


Graph C.50: Work Input and Efficiency variation with different Expander pressure ratios for R245ca

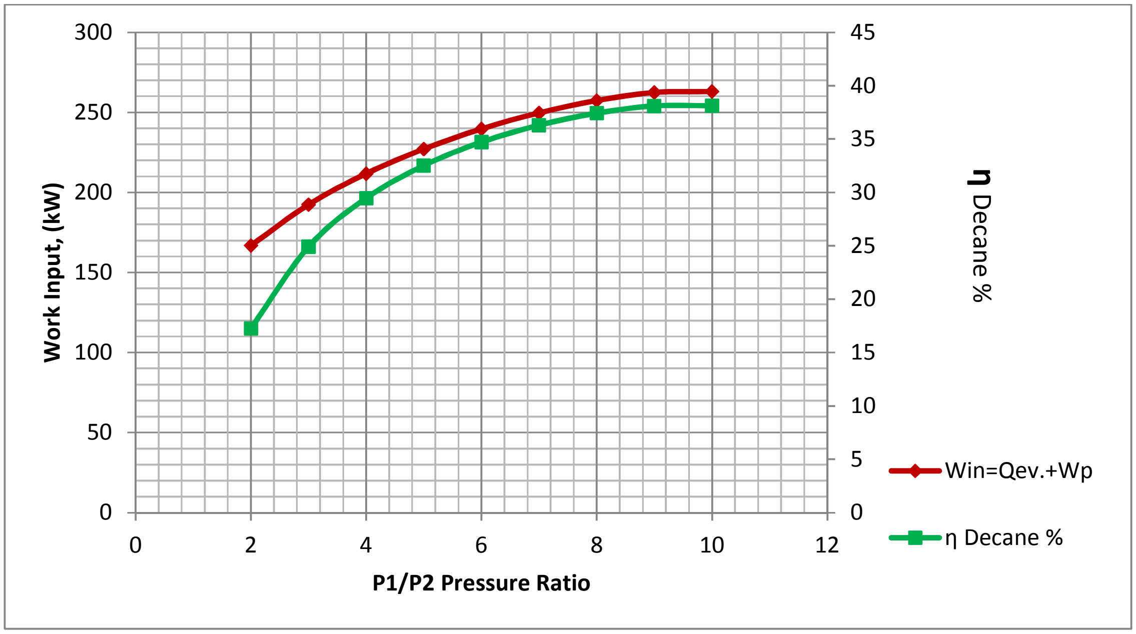


Graph C.51: Work Input and Work Output variation with different Expander pressure ratios for R245ca

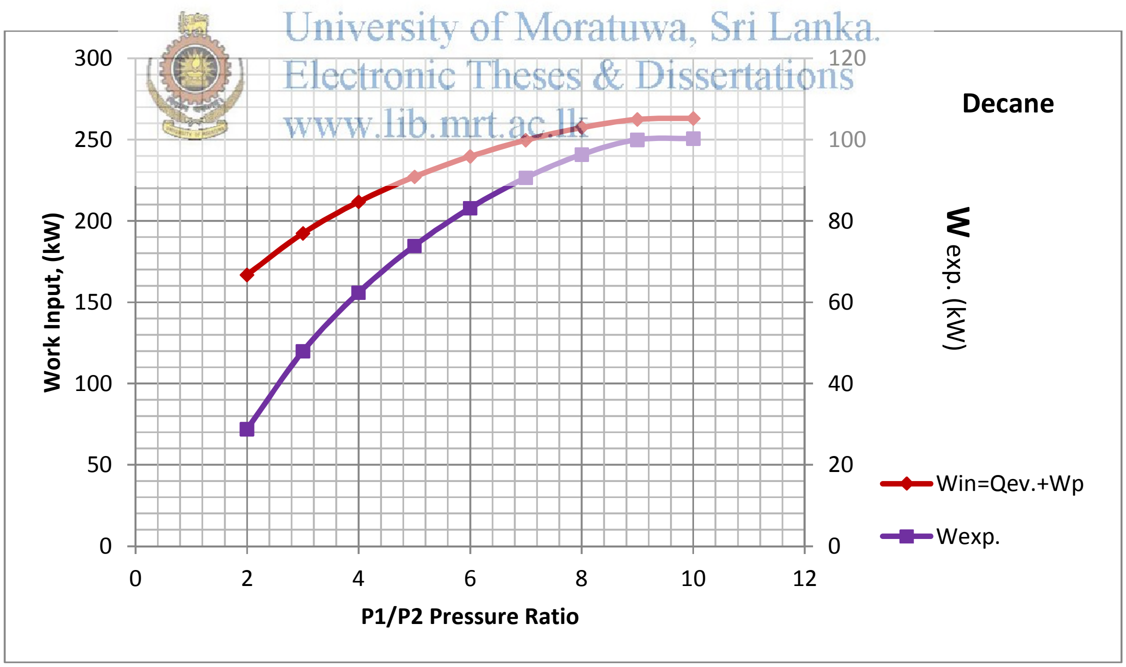
Decane as working fluid



Graph C.52: Variation of Work component with different Expander pressure ratios for Decane

