

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

ORGANIC RANKINE CYCLE EFFICIENCY OPTIMIZATION
BY USING AN ALGORITHMIC APPROACH

POYRAZ ATALAY
A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE
DEPARTMENT OF MECHANICAL ENGINEERING

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T.C.
GEBZE TEKNİK ÜNİVERSİTESİ
FEN BİLİMLERİ ENSTİTÜSÜ

ALGORİTMİK BİR YAKLAŞIM
KULLANILARAK ORGANİK RANKINE
ÇEVİRİMİ VERİMİNİN OPTİMİZASYONU

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YÜKSEK LİSANS TEZİ
MAKİNE MÜHENDİSLİĞİ ANABİLİM DALI

DANIŞMANI
PROF. DR. ALP ER ŞEVKİ KONUKMAN

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YÜKSEK LİSANS JÜRİ ONAY FORMU

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...../...../..... tarih ve/..... sayılı kararı.

SUMMARY

In this work, the efficiency optimization of the Organic Rankine Cycle was studied by creating an algorithm. Firstly, information was given about energy sources would be suitable for Organic Rankine Cycle. Similarities and differences between the Standard Rankine Cycle using water vapor as working fluid and the Organic Rankine Cycle using an organic fluid as working fluid were explained.

In the section where the calculations were made, cycle modeling was mentioned. Cycle and heat exchanger sizing codes were written in MATLAB. The thermodynamic properties of fluids were found using CoolProp libraries. The parameters affecting the cycle efficiency were taken as evaporation pressure, condensation pressure, superheating value, turbine and pump efficiencies. In order to make economic calculations after thermodynamic calculations, it was necessary to find the heat transfer area of the evaporator, condenser and regenerator. For that, the total heat transfer coefficients were calculated. Thus, required heat exchanger areas and purchased equipment costs were found.

The relationship between evaporation pressure with heat transfer areas of evaporator and condenser were studied for several organic fluids. The effect of evaporation pressure with the cost of evaporator and condenser were also studied. A novel finned tube type heat exchanger cost estimation method was introduced.

Key Words: Organic Rankine Cycle, Optimization, Heat Exchanger, Evaporator, Condenser, Regenerator

ÖZET

Bu çalışmada Organik Rankine Çevrimi'nin optimizasyonu bir algoritma yazılarak incelendi. Çalışmada öncelikle Organik Rankine Çevrimi'nin, hangi enerji kaynaklarının kullanımı için uygun olabileceği hakkında bilgi verildi. İş akışkanı olarak su buharı kullanan Standart Rankine Çevrimi ile iş akışkanı olarak organik bir akışkan kullanan Organik Rankine Çevrimi'nin arasındaki benzerlikler ve farklar belirtildi.

Hesapların anlatıldığı kısımda, çevrimin nasıl modellendiğinden bahsedildi. Çevrim ve eşanjör boyutlandırma kodlamaları MATLAB programında yapıldı. Akışkanların termodinamik özellikleri CoolProp kütüphanelerinden bulundu. Çevrim verimini etkileyen parametreler, evaporasyon basıncı, kondenzasyon basıncı, aşırı kızdırma miktarı, türbin ve pompa verimleri olarak alındı. Termodinamik hesaplardan sonra ekonomik hesapların yapılabilmesi için, evaporatör, kondenser ve rejeneratörün ısı transfer alanını bulmak gerekli idi. Bu katsayılar hesaplandı. Böylece gerekli alanlar ve satın alınan ekipman maliyetleri hesaplanmış oldu.

Evaporasyon basıncı ile evaporatörün ve kondenserin ısı transfer alanları arasındaki ilişki birkaç organik akışkan için çalışıldı. Evaporatör ve kondenserin maliyetleri ile evaporasyon basıncı arasındaki ilişki de ayrıca çalışıldı. Kanatlı borulu eşanjörler için yeni bir maliyetlendirme metodu tanıtıldı.

Anahtar Kelimeler: Organik Rankine Çevrimi, Optimizasyon, Eşanjör, Evaporatör, Kondenser, Rejeneratör

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LIST of ABBREVIATIONS and ACRONYMS

Abbreviations Explanations and Acronyms

GTÜ	: Gebze Teknik Üniversitesi
GTU	: Gebze Technical University
°C	: Degree Celcius
ORC	: Organic Rankine Cycle
HVAC	: Heating, Ventilating and Air Conditioning
ODP	: Ozone Depleting Potential
GWP	: Global Warming Potential
K	: Kelvin
mol	: Molecular Weight
sORC	: Simple Organic Rankine Cycle
rORC	: Regenerative Organic Rankine Cycle
0	: 0 denotes none regenerative ORC namely sORC
1	: 1 denotes regenerative ORC namely rORC
C	: Critical
el	: Evaporation Liquid Phase
esl	: Evaporator Sensible Liquid Phase Capacity
wf	: Working Fluid
m	: Flow Rate of the Working Fluid
po	: Pump Outlet
etp	: Evaporator Two Phase Capacity
ev	: Evaporation Vapor Phase
el	: Evaporation Liquid Phase
esv	: Evaporator Sensible Vapor Phase Capacity
ti	: Turbine Inlet
te	: Total Evaporator Capacity
tur	: Turbine Capacity
csv	: Condenser Sensible Vapor Phase Capacity

cv	: Condensation Vapor Phase
ctp	: Condenser Two Phase Capacity
cl	: Condensation Liquid Phase
csl	: Condenser Sensible Liquid Phase Capacity
pi	: Pump Inlet
tc	: Total Condenser Capacity
p	: Pump Power
η	: Cycle Efficiency
ro	: Regenerator Outlet
reg	: Regenerator Capacity

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1. INTRODUCTION

As the fossil fuel reserves in the world declines day by day, fuel price increases. Industry consumes a great percentage of the fossil fuels worldwide to manufacture products or produce electricity. This not only causes massive fuel consumption and depletion of the natural sources but also contributes to the global warming and air pollution as well. To diminish these effects, some kind of precautions must be applied. Waste heat recovery from industrial energy sources is one way to slow down the global warming and air pollution. By using waste heat, energy stored in the waste heat sources could be used in cooling, heating applications and to produce electricity.

One of the main classification methods of waste heat is to classify them by their temperature range. High temperature ($>650\text{ }^{\circ}\text{C}$), medium temperature ($230\text{--}650^{\circ}\text{C}$) and low temperature ($<230\text{ }^{\circ}\text{C}$) [1]. In the conversion of low temperature heat into electricity, among various cycles, a special Rankine Cycle could be utilised called Organic Rankine Cycle.

A typical low temperature heat source is the waste heat in combustion engines and in industrial processes, hot gases from furnaces in the steel and aluminum industry and from kilns in the ceramics industry, exhaust gases of diesel engines and gas turbines, hot liquids used to cool kilns or furnaces and so on.

Organic Rankine cycle (ORC) is a Rankine cycle, where an organic fluid is used instead of water as the working fluid. Particularly in low temperature applications many benefits may be obtained by using ORC instead of steam Rankine process [2]. In recent years' studies have shown that electricity production with Organic Rankine cycle provides an important percentage of the total power demand [3].

In this study, firstly thermodynamic calculations were made and then heat transfer area calculations were made for sizing evaporator, condenser and regenerator. As heat exchanger's working principles are quite simple, its difficult to make performance analysis due to the multiplicity of the parameters that affect heat transfer, like fin spacing, number of tubes, number of rows, plate geometry, tube material, fin material, working fluid, pass number and so on [4], [5]. Capacity calculations cannot be made precisely without an interpolative computer program and also these calculations must be verified with experiments.

In this study, overall heat transfer coefficients for both evaporator and condenser were thoroughly calculated. The goal of this research is to investigate how to reach thermodynamically maximum efficiency results when designing an ORC system and to study the effects of evaporation pressure on heat exchanger areas and costs. To realise it, Firstly, it is intended to use a commercial software by Friterm A.Ş [6]. To make preliminary thermal and constructive design of the heat exchangers. But the program was made to satisfy HVAC applications. Therefore the program won't work with ORC, because of the temperature range or fin type etc. Therefore, a Matlab program was created to model ORC and to find heat exchangers' areas. In the second part of the calculation section economical influences of the system were also taken into account.

1.1. Thermodynamic Background and Commercial Applications

In general, thermodynamics is the science that deals with energy production, storage, transfer and conversion. At present, fossil fuel is still the world's predominant energy source. But the burning of fossil fuels generates only thermal energy, therefore these energy sources are so called "primary energy sources", that must be converted to secondary energy source, so called energy carriers (electrical energy etc.).

To convert thermal energy into another form of energy a heat engine must be used. To utilise a heat engine, thermodynamic cycles are used. The thermodynamic cycles can be divided into two primary classes;

- Power cycles: Power cycles are cycles which convert some heat input into a mechanical work output.
- Heat pump cycles: Heat pump cycles transfer heat from low to high temperatures using mechanical work input [7].

1.2. Rankine Cycle

In general, the Rankine Cycle is an idealized thermodynamic cycle of a constant pressure heat engine that converts part of heat into mechanical work. In this cycle the heat is supplied externally to a closed loop, which usually uses water (in a liquid and vapor phase) as the working fluid. The working fluid in the Rankine Cycle undergoes the phase change from a liquid to vapor phase and vice versa [8].

1.3. Organic Rankine Cycle

Electricity could be produced at low temperatures by waste heat recovery or from the geothermal sources by using Organic Rankine Cycle. In addition, at the conditions which the temperature is below 500°C, both for low efficiency and technical difficulties, standard Rankine Cycle (steam process) applications are not suitable.

Waste heat sources which have low temperature energy, Organic Rankine Cycle applications are used instead of Standard Rankine Cycle over years and many studies have been made in this area. ORC, is a typical Rankine Cycle in a thermodynamical way, but the working fluid is an organic fluid instead of water. For this cycle, organic fluids are chosen because their boiling temperature is lower than water. Also organic fluid's molecular weight is heavier than water. This property enables the electricity production for low temperature heat sources which cannot be used with standard Rankine Cycle. These fluids are called working fluids. With organic fluids, electricity production could be made nearly for 500°C waste heat energy sources.

To convert low temperature heat sources to electricity, have some difficulties in both technical and economical ways. The traditional technology to produce electricity from waste heat sources, is steam turbine, however it is suitable for high temperature and high pressure applications. The optimal technology for low temperature heat sources is ORC. ($<150^{\circ}\text{C}$)

Industrial waste heat, internal combustion engine cooling water, geothermal heat, solar pond, fossil fuel waste heat, etc. are considered as heat sources for ORC.

1.4. Commercial Applications

There are some firms that sell turnkey ORC systems worldwide which could be listed as below.

- Siemens (Germany)
- Enogia (France)
- Exergy (Italy)
- General Electric (USA)
- GMK (Germany)
- GT-Energy Tech (Australia)
- Turboden (Italy)

2. LITERATURE RESEARCH

In the area of ORC modelling and equipment cost estimation for ORC equipment, many studies were made in the literature. Mostly there are studies regarding shell and tube heat exchanger cost estimation exist in the literature but also there are a few studies about finned tube heat exchanger cost estimation exist. Also in this chapter, heat transfer correlations used in this study were found separately and got together from several articles especially for condensation and evaporation correlations, fin efficiency formulations and so on.

The most frequent cost estimation method for heat exchanger was the logarithmic cost estimation by Turton method in the literature. It is expressed similar to given expression below. Often, this expression is multiplied with pressure and temperature dependant factors.

$$\log C_{hx}^0 = (K_1 + K_2 + K_3 \cdot \log(A)^2) \quad (2.1)$$

Lai and Alwi developed a new approach for heat exchanger network capital and utility costs. T-S diagrams and A-H diagrams were plotted for heat exchanger networks to find costs. Annualized costs were found using a correlation multiplied with heat exchanger areas [9].

$$\text{Annualized Cost} = \text{Annualized Factor} \cdot (1.3 \cdot 1.000 \cdot A^{0.83}) \quad (2.2)$$

Smith's book is a valuable source, used in this study. Capital cost correlations and base costs, base exponents and so on. were given in this book. At the end of this study, case study costs were compared with correlations used in Smith's book [10].

Kirmse and Oyewunmi used a logarithmic cost correlation to predict heat exchanger costs. K and F coefficients were selected for equipment type, material and pressure [11].

Aliyon and Mehrpooya used reference costs of the equipments and to bring the results up to date, used Chemical Engineering Plant Cost Index [12].

Mignard developed a correlation to predict heat exchanger costs using Chemical Engineering Plant Cost Index in the form of a linear equation [13].

Andreasen and Kærn studied multi-objective optimization of ORC using pure and mixed working fluids. Heat exchanger cost method were same as in the Smith's book [14].

Lemmens studied the cost engineering and estimation techniques and their applicability on ORC systems and expected between 10% to 30% accuracy. Logarithmic cost method used [15].

Heberle and Bassermann studied equipment costs and annual capital investment costs. Heat exchanger cost method was logarithmic and formulation have K, F factors and heat exchanger area [16].

Ehyaei and Ahmadi used cost correlations for evaporator and condenser as below [17].

$$C_{\text{evap}} = 216.6 + 353.4A_{\text{evap}} \quad (2.3)$$

$$C_{\text{cond}} = 338.6.A_{\text{evap}} \quad (2.4)$$

In Tian and Yue's study, they constituted an economical model. This model is also a logarithmic model and for the all equipment in the cycle cost correlations and coefficients were stated [18].

Valencia and Nunez used the same cost correlations for heat exchangers as in the Pierobon and Nyugen's study [19].

Koo and Oh used the cost correlation for heat exchangers as below [20].

$$PC_{\text{cond}} = 37940.A^{0.5155} \cdot \frac{CEPCI_{2017}}{CEPCI_{1998}} \quad (2.5)$$

$$PC_{\text{heater}} = (198.36.A + 52635) \cdot \frac{CEPCI_{2017}}{CEPCI_{1998}} \quad (2.6)$$

Weiland and Lance developed a cost correlation for different equipments for power cycle components. For example for dry cooler the equation form below were used [21]. This correlation is more realistic because it takes overall heat transfer coefficient into account.

$$C = a.U.A^b \quad (2.7)$$

Wang and Liu prepared an extensive table for recommended working fluids for different applications. In this study R123, R245fa and R601a fluids were selected. That table is quite useful in this study because, they list the candidate working fluids with waste heat source temperature [22].

Abrosimov and Baccioli studied the techno-economic analysis of combined inverted Brayton-Organic Rankine Cycle for high-temperature waste heat recovery. They used the Turton's method (logarithmic) for calculating the equipment costs. In this study turbine cost correlation is taken from Abrosimov and Baccioli's study [23].

$$C_{hex} = 190 + 310(A_{evap} + A_{cond}) \quad (2.8)$$

Ng and Tam were used the Turton method for costing the heat exchangers as well [24]. According to the Caputo and Pelagagge's studies, heat exchanger cost was correlated as below [25].

$$C_E = 13324 + 431.A^{0.91} \quad (2.9)$$

Sun and Wang studied the thermodynamic and economic optimization of a double pressure ORC system. Turton method were used, pressure and working temperature factors were added as well [26].

Alibaba and Pourdarbani used a different cost correlation method for heat exchangers and considered the heating capacity, temperature difference and flow rate in this formula [27].

$$C_{hx} = 6570. \left(\frac{\dot{Q}}{\Delta T} \right)^{0.8} + 21276. \dot{w}_t + 1184.4 m_{fg}^{1.2} \quad (2.10)$$

$$C_{cond} = 1773. m_{steam} \quad (2.11)$$

Mohammadi and Ashouri calculated the evaporator and superheater costs using heat transfer area multiplied with a coefficient [28]. Imran and Haglind used cost correlation for heat exchangers and dependant variable was heat exchanger mass. They used the formula below [29].

$$C_{hx} = 250 + 5. M_{hx} \quad (2.12)$$

Symister made a total overview of the cost estimation techniques used for chemical engineering. Different cost estimation methods and commercial softwares were used to predict U tube, floating head, U tube kettle type shell and tube heat exchangers and pumps, compressors, mixers [30].

Roumpedakis and Loumpardis studied exergetic and economic analysis of a solar driven small scale ORC. Heat exchanger cost estimation were used as below [31].

$$C_{hex} = 190 + 310. (A_{evap} + A_{cond}) \quad (2.13)$$

Castelli and Elsidio studied the optimization of the ORC system which utilizes waste heat from aluminium production plants and stated that exhaust gas temperature was between 200°C and 300°C [32]. Also in this study, waste heat was recovered from Aluminium melting furnace exhaust gas and its temperature was taken as 260°C. Song and Loo studied the thermos-economic optimization of ORC systems for geothermal power generation and used the logarithmic cost estimation method [33].

La Seta and Meroni studied combined turbine and cycle optimization for ORC systems. R245fa organic fluid were used and the cycle optimization were tried to increase by designing turbine blade [34].

Pierobon and Nyugen calculated heat exchanger costs as below [35].

$$PEC_{ht} = 10000 + 324A^{0.91} \quad (2.14)$$

These literature research shows that, there is no precise cost calculation method for finned tube heat exchangers. All articles that investigated, provide limited benefits. Because most of the correlations consist of heat transfer area multiplication with some coefficients or logarithmic expressions. Hence, there is a need for a new precise cost calculation method.

In this study a cost calculation method for finned tube heat exchangers was developed. This method is based on a finned tube type heat exchanger firm's costing method. This method takes account of number of tubes, number of rows, fin pitch, length of heat exchanger, material and so on. That means all parameters to design heat exchangers were taken into account.

After the search for cost correlation methods in the open literature, heat transfer area calculation correlations were used in the study also researched. In the study of Li and Hao, R123 working fluid were used for a ORC system and the fluid's superheated steam state temperature was taken as 180 °C. Fluid properties were divided into superheated, saturated and unsaturated states [36].

In Coulson and Richardson's book heat transfer correlations for shell and tube heat exchangers exist. Also there are information about finned tube heat exchangers and overall heat transfer coefficients for such devices available as list in that source [37]. Information about available waste heat sources and their applicability in the industry were found in the literature [38].

In Jung and Krumdieck's study was a quite useful source used in this study. Equipments, T-S diagrams, preheater usage, pressure selection, working fluid deployment were quite useful. In that study exhaust gas was modelled as air. R123 fluid were used 25 bar evaporation pressure and 10 bar condensation pressure were used and thermodynamically 14.9% efficiency was obtained [39].

Bălănescu and Homutescu studied R134a and R123 working fluids in a ORC and achieve the 11.6% efficiency with R123 fluid [40]. The main source for heat transfer calculations and sizing the heat exchanger in this study, several sources in the literature have a crucial role. First one is the VDI Heat Atlas Book [41]. The correlations of several chapters were used.

- Heat Transfer in Cross-Flow Around Single Rows of Tubes and Through Tube Bundle
- Heat Transfer in Pipe Flow
- Part M Specific Heat Transfer Problems

- Heat Transfer to Finned Tubes

For evaporation and condensation, Shah Correlations were used [42], [43]. LMTD calculation of regenerator in this study was formulized using Genceli's book [44]. In addition to the VDI Heat Atlas Book for getting the necessary correlations, a handbook of Zeller's was useful to visualize the formulas for pipe inner and outer areas [45].

3. EQUIPMENT USED IN ORC WASTE HEAT RECOVERY

The essential equipment used with ORC system to produce electricity is described below.

- Turbine (Expansion device) + Generator
- Condenser
- Pump
- Evaporator
- Working Fluid
- Waste Heat Source

3.1. Turbine and Generator

The purpose of using a compressor is to increase the fluid's pressure so it could be said that a compressor increases the energy of a fluid. An Expansion device (scroll or turbine) has the opposite purpose of a compressor. The working fluid enters an expansion device in high temperature and high pressure is expanded in this device and some amount of fluid's energy transformed to mechanical energy. Expansion device's output shaft is connected to generator and electricity is produced in the generator. The reason of the working fluid selection of heavy molecular weighted fluids in the ORC are, these fluids have lower boiling point than water and lower boiling pressure than water, and this fluid flow to the expander blades does not cause damage.

3.2. Condenser, Evaporator and Heat Exchangers

The evaporator and condenser type could be selected and vary according to designer or the system but it could be classified basically as the waste heat physical phase and cooling media phase of condenser.

If cooling water is abundant, water cooled condenser could be used. In this case mostly shell and tube type condenser is used. If cooling water is not abundant, air cooled condenser could also be used.

If the waste heat is in the vapor state, shell and tube or finned tube type evaporator could be used, but heat source temperature must be considered with using with finned and tube type evaporator because mostly fin material is aluminum or copper alloys and the pipe material is copper. This material selection limits the usage of waste heat temperature range because it is not recommended to use these equipment parts near melting point or near homologous temperature which is described below.

$$T_{\text{homologous}} = \frac{T_{\text{working}}}{T_{\text{melting}}} \quad (3.1)$$

For most of the metals this rate must be below 0.4. Also fin pitch is also another important selection criteria while if fin pitch is smaller, particules in the exhaust flue gas could accumulate between the fins and this hinder the heat transfer. Therefore, if finned and tube type evaporator is used in a rankine cycle fin, tube and header material, fin pitch and so on must be taken into account.

