

THERMODYNAMIC AND ECONOMIC ANALYSIS OF ORC WITH
OPTIMIZATION OF SYSTEM COMPONENTS

by

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B.S., Mechanical Engineering, Boğaziçi University, 2016

Submitted to the Institute for Graduate Studies in
Science and Engineering in partial fulfillment of
the requirements for the degree of
Master of Science

Graduate Program in Mechanical Engineering
Boğaziçi University

2019

ACKNOWLEDGEMENTS

I would like to express my sincere gratitude to my supervisor and co-supervisor Assoc. Prof. Hasan Bedir and Prof. Günay Anlaş for their support, advice, guidance and motivation to struggle against difficulties I have been faced during this study. It is an honor to be with them in this study. I would also like to acknowledge to Prof. Salim Kunt Atalık, Assoc. Prof. Emre Otay and Assist. Prof. Hatice Mercan for their precious time to review of this thesis as the committee members.

I would like to thank Cansu Özerim and Ozan Burak Eriçok for their friendship, support and helpful discussions on my study.

I am very grateful to Gülcan Özerim for her unfailing support and encouragement during my study. Her support has a crucial role on this accomplishment.

I wish also to thank my friends Adil Han Orta, Coşkun Bilgi and Fatma Mutlu for their friendship and feedback.

I must point out that my parents M. Vildan Bekiloğlu and Ufuk Bekiloğlu have always encouraged and supported me throughout my education life, and I want to thank them for their support and love.

Finally, financial support provided by TUBİTAK BİDEB 2210-A is acknowledged.

ABSTRACT

THERMODYNAMIC AND ECONOMIC ANALYSIS OF ORC WITH OPTIMIZATION OF SYSTEM COMPONENTS

Organic Rankine cycles (ORCs) are used to generate power from low temperature heat sources. There are two main chapters in this study. In both parts, 28 different working fluids are used for three different heat source temperatures (90, 120 and 150 °C). In the first part, a preliminary radial-inflow turbine (RIT) design with loss calculations is incorporated into the model to add dynamic turbine efficiency. The turbine model is run together with the ORC, and real gas properties are used. A genetic multi-objective optimization algorithm (NSGA-II) is used to obtain an ORC and design of system components according to minimum thermal conductance per net power output and maximum performance factor (PF). The decision variables are specific speed, condensation pressure, pressure ratio through turbine, radius ratio at the nozzle, degree of superheating and pinch point temperature difference (PPTD) in the evaporator. Pareto frontiers are obtained as a result, and a decision-making method (TOPSIS) is used to select an optimum solution for each working fluid. R1234yf, R1234ze(e) and isobutane are found to be the optimum working fluids for 90, 120 and 150 °C respectively. For each solution, turbine geometry, fluid velocities and rotor rotational speed are calculated. In the second part, a heat transfer model is added into the model used in the first part for the evaporator preferred as a plate heat exchanger (PHE). The first objective function is redefined, and evaporator heat transfer area is used instead of minimum thermal conductance. The length, the width of the plates and the spacing between the plates in PHE are added as new decision variables to optimize evaporator geometry. R1234yf is found to be the optimum working fluid for 90 °C source temperature, and butane is obtained to be ideal for both 120 and 150 °C unlike the first chapter.

ÖZET

SİSTEM KOMPONENTLERİ ENİYİLEMESİ İLE TERMODİNAMİK VE EKONOMİK AÇIDAN ORÇ ANALİZİ

Organik Rankine çevrimleri (ORÇ) düşük sıcaklık kaynaklarından güç üretmek amacıyla kullanılırlar. Bu çalışma iki ana bölüm içermektedir. Her iki bölümde de 28 farklı çalışma akışkanı üç farklı sıcaklık kaynağı için kullanılmaktadır (90, 120 ve 150 °C). İlk kısımda, değişken türbin verimini eklemek için kayıp hesapları ile radyal içe akış türbininin (RİT) ön dizayn modele dahil edilmiştir. Bu türbin modeli, gerçek gaz özellikleri kullanılarak ORÇ ile birlikte çalıştırılmaktadır. ORÇ'yi dizayn ederken üretilen net güç başına minimum ısıl iletkenliğini ve maksimum performans faktörünü bulmak için genetik algoritmaya dayalı çok amaçlı eniyileme kullanılmaktadır. Özgül hız, yoğunlaşma basıncı, türbindeki basınç oranı, aşırı ısıtma derecesi ve buharlaştırıcıdaki sıkışma noktası sıcaklık farkı karar değişkenleridir. Pareto sınırları elde edilmekte ve bir karar verme methodu (TOPSIS) kullanılarak her çalışma akışkanı için en iyi çözüm seçilmektedir. R1234yf, R1234ze(e) ve izobütan sırasıyla 90, 120 ve 150 °C sıcaklık kaynakları için en iyi çalışma akışkanları olarak bulunmuşlardır. Her bir çözüm için, türbin geometrisi, akışkan hızları ve rotor açısal hızı hesaplanmaktadır. Tezin ikinci kısmında, RİT ile birlikte çalışan ORÇ modeline plakalı ısı değiştiricisi (PID) olarak tercih edilen buharlaştırıcı için ısı transfer modeli eklenmiştir. Amaç fonksiyonları yeniden tanımlanmış ve üretilen net güç başına minimum ısıl iletkenliği yerine üretilen net güç başına buharlaştırıcı ısı transfer alanı kullanılmıştır. Buharlaştırıcı geometrisini eniyilemek için PID'deki plakaların boyu, genişliği ve aralarındaki mesafe yeni karar değişkenleri olarak eklenmiştir. İkinci kısımda ise 90 °C sıcaklık kaynağı için R1234yf en iyi çalışma akışkanı olarak bulunmuş ve bütan ise hem 120 hem de 150 °C sıcaklık kaynakları için en iyi çalışma akışkanı olarak elde edilmiştir.