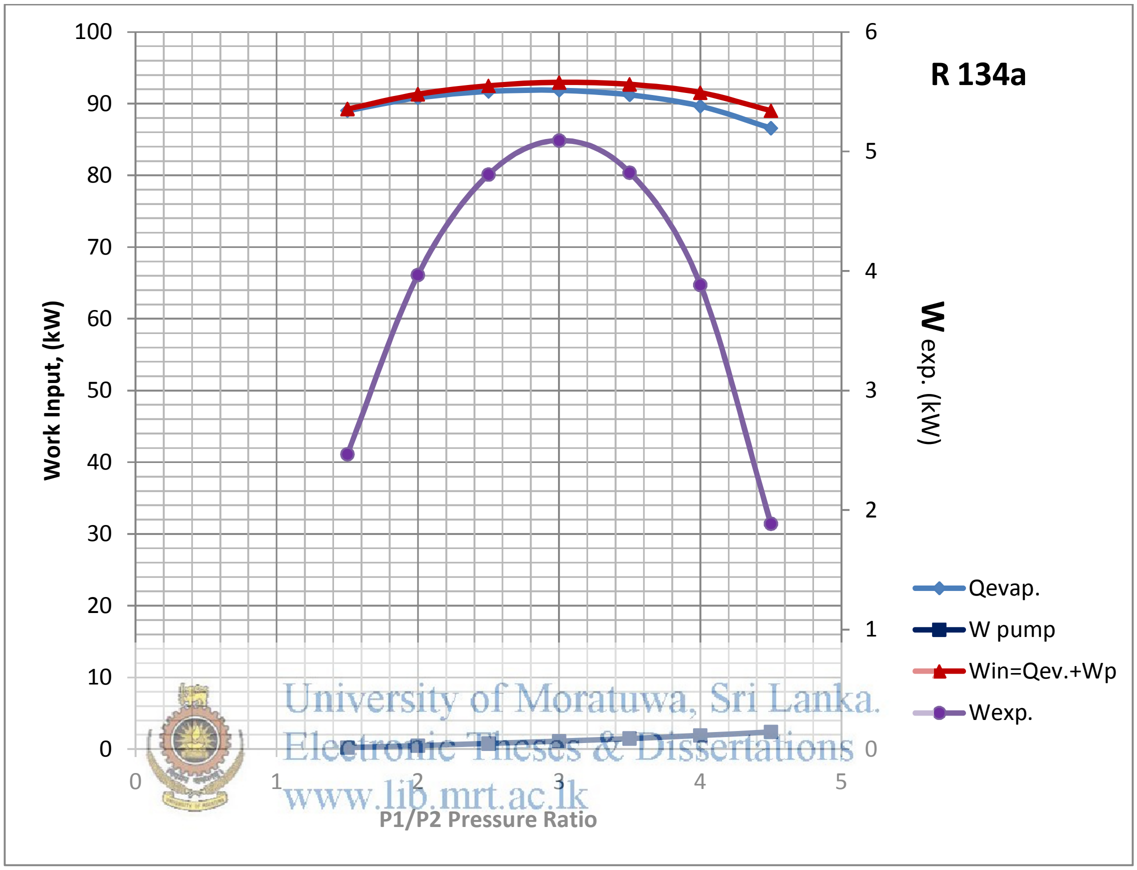


Graph C.53: Work Input and Efficiency variation with different Expander pressure ratios for Decane

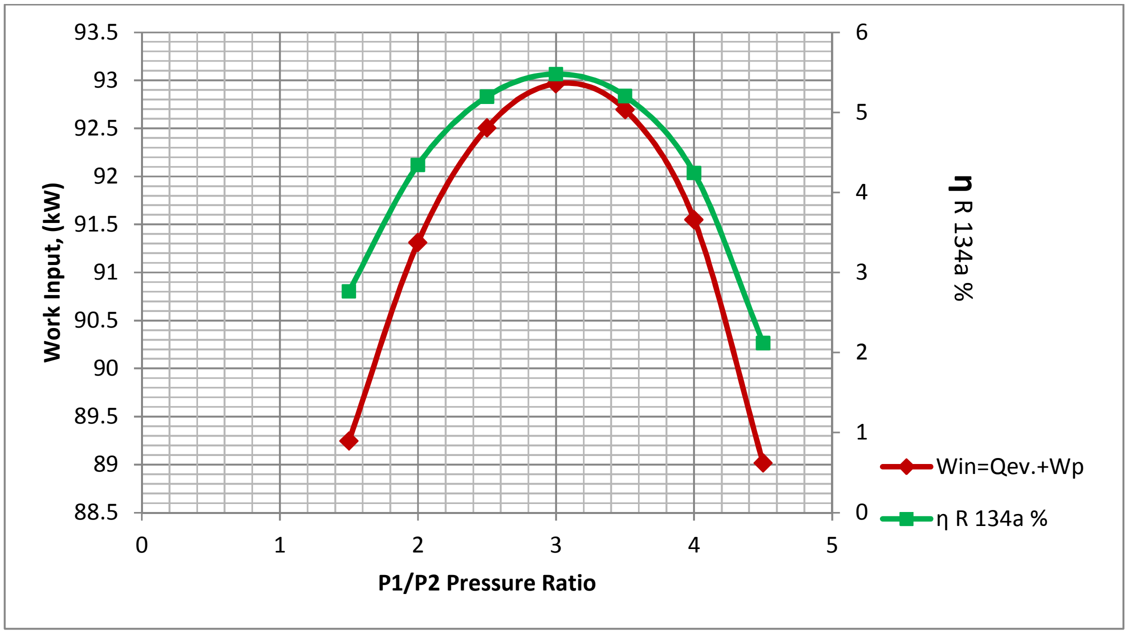


Graph C.54: Work Input and Work Output variation with different Expander pressure ratios for Decane

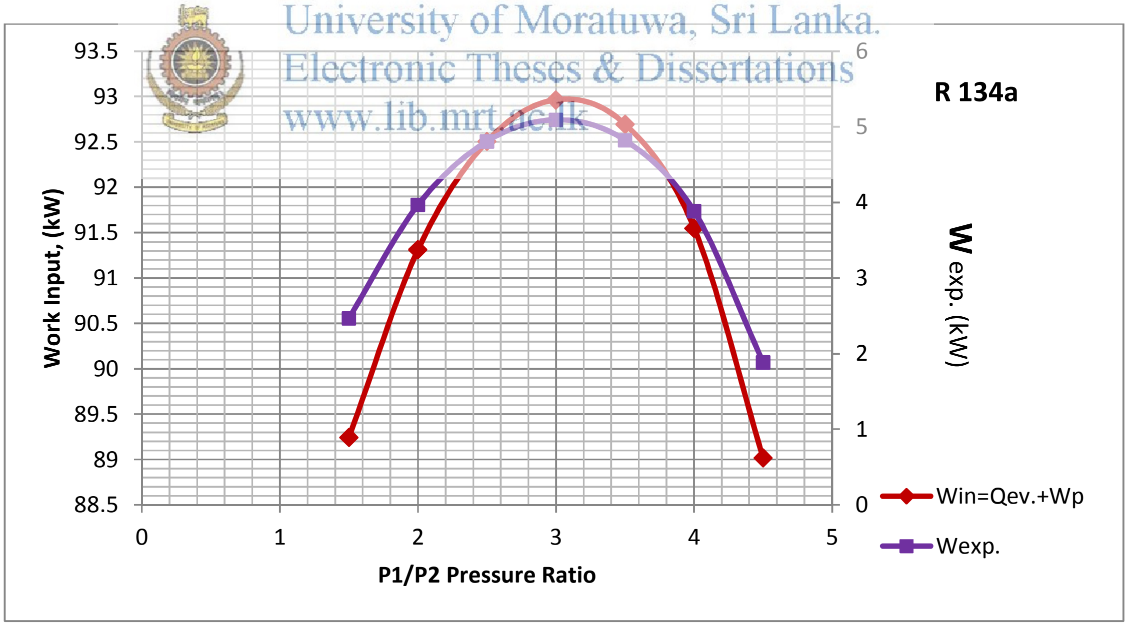
R134a as working fluid



Graph C.55: Variation of Work component with different Expander pressure ratios for R134a