3.3. Working Fluid Pump

To compensate the pressure drop across the cycle a working fluid pump must be connected at the condenser outlet. Pump type is centrifugal.

There are two points must be taken into account in ORC in pump and turbine in the same manner. First one is, the working fluid must be in superheated vapor phase entering the turbine, because if there are any condense drop inside the working fluid, that damage the turbine blades. Second one is about working fluid pump. After condenser outlet, (in the pump inlet) there must be no gas phase working fluid must be remained. If so, working fluid could not be pumped fully and that, eventually decrease the cycle efficiency. Therefore, superheating process is applied before turbine inlet and subcooling process is applied before pump inlet.

3.4. Working Fluid

During system premodelling, parameters like working fluid's evaporation temperature and evaporation pressure, condensation temperature and condensation pressure, critical point temperature and critical point pressure, environmental effects of working fluid and cost must be considered. Like as in the simple selection figure below, according to the waste heat temperature, suitable working fluid could be seen.

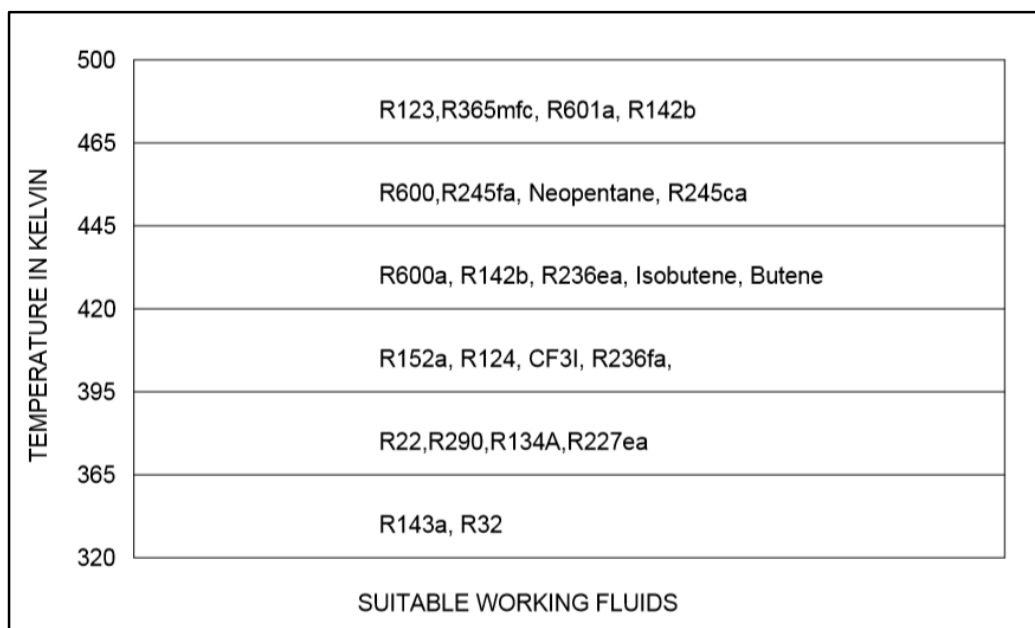


Figure 3.1: Waste heat source temperature vs. suitable working fluids.

In this study 3 organic fluids were used for several case studies which are R123, R601a and R245fa. Information about these fluids were given below.

- R123

2,2-Dichloro-1,1,1-trifluoroethane or HCFC-123 is considered as an alternative to CFC-11 in low pressure refrigeration and HVAC systems. Properties regarding this fluid is given below [46].

Ozone depletion potential (ODP): 0.012

Global warming potential (GWP): 76

Chemical formula: $\text{C}_2\text{HCl}_2\text{F}_3$

Molar mass: $152.93 \text{ g}\cdot\text{mol}^{-1}$

Appearance: Colorless liquid

Density: 1.4638 g/cm^3

Melting point: -107°C (166 K)

Boiling point : 27.82°C (300.97 K)

HCFC-123 will eventually be phased out under the current schedule of the Montreal Protocol, but can continue to be used in new HVAC equipment until 2020 in developed countries, and will still be produced for service use of HVAC equipment until 2030. Developing countries can use in new equipment until 2030, and can be produced for use in service until 2040.

HCFC-123 is used in large tonnage centrifugal chiller applications, and is the most efficient refrigerant currently in use in the marketplace for HVAC applications.

- R245fa

1,1,1,3,3-Pentafluoropropane (HFC-245fa) is a hydrofluorocarbon used primarily for closed-cell spray foam insulation produced by Honeywell and in Asia by Sinochem. Honeywell markets HFC-245fa under the Enovate and Genetron 245fa brand names. HFC-245fa is also known as pentafluoropropane and by its chemical name 1,1,1,3,3,-pentafluoropropane. Unlike CFC and HCFC blowing agents formerly used for this purpose, it has no ozone depletion potential and is nearly non-toxic [47], [48].

Ozone depletion potential is ODP: Non-ozone depleting

Global warming potential is GWP: 1030

Chemical formula: $\text{CF}_3\text{CH}_2\text{CHF}_2$ (1,1,1,3,3- pentafluoropropane)

Molar mass: 134,03 g/mol

Appearance: colourless

Density: 1,32 g/cm³ at 20 °C

Melting point: -103 °C

Boiling point : 15,3 °C

- R601a

Isopentane, also called methylbutane or 2-methylbutane, is a branched-chain saturated hydrocarbon (an alkane) with five carbon atoms, with formula C_5H_{12} or $\text{CH}(\text{CH}_3)_2(\text{C}_2\text{H}_5)$. Isopentane is an extremely volatile and extremely flammable liquid at room temperature and pressure. It is also the least dense liquid at standard conditions. The normal boiling point is just a few degrees above room temperature and isopentane will readily boil and evaporate away on a warm day. Isopentane is used in a closed loop in geothermal power production to drive turbines. Isopentane is a major component (sometimes 30% or more) of natural gasoline, an analog of common petroleum-derived gasoline that is condensed from natural gas. It has a substantially higher octane rating (RON 93.7) than n-pentane (61.7), and therefore there is interest in conversion from the latter. [49]

Ozone depletion potential is ODP: Not classified as dangerous for the ozone layer

Global warming potential is GWP: Not included in the list of fluorinated greenhouse gases

Chemical formula: C_5H_{12}

Molar mass: 72.151 g·mol⁻¹

Appearance: Colorless liquid

Density: 72.15 g/mol

Melting point: -161 to -159 °C; (112 to 114 K)

Boiling point : 27.8 to 28.2 °C; (300.9 to 301.3 K)

4. WASTE HEAT SOURCES CLASSIFICATION

Waste heat in the industry arises from industrial systems in facilities. Major industrial waste heat sources are exhaust or flue gases. Additionally, hot air used in the processes, dryer, heat treatment furnaces, heat exchangers, cooling fluids, various gases, could be classified as waste heat sources. From the table 4.1, waste heat sources, temperature range, fouling states are listed [38].

Table 4.1: Temperature range for some waste heat sources in industry.

WASTE HEAT SOURCE	TEMPERATURE RANGE (°C)	FOULING
Furnace or Heating System Exhaust Gases	300-1100	Various
Gas turbine exhaust gases	480-600	Clean
Engine cooling water	85-95	Clean
Exhaust gases (general)	450-600	Mostly clean
Hot Surfaces	65-315	Clean
Compressor intercooler water	35-85	Clean
Vapor passages and leaks	120-320	Mostly clean
Condense Products	65-260	Clean
Emission Control Devices	60-800	Mostly clean

In the figure 4.1, industrial waste heat sources and estimated energy losses in BTU unit are given [38]. What draws attention is, in the processes which combustion occurs, have the higher waste heat energy potential.

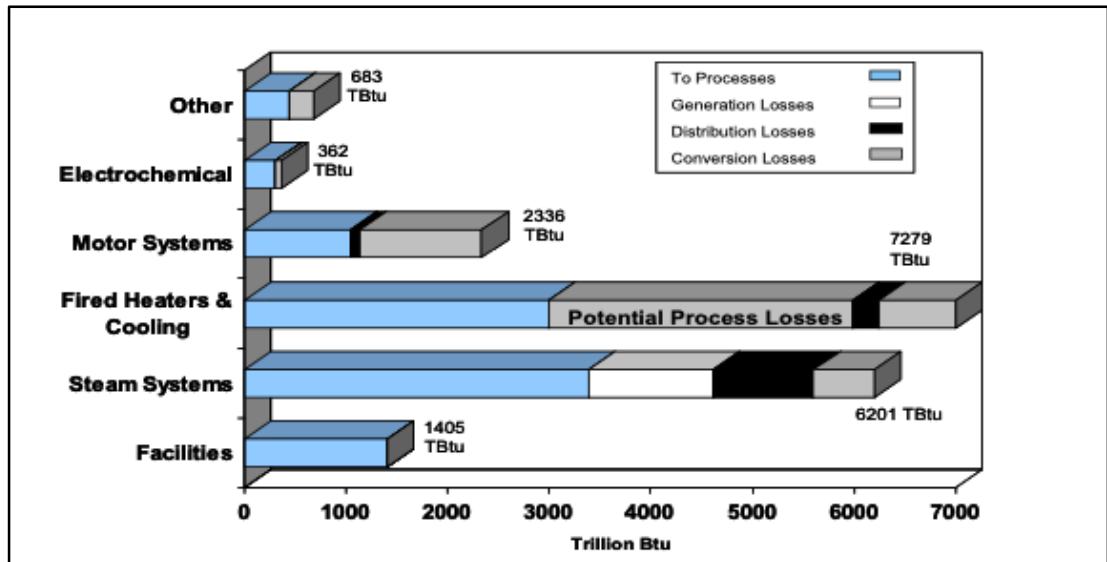


Figure 4.1: Estimates of energy losses for major energy use areas in manufacturing.

In the table 4.2, suitability for waste heat recovery of iron-steel, aluminum, glass or cement production industries can be seen [38].

Table 4.2: Temperature range and characteristics for industrial waste heat sources

Industry	Waste Heat Source	Temperature Range (°C)	Recovery Usage
Iron-Steel	Blast furnace gases	400-600	Wide spread
	Ark furnace gases	1500-1600	Wide spread
	Simple O ₂ processes	1250-1700	Wide spread
Glass	Glass, Fiber	980-1430	Usage is limited
	Glass melting furnaces	480-1430	Usage is limited
Aluminum	Aluminum melting furnaces exhaust gases	300-500	Possible, but not enough application
Cement	Cement Production facilities exhaust gases	200-400	Possible, but not enough application

As it could be seen from the table, the most suitable waste heat energy sources for ORC are below ≤ 500 (°C). Also these energy sources must have constant flow rate, fluctuations of exhaust gas temperature must be minimum to ensure continuous electricity production and cycle efficiency.

The most suitable candidates for ORC seems to be the systems running with fossil fuel burners, because, for example as in the aluminum melting furnaces or steel melting furnaces, combustion air is drawn via a fan to the burner and these conditions are convenient to model a cycle. Because to keep the system running, burner must be provided by combustion air and this leads to the constant rate and temperature of exhaust flue gas. Also these type of systems run through the year. Only for urgent maintenance reasons furnaces are stopped.

5. ALGORITHM CONSTRUCTION

To model an ORC system and make thermodynamic calculations, commercial programs like Aspen Plus, DWSIM, EES, EES based CoolPack or web based SmoWeb could be used. However, in this study an original code is developed according to address this studie's needs.

Nowadays, to size heat exchangers used in ORC, capacity guaranteed, iterative, commercial computer programs are used. The important steps in heat exchanger sizing are using the correct heat transfer correlations for each working fluid and temperaure range or fluid regime. As heat exchanger modelling is also part of this study, overall heat transfer coefficients were calculated thoroughly by an algorithm.

Turbine and pump modelling are not part of this study. Only, the theoretical power output from thermodynamic calculations multiplied with efficiency of each device to find real power outputs for pump and turbine.

To succefully make thermodynamical, heat transfer and cost calculations, an algorithm was constructed. This algorithm was divided into three parts. In the first part, only thermodynamic calculations were made undependent from heat exchanger sizing or pump and turbine sizing. The second part of the algorithm were used to calculate heat exchanger's overall heat transfer coefficients and heat transfer areas. The third part of the algorithm were used to sense the economical influence of the cycle. In that part a realistic equipment costing method by a commercial heat exchanger firm for both condenser and evaporator were used.

Economical cost correlations were used for pump, turbine and regenerator found in the literature. Heat exchanger type assumed in this study for evaporator and condenser are finned tube type heat exchangers and for regenerator, shell and pipe type heat exchanger.

5.1. Method

System algorithm, modelling and coding were created in MATLAB. The reason of selection of this software is first of all free licensing for GTU students and coding is much easier than other standard programming languages.

Thermodynamic properties of the fluids at each point were found by an external library instead of looking up from charts. These type of library usage are common in commercial heat exchanger sizing or cycle softwares. Generally, Refprop library is used in such software but this library is not freeware. Therefore, in this study a similar freeware library called Coolprop were used.

Usage of this library is not direct for MATLAB software. To use Coolprop libraries in MATLAB, first of all, Python programming language must be downloaded and installed. Then, in Python terminal, Coolprop libraries are installed. After these steps, Coolprop library is ready to use in MATLAB as in below. For example, to find the evaporation enthalpy in liquid phase of R123 fluid the following syntax is used.

$$H_{el}=py.CoolProp.CoolProp.PropsSI("H","P", P_{evap}, "Q", 0, "R123") \quad (5.1)$$

After the decision of the method, all equipment are modelled, and thermodynamic points are found by using Coolprop library. The most obvious step for efficiency optimization is to use a internal heat exchanger namely a regenerator between pump outlet and turbine outlet (Regenerative ORC). Algorithm's general flow diagram was made according to the regenerator included version but in conclusion section of this study non regenerator vs. regenerator included version was compared.

The evaporation pressure increase also increases the efficiency until maximum point. After that point efficiency decreases so an iterative code was created to find the maximum of these iterations. The other step for optimization of efficiency is to increase the turbine inlet temperature. This could be done by increasing the superheating temperature in evaporator but the turbine inlet temperature must not exceed the critical temperature of the working fluid. Therefore, turbine inlet

temperature is parametrically fixed to (T_c-1) . $[^{\circ}\text{C}]$ The last parameter for efficiency optimization is to decrease condensation temperature but in this study condensation pressure was kept constant. After these assumptions maximum efficiency was calculated. Thus, the first objective of this study was achieved. The other objectives of this study are to find out the relationship between evaporation pressure with heat exchanger areas and costs.

After cycle efficiency maximization, heat exchanger areas were calculated corresponding to the evaporation pressures in case studies and cost of heat exchangers were found. Total cost influence of the cycle was investigated by computing pump, regenerator and turbine costs. Turbine, pump and regenerator equipment costs were calculated with the correlations and coefficients, found in the literature and thus the this objective were also achieved.

5.2. ORC Modelling Algorithm

ORC modelling code in MATLAB, created for both simple ORC and for regenerative ORC case. It is obvious that, for regenerative ORC case, system efficiency is higher. From this point on, regenerative ORC will be called rORC, and non regenerative ORC will be called sORC. In this section, for both cases, thermodynamical points will be explained. From the figure below, sORC cycle utilized in this study could be seen.

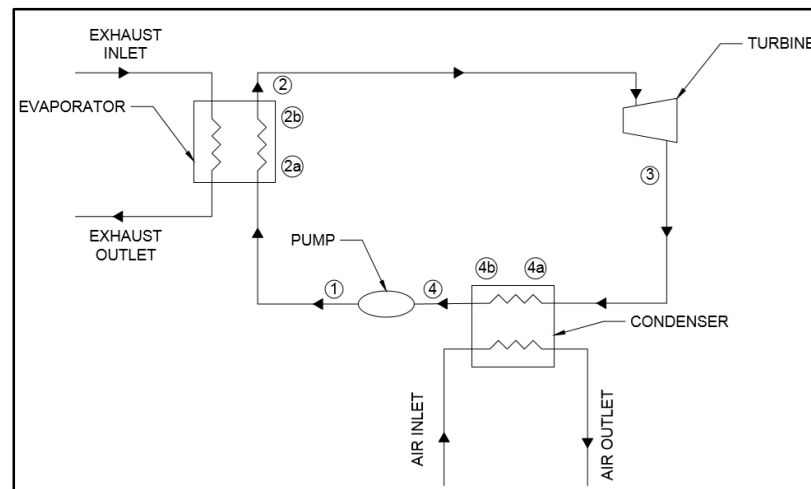


Figure 5.1: Simple ORC cycle.

Between points 1&2a: Working fluid is in the liquid phase. Working fluid is heated in the evaporator until the boiling point.

$$Q_{esl} = m_{wf0} \cdot (h_{el0} + Q_{p00}) \quad (5.2)$$

Between points 2a&2b: Working fluid starts to evaporate at point 2a. Vapor quality is increased from 0 to 1. At point 2b working fluid is fully evaporated.

$$Q_{etp0} = m_{wf0} \cdot (h_{ev0} + h_{el0}) \quad (5.3)$$

Between points 2b&2: From point 2b to 2, working fluid is superheated. In the 2b point, working fluid is in the saturated vapor phase. Then working fluid is superheated to the turbine inlet temperature. After point 2 working fluid leaves the evaporator.

$$Q_{esv0} = m_{wf0} \cdot (h_{ti0} + h_{ev0}) \quad (5.4)$$

Therefore, total evaporator power calculated as:

$$Q_{te0} = Q_{esl0} + Q_{etp0} + Q_{esv0} \quad (5.5)$$

Between points 2&3: Working fluid, hits the turbine blades and fluid's energy is converted to mechanical energy. A generator is connected to the turbine output shaft and thus electrical energy is created.

$$Q_{tur0} = m_{wf0} \cdot (h_{turi0} - h_{turo0}) \quad (5.6)$$

Between points 3&4a: At point 3 working fluid enters the condenser. Working fluid is still in the superheated vapor phase. Between these points desuperheating process occurs. At point 4a working fluid is in the saturated vapor phase.

$$Q_{csv0} = m_{wf0} \cdot (h_{turo0} - h_{cv0}) \quad (5.7)$$

Between points 4a&4b: After point 4a, working fluid starts to condense. Vapor quality decreases from 1 to 0. At point 4b, working fluid is in the saturated liquid phase.

$$Q_{ctp0} = m_{wf0} \cdot (h_{cv0} - h_{cl0}) \quad (5.8)$$

Between points 4b&4: Between these points, working fluid subcooling process occurs. If there's no subcooling process made, working fluid inside the pump may change its phase to vapor again and vapor can not be pumped properly. Thus all working fluid inside the pump must be in liquid phase.

$$Q_{csl0} = m_{wf0} \cdot (h_{cl0} - h_{pi0}) \quad (5.9)$$

Therefore, condenser total power is calculated as:

$$Q_{tc0} = Q_{csv0} + Q_{ctp0} + Q_{csl0} \quad (5.10)$$

Between points 4&1: After point 4b, working fluid leaves the condenser and enters the pump. Inside pump, working fluid pressure increased and pumped to the evaporator to keep the cycle running.

$$Q_{p0} = m_{wf0} \cdot (h_{po0} - h_{pi0}) \quad (5.11)$$

Total system efficiency in this simple ORC is calculated as:

$$\eta_{sORC} = \frac{Q_{turo} - Q_{p0}}{Q_{te0}} \quad (5.12)$$

From the figure below, regenerative ORC cycle utilized in this study could be seen.

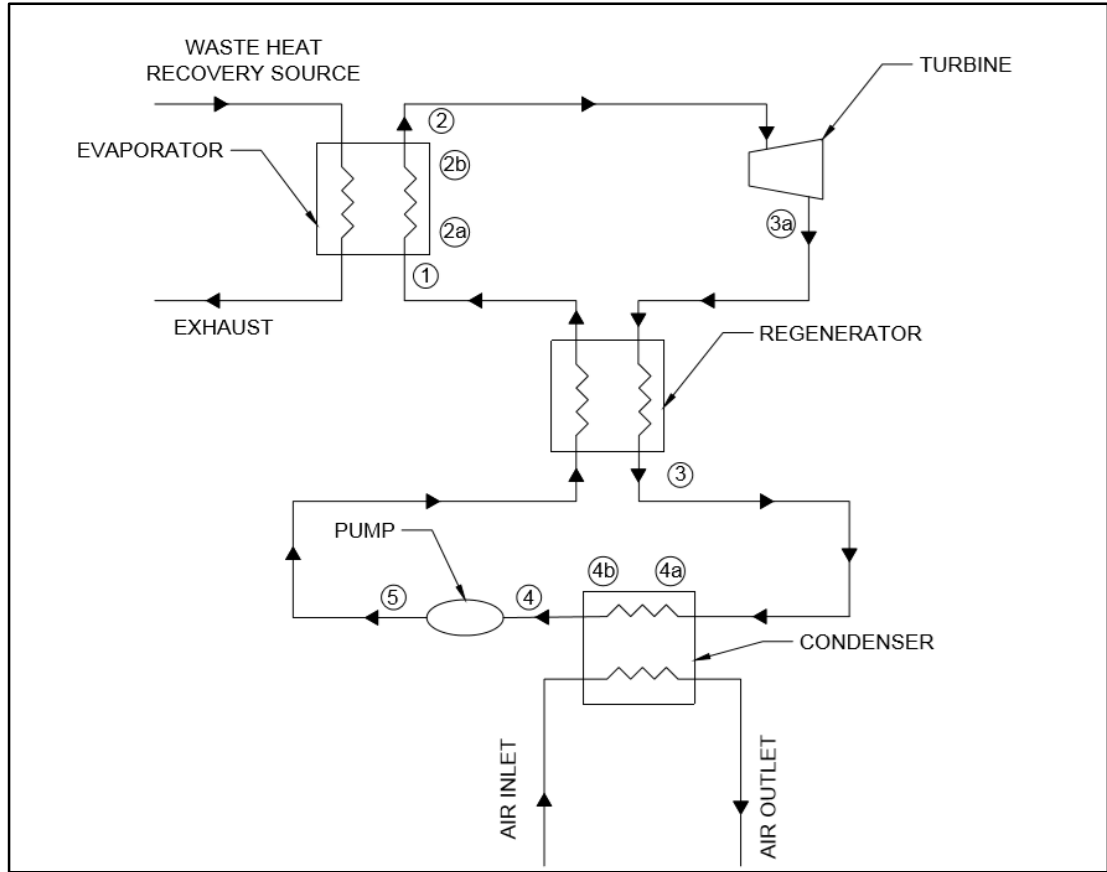


Figure 5.2: Regenerative ORC cycle.