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LIST OF SYMBOLS

0	Stagnation (Subscript)
$1, 2, \dots, 11$	Stations in the system (Subscript)
a	Speed of sound
A	Area
b	Plate spacing
bp	Boiling point (Subscript)
C	Absolute velocity
C_p	Specific heat capacity
c_0	Spouting velocity
ch	Channel (Subscript)
$cond$	Condenser (Subscript)
cr	Critical (Subscript)
cw	Cold water (Subscript)
D	Rotor diameter
Δh_{act}	Actual enthalpy drop
e	Effective (Subscript)
E	Elastic modulus
eq	Equivalent (Subscript)
$evap$	Evaporator (Subscript)
f	Fluid (Subscript)
g	Gas (Subscript)
G	Mass velocity
h	Static enthalpy
H	Stagnation enthalpy
hw	Hot water (Subscript)
i	Dummy index (Subscript)
i_{fg}	Enthalpy of vaporization
in	Inlet; Input (Subscript)

i_s	Isentropic (Subscript)
k	Blockage factor; Thermal conductivity
l	Liquid (Subscript)
L	Blade width; Length
L_{ax}	Axial length
m	Mass flow rate; Meridional (Subscript); Material (Subscript)
Ma	Mach number
n	Normalized (Subscript)
N	Number
n_s	Specific speed
out	Outlet (Subscript)
p	Pump; Plate (Subscript)
P	Pressure
P_{co}	Corrugation pitch
pp	Pinch point (Subscript)
$\dot{Q}$	Heat transfer rate
q''	Heat flux
r	Radius; Region (Subscript)
r_{23}	Radius ratio
s	Entropy
sat	Saturated (Subscript)
sp	Single phase (Subscript)
SP	Size parameter
T	Temperature
t	Blade thickness; Plate thickness; Turbine (Subscript)
tot	Total (Subscript)
U	Rotor blade velocity; Overall heat transfer coefficient
UA	Thermal conductance
VR	Volumetric expansion ratio
$\dot{V}$	Volumetric flow rate
$\dot{W}$	Power

W	Relative velocity; Width
wf	Working fluid (Subscript)
x	Quality
Z_{rotor}	Rotor blade number
α	Absolute flow angle; Convective heat transfer coefficient
β	Relative flow angle; Chevron angle
ϵ	Effective roughness
ε_b	Back plate clearance
$\eta_{thermal}$	Cycle thermal efficiency
η_{ts}	Total-to-static efficiency
Ω	Angular velocity
ρ	Density
σ	Nozzle solidity
σ_{rr}	Stress
v	Total-to-static velocity ratio
ν	Poisson's ratio
μ	Dynamic viscosity
ω_{n1}	First bending mode
θ	Tangential component (Subscript)

LIST OF ACRONYMS/ABBREVIATIONS

ANN	Artificial Neural Network
CFC	Chlorofluorocarbon
ED	Euclidean Distance
EES	Engineering Equation Solver
FA	Firefly Algorithm
GA	Genetic Algorithm
GWP	Global Warming Potential
HC	Hydro Carbon
HCFC	Hydro Chlorofluorocarbon
HFC	Hydro Fluorocarbon
HPCD	Horizontal Port Center Distance
LEC	Levelized Electricity Cost
LMTD	Logarithmic Mean Temperature Difference
MNLP	Mixed Integer Non-linear Programming
MOGA-II	Multi-objective Genetic Algorithm-II
NSGA-II	Non-dominated Sorting Genetic Algorithm-II
Obj	Objective Function
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PCF	Perfluorocarbon
PF	Performance Factor
PPTD	Pinch Point Temperature Difference
RANS	Reynolds Average Numerical Method
RIT	Radial Inflow Turbine
VPCD	Vertical Port Center Distance
TOPSIS	Technique for Order Preference by Similarity to Ideal Situation