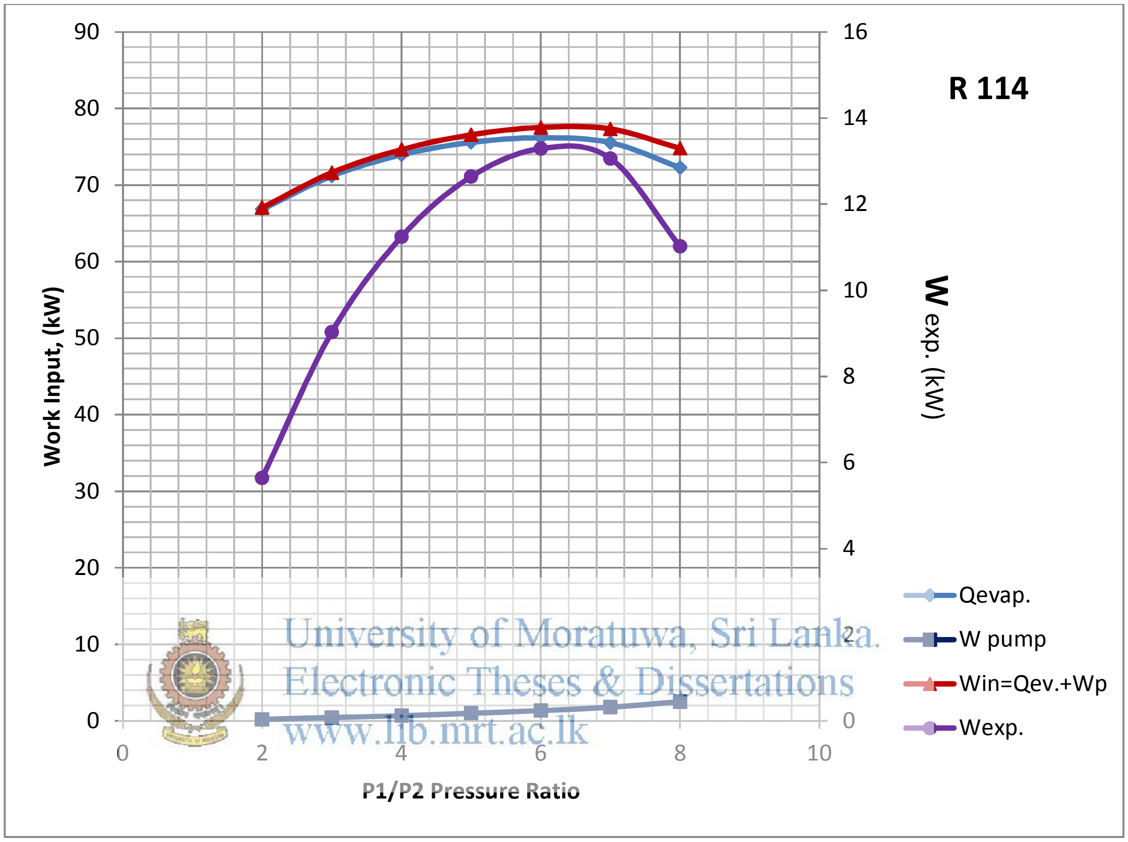


Graph C.56: Work Input and Efficiency variation with different Expander pressure ratios for R134a

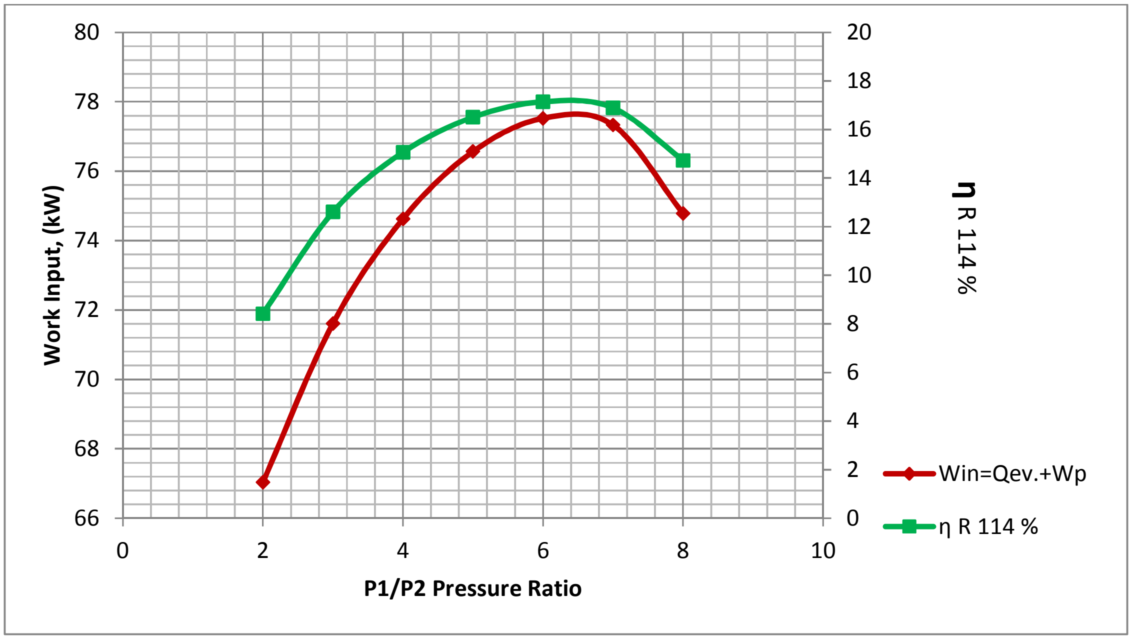


Graph C.57: Work Input and Work Output variation with different Expander pressure ratios for R134a