Between points 1&2a: Working fluid is in the liquid phase. Working fluid is heated in the evaporator until the boiling point.

$$Q_{esl1} = m_{wf1} \cdot (h_{el1} - h_{ro}) \quad (5.13)$$

Between points 2a&2b: Working fluid starts to evaporate at point 2a. Vapor quality is increased from 0 to 1. At point 2b working fluid is fully evaporated.

$$Q_{etp1} = m_{wf1} \cdot (h_{ev1} - h_{el1}) \quad (5.14)$$

Between points 2b&2: From point 2b to 2, working fluid is superheated. In the 2b point, working fluid is in the saturated vapor phase. Then working fluid is superheated to the turbine inlet temperature. After point 2 working fluid leaves the evaporator.

$$Q_{esv1} = m_{wf1} \cdot (h_{ti1} - h_{ev1}) \quad (5.15)$$

Therefore, total evaporator power outlet is

$$Q_{te1} = Q_{esl1} + Q_{etp1} + Q_{esv1} \quad (5.16)$$

Between points 2&3a: Working fluid, hits the turbine blades and fluid's energy is converted to mechanical energy. A generator is connected to the turbine output shaft and thus electrical energy is created.

$$Q_{tur1} = m_{wf1} \cdot (h_{turi1} - h_{turo1}) \quad (5.17)$$

Between 3a&3 (Also 5&1): To increase the evaporator inlet temperature and to decrease condenser inlet temperature, an heat exchanger (regenerator) between pump outlet and turbine outlet is utilized.

$$Q_{reg} = m_{wf1} \cdot (h_{turo1} - h_{ro}) \quad (5.18)$$

Between points 3&4a: At point 3 working fluid enters the condenser. Working fluid is still in the superheated vapor phase. Between these points desuperheating process occurs. At point 4a working fluid is in the saturated vapor phase.

$$Q_{csv1} = m_{wf1} \cdot (h_{ro} - h_{cv}) \quad (5.19)$$

Between points 4a&4b: After point 4a, working fluid starts to condense. Vapor quality decreases from 1 to 0. At point 4b, working fluid is in the saturated liquid phase.

$$Q_{ctp1} = m_{wf1} \cdot (h_{cv1} - h_{cl1}) \quad (5.20)$$

Between points 4b&4: Between these points, working fluid subcooling process occurs. If there's no subcooling process made, working fluid inside the pump may change it's phase to vapor again and vapor can not be pumped. Thus all working fluid inside the pump must be in liquid phase.

$$Q_{cs1} = m_{wf1} \cdot (h_{cl1} - h_{pi1}) \quad (5.21)$$

Therefore, condenser total power is calculated as:

$$Q_{tc1} = Q_{csv1} + Q_{ctp1} + Q_{cs1} \quad (5.22)$$

Between points 4&5: After point 4, working fluid leaves the condenser and enters the pump. Inside pump, working fluid pressure increased and pumped to the evaporator to keep the cycle running.

$$Q_{p1} = m_{wf1} \cdot (h_{po1} - h_{pi1}) \quad (5.23)$$

Total system efficiency in this regenerative ORC is calculated as:

$$\eta_{rORC} = \frac{Q_{tur1} - Q_{p1}}{Q_{te1}} \quad (5.24)$$

In this study, to model the ORC cycle, the algorithm below was utilized.

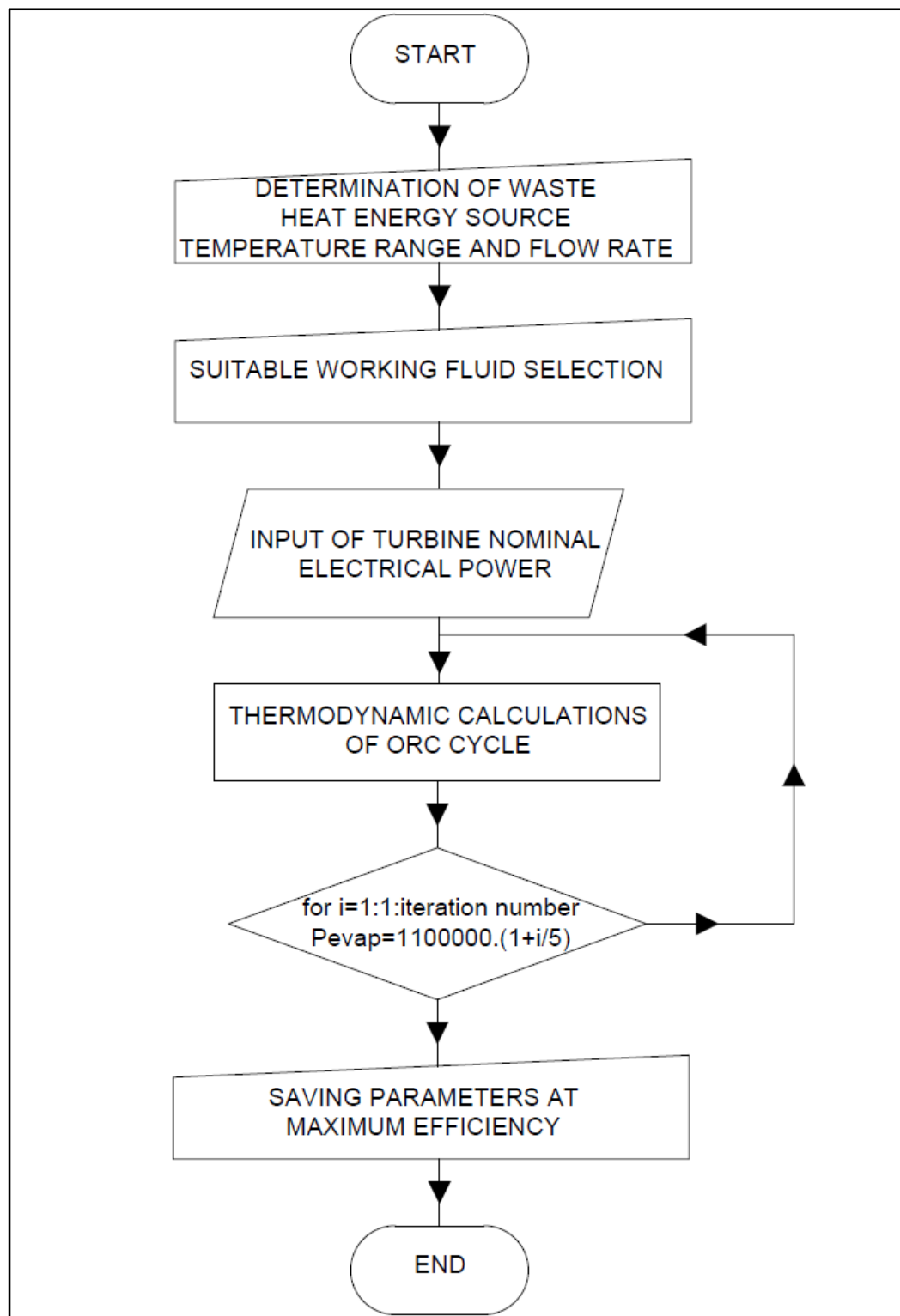


Figure 5.3: ORC cycle algorithm.

5.3. Heat Exchanger Area Calculation Algorithm

In this section of the algorithm, evaporator, condenser and regenerator heat transfer areas calculation method were described. To find heat transfer areas, the parameters below were needed. For exhaust gas side, volumetric flow rate, inlet temperature, fluid type. In this study exhaust gas was modeled as air [39]. For tube side, necessary parameters were obtained from the first part of the algorithm. For evaporator and condenser all correlations used in the algorithm explained below.

To calculate evaporator and condenser heat transfer areas, devices must be divided into 3 parts. For evaporator, parts are called as heating zone (working fluid liquid phase), phase change zone (working fluid evaporation) and superheating zone (working fluid is in vapor phase). For condenser, parts are called as desuperhating zone (working fluid is in vapor phase), phase change zone (working fluid condensation), subcooling zone (working fluid is in liquid phase). Heat exchanger working schemes are presented below.

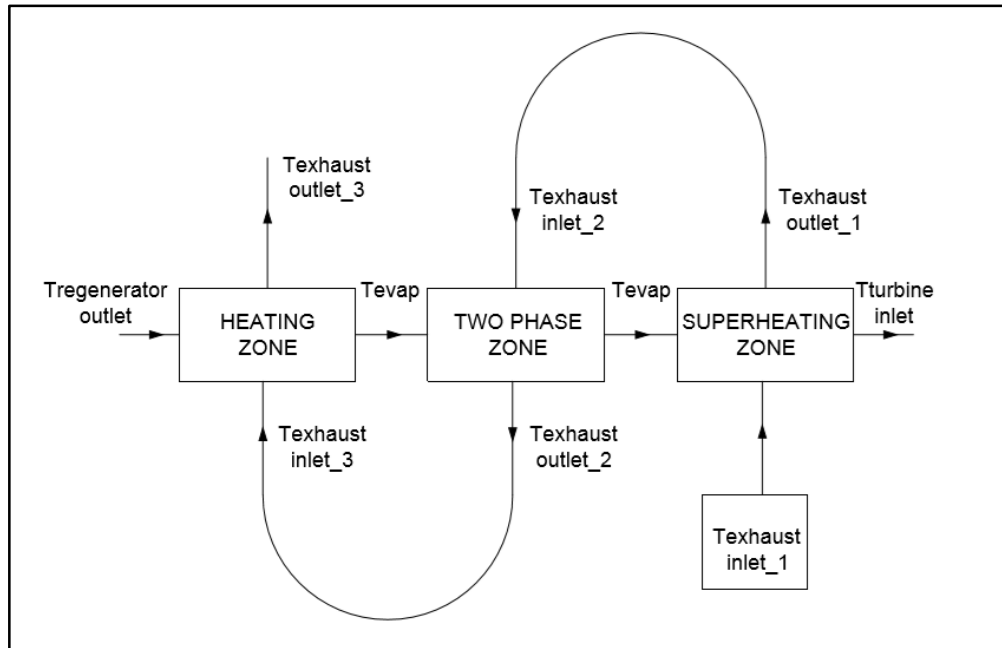


Figure 5.4: Evaporator working scheme.

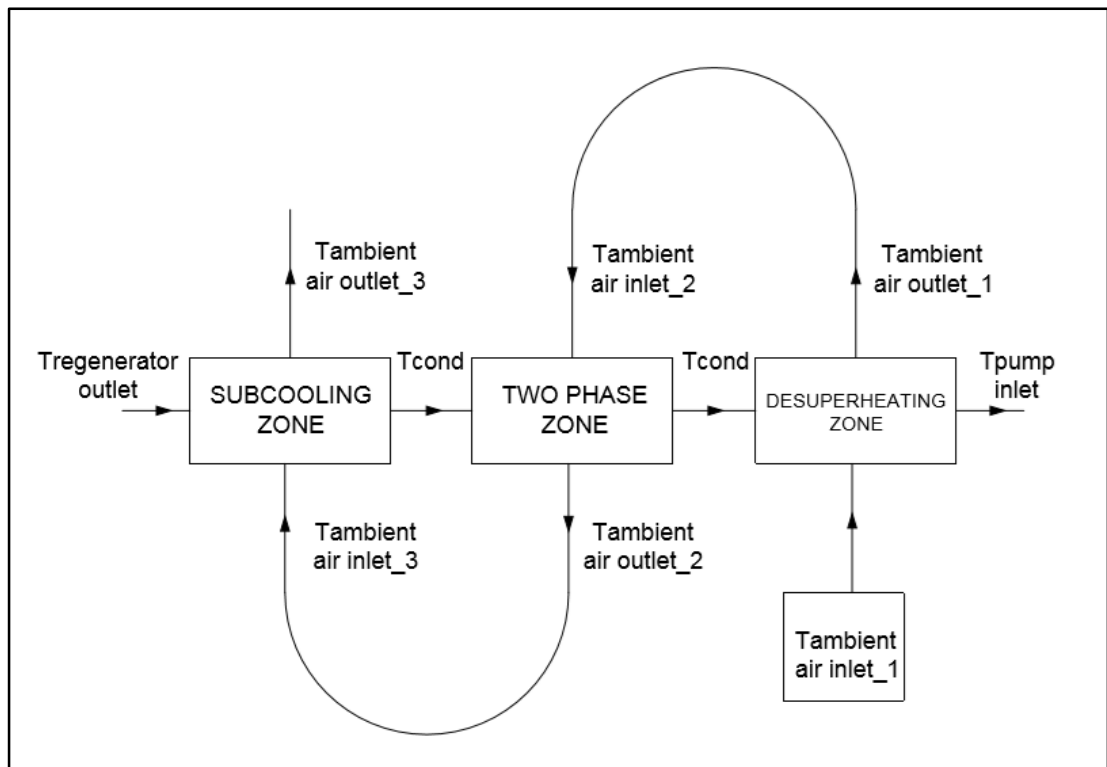


Figure 5.5: Condenser working scheme.

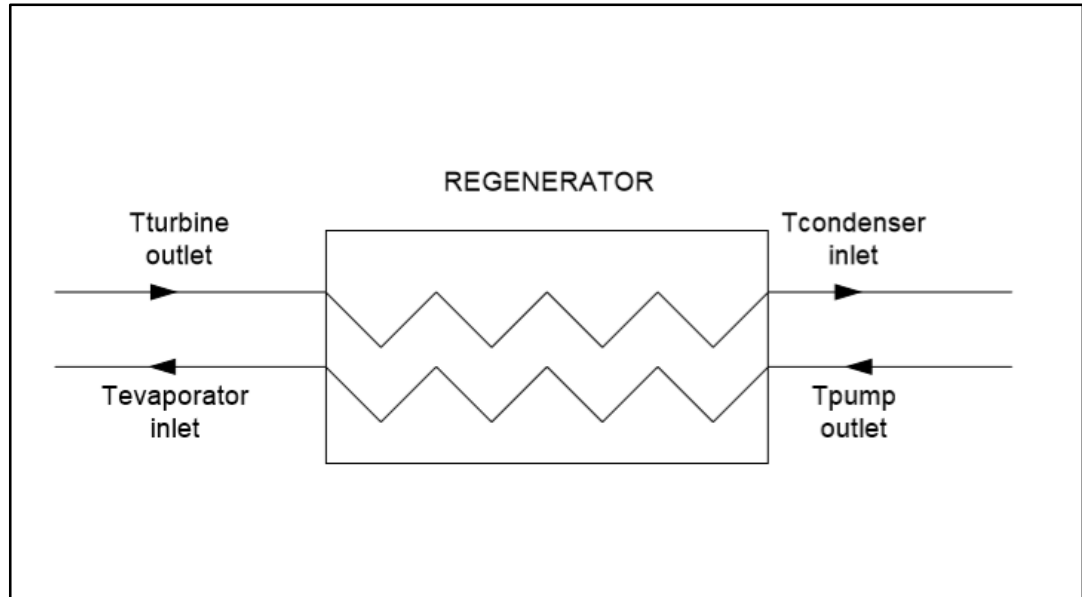


Figure 5.6: Regenerator working scheme.

All correlations used in the heat exchanger sizing are presented below.

- Exhaust Side Correlations are listed below [41].

Streamed length, i.e., the length of the flow path traversed over a single tube is calculated as:

$$I = D_o \cdot \frac{\pi}{2} \quad (5.25)$$

Void fraction ratio is calculated as:

$$a = \frac{B_{st}}{D_o} \quad (5.26)$$

$$b = \frac{B_{sl}}{D_o} \quad (5.27)$$

$$\Psi = 1 - \frac{\pi}{4 \cdot a} \text{ for } b \geq 1 \quad (5.28)$$

$$\Psi = 1 - \frac{\pi}{4 \cdot a \cdot b} \text{ for } b < 1 \quad (5.29)$$

Reynolds Number Calculation:

$$Re_{fin} = \frac{G_{fin} \cdot I}{\vartheta_{fin} \cdot \Psi} \quad (5.30)$$

Nusselt number Calculations:

Nusselt number for both laminar and turbulent flow calculated and these results put in the average Nusselt number formula.

$$Nu_{fin,l} = 0.664 \cdot \sqrt{Re_{fin}} \cdot \sqrt[3]{Pr_{fin}} \quad (5.31)$$

$$Nu_{fin,t} = \frac{0.037 \cdot Re_{fin}^{0.8} \cdot Pr}{1 + 2.443 \cdot Re_{fin}^{0.1} \cdot (Pr_{fin}^{2/3} - 1)} \quad (5.32)$$

Average Nusselt number is calculated as:

$$Nu_{fin,avg} = 0.3 + \sqrt{Nu_{fin,l}^2 + Nu_{fin,t}^2} \quad (5.33)$$

Average Nusselt number must be multiplied with inline or staggered configuration factor (f_a), temperature dependent variations (f_{tw}), effect of number of rows (f_k),

$$Nu_{fin} = Nu_{fin,avg} \cdot f_a \cdot f_{tw} \cdot f_k \quad (5.34)$$

Mean fin side heat transfer coefficient calculated as:

$$h_{fin} = Nu_{fin} \cdot \frac{k_{fin}}{I} \quad (5.35)$$

Exhaust side heat transfer coefficient enhancement because of fins, must be multiplied with the h_{fin} . Fin efficiency calculations were presented below.

$$I_f = \sqrt{B_{sl}^2 + \frac{B_{st}^2}{4}} \quad (5.36)$$

$$\varphi' = 1.27 \cdot \frac{B_{st}}{D_o} \cdot \sqrt{\frac{I_f}{B_{st}}} - 0.3 \quad (5.37)$$

$$\varphi = (\varphi' - 1) \cdot (1 + 0.35 \cdot \ln(\varphi')) \quad (5.38)$$

$$X = \varphi' \cdot \frac{D_o}{2} \sqrt{\frac{2 \cdot h_{fin}}{k_{fin} \cdot \delta}} \quad (5.39)$$

$$\eta_f = \frac{\tanh(X)}{X} \quad (5.40)$$

$$\eta_{ff} = \frac{A_{ss} \cdot \eta_f \cdot A_{sp}}{A_{tot}} \quad (5.41)$$

- Tube Side Correlations

Reynolds Number Calculation:

$$Re_{tube} = \frac{V_{tube} \cdot D_i}{\vartheta_{tube}} \quad (5.42)$$

Friction coefficient:

$$\xi = (1.8 \cdot \log_{10}^{Re_{tube}} - 1.5)^{-2} \quad (5.43)$$

Nusselt Number calculation:

For laminar flow:

$$Nu_{t,1} = 3.66 \quad (5.44)$$

$$Nu_{t,2} = \sqrt[3]{Re_{tube} \cdot Pr_{tube} \cdot \frac{D_i}{L}} \quad (5.45)$$

$$Nu_{t,3} = \left(\frac{2}{1 + 22 \cdot Pr_{tube}} \right)^{1/6} \cdot \sqrt{Re_{tube} \cdot Pr_{tube} \cdot \frac{D_i}{L}} \quad (5.46)$$

$$Nu_{l,tube} = \sqrt[3]{Nu_{t,1}^3 + 0.7^3 + (Nu_{t,2} - 0.7)^3 + Nu_{t,3}^3} \quad (5.47)$$

For turbulent flow:

$$Nu_{tur} = \frac{(\xi/8) \cdot Re_{tube} \cdot Pr_{tube}}{1 + 12.7 \cdot \sqrt{\frac{\xi}{8}} \cdot (Pr_{tube}^{\frac{2}{3}} - 1)} \cdot \left(1 + \frac{D_i}{L}\right)^{2/3} \quad (5.48)$$

For transient flow: Nusselt number will be interpolated. First, Nusselt number for Re=2300 is calculated.

$$Nu_{2300,2} = 1.1615 \cdot \sqrt[3]{2300 \cdot Pr_{\text{tube}} \cdot \frac{D_i}{L}} \quad (5.49)$$

$$Nu_{2300,3} = \left(\frac{2}{1 + 22 \cdot Pr_{\text{tube}}} \right)^{1/6} \cdot \sqrt{2300 \cdot Pr_{\text{tube}} \cdot \frac{D_i}{L}} \quad (5.50)$$

$$Nu_{l,2300} = \sqrt[3]{Nu_{t,1}^3 + 0.7^3 + (Nu_{2300,2} - 0.7)^3 + Nu_{2300,3}^3} \quad (5.51)$$

Then Nusselt number for Re=10000 is calculated.

$$Nu_{\text{tur},10000} = \frac{(\xi/8) \cdot 10000 \cdot Pr_{\text{tube}}}{1 + 12.7 \cdot \sqrt{\xi/8} \cdot (Pr_{\text{tube}}^{1/4} - 1)} \cdot \left(1 + \frac{D_i}{L}\right)^{2/3} \quad (5.52)$$

$$Nu_{\text{tube}} = Nu_{l,\text{tube}} \text{ for } Re_{\text{tube}} \leq 2300 \quad (5.53)$$

If Reynolds number is between 2300 and 10000, then an interpolation factor for Nusselt number calculations is needed.

$$\gamma = \frac{Re_{\text{tube}} - 2300}{10000 - 2300} \quad (5.54)$$

for $2300 < Re_{\text{tube}} \leq 10000$

$$Nu_{\text{tube}} = (1 - \gamma) \cdot Nu_{l,2300} + \gamma \cdot Nu_{\text{tur},10000} \quad (5.55)$$

for $Re_{\text{tube}} > 10000$

$$Nu_{\text{tube}} = Nu_{\text{tur}} \quad (5.56)$$

$$h_{\text{tube}} = Nu_{\text{tube}} \cdot \frac{k_{\text{tube}}}{D_i} \quad (5.57)$$

For evaporation, Shah chart correlation will be used [42].

$$Re_{tube} = \frac{D_i \cdot (1 - x)}{\vartheta_{liq, phase}} \quad (5.58)$$

Liquid phase heat transfer coefficient is calculated as above and then will be used for calculating two phase heat transfer coefficient.

Convection number calculation:

$$Co = \left(\frac{1}{x} - 1\right)^{0.8} \cdot \sqrt{\frac{\rho_v}{\rho_l}} \quad (5.59)$$

Heat flux calculation:

$$q = \frac{q_{tp}}{\pi \cdot D_i \cdot L \cdot N_t \cdot N_r} \quad (5.60)$$

Boiling number calculation:

$$Bo = \frac{q}{V_{tp} \cdot (h_v - h_l)} \quad (5.61)$$

Froude number calculation:

$$Fr_l = \frac{V_{tp}^2}{\rho_l^2 \cdot g \cdot D_i} \quad (5.62)$$

Ratio of liquid phase heat transfer coefficient to two phase heat transfer coefficient is defined as below.

$$\Psi_{tp} = \frac{h_{tp}}{h_l} \quad (5.63)$$

After that, N coefficient is calculated.

$$N = Co \text{ for } Fr_l > 0.04 \quad (5.64)$$

$$N = 0.38. Fr_l^{-0.3} \text{ for } Fr_l \leq 0.04 \quad (5.65)$$

$$h_{cb} = 1.8. N^{-0.8} \quad (5.66)$$

$$h_{nb} = 230. \sqrt{Bo} \text{ for } Bo \geq 0.3. 10^{-4} \quad (5.67)$$

$$h_{nb} = 1 + 46. \sqrt{Bo} \text{ for } Bo < 0.3. 10^{-4} \quad (5.68)$$

$$h_{bs1} = 14.7. \sqrt{Bo}. \ln(2.74. N^{-0.1}) \text{ for } Bo \geq 11. 10^{-4} \quad (5.69)$$

$$h_{bs1} = 15.43. \sqrt{Bo}. \ln(2.74. N^{-0.1}) \text{ for } Bo < 11. 10^{-4} \quad (5.70)$$

$$h_{bs2} = 14.7. \sqrt{Bo}. \ln(2.47. N^{-0.15}) \text{ for } Bo \geq 11. 10^{-4} \quad (5.71)$$

$$h_{bs2} = 15.43. \sqrt{Bo}. \ln(2.47. N^{-0.15}) \text{ for } Bo < 11. 10^{-4} \quad (5.72)$$

$$\Psi_{tp} = \max(h_{cb}, h_{bs2}) \text{ for } N \leq 0.1 \quad (5.73)$$

$$\Psi_{tp} = \max(h_{cb}, h_{bs1}) \text{ for } 0.1 < N \leq 1 \quad (5.74)$$

$$\Psi_{tp} = \max(h_{cb}, h_{nb}) \text{ for } N > 0.1 \quad (5.75)$$

After these calculations, liquid phase heat transfer coefficient is calculated with Ψ_{tp} to determine two phase heat transfer coefficient.

$$h_{tp} = \Psi_{tp}. h_l \quad (5.76)$$

For condensation Shah correlation will be used [43].

$$PR = \frac{Pr_l}{Pr_c} \quad (5.77)$$

$$h_{tp} = h_l \cdot (1 - x)^{0.8} + \frac{3.8 \cdot x^{0.76} \cdot (1 - x)^{0.04}}{PR^{0.38}} \quad (5.78)$$

To find heat exchanger areas from the correlations above, LMTD (logarithmic mean temperature difference) must be calculated.

$$Q = U \cdot A \cdot \text{LMTD} \quad (5.79)$$

$$\text{LMTD} = \frac{\Delta T_i - \Delta T_e}{\ln \frac{T_i}{T_e}} \quad (5.80)$$

By using these two formula, the expression below could be derived.

$$A = \frac{Q}{U \cdot \text{LMTD}} \quad (5.81)$$

To correctly calculate LMTD value flow type of the heat exchangers must be selected. For this study counterflow heat exchangers were selected. Flow scheme could be seen below.

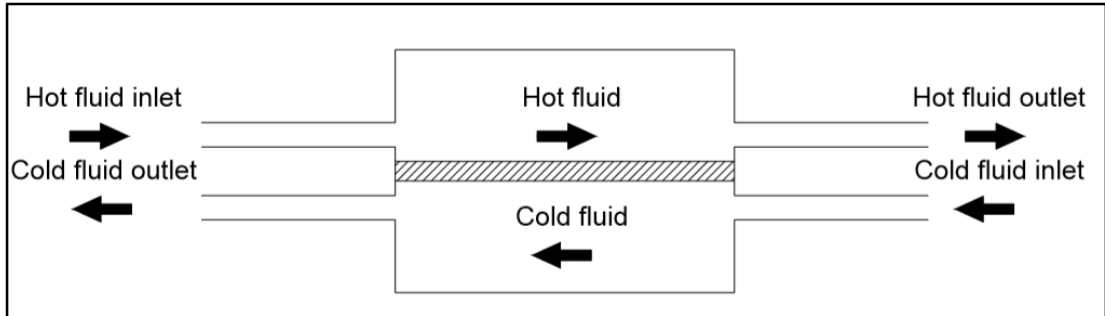


Figure 5.7: Counterflow heat exchanger flow scheme.

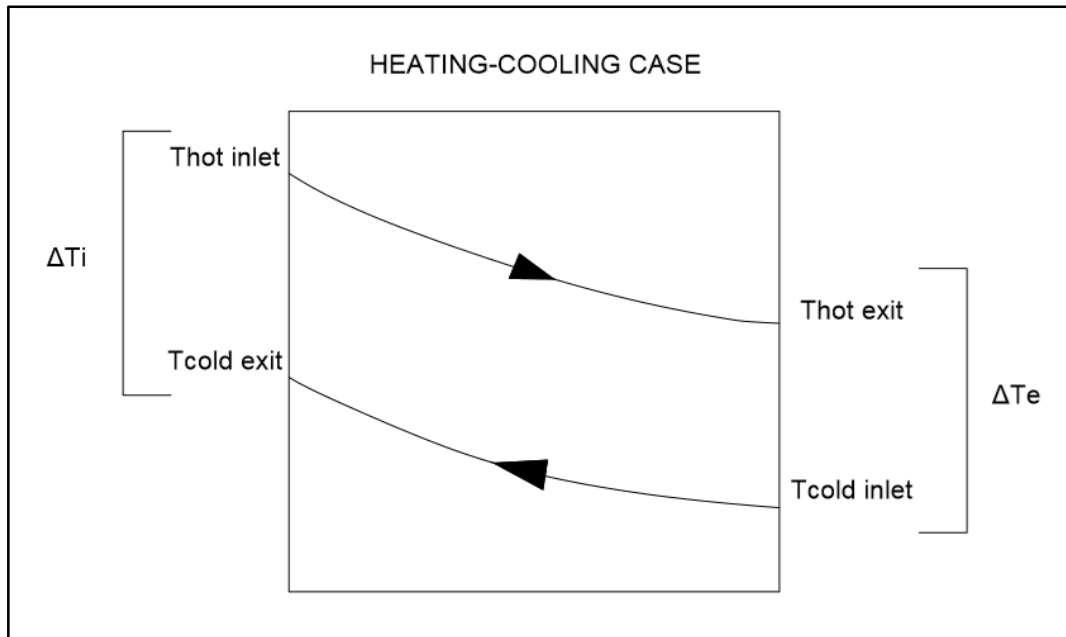


Figure 5.8: Counterflow heat exchangers heating&cooling case.

For the heating or cooling processes, scheme above was utilized as in the evaporator sensible liquid heat transfer section, evaporator superheating section, condenser desuperheating section and condenser subcooling section heat transfer section.

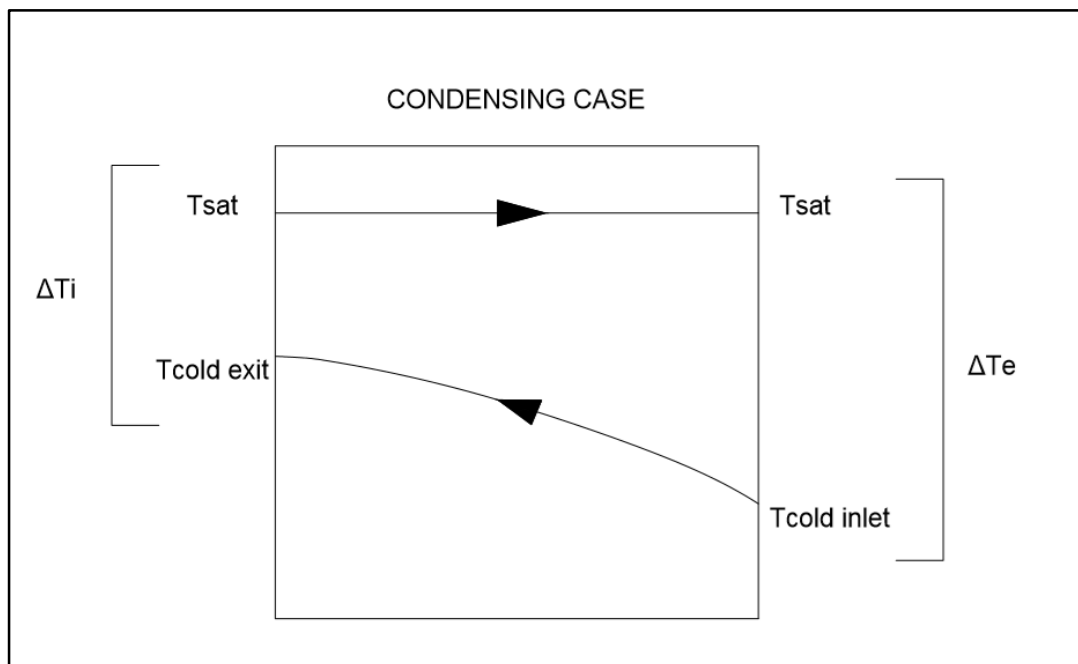


Figure 5.9: Counterflow heat exchangers condensing case.

For the condensation process, scheme above was utilized. As it could be seen from the scheme, as condensation occurs, temperature does not change and cooling media temperature increases.

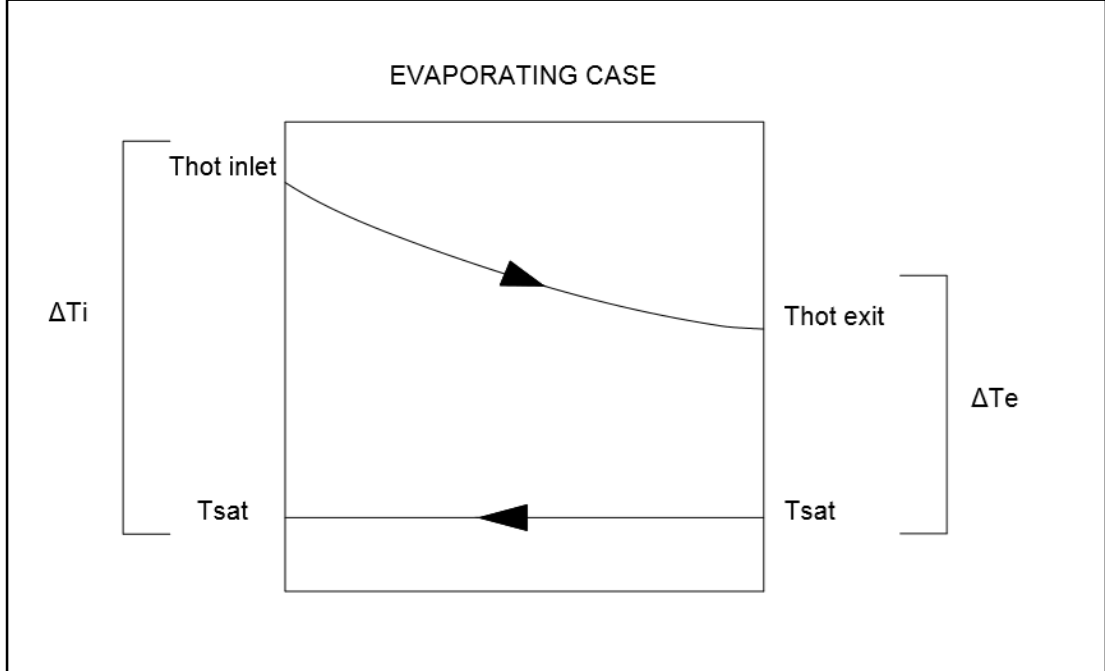


Figure 5.10: Counterflow heat exchangers evaporating case.

For the evaporation process, scheme above was utilized. As it could be seen from the scheme, as evaporation occurs, temperature does not change and heating media temperature decreases.

For regenerator; When both the fluids have equal capacity rates (i.e. $m_h.Cp_h = m_c.Cp_c$) in counter flow heat exchangers LMTD is expressed as below [44].

$$LMTD = T_{inlet} - T_{outlet} \quad (5.82)$$

5.4. System Cost Calculation Algorithm

After obtaining the heat transfer areas of all equipment economical influence of the cycle could be investigated.

For evaporator and condenser a realistic costing method was created to estimate equipment costs. It was based on a heat exchanger firm's costing method. It was developed for finned tube type heat exchangers.

For regenerator, pump and turbine, cost correlations were used found in the literature. Evaporator and condenser equipment cost method and pump, turbine and regenerator cost method are presented below.

- Evaporator and Condenser Equipment Cost Method

Finned tube type heat exchanger and parts are introduced below. Isometric view of a finned tube type heat exchanger is presented below.

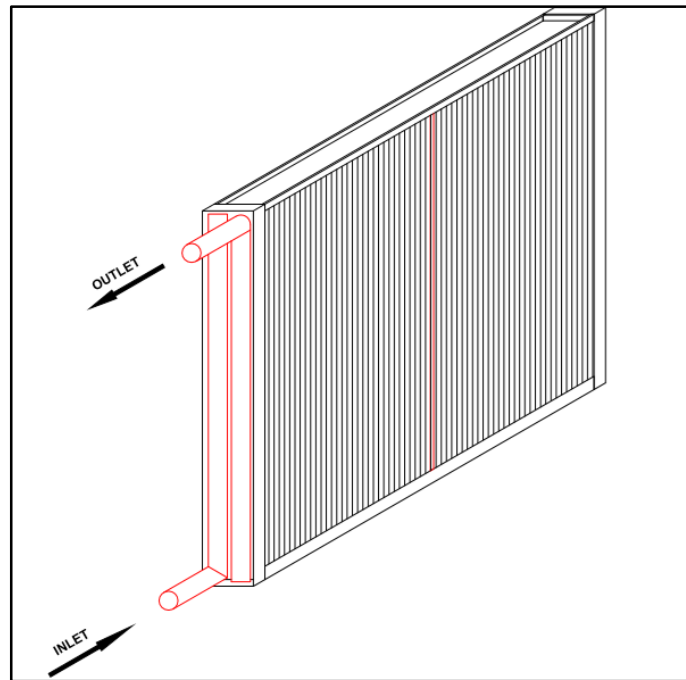


Figure 5.11: Finned tube type heat exchanger.

Finned tube type heat exchangers used in this study basically consists of front, rear plates and middle plates to support the fin and tube bundle, top and bottom lids, fins and tubes and headers.

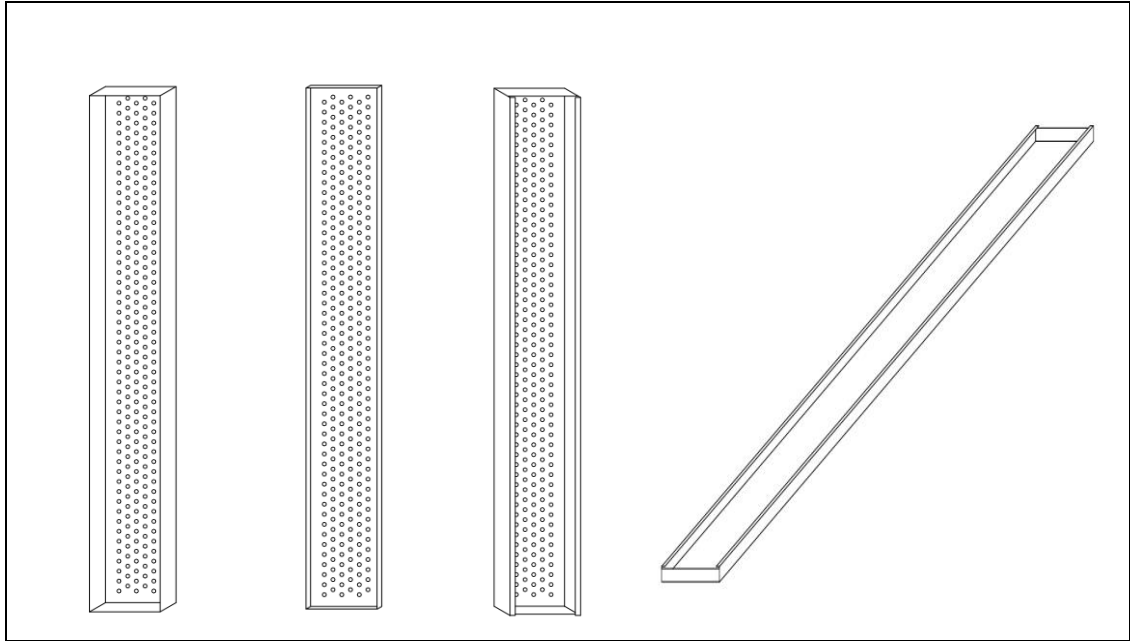


Figure 5.12: Front, middle, rear plate, top and bottom lids.

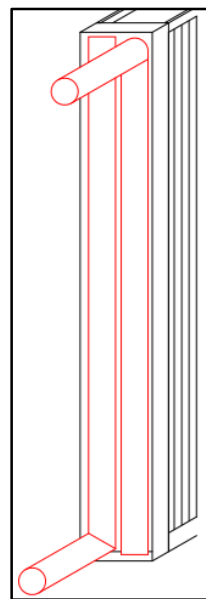


Figure 5.13: Inlet and outlet headers.

In practice, fin material selected as copper or aluminum because it is easy to form these materials and these materials have high heat conductivity. Pipes are usually selected as stainless steel or copper.

To find the cost of these type of heat exchangers a novel type of cost method utilized because in literature there are only correlations for shell and tube type heat exchangers exist.

Parameters for cost of evaporator and condenser are listed below.