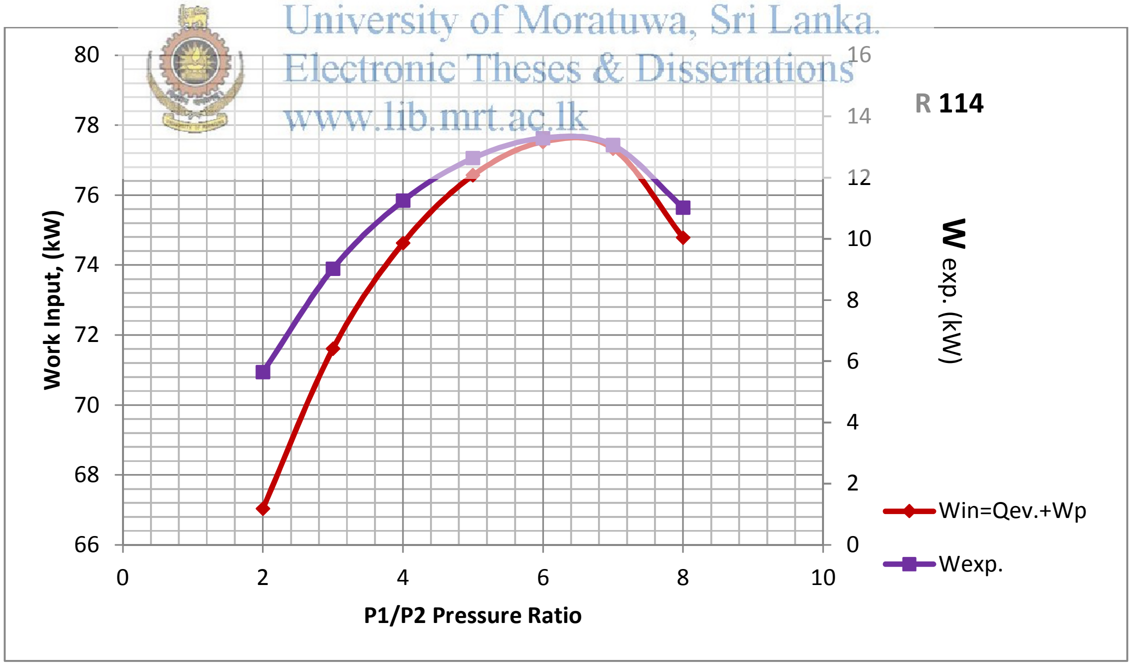
R114 as working fluid



Graph C.58: Variation of Work component with different Expander pressure ratios for R114

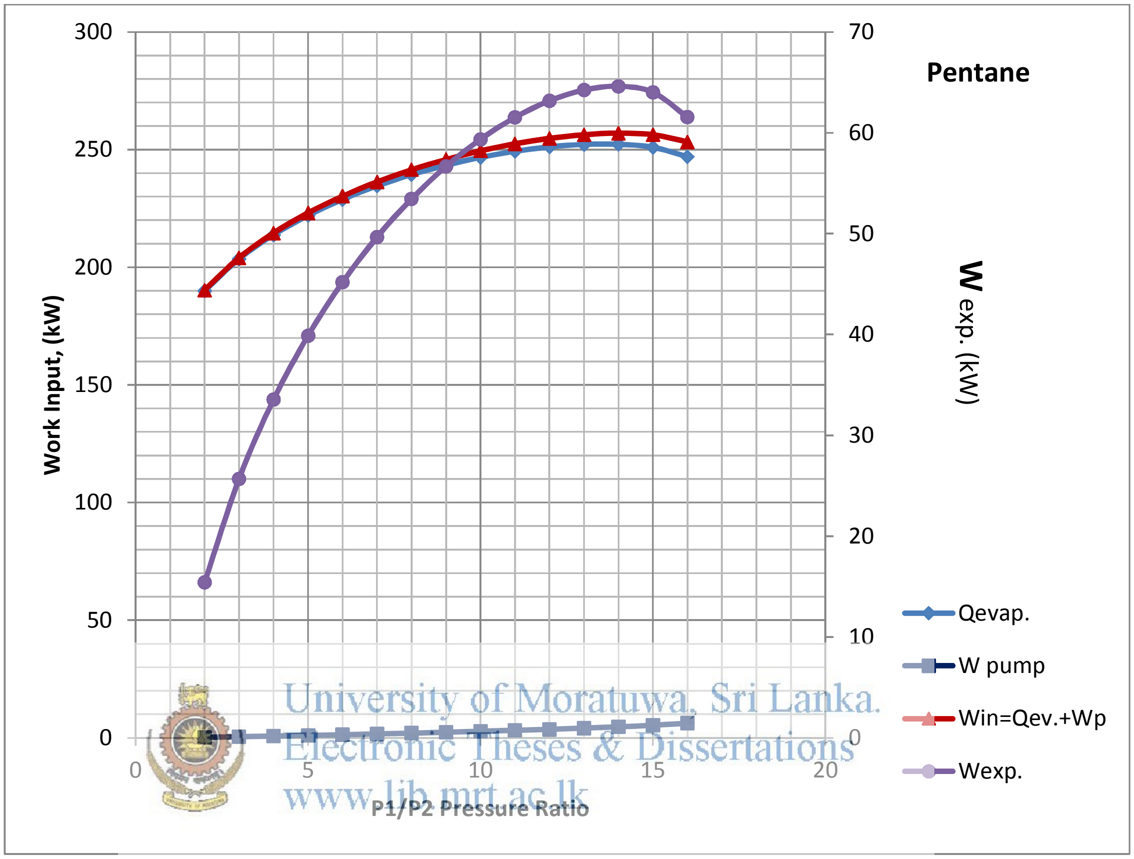


Graph C.59: Work Input and Efficiency variation with different Expander pressure ratios for R114

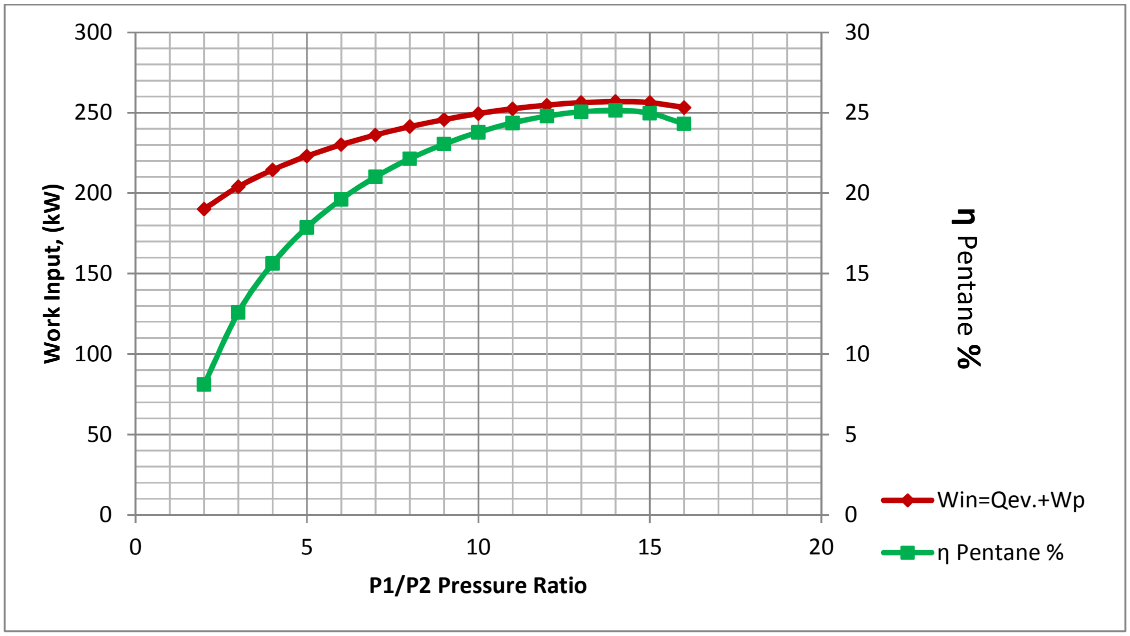


Graph C.60: Work Input and Work Output variation with different Expander pressure ratios for R114

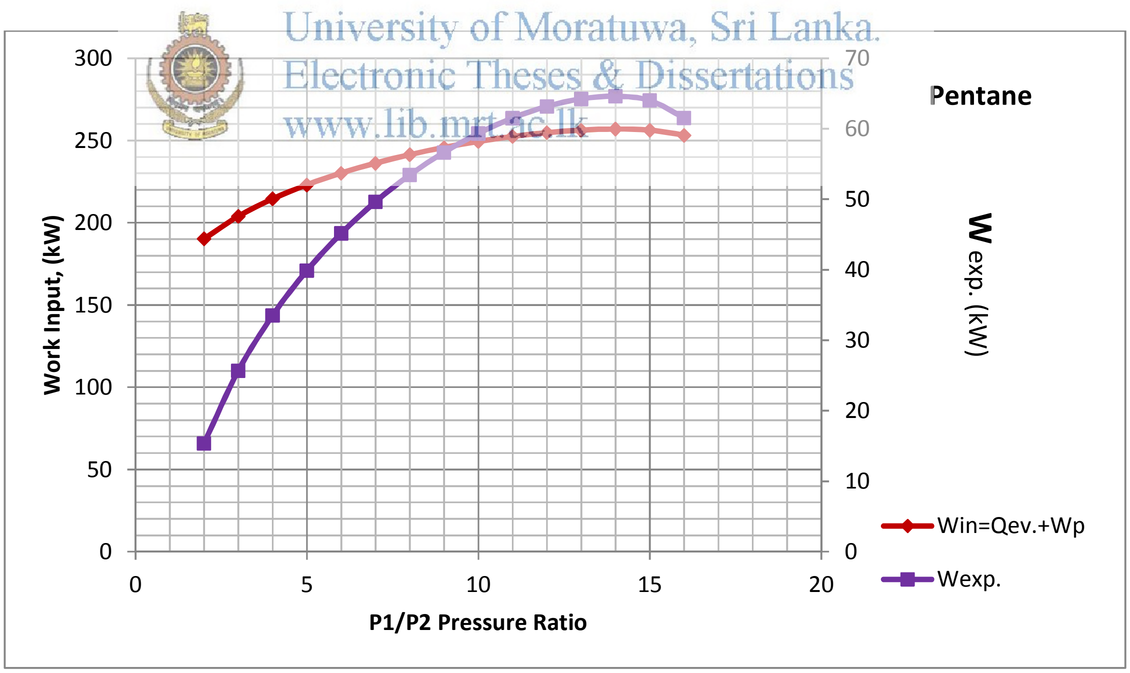
Pentane as working fluid



Graph C.61: Variation of Work component with different Expander pressure ratios for Pentane

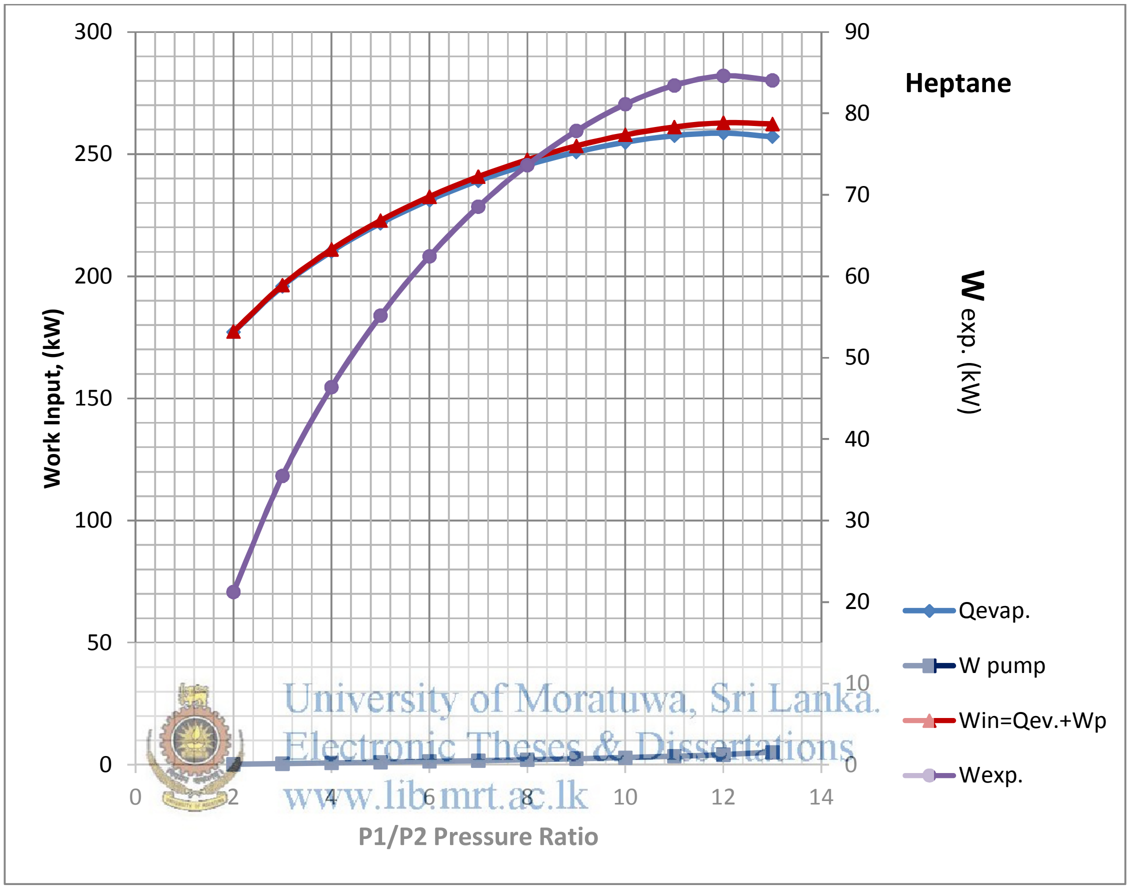


Graph C.62: Work Input and Efficiency variation with different Expander pressure ratios for Pentane