- Number of tubes,
- Number of rows,
- Geometry (Longitudinal and transverse distances between tubes)
- Density of sheet metal parts, fins and pipes,
- Length of the coil,
- Lid height,
- Width of the coil,
- Cost of stainless steel, copper and aluminum (kg/\$)
- Fin pitch,
- Fin thickness,

Basic formulations for costing are listed below. Thickness of all parts are 2 mm. For front and rear plate volumes:

$$A_{f,r} = (B_{sl} \cdot N_r + 70) \cdot (B_{st} \cdot N_t + 70) + 2.60 \cdot (B_{st} \cdot N_t + 70) + 2.20 \cdot (B_{st} \cdot N_t + 70) \quad (5.83)$$

For middle plate volume:

$$A_{mid} = (B_{sl} \cdot N_r + 70) \cdot (B_{st} \cdot N_t + 4) + 2.40 \cdot (B_{st} \cdot N_t + 70) + 2.40 \cdot (B_{sl} \cdot N_r + 70) \quad (5.84)$$

Number of middle plates ($N_{m,p}$) depends on the coil length. To prevent fin block deflection it is advised to place middle plates on each 1500 mm. For lids volume:

$$A_{lid} = (B_{sl} \cdot N_r + 70) \cdot L + 2.35 \cdot L + 2.15 \cdot L + 2.30 \cdot (B_{sl} \cdot N_r + 70) \quad (5.85)$$

Number of lids are 2, top and bottom lids. Total volume is calculated as below.

$$A_{\text{total}} = (A_{f,r} + A_{f,r} + N_{m,p} \cdot A_{\text{mid}} + 2 \cdot A_{\text{lid}}) / 10^6 \quad (5.86)$$

$$C_{\text{sheet}} = R_w \cdot A_{\text{total}} \cdot \rho_{\text{sheet}} \cdot C_{\text{st}} \quad (5.87)$$

Total cost of fins is calculated as below.

$$C_{\text{fin}} = (R_w \cdot B_{\text{st}} \cdot N_t \cdot B_{\text{sl}} \cdot N_r \cdot \delta \cdot \frac{L}{DL} \cdot \rho_{\text{fin}} \cdot C_{\text{fin}}) / 10^6 \quad (5.88)$$

Total cost of pipes is calculated as below.

$$C_{\text{pipe}} = (R_w \cdot \frac{\pi \cdot (D_o^2 - D_i^2)}{4} \cdot L \cdot \rho_{\text{pipe}} \cdot N_t \cdot N_r \cdot C_{\text{pipe}}) / 10^6 \quad (5.89)$$

In addition to these costs, labor costs (C_{labor}), paint cost (C_{paint}) and header cost (C_{header}) must be added. Each firm's production method, labor costs, painting method or material selection could be different. So these costs at large depend on the factors aforementioned. So general total cost estimation of a finned pipe heat exchanger calculated as below.

$$C_{\text{tot}} = (C_{\text{sheet}} + C_{\text{fin}} + C_{\text{pipe}} + C_{\text{labor}} + C_{\text{paint}} + C_{\text{header}}) \cdot R_p \quad (5.90)$$

For regenerator, the correlation below utilized [23].

$$\log C_{\text{regen}}^0 = B_1 \cdot (K_1 + K_2 + K_3 \cdot \log(A_{\text{reg}})^2) \quad (5.91)$$

For turbine and pump the correlation below utilized [23].

$$C_{\text{pump}} = C_{B,\text{pump}} + \left(\frac{Q_{\text{pump}}}{Q_{B,\text{pump}}} \right)^M \cdot f_m \cdot f_p \cdot f_t \quad (5.92)$$

$$C_{\text{turb}} = C_{B,\text{turb}} + \left(\frac{Q_{\text{turb}}}{Q_{B,\text{turb}}} \right)^M \cdot f_m \cdot f_p \cdot f_t \quad (5.93)$$

The correction factors of f_m , f_p and f_t were interpolated for different pressure and temperature conditions in the developed MATLAB program.

Regenerator, turbine and pump costs above were calculated according to the January 2000 index of equipment and must be brought up-to-date by using cost indexes. January 2000 index of equipment is 435.8 and December 2019 [10]. index of equipment is 559.27 [12].

$$C_{\text{reference}} = C_{\text{origin}} \cdot \left(\frac{CEPCI_{\text{reference}}}{CEPCI_{\text{origin}}} \right) \quad (5.94)$$

For evaporator and condenser cost method is already up to date. So total cost of the system could be calculated as below. For regenerative ORC system:

$$C_{\text{tot}} = C_{\text{evap}} + C_{\text{cond}} + C_{\text{regen}} + C_{\text{pump}} + C_{\text{turb}} \quad (5.95)$$

For simple ORC system:

$$C_{\text{tot}} = C_{\text{evap}} + C_{\text{cond}} + C_{\text{pump}} + C_{\text{turb}} \quad (5.96)$$

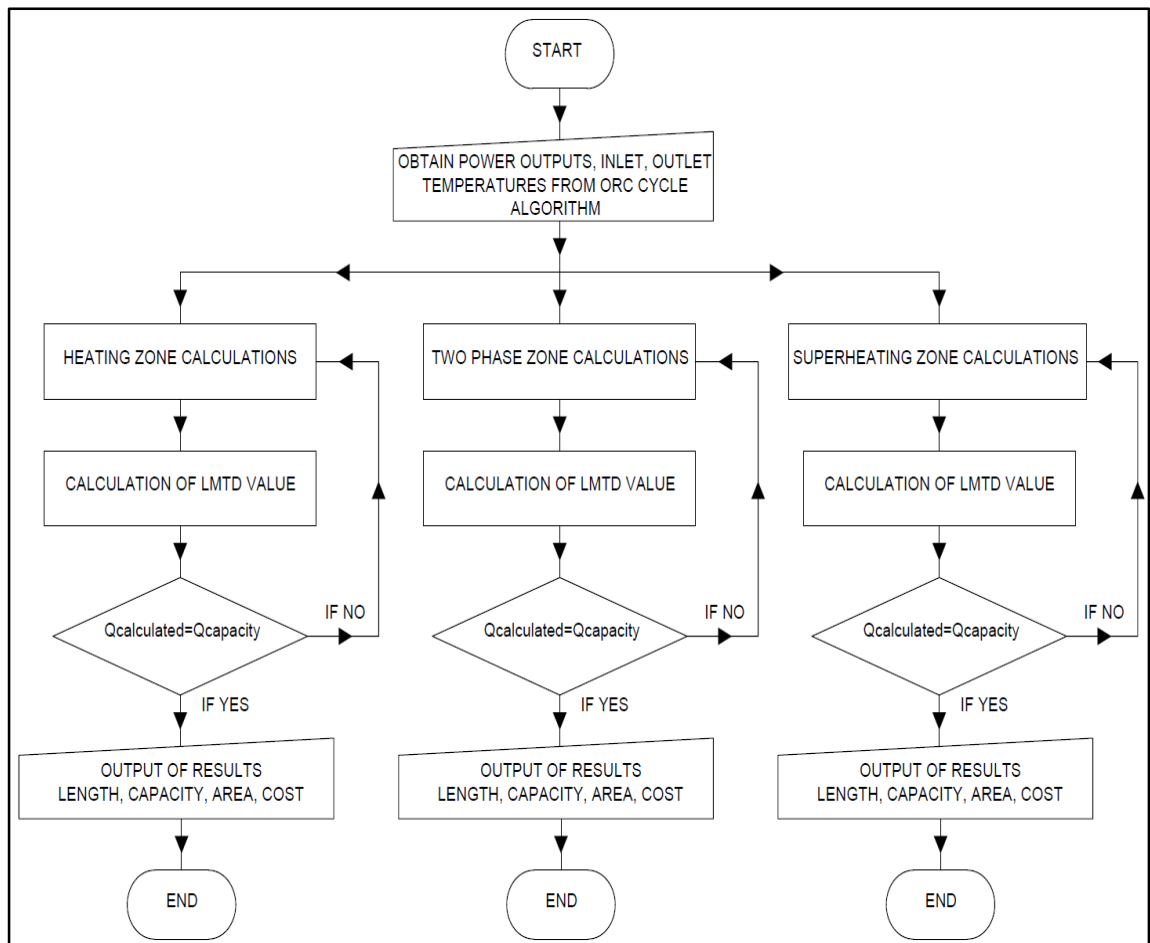


Figure 5.14: Evaporator modelling algorithm.

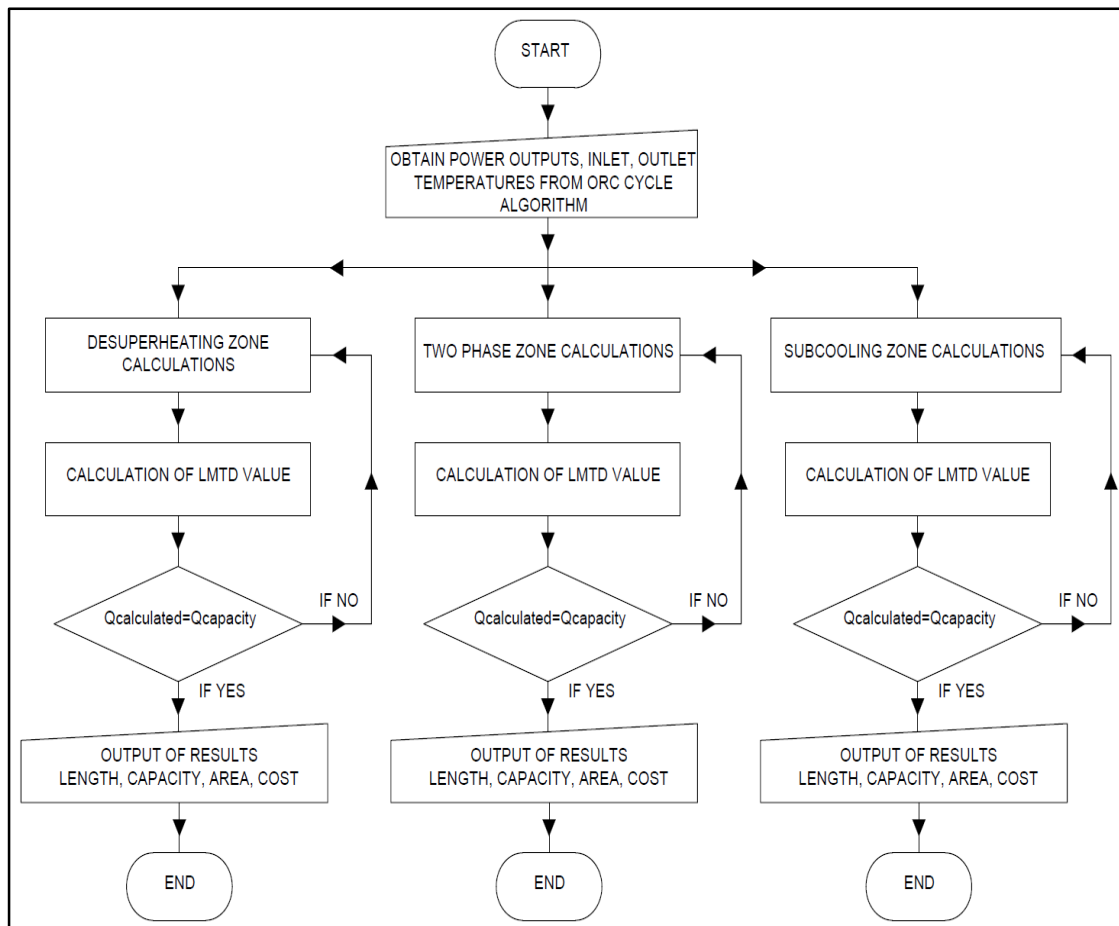


Figure 5.15: Condenser modelling algorithm.

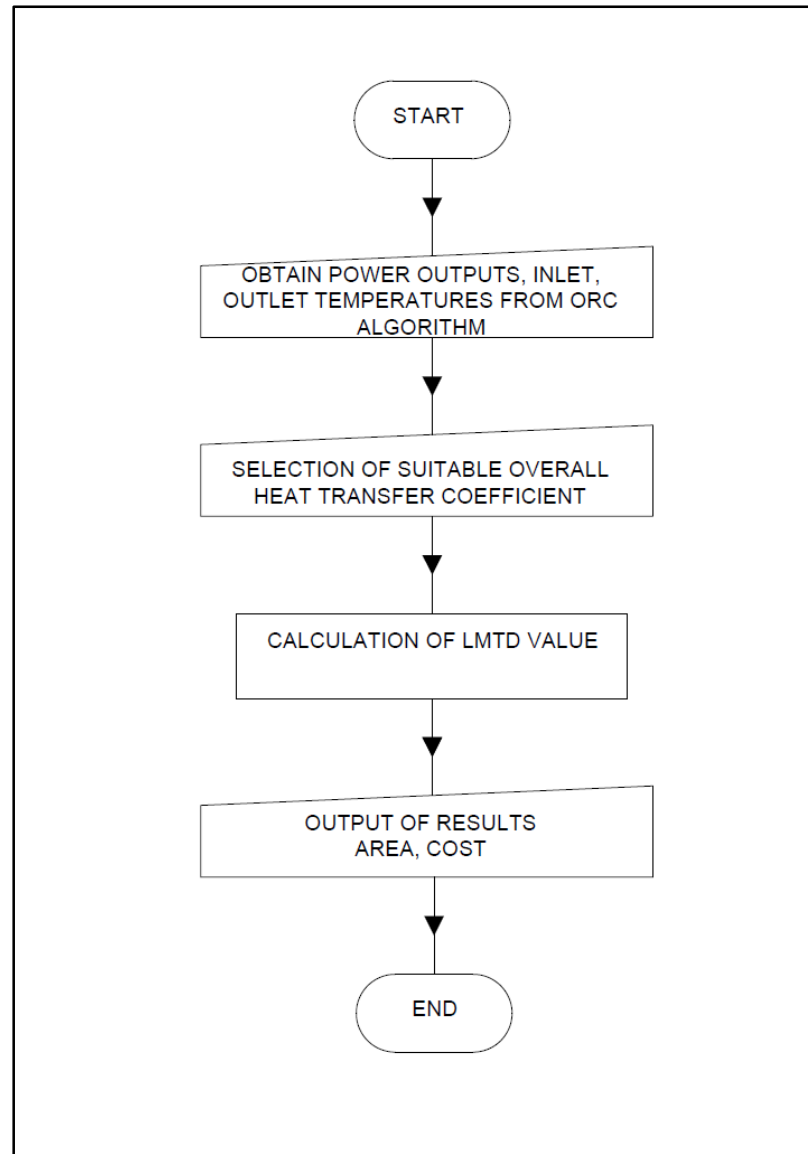


Figure 5.16: Regenerator modelling algorithm.

5.5. Case Study Scenarios

The case study scenarios studied in this thesis: Modelling and optimization studies of an ORC system which uses exhaust gases of an aluminum melting furnace as primary energy source. Detailed information about furnace is given below.

Furnace Type = Aluminum melting furnace

Air-Fuel Ratio of the burner= 12:1

$P_{\text{burner}} = 10$ [MW]

$P_{\text{turbine nominal}} = 100$ [kW]

$p_{\text{exhaust gas}} = 1$ [bar]

Working fluids=R123, R245fa, R601a

Unknowns are given below:

P_{pump} ,

$P_{\text{evaporator}}$,

$P_{\text{condenser}}$,

$P_{\text{regenerator}}$,

P_{turbine} ,

η_{cycle} ,

Natural gas lower heating value is taken as $H_u = 9,59$ [kWh]. As burner power is specified, natural gas flow rate could be found by dividing burner nominal power to natural gas lower specific heat.

$$m_{\text{ng}} = \frac{P_{\text{burner}}}{H_u} = \frac{10000}{9,59} = 1043 \text{ [Nm}^3\text{/h]} \quad (5.97)$$

Air-Fuel ratio is taken 12:1. So combustion air fan flow rate in room temperature could be found as:

$$m_{\text{cafr}} = 10423.12 = 12516 \text{ [Nm}^3\text{/h]} \quad (5.98)$$

Exhaust fan flow rate in room temperature equals to

$$m_{\text{efr}} = m_{\text{ng}} + m_{\text{cafr}} = 1043 + 12516 = 13559 \text{ [Nm}^3\text{/h]} \quad (5.99)$$

Exhaust gas temperature of aluminum melting furnace is taken as approximately 260 °C. The expansion caused by heating must be taken into account when calculating the exhaust flowrate. From the relationship below V_2 could be found.

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \quad (5.100)$$

$$\frac{13559}{25 + 273} = \frac{V_2}{260 + 273} \quad (5.101)$$

$V_2 = 24252 \text{ m}^3/\text{h}$ found.

Exhaust flow rate is taken as:

$$V_2 = 2,5 \cdot 10^4 \text{ m}^3/\text{h} \quad (5.102)$$

Exhaust gas will be modelled as air [39]. All assumptions made in the algorithm are given below.

$$p_{\text{condensation}} = 2 \text{ [bar]} \quad (5.103)$$

$$W_{\text{turbine_nominal}} = 100 \text{ [kW]} \quad (5.104)$$

Working fluids are R123, R245fa or R601a

$$\eta_{\text{pump}} = 0,75 \quad (5.105)$$

$$\eta_{\text{turbine}} = 0,8 \quad (5.106)$$

$$\Delta T_{\text{ConSH}} = 10 \quad (5.107)$$

$$\Delta T_{\text{ConSC}} = 10 \quad (5.108)$$

$$\Delta T_{\text{SHEVAP}} = T_{\text{critical}} - T_{\text{evaporation}} - 1 \quad (5.109)$$

In evaporator and condenser vapor quality is taken as:

$$x = 0.5 \quad (5.110)$$

6. RESULTS AND DISCUSSIONS

In this study, a computer program was created to predict heat exchanger sizing, each and total equipment costs for heat exchangers, turbine and pump could be utilized in an ORC system. There are total 6 case study type that were defined. 3 different organic fluids were used and for each fluid, 2 cycle types were defined as rORC (regenerative Organic Rankine Cycle) and sORC (simple Organic Rankine cycle). In each case there are 3 evaporation pressures were defined. So totally there are 18 cases were studied.

The graphical representations of these cases that constitute from the temperature and entropy data were given below as T-S diagrams, were created in MATLAB interface as below. Lower pressure is 2 bar. Higher pressures are 20 bar, 25 bar and 30 bar. All cases yields similar plots so, general form of the T-S diagrams for both regenerative and simple cases were given below.

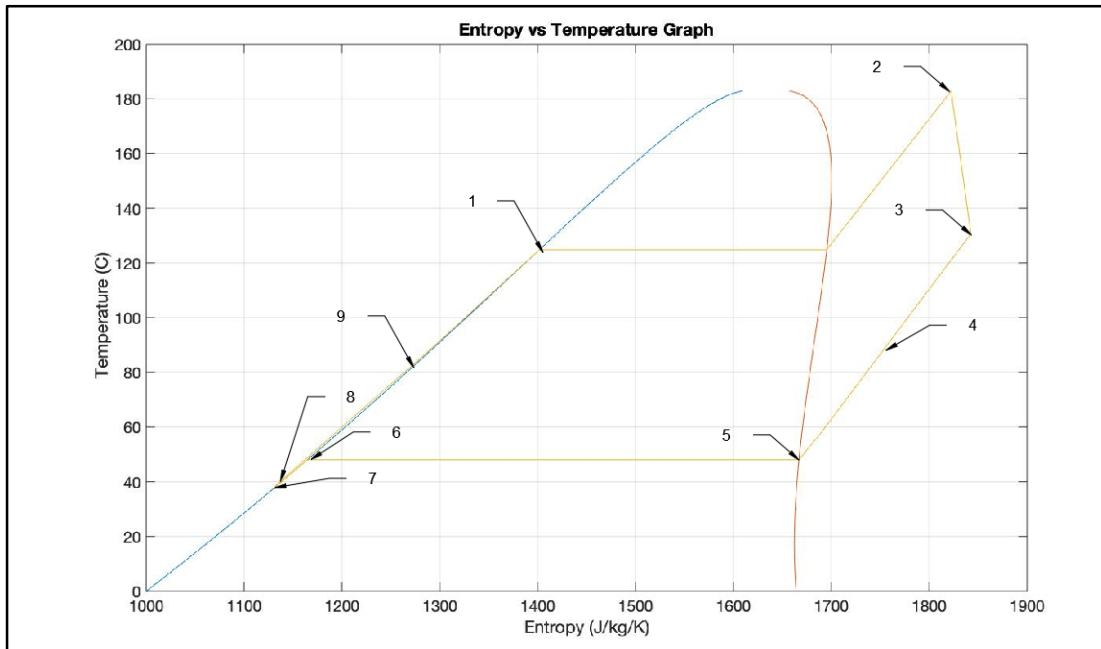


Figure 6.1: T-S diagram of rORC cases.

- 1st point is evaporation temperature.
- 2nd point is turbine inlet temperature.
- 3rd point is turbine outlet temperature.
- 4th point is regenerator inlet temperature.
- 5th point is condensation temperature.
- 6th point is condensation temperature.
- 7th point is subcooling temperature.
- 8th point is pump outlet temperature
- 9th point is evaporator inlet temperature.