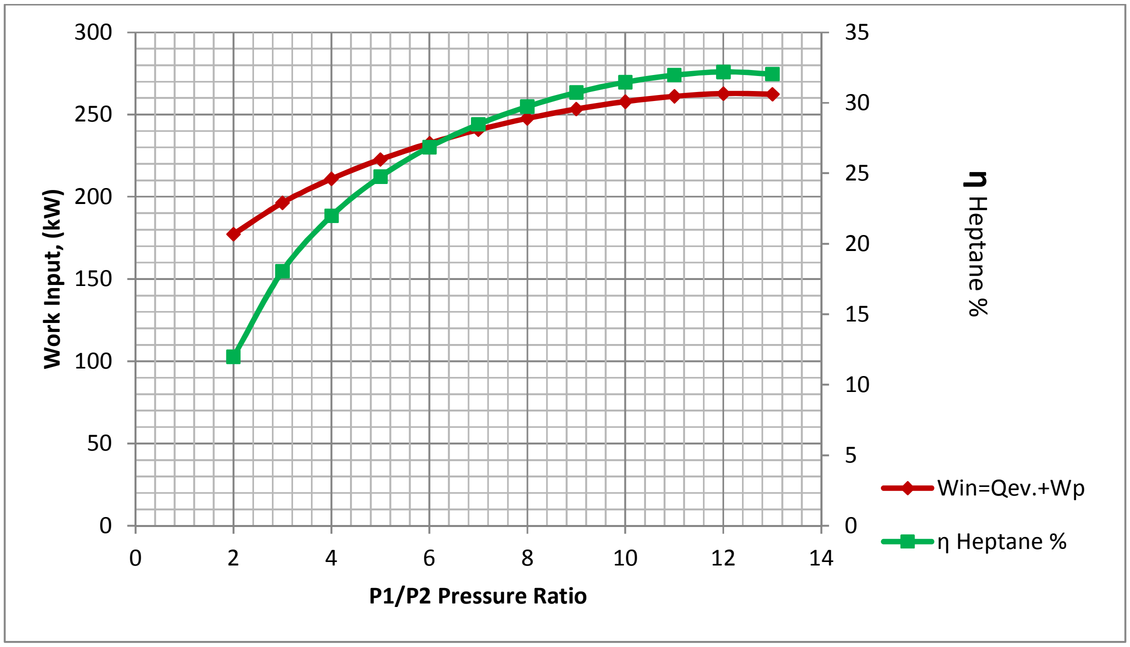


Graph C.63: Work Input and Work Output variation with different Expander pressure ratios for Pentane

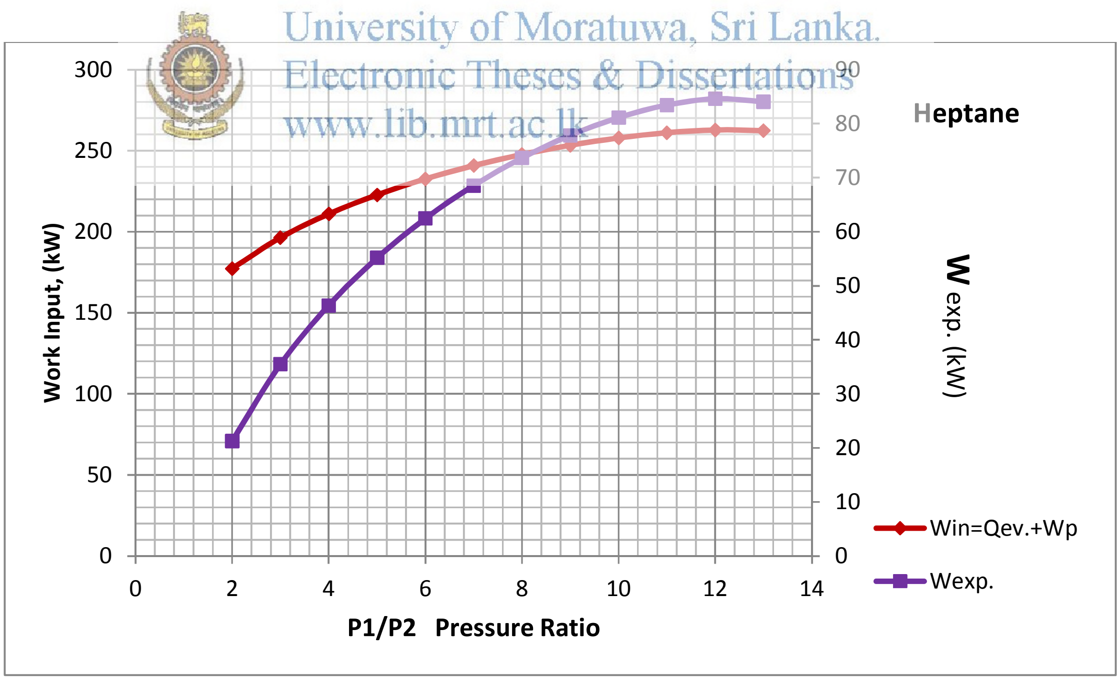
Heptane as working fluid



Graph C.64: Variation of Work component with different Expander pressure ratios for Heptane



Graph C.65: Work Input and Efficiency variation with different Expander pressure ratios for Heptane



Graph C.66: Work Input and Work Output variation with different Expander pressure ratios for Heptane

APPENDIX D

WORK OUTPUT CALCULATIONS FOR CASE STUDY

Following table shows the detailed calculations on work output of the expander.

Where;

Exh. Energy – Exhaust energy from heat source (kW)

Exh. T – Exhaust heat source average temperature (^oC)

Evap. Ext. T – Evaporator exiting/leaving temperature of organic fluid (^oC)

Evp. In – Evaporator in/entering temperature of organic fluid (^oC)

Eff. Evp. – Efficiency of evaporator

Q evap. In – Heat absorbed at the evaporator (kW)

hg evp. out – Saturated enthalpy at gaseous phase in the evaporator leaving fluid (KJ/kg)

hf evp. in - Saturated enthalpy at liquid phase in the evaporator entering fluid (KJ/kg)

Mass – Mass flow rate of the organic fluid (kg/s)

hg exp. out - Enthalpy at gaseous phase of the fluid leaving expander (KJ/kg)

η is – Isentropic efficiency of the expander

Wexp. – Work done at the expander (kW)

Table D.1: Sapugaskanda Power Station Exhaust Heat Recovery

Fluid	Exh. Enrgy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
R123	10799.13	430-180	165	45	0.2	2159.826	462.95	248.325	10.06325	408.52	0.75	410.8072
R113	10799.13	430-180	165	45	0.2	2159.826	455.975	240.31	10.01473	388.1	0.75	509.8122
Pentane	10799.13	430-180	165	45	0.2	2159.826	719.47	189.585	4.076028	589.78	0.75	396.465

Table D.2: Lakvijaya Coal Plant Exhaust Heat Recovery

Fluid	Exh. Energy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
R134a	17916.67	150-90	80	45	0.2	3583.334	280.67	115.8	21.7343	273.38	0.75	118.8323
R114	17916.67	150-90	80	45	0.2	3583.334	383.67	244.62	25.77011	364.365	0.75	373.119
R245fa	17916.67	150-90	80	45	0.2	3583.334	464.31	261.28	17.64928	439.3	0.75	331.0564
R245ca	17916.67	150-90	80	45	0.2	3583.334	326.37	109.365	16.51268	299.11	0.75	337.6017

Table D.3: Lakvijaya Coal Plant Continuous Blow down Heat Recovery

Fluid	Exh. Energy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
R114	1723.446	275-100	100	45	0.200	344.689	392.960	244.620	2.324	364.365	0.750	49.833
R245fa	1723.446	275-100	100	45	0.200	344.689	477.670	261.280	1.593	439.300	0.750	45.840
R245ca	1723.446	275-100	100	45	0.200	344.689	341.040	109.365	1.488	299.110	0.750	46.788
R123	1723.446	275-100	100	45	0.200	344.689	439.770	248.770	1.805	408.520	0.750	42.297

Table D.4: Jaffna Diesel Plant Exhaust Heat Recovery

Fluid	Exh. Energy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
Heptane	384.44	420-240	220	45	0.2	76.888	857.87	184.3	0.11415	536.915	0.75	27.47775
Benzene	384.44	420-240	220	90	0.2	76.888	562.19	18.947	0.141535	406.25	0.75	16.55325
Toluene	384.44	420-240	220	45	0.2	76.888	777.06	139.33	0.120565	540.375	0.75	21.40197

Table D.5: Keravalapitiya Plant Exhaust Heat Recovery

Fluid	Exh. Energy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
R245ca	163838.43	500-160	145	45	0.2	32767.686	366.555	109.365	127.4065	299.11	0.75	6444.7
R123	163838.43	500-160	145	45	0.2	32767.686	459.495	248.325	155.1721	408.52	0.75	5932.422
R113	163838.43	500-160	145	45	0.2	32767.686	446.56	240.31	158.8736	388.1	0.75	6965.814
Pentane	163838.43	500-160	145	45	0.2	32767.686	695.16	189.585	64.81271	539.78	0.75	7552.949

Table D.6: Kelanithissa GT Plant Exhaust Heat Recovery

Fluid	Exh. Enrgy	Exh. T	Evap. Ext T	Evp. In	Eff. Evp.	Qevp. In	hg evp. Out	hf evp. In	Mass kg/s	hg exp. Out	η is	W exp.
R123	32790.3	470-180	165	45	0.2	6558.06	462.95	248.325	30.5559	408.52	0.75	1247.368
R113	32790.3	470-180	165	45	0.2	6558.06	455.975	240.31	30.40855	388.1	0.75	1547.985
Pentane	32790.3	470-180	165	45	0.2	6558.06	719.47	189.585	12.37638	589.78	0.75	1203.82

APPENDIX E

NET POSITIVE VALUE CALCULATIONS

Expected Turnover Calculation

Based on different tariff rates for unit price and expected annual running hours, expected annual turnover would change. Following tables are related to annual turnover calculation in different situation.

Table E.18: Expected annual turnover at 60% running hours & Rs. 14.00/kWh

Waste Heat Recovery Opportunity	**Exp. Elec. Output (kW)**	**Exp. Running Hours per Year**	**Exp. Generation kW/yr**	**Unit Selling Price Rs.**	**Exp. Annual Turnover Rs.**
Sapugaskanda Power Station	458.831	5256	2411615.504	14.00	33,762,617.05
Lakvijaya Coal Plant	335.807	5256	1765002.124	14.00	24,710,029.74
Jaffna Diesel Plant	24.730	5256	129980.769	14.00	1,819,730.77
Keravalapitiya Plant	6797.654	5256	35728470.860	14.00	500,198,592.04
Kelanithissa GT Plant	1393.187	5256	7322589.491	14.00	102,516,252.87
Lakvijaya Blowdown	44.850	5256	235732.026	14.00	3,300,248.37

Table E.19: Expected annual turnover at 60% running hours & Rs. 15.00/kWh

Waste Heat Recovery Opportunity	Exp. Elec. Output (kW)	Exp. Running Hours per Year	Exp. Generation kW/yr	Unit Selling Price Rs.	Exp. Annual Turnover Rs.
Sapugaskanda Power Station	458.831	5256	2411615.504	15.00	36,174,232.55
Lakvijaya Coal Plant	335.807	5256	1765002.124	15.00	26,475,031.87
Jaffna Diesel Plant	24.730	5256	129980.769	15.00	1,949,711.54
Keravalapitiya Plant	6797.654	5256	35728470.860	15.00	535,927,062.90
Kelanithissa GT Plant	1393.187	5256	7322589.491	15.00	109,838,842.36
Lakvijaya Blowdown	44.850	5256	235732.026	15.00	3,535,980.40

Table E.20: Expected annual turnover at 60% running hours & Rs. 15.40/kWh

Waste Heat Recovery Opportunity	Exp. Elec. Output (kW)	Exp. Running Hours per Year	Exp. Generation kW/yr	Unit Selling Price Rs.	Exp. Annual Turnover Rs.
Sapugaskanda Power Station	458.831	5256	2411615.504	15.40	37,138,878.76
Lakvijaya Coal Plant	335.807	5256	1765002.124	15.40	27,181,032.72
Jaffna Diesel Plant	24.730	5256	129980.769	15.40	2,001,703.85
Keravalapitiya Plant	6797.654	5256	35728470.86	15.40	550,218,451.25
Kelanithissa GT Plant	1393.187	5256	7322589.491	15.40	112,767,878.16
Lakvijaya Blowdown	44.850	5256	235732.026	15.40	3,630,273.21