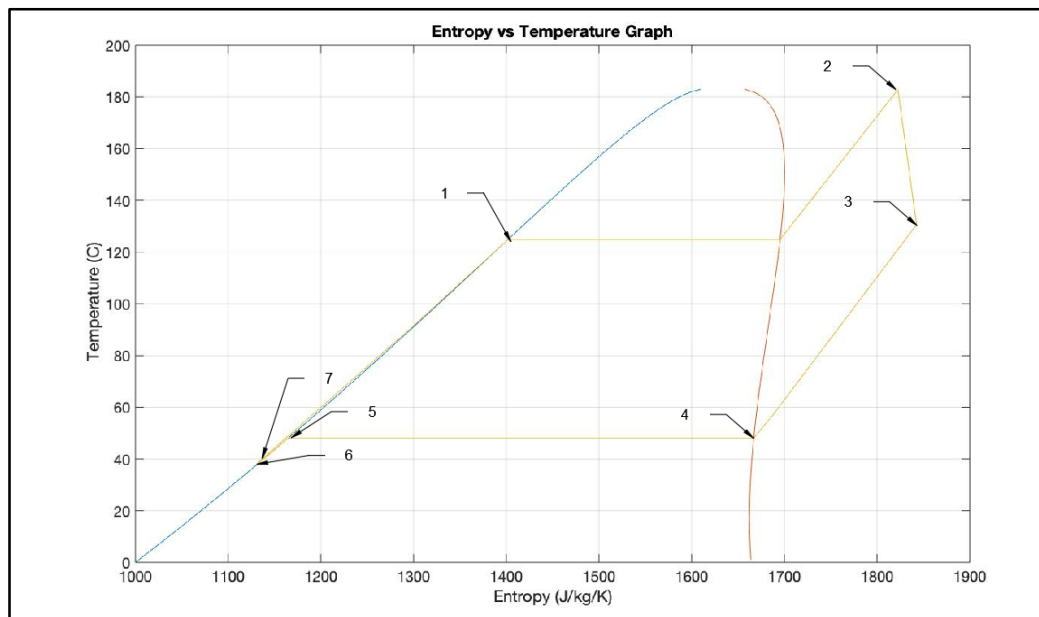


Figure 6.2: T-S diagram of sORC cases.

- 1st point is evaporation temperature.
- 2nd point is turbine inlet temperature.
- 3rd point is turbine outlet temperature.
- 4th point is condensation temperature.
- 5th point is condensation temperature.
- 6th point is subcooling temperature.
- 7th point is pump outlet temperature

6.1. Case 1: rORC with R123

In this case, R123 fluid with regenerative ORC cases were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.1: Temperature and efficiencies in case 1 [$^{\circ}\text{C}$].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R123-20bar-rORC	147.3	182.3	114	58.05	48.05	38.05	39.15	79.58	12.22
R123-25bar-rORC	160.2	182.2	102.7	58.05	48.05	38.05	39.45	71.76	12.66
R123-30bar-rORC	171.3	182.3	90.04	58.05	48.05	38.05	39.75	62.92	12.55

The effect of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 25 bar.

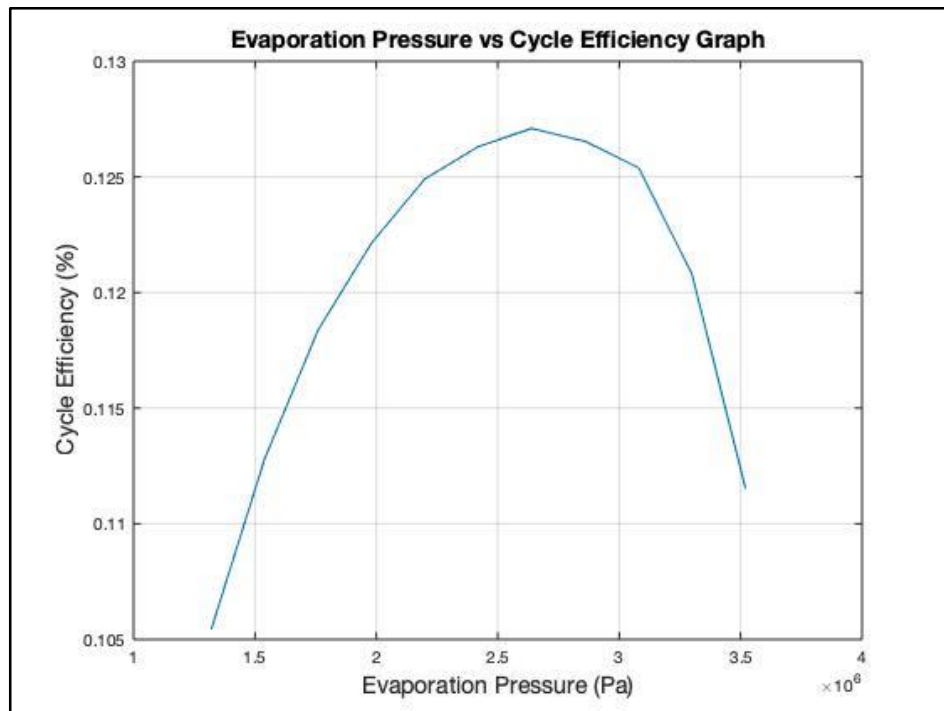


Figure 6.3: R123 fluid with rORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

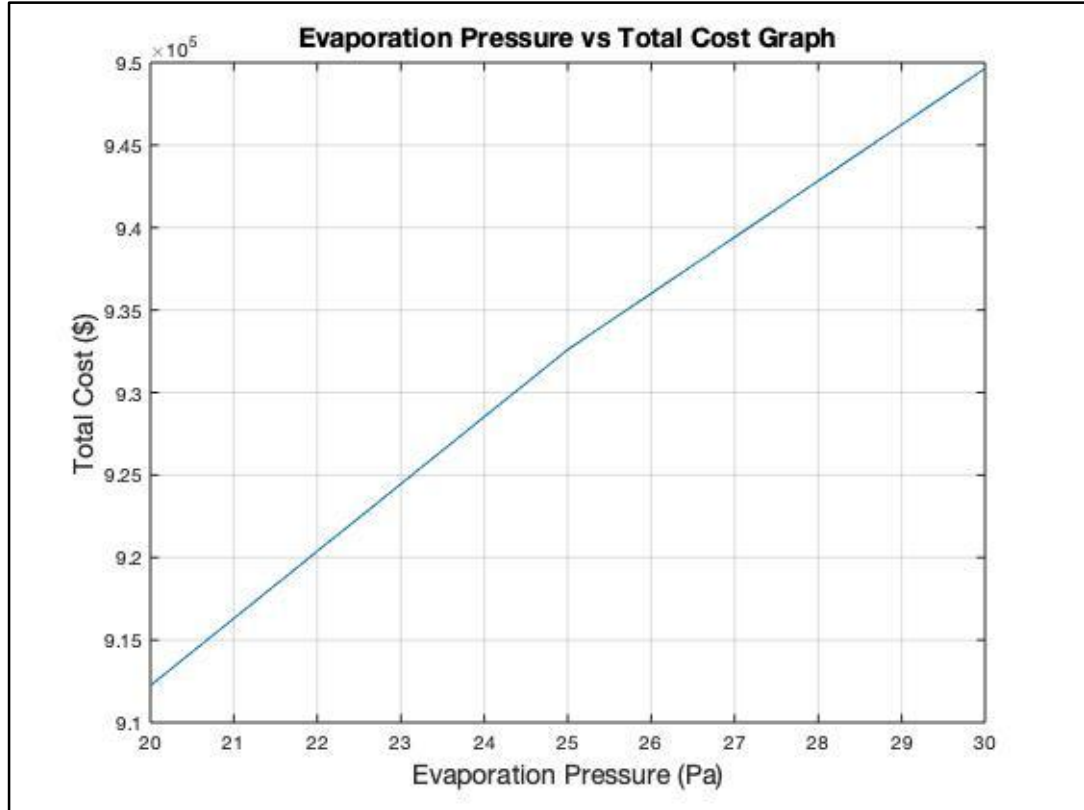


Figure 6.4: R123 fluid with rORC cases.

6.2. Case 2: rORC with R245fa

In this case, R245fa fluid with regenerative ORC cases were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.2: Temperature and efficiencies in case 2 [°C].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R245fa-20bar-rORC	121.9	152.9	94.94	43.46	33.46	23.46	24.41	61.34	11.18
R245fa-25bar-rORC	133.4	153.4	85.29	43.46	33.46	23.46	24.67	54.73	11.66
R245fa-30bar-rORC	143.2	153.2	73.21	43.46	33.46	23.46	24.93	46.33	11.54

The effects of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 25 bar.

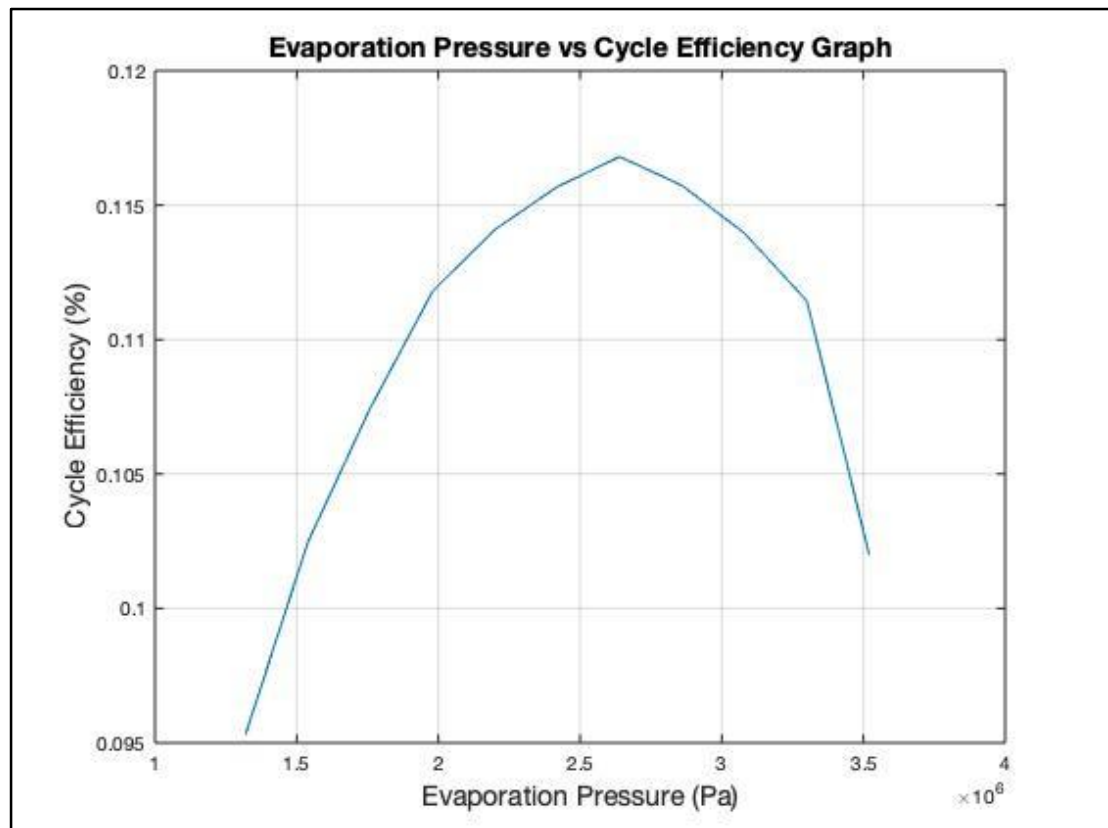


Figure 6.5: R245fa fluid with rORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

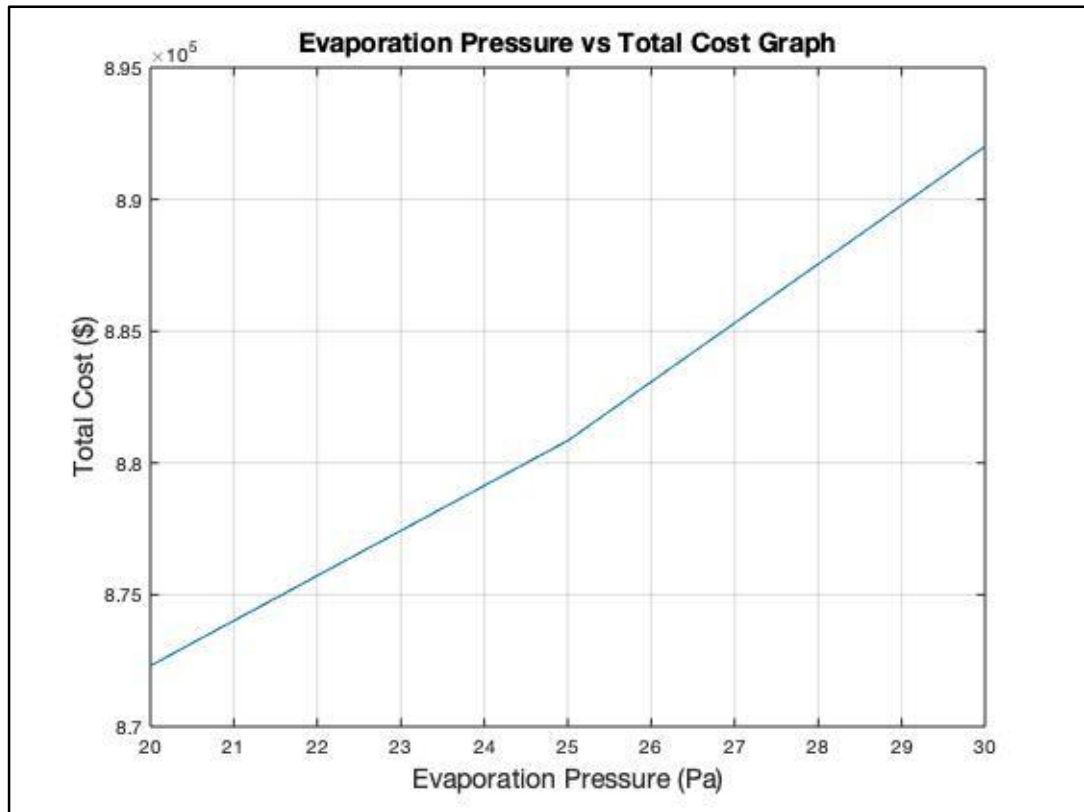


Figure 6.6: R245fa fluid with rORC cases.

6.3. Case 3: rORC with R601a

In this case, R601a fluid with regenerative ORC cases were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.3: Temperature and efficiencies in case 3 [°C].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R601a-20bar-rORC	154.1	186.1	132.4	59.08	49.08	39.08	40.02	99.14	13.04
R601a-25bar-rORC	167.8	185.8	122.2	59.08	49.08	39.08	40.51	91.28	13.51
R601a-30bar-rORC	179.5	186.5	110.5	59.08	49.08	39.08	40.82	82.2	13.40

The effects of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 25 bar.

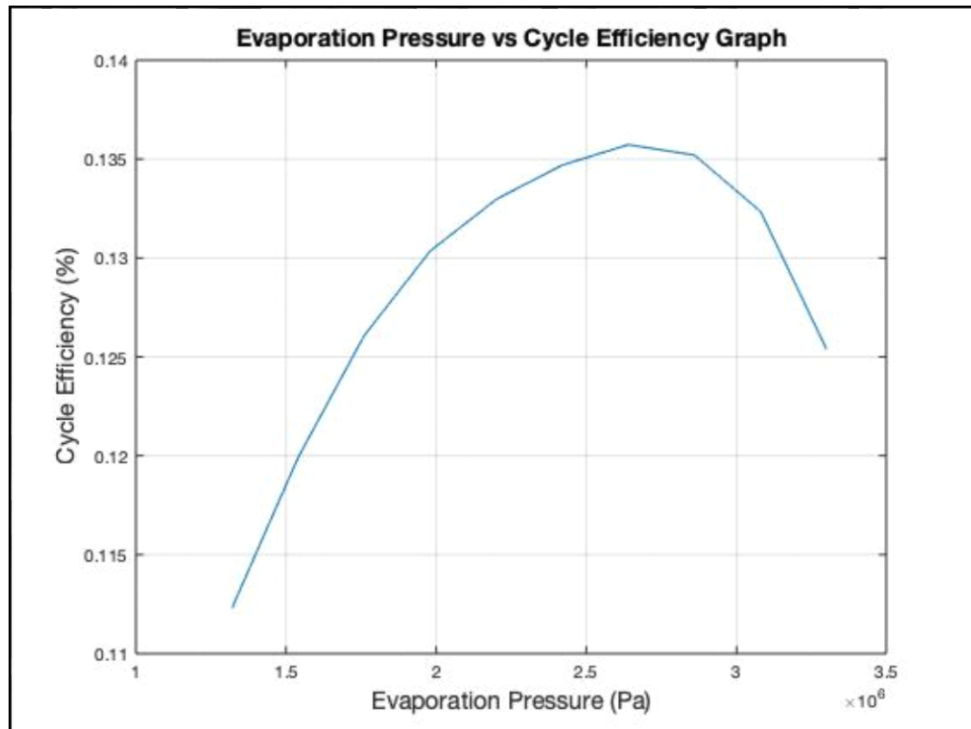


Figure 6.7: R601a fluid with rORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

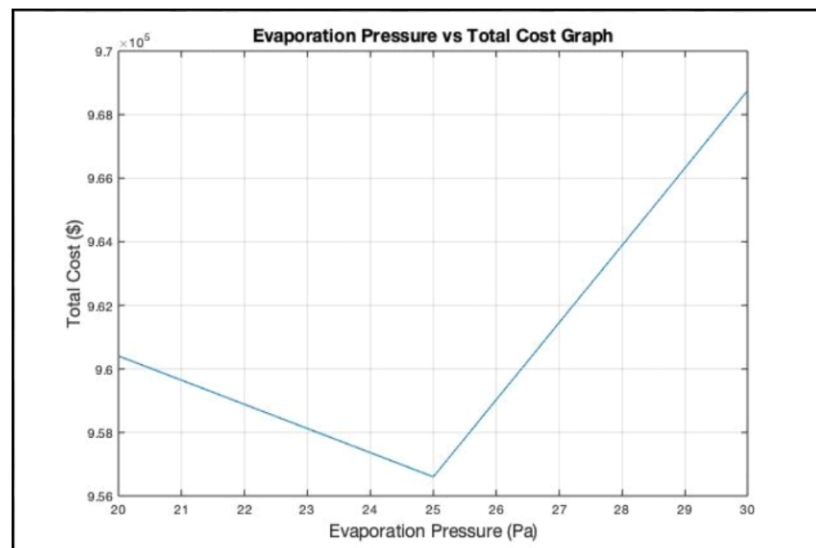


Figure 6.8: R601a fluid with rORC cases.

6.4. Case 4: sORC with R123

In this case, R123 fluid with simple ORC cases were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.4: Temperature and efficiencies in case 4 [$^{\circ}\text{C}$].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R123-20bar-sORC	147.3	182.3	114	-	48.05	38.05	39.15	-	9.35
R123-25bar-sORC	160.2	182.2	102.7	-	48.05	38.05	39.45	-	10.25
R123-30bar-sORC	171.3	182.3	90.04	-	48.05	38.05	39.75	-	10.78

The effects of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 33 bar.