Table E.21: Expected annual turnover at different running hours & Rs. 15.00/kWh

Waste Heat Recovery Opportunity	Exp. Elec. Output (kW)	Exp. Running Hours per Year	Exp. Generation kW/yr	Unit Selling Price Rs.	Exp. Turnover Rs.
Sapugaskanda Power Station	458.831	5256	2411615.504	15.00	36,174,232.55
Lakvijaya Coal Plant	335.807	6132	2059169.145	15.00	30,887,537.18
Jaffna Diesel Plant	24.730	5256	129980.769	15.00	1,949,711.54
Keravalapitiya Plant	6797.654	5256	35728470.860	15.00	535,927,062.90
Kelanithissa GT Plant	1393.187	3504	4881726.327	15.00	73,225,894.91
Lakvijaya Blowdown	44.850	6132	275020.698	15.00	4,125,310.46

Net Positive Value (NPV) Calculations

NPV calculations were done under 07 scenarios to investigate the feasibility of implementing WHR systems in identified heat sources. The identifies heat sources were mentioned in the tables as follows;

A - Sapugaskanda Power Station
B - Lakvijaya Coal Plant
C - Jaffna Diesel Plant
D - Keravalapitiya Plant
E - Kelanithissa GT Plant
F - Lakvijaya Boiler Blow Down

Table E.22: Scenario 1 – Electricity unit selling price Rs. 14.00, Interest Rate 8%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	33,762,617	32,447,017	8	129,551,531	(2,008,469)
B	97,240,000	972,400	24,710,030	23,737,630	8	94,777,473	(2,462,527)
C	8,125,000	81,250	1,819,731	1,738,481	8	6,941,250	(1,183,750)
D	1,591,200,000	15,912,000	500,198,592	484,286,592	8	1,933,615,937	342,415,937
E	327,600,000	3,276,000	102,516,253	99,240,253	8	396,237,554	68,637,554
F	14,625,000	146,250	3,300,248	3,153,998	8	12,593,001	(2,031,999)

Table E.23: Scenario 2 – Electricity unit selling price Rs. 15.00, Interest Rate 8%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	36,174,233	34,858,633	8	139,180,412	7,620,412
B	97,240,000	972,400	26,475,032	25,502,632	8	101,824,614	4,584,614
C	8,125,000	81,250	1,949,712	1,868,462	8	7,460,225	(664,775)
D	1,591,200,000	15,912,000	535,927,063	520,015,063	8	2,076,269,361	485,069,361
E	327,600,000	3,276,000	109,838,842	106,562,842	8	425,474,530	97,874,530
F	14,625,000	146,250	3,535,980	3,389,730	8	13,534,211	(1,090,789)

Table E.24: Scenario 3 – Electricity unit selling price Rs. 15.40, Interest Rate 8%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	37,138,879	35,823,279	8	143,031,965	11,471,965
B	97,240,000	972,400	27,181,033	26,208,633	8	104,643,471	7,403,471
C	8,125,000	81,250	2,001,704	1,920,454	8	7,667,815	(457,185)
D	1,591,200,000	15,912,000	550,218,451	534,306,451	8	2,133,330,731	542,130,731
E	327,600,000	3,276,000	112,767,878	109,491,878	8	437,169,321	109,569,321
F	14,625,000	146,250	3,630,273	3,484,023	8	13,910,694	(714,306)

Table E.25: Scenario 4 – Electricity unit selling price Rs. 14.00, Interest Rate 10%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	33,762,617	32,447,017	10	122,999,723	(8,560,277)
B	97,240,000	972,400	24,710,030	23,737,630	10	89,984,293	(7,255,707)
C	8,125,000	81,250	1,819,731	1,738,481	10	6,590,210	(1,534,790)
D	1,591,200,000	15,912,000	500,198,592	484,286,592	10	1,835,827,206	244,627,206
E	327,600,000	3,276,000	102,516,253	99,240,253	10	376,198,638	48,598,638
F	14,625,000	146,250	3,300,248	3,153,998	10	11,956,135	(2,668,865)

Table E.26: Scenario 5 – Electricity unit selling price Rs. 15.00, Interest Rate 10%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	36,174,233	34,858,633	10	132,141,643	581,643
B	97,240,000	972,400	26,475,032	25,502,632	10	96,675,039	(564,961)
C	8,125,000	81,250	1,949,712	1,868,462	10	7,082,939	(1,042,061)
D	1,591,200,000	15,912,000	535,927,063	520,015,063	10	1,971,266,220	380,066,220
E	327,600,000	3,276,000	109,838,842	106,562,842	10	403,957,013	76,357,013
F	14,625,000	146,250	3,535,980	3,389,730	10	12,849,745	(1,775,255)

Table E.27: Scenario 6 – Electricity unit selling price Rs. 14.00, Interest Rate 12%

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Return	PV of Inv. After 5 years
A	131,560,000	1,315,600	33,762,617	32,447,017	12	116,964,235	(14,595,765)
B	97,240,000	972,400	24,710,030	23,737,630	12	85,568,843	(11,671,157)
C	8,125,000	81,250	1,819,731	1,738,481	12	6,266,834	(1,858,166)
D	1,591,200,000	15,912,000	500,198,592	484,286,592	12	1,745,744,782	154,544,782
E	327,600,000	3,276,000	102,516,253	99,240,253	12	357,738,902	30,138,902
F	14,625,000	146,250	3,300,248	3,153,998	12	11,369,458	(3,255,542)

Table E.28: Scenario 7 – Electricity unit selling price Rs. 15.00, Interest Rate 10% and Running hours are varied depending on actual situation

WHR Opp.	Total Investment Rs.	Total Overhead (OH) Cost 0.1% from Inv.	Exp. Turnover (TO) Rs.	Annual Return (TO-OH) Rs.	Int. Rate %	NPV of Income	PV of Inv. After 5 years
A	131,560,000	1,315,600	36,174,232	34,858,633	10	132,141,643	581,643
B	97,240,000	972,400	30,887,537	29,915,137	10	113,401,906	16,161,906
C	8,125,000	81,250	1,949,711	1,868,462	10	7,082,939	(1,042,061)
D	1,591,200,000	15,912,000	535,927,062	520,015,063	10	1,971,266,220	380,066,220
E	327,600,000	3,276,000	73,225,894	69,949,895	10	265,165,136	(62,434,864)
F	14,625,000	146,250	4,125,310	3,979,060	10	15,083,770	458,770