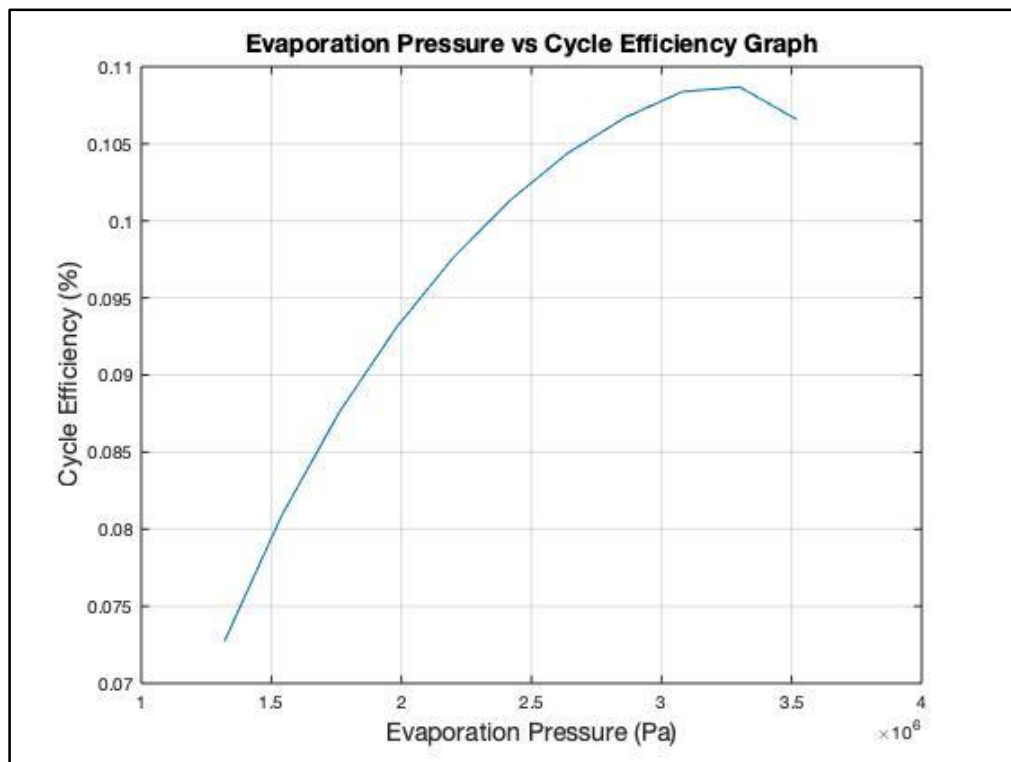


Figure 6.9: R123 fluid with sORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

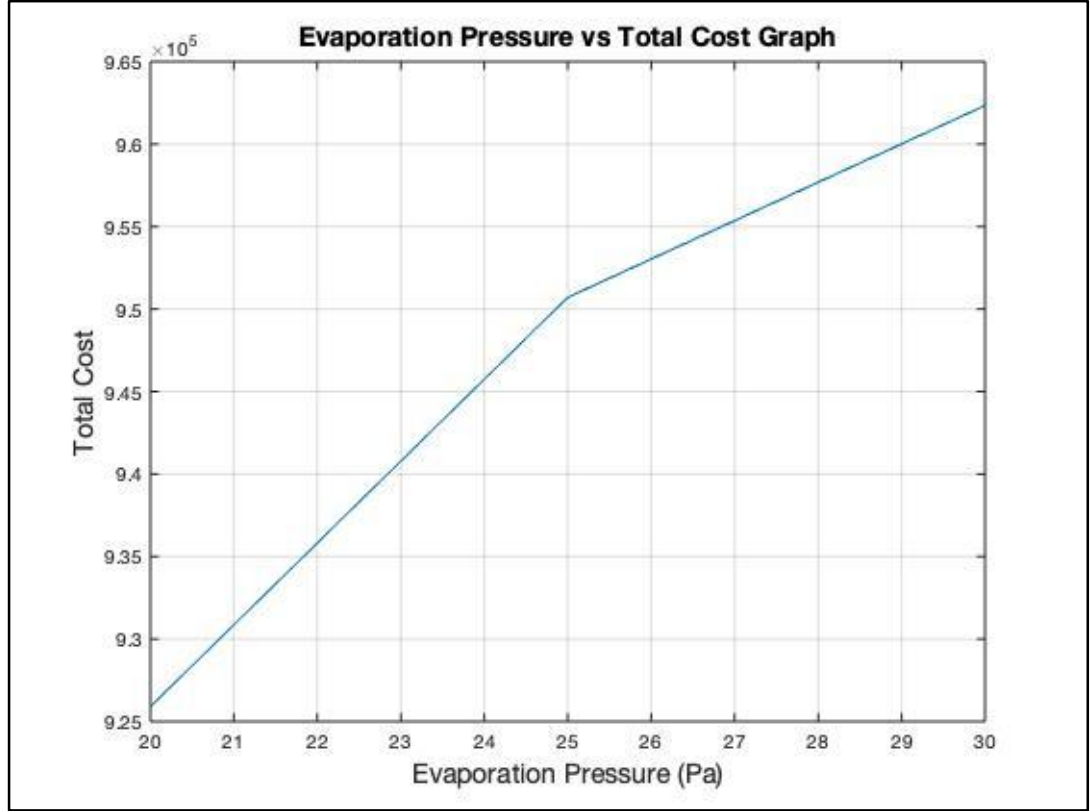


Figure 6.10: R123 fluid with sORC cases.

6.5. Case 5: sORC with R245fa

In this case, R245fa fluid with simple ORC cases were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.5: Temperature and efficiencies in case 5 [$^{\circ}\text{C}$].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R245fa-20bar-sORC	121.9	152.9	94.94	-	33.46	23.46	24.41	-	8.45
R245fa-25bar-sORC	133.4	153.4	85.29	-	33.46	23.46	24.67	-	9.33
R245fa-30bar-sORC	143.2	153.2	71.21	-	33.46	23.46	24.93	-	9.84

The effect of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 33 bar.

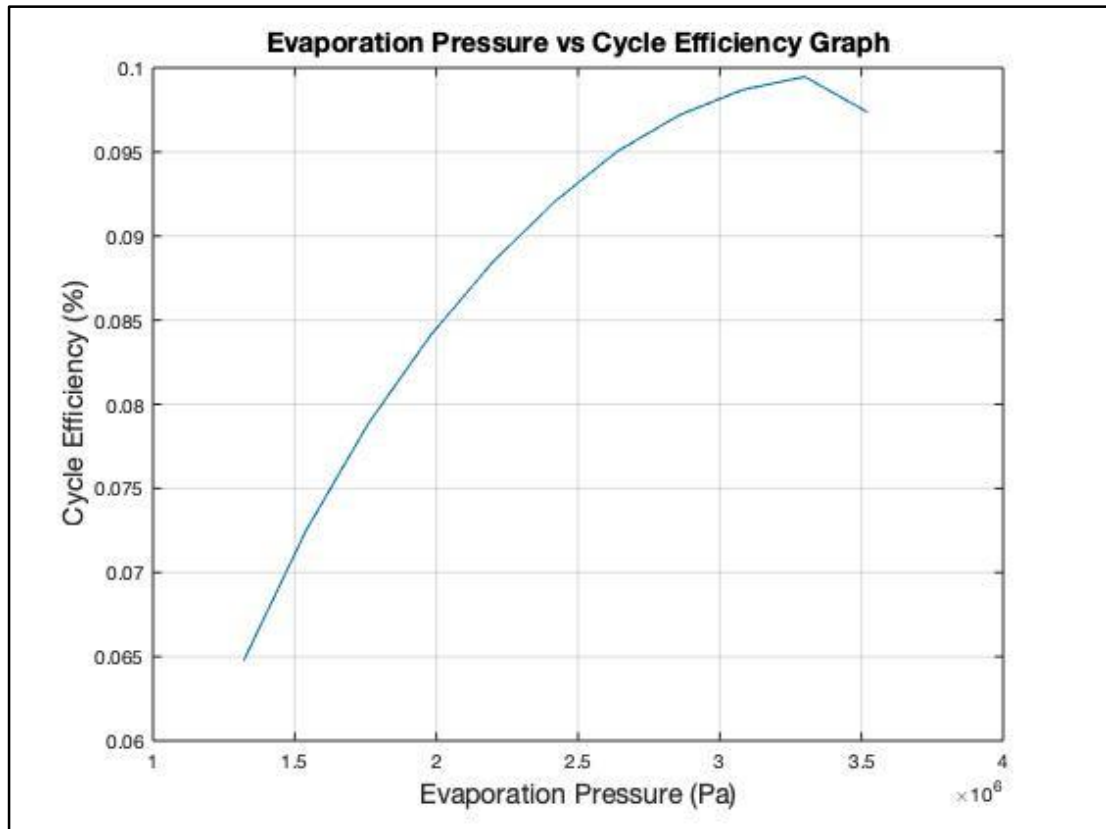


Figure 6.11: R245fa fluid with sORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

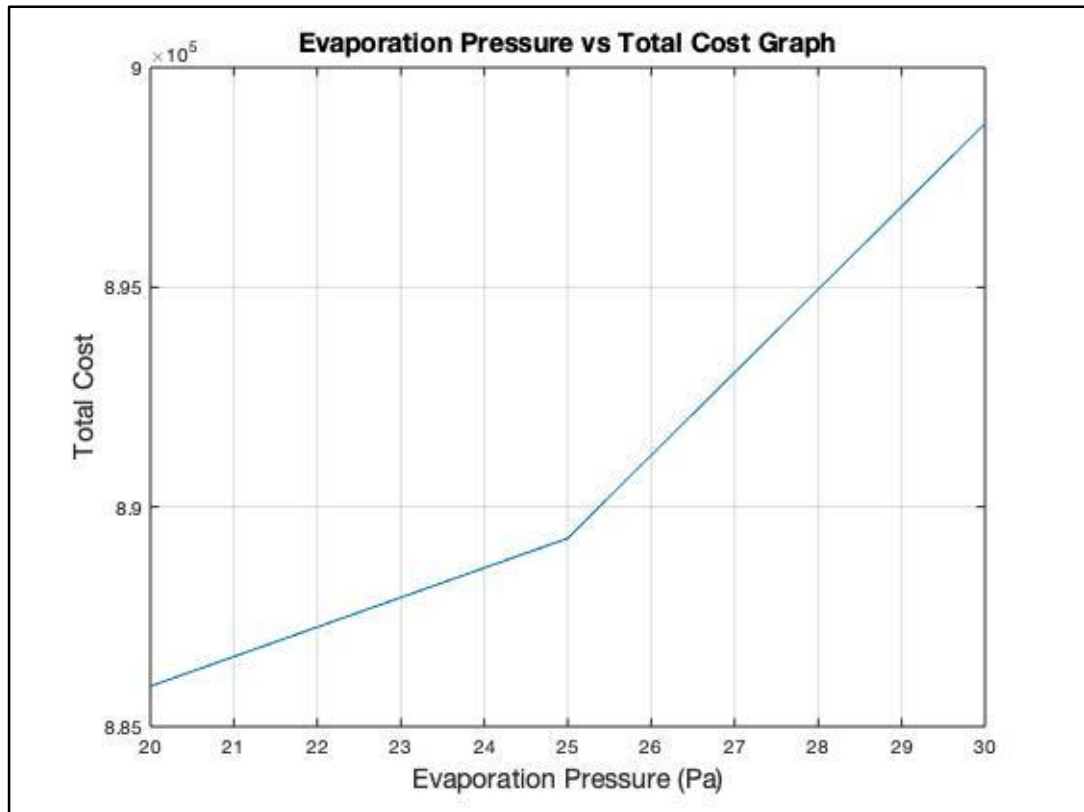


Figure 6.12: R245fa fluid with sORC cases.

6.6. Case 6: sORC with R601a

In this case, simple ORC cases with R601a fluid were studied. Evaporation pressures of 20, 25 and 30 bar and condensation pressure of 2 bar were used. In the table below, temperature and efficiencies were given.

Table 6.6: Temperature and efficiencies in case 6 [°C].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R601a-20bar-sORC	154.1	186.1	132.4	-	49.08	39.08	40.02	-	8.53
R601a-25bar-sORC	167.8	185.8	122.2	-	49.08	39.08	40.51	-	9.43
R601a-30bar-sORC	179.5	186.5	110.5	-	49.08	39.08	40.82	-	9.99

The effects of changing evaporation pressure from 11 bar to 35 bar for cycle efficiencies in this case were given below. The maximum efficiency is obtained around 30 bar.

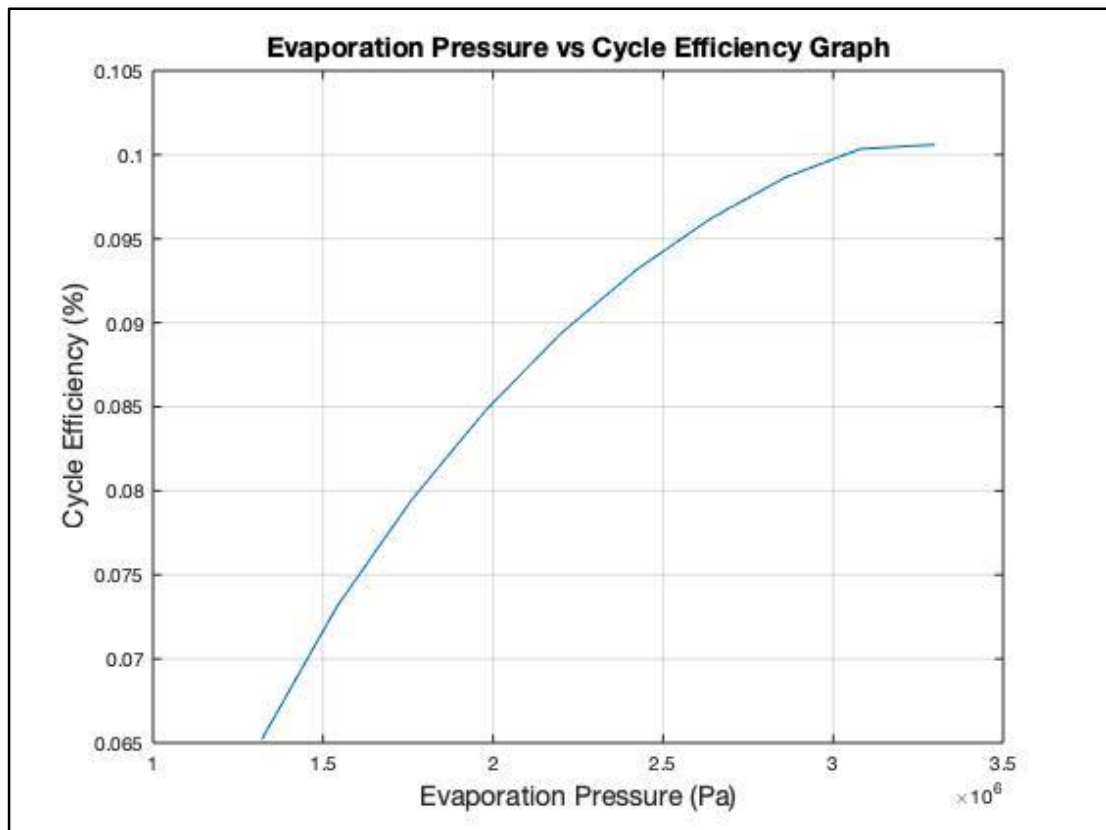


Figure 6.13: R601a fluid with sORC cases.

The effects of changing evaporating pressures from 20 bar to 25 and 30 bar to total costs were given below.

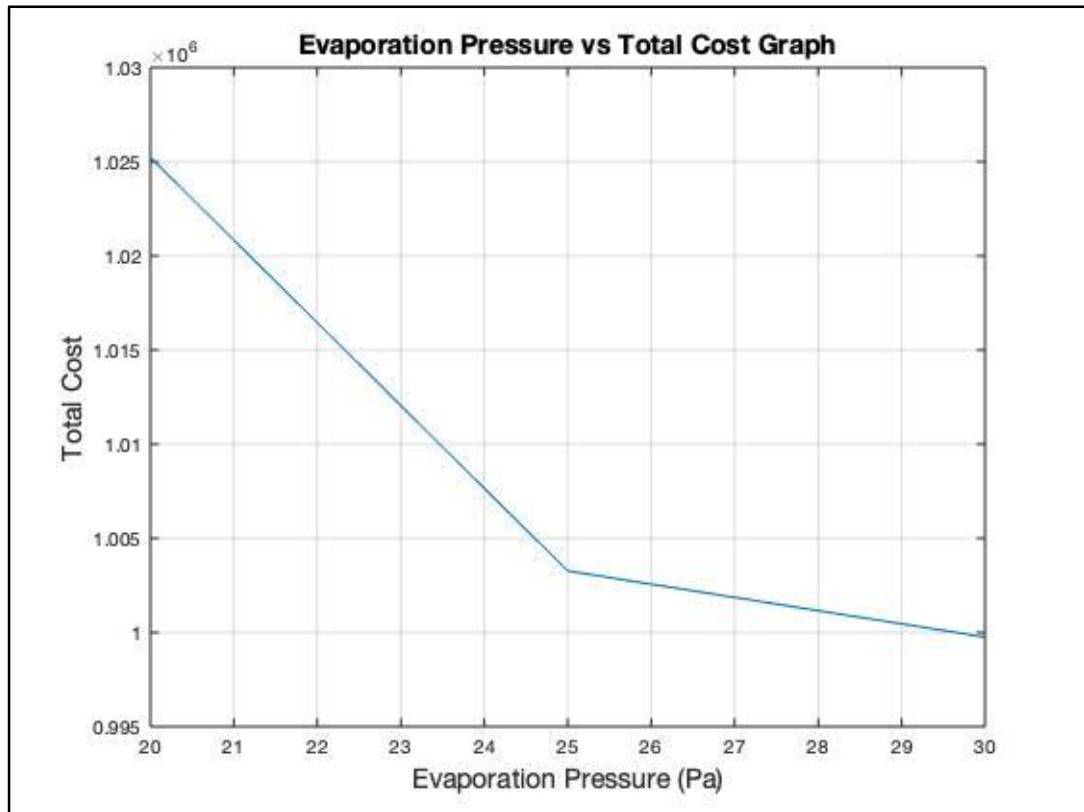


Figure 6.14: R601a fluid with sORC cases.

6.7. General Results of the Cases

For ease and comparison, all temperature and efficiency results were given below.

Table 6.7: Temperature and efficiencies in all cases [°C].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R123-20bar-rORC	147.3	182.3	114	58.05	48.05	38.05	39.15	79.58	12.22
R123-20bar-sORC	147.3	182.3	114	-	48.05	38.05	39.15	-	9.35
R123-25bar-rORC	160.2	182.2	102.7	58.05	48.05	38.05	39.45	71.76	12.66
R123-25bar-sORC	160.2	182.2	102.7	-	48.05	38.05	39.45	-	10.25
R123-30bar-rORC	171.3	182.3	90.04	58.05	48.05	38.05	39.75	62.92	12.55
R123-30bar-sORC	171.3	182.3	90.04	-	48.05	38.05	39.75	-	10.78
R245fa-20bar-rORC	121.9	152.9	94.94	43.46	33.46	23.46	24.41	61.34	11.18
R245fa-20bar-sORC	121.9	152.9	94.94	-	33.46	23.46	24.41	-	8.45
R245fa-25bar-rORC	133.4	153.4	85.29	43.46	33.46	23.46	24.67	54.73	11.66

Table 6.7 Cont.: Temperature and efficiencies in all cases [°C].

Case	Tevap	Tti	Tto	Trego	Tcon	Tpi	Tregi	Trego	η (%)
R245fa-25bar-sORC	133.4	153.4	85.29	-	33.46	23.46	24.67	-	9.33
R245fa-30bar-rORC	143.2	153.2	73.21	43.46	33.46	23.46	24.93	46.33	11.54
R245fa-30bar-sORC	143.2	153.2	71.21	-	33.46	23.46	24.93	-	9.84
R601a-20bar-rORC	154.1	186.1	132.4	59.08	49.08	39.08	40.02	99.14	13.04
R601a-20bar-sORC	154.1	186.1	132.4	-	49.08	39.08	40.02	-	8.53
R601a-25bar-rORC	167.8	185.8	122.2	59.08	49.08	39.08	40.51	91.28	13.51
R601a-25bar-sORC	167.8	185.8	122.2	-	49.08	39.08	40.51	-	9.43
R601a-30bar-rORC	179.5	186.5	110.5	59.08	49.08	39.08	40.82	82.2	13.40
R601a-30bar-sORC	179.5	186.5	110.5	-	49.08	39.08	40.82	-	9.99

Also for each case for evaporator and condenser heat transfer areas were given below for each zone.

Table 6.8: Evaporator liquid phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	266	356	431	404	463	504
R245fa	179	239	339	293	331	415
R601a	296	366	444	554	549	578

Table 6.9: Evaporator two phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	164	139	108	164	139	108
R245fa	112	89	70	112	89	70
R601a	162	119	79	162	119	79

Table 6.10: Evaporator vapor phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	34	25	13	34	25	13
R245fa	23	16	11	23	16	11
R601a	45	24	10	45	24	10

Table 6.11: Condenser liquid phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	27	32	32	27	32	32
R245fa	197	190	191	197	190	191
R601a	63	65	65	63	65	65

Table 6.12: Condenser two phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	2142	2259	2338	2142	2259	2338
R245fa	1675	1680	1675	1675	1680	1675
R601a	2600	2490	2516	2600	2490	2516

Table 6.13: Condenser vapor phase area [m²].

Working Fluid	rORC cases			sORC cases		
	20 bar	25 bar	30 bar	20 bar	25 bar	30 bar
R123	99	105	115	339	312	298
R245fa	83	102	96	235	212	138
R601a	120	99	100	749	583	444

The evaporation pressure to system working fluid flow rate behaviour was also plotted in MATLAB interface as below. The working fluid flow rate change, reaches a minimum with the change of evaporation pressure and then increases again as the opposite of the cycle efficiency plots.

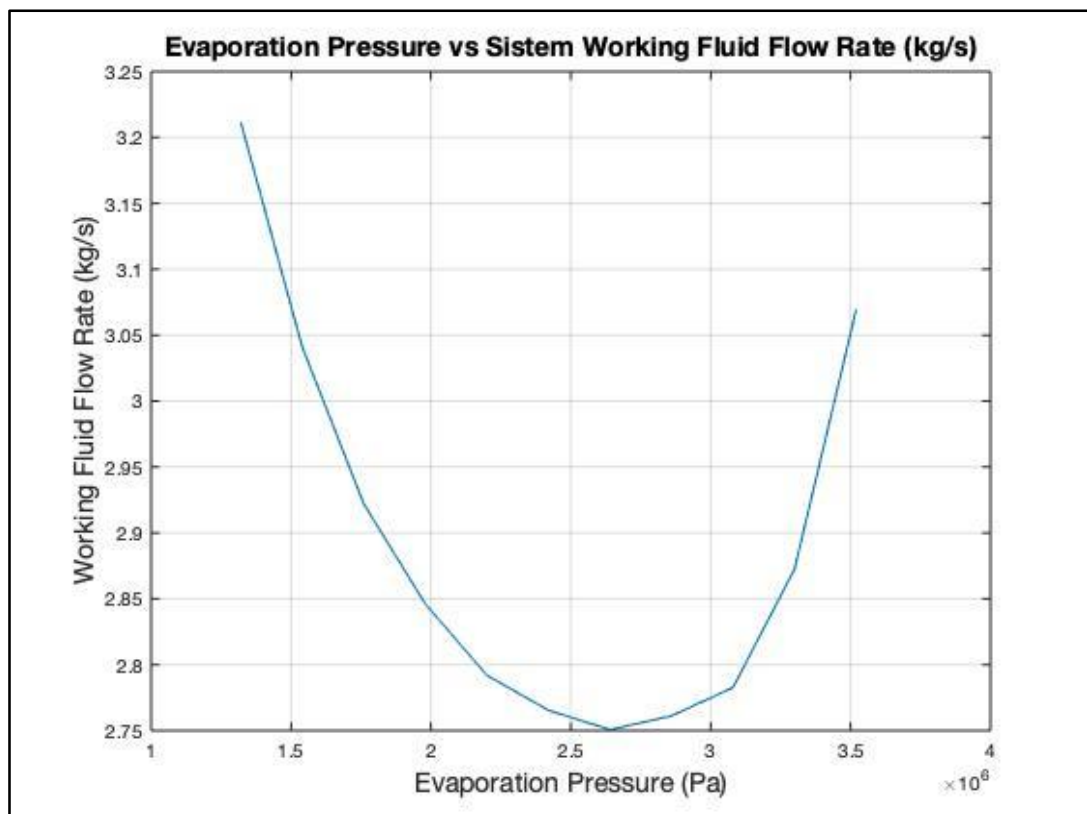


Figure 6.15: R123 Evaporation Pressure vs working fluid flow rate [kg/s].

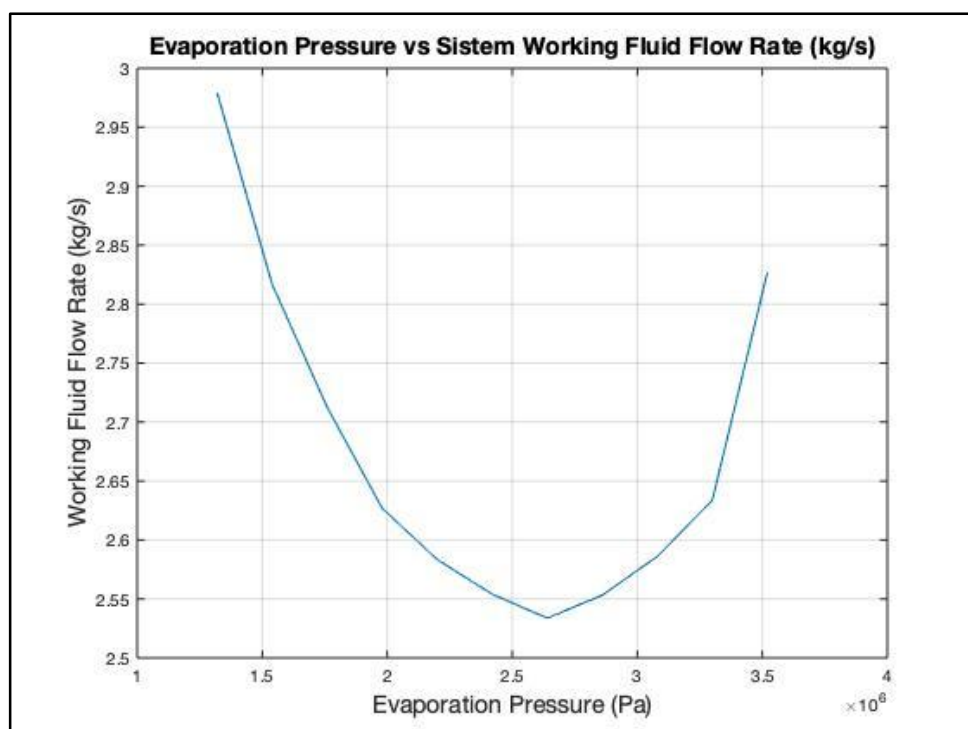


Figure 6.16: R245fa Evaporation Pressure vs working fluid flow rate [kg/s].

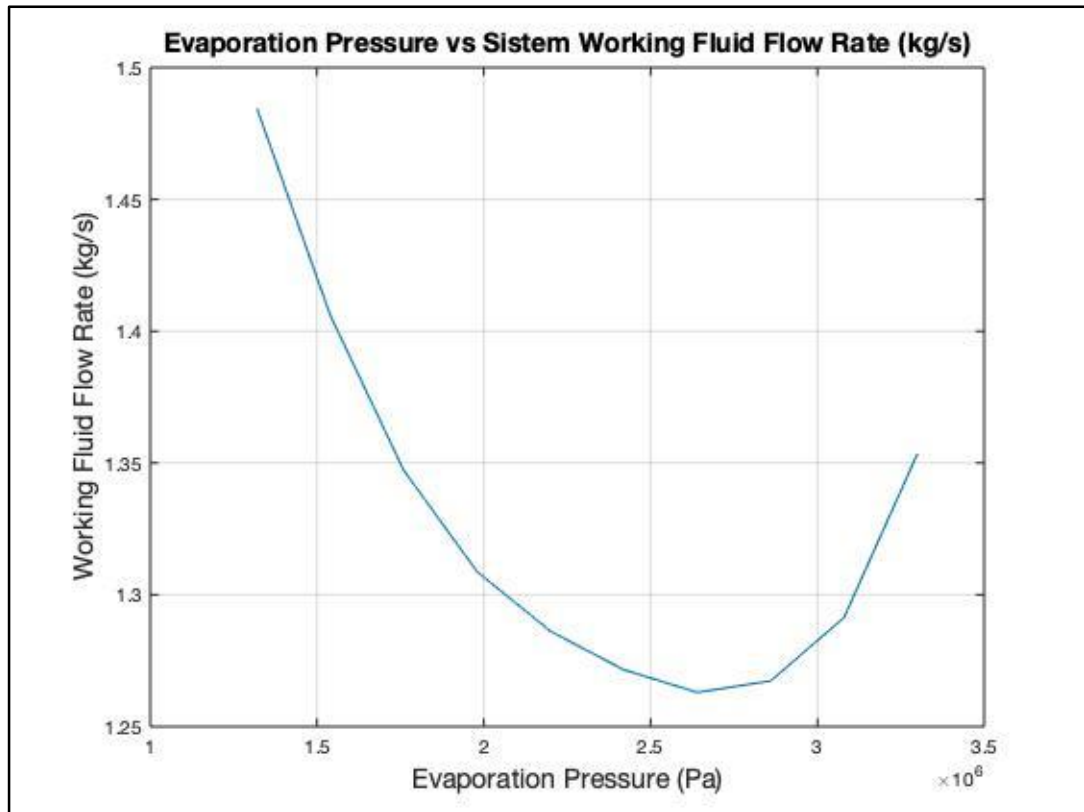


Figure 6.17: R601a Evaporation Pressure vs working fluid flow rate [kg/s].

Table 6.14: Total System Costs [\$].

Working Fluid	rORC cases				sORC cases			
	20 bar	25 bar	30 bar	Average	20 bar	25 bar	30 bar	Average
R123	912200	932600	949600	931467	925900	950700	962300	946300
R245fa	872300	880800	892000	881700	885900	889300	898700	891300
R601a	960400	956600	968800	961933	1025000	1003000	999800	1009267

As system total costs vs evaporation pressure change is not great, average cost values could be helpful to choose such systems.

First output of this study shows how to maximize the cycle efficiency by changing organic fluids and evaporation pressure. 3 different organic fluids were used and for each fluid, 2 cycle types were defined as rORC (regenerative Organic Rankine Cycle) and sORC (simple Organic Rankine cycle).

As it could be seen from the plots were given in this chapter, the efficiency of the regenerative cases are higher than the simple cases. So it could be interpreted that placing an internal heat exchanger between the turbine outlet and pump outlet are useful to increase cycle efficiency.

In case of the choosen exhaust gas inlet temperature, flow rate and energy, the maximum efficiency is obtained from the R601a fluid with rORC case between 0.13-0.14% with the evaporation pressure of circa 25 bar.

R123 fluid have both ODP and GWP, R245fa doesn't have ODP but have GWP but the R601a fluid have no ODP and GWP. So the results show that efficiency obtained with R601a fluid is maximum and also enviromental friendly. The reason of the usage of an ORC system is to make it possible to use the waste or low heat recovery and not to use fossil fuels. So, efficiency-wise R601a fluid could be selected for such system.

Second output of this study shows the effect of changing evaporation pressure and working fluids with total system costs. As in the first output, there are 3 working fluids used and each fluids were used in total 6 cases. 3 of them used in in a rORC system with evaporation pressures of 20 bar, 25bar and 30 bar. The other 3 of them used in a sORC system with evaporation pressures of 20 bar, 25bar and 30 bar. For regenerator, as evaporation pressure increases, heat exchanger area and thus heat exchanger cost is decreased. Because as evaporation pressure increases turbine outlet temperature and enthalpy decreases and regenerator outlet temperature remains constant while it is assumed $T_{con}+10\text{ }^{\circ}\text{C}$.

With the help of the regenerator, working fluid evaporator inlet temperature increases, so it helps to reduce the evaporator liquid phase zone heat transfer area and costs and also it decreases the vapor side zone heat transfer area of condenser but regenerator cost itself must be considered before desinging such systems.

To be able to answer the design load for vapor phase zone of condenser and liquid phase zone of evaporator for both simple and regenerative cases, design characteristics like fin pitch or length were changed. Because these zones have different capacities resulting from the evaporation pressure change and regenerator. Even though the two phase and liquid phase capacities are the same, since vapor phase design characteristics were changed for condenser and liquid phase zone heat design characteristics were changed for evaporator, total heat tranfer areas and costs were changed as well.

Latent heat transfer area is much greater than vapor and liquid phase zones (sensible) in the condenser. So cost-wise, two phase heat transfer area is a distinctive property for total system costs.

R245fa with rORC mean cost yields lower equipment costs. So R245fa-rORC case with 25 bar evaporation pressure is cost-wise suitable selection between these case studies. Even though the maximum efficiency was obtained with R601a-rORC with 25 bar case, cost-wise R245fa-rORC with 25 bar case was the cheapest. Because total two phase heat transfer area of R601a-rORC with 25 bar case is much higher than the R245fa-rORC with 25 bar case.

To compare the results with the literature, a different costing method compared with the results in this study. According to the source, typical equipment capacity delivered capital cost parameters for air-cooled heat exchangers (which is finned tube heat exchanger) are given below [10], [15].

Capacity measure: Plain tube heat transfer area (m^2)

Base size (Q_B): 200 (m^2)

Base cost (C_B): $1.56 \cdot 10^5$ (\$)

Cost exponent (M): 0.89

$$C_{\text{reference}} = C_{\text{origin}} \cdot \left(\frac{CEPCI_{\text{reference}}}{CEPCI_{\text{origin}}} \right) \quad (6.1)$$

$$C_{\text{achx}} = C_B + \left(\frac{Q}{Q_B} \right)^M \cdot f_m \cdot f_p \cdot f_t \quad (6.2)$$

The correction factors of f_m , f_p and f_t were interpolated for different pressure and temperature conditions. To compare the results with literature another computer program was developed in MATLAB. Results using the correlation above, are given in a table below.

Table 6.15: Calculated total evaporator cost according to the source [\$].

Working Fluid	rORC cases				sORC cases			
	20 bar	25 bar	30 bar	Average	20 bar	25 bar	30 bar	Average
R123	53354	76264	102320	77313	63209	86596	111430	87078
R245fa	59371	82361	110360	84031	74950	96297	121860	97702
R601a	33089	44560	62736	46795	40098	52261	70833	54397
Total Average				69379	Total Average			79726

Table 6.16: Calculated total evaporator cost in this study [\$].

Working Fluid	rORC cases				sORC cases			
	20 bar	25 bar	30 bar	Average	20 bar	25 bar	30 bar	Average
R123	44070	48700	51300	48023	55600	57500	57370	56823
R245fa	31350	33870	40070	35097	40940	41450	46320	42903
R601a	47250	47850	49820	48307	68730	63020	60910	64220
Total Average				43809	Total Average			54649

Table 6.17: Calculated total condenser cost according to the source [\$].

Working Fluid	rORC cases				sORC cases			
	20 bar	25 bar	30 bar	Average	20 bar	25 bar	30 bar	Average
R123	409530	504870	624270	512890	539160	596200	702120	612493
R245fa	526390	638100	772690	645727	631040	761250	955330	782540
R601a	386240	483230	590700	486723	499170	570390	705140	591567
Total Average				548447	Total Average			662200

Table 6.18: Calculated total condenser cost in this study [\$].

Working Fluid	rORC cases				sORC cases			
	20 bar	25 bar	30 bar	Average	20 bar	25 bar	30 bar	Average
R123	193600	203700	211100	202800	200400	221100	225500	215667
R245fa	168100	169500	168700	168767	180800	178800	177300	178967
R601a	236900	226200	228500	230533	289200	266500	257000	270900
Total Average				200700	Total Average			221844

The total average cost results show that, there are discrepancies between the cost calculation method in the literature with the proposed one. For evaporator total average cost for regenerative cases, there is approximately 1.58 times difference exist. For simple cases, there is 1.46 times difference exist. For condenser total average cost for regenerative cases, there is approximately 2.73 times difference exist. For simple cases, there is 2.98 times difference exist.

Therefore, it could be said that, the proposed cost calculation method for finned tube type heat exchangers yields lower and precise cost results.

7. CONCLUSION AND FUTURE WORK

In this study, a computer program was created to predict heat exchanger sizing, each and total equipment costs for heat exchangers, turbine and pump could be utilized in an ORC system. A novel cost calculation method for finned tube type heat exchangers was introduced because in the open literature there are detailed cost correlations for shell and tube type heat exchangers but for finned tube type heat exchangers there no detailed cost correlations exist.

With this novel cost calculation method, possible errors regarding costing of the system could be reduced and more precise butgeting of the heat exchangers could be ensured. This calculation method could be used not only for ORC systems but also all different kinds of systems which use finned tube type heat exchangers. In addition to that, for each zone for evaporator and condenser which are vapor phase zone, two phase zone and liquid zone, overall heat transfer coefficients were calculated, thus, more realistic heat exchanger areas and costs were calculated. The cycle efficiencies were calculated as well.

As previously mentioned, cost method for evaporator and condenser could be considered novel in this study. Because with the help of the developed program, not only overall heat transfer coefficients for each zone were calculated, but also necessary heat transfer areas for each zone, were calculated separately and thoroughly as well. In the open literature, there are no detailed cost calculation method exist for finned tube type heat exchangers. In the literature, there are statistical based costing methods exist and these methods may not always reflect the real equipment costs, manufacturing costs and so on [29].

The total average cost comparison between the literature with the proposed one have discrepancies. The cost calculation method yields much higher results than the proposed one. For evaporator total average cost for regenerative cases, there is approximately 1.58 times difference exists. For simple cases, there is 1.45 times difference exist. For condenser total average cost for regenerative cases, there is approximately 2.73 times difference exists. For simple cases, there is 2.98 times difference exists.

Therefore, it could be said that, the proposed cost calculation method for finned tube type heat exchangers yields more realistic results.

For more accurate results the future work below is proposed. To precisely calculate the heat transfer coefficient on the fin side, exhaust gas composition must be known. With the help of a combustion equation it could be done. Because its density and specific heat value, affects the total heat capacity. In this study, because of the time shortage, exhaust gas was modelled as air.

For two phase heat transfer coefficient, different evaporation and condensation correlations must be tried and results must be recorded and compared. This study could be widened by preparing an experimental system for small capacities. For different capacities, working fluids and turbine power etc. experiments could be done. Equations for different fin types (flat, corrugated), pipe types (smooth, grooved) etc. could be added to the program.

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BIOGRAPHY

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APPENDICES

Appendix A: Program File Hierarchy

R601a cases

R601a-Cases for 100 kW Electricity with Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

R601a-Cases for 100 kW Electricity without Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m

Condenser_ORC.m

Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m

Condenser_ORC.m

Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m
Condenser_ORC.m
Cost_Calculations.m

R245fa cases

R245fa-Cases for 100 kW Electricity with Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m
Condenser_ORC.m
Regenerator_ORC.m
Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m
Condenser_ORC.m
Regenerator_ORC.m
Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m
Condenser_ORC.m
Regenerator_ORC.m
Cost_Calculations.m

R245fa-Cases for 100 kW Electricity without Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m
Condenser_ORC.m
Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m
Condenser_ORC.m
Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m
Condenser_ORC.m

Cost_Calculations.m

R123 cases

R123-Cases for 100 kW Electricity with Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m

Condenser_ORC.m

Regenerator_ORC.m

Cost_Calculations.m

R123-Cases for 100 kW Electricity without Regenerator

ORC_Efficiency_calc_Evaporation_Pressure.m

Pevap=20 bar

Evaporator_ORC.m

Condenser_ORC.m

Cost_Calculations.m

Pevap=25 bar

Evaporator_ORC.m

Condenser_ORC.m

Cost_Calculations.m

Pevap=30 bar

Evaporator_ORC.m

Condenser_ORC.m

Cost_Calculations.m

Graphs

R601a

Regenerative

Graphs.m

Simple

Graphs.m

R245fa

Regenerative

Graphs.m

Simple

Graphs.m

R123

Regenerative

Graphs.m

Simple

Graphs.m

Cost Comparison Module

R601a

Regenerative

che_pro_r601a_regen_evap_20bar.m

che_pro_r601a_regen_evap_25bar.m

che_pro_r601a_regen_evap_30bar.m

che_pro_r601a_regen_cond_20bar.m

che_pro_r601a_regen_cond_25bar.m

che_pro_r601a_regen_cond_30bar.m

Simple

che_pro_r601a_nonregen_evap_20bar.m

che_pro_r601a_nonregen_evap_25bar.m

che_pro_r601a_nonregen_evap_30bar.m

che_pro_r601a_nonregen_cond_20bar.m

che_pro_r601a_nonregen_cond_25bar.m

che_pro_r601a_nonregen_cond_30bar.m

R245fa

Regenerative

che_pro_r245fa_regen_evap_20bar.m
che_pro_r245fa_regen_evap_25bar.m
che_pro_r245fa_regen_evap_30bar.m
che_pro_r245fa_regen_cond_20bar.m
che_pro_r245fa_regen_cond_25bar.m
che_pro_r245fa_regen_cond_30bar.m

Simple

che_pro_r245fa_nonregen_evap_20bar.m
che_pro_r245fa_nonregen_evap_25bar.m
che_pro_r245fa_nonregen_evap_30bar.m
che_pro_r245fa_nonregen_cond_20bar.m
che_pro_r245fa_nonregen_cond_25bar.m
che_pro_r245fa_nonregen_cond_30bar.m

R123

Regenerative

che_pro_r123_regen_evap_20bar.m
che_pro_r123_regen_evap_25bar.m
che_pro_r123_regen_evap_30bar.m
che_pro_r123_regen_cond_20bar.m
che_pro_r123_regen_cond_25bar.m
che_pro_r123_regen_cond_30bar.m

Simple

che_pro_r123_nonregen_evap_20bar.m
che_pro_r123_nonregen_evap_25bar.m
che_pro_r123_nonregen_evap_30bar.m
che_pro_r123_nonregen_cond_20bar.m
che_pro_r123_nonregen_cond_25bar.m
che_pro_r123_nonregen_cond_30bar.m

Appendix B: Heat Exchanger Module Interface

Input for capacity calculations in heat exchanger modules was given below.

```
working_fluid='R123';
T_fin_side=260; % [°C]
P_fin_side=101325; % [Pa]
Nt=80; % Vertical number of tubes
Nr=5; % Horizontal number of rows
Length_1=1165; % [mm] First estimation of coil length
NC_1=50; % Number of Circuit
DL_1=4*10^-3; % [m] Fin pitch
Length_2=1035; % [mm] First estimation of coil length
NC_2=50; % Number of Circuit
DL_2=6*10^-3; % [m] Fin pitch
Length_3=660; % [mm] First estimation of coil length
NC_3=50; % Number of Circuit
DL_3=25*10^-3; % [m] Fin pitch
V_fin_side=2.5*10^4; % [m^3/h] Volumetric Flow Rate of fin side
Configuration="Triangular";
Bst=38.1*10^-3; % Vertical pitch between pipes [m]
Bsl=33*10^-3; % Horizontal pitch between pipes [m]
Di=15*10^-3; % Pipe inner diameter [m]
ttube=0.35*10^-3; % Pipe thickness [m]
ktube=16.2; % [W/mK] Thermal conductivity of pipes (Stainless Steel)
density_pipe=8000; % [kg/m^3]
delta=1*10^-3; % [m] Fin Thickness
kfin=16.2; % [W/mK] Thermal conductivity of fins (Stainless Steel)
density_fin=8000; % [kg/m^3]
Pcon=200000; % [Pa] Condensation Pressure
Pevap(i)= 2000000 % [Pa] Evaporation Pressure
W=100000; % [W] Turbine Power
Etap=0.75; % Pump Efficiency
```

EtaT=0.8; % Turbine Efficiency

DtConSH=10; % [°C] Condenser desuperheating

DtConSC=10; % [°C] Condenser Subcooling