1. INTRODUCTION

Demand on the energy is constantly increasing due to growing population and improvements in the industry. On the other hand, environmental issues such as global warming, ozone layer depletion and air pollution have seriously occupied the agenda of politicians for about 20 years, and some protocols have been constituted to decrease the emission of the greenhouse gases. Therefore, people tend to sustainable and low-grade heat sources such as geothermal and solar energy, waste heat recovery systems and likewise to meet the energy need and establish low carbon economies instead of fossil fuels [1]. Different thermodynamic cycles can be preferred with this aim and the most popular ones are Kalina and organic Rankine cycles (ORCs) [2]. There are a great number of studies on comparison of ORC and Kalina cycles in the literature. Yari *et al.* [3] thermoeconomically compared two systems for geothermal resources have low temperature level, and the optimizations were run to maximize the net output power and to minimize the total cost separately. According to the results, these objectives are achieved at different operating conditions for both systems, and they suggested ORC because it is found to be a better choice economically. Guzovic *et al.* [4] examined both ORC and Kalina cycle for a geothermal resource in Croatia. According to their study, ORC generates more output power and has better the first and second law efficiencies. Furthermore, ORC has also advantages in aspects of complexity and safety since it has the same components and the technology with the conventional Rankine cycle addition to lower pressure and temperature levels. The main difference between Clasius-Rankine cycle and ORC is the working fluid because fluids used in ORC have lower boiling point, viscosity and critical temperature; higher molecular weight and thermal conductivity compared to water [5–7].

The most important subject about the comparison is the highest pressure levels in the Kalina cycle. The highest pressure is observed at the inlet turbine; hence, this pressure value should be examined to control suitability of conventional equipment such as pipes, turbine blades. There are different considerations about maximum pressure for durability and safety of pipes in the literature. According to Marcuccilli

and Thiolet [8], if the pressure level is higher than 60 bar, it is not convenient for a real power plants. On the other hand, 44 bar is offered for engineering applications in [9]. Additionally, 20 bar is determined as the limit with laws in some countries, and it is usually called as Dampfkesselverordnung [10]. According to the output power and economic results of [11], Kalina cycle is a more logical choice, but the maximum pressure levels are approximately 23 bar and 122 bar for ORC and Kalina cycle respectively. Therefore, it is not appropriate for real power plants with the conventional pipes.

Another difference in applications is blade material of the turbine, and conventional copper and aluminum blades are used in ORC; however, titanium blades should be used instead of conventional ones so as to prevent corrosion problem in Kalina cycle [9].

In addition to all these, ORC has the simple plant scheme, robust characteristic and easy sustainability [1]; thus, ORC is selected to examine in this study.

A schematic demonstration of ORC drawn in Aspen Plus v8.8 is shown in Figure 1.1. The working fluid is vaporized in the evaporator and the vapor has high enthalpy at the inlet of the turbine. This enthalpy is converted into power thanks to the turbine during expansion period and the pressure decreases whilst the vapor is expanded. After the expansion, the fluid enters into the condenser to form liquid because only liquid form is desired for pump operation. The fluid is pressurized in the pump before evaporation, and the cycle is completed.

1.1. Literature Survey on Organic Rankine Cycle (ORC)

There are a great number studies on the ORC in the literature [5], for instance, there are more than 1000 papers about only working fluid screening for the cycle [8]. Constitutively, these studies can be classified as follows:

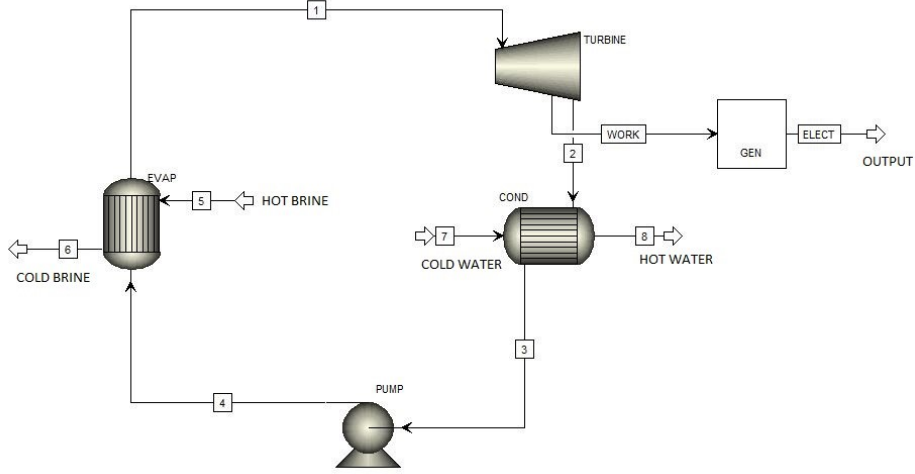


Figure 1.1. An example configuration for ORC.

1.1.1. Working Fluid Screening

Working fluid selection is the crucial point in the application of ORC, because it influences efficiency, net power output, size of heat exchanger and expander, environment and cost; they can be classified with respect to two ways [12].

First, working fluids can be categorized into three groups according to the T-s diagrams: wet, isentropic and dry fluids. As it is seen in Figure 1.2, wet, isentropic and dry fluids have negative ($ds/dT < 0$), approximately infinite ($ds/dT \approx 0$) and positive ($ds/dT > 0$) slopes respectively [13]. Additionally, there is an analytical approach to determine the type of working fluid in the literature [14],

$$\frac{ds}{dT} = \frac{C_p}{T_{cond}} - \frac{(n \cdot T_{cond}/T_{cr}) / (1 - T_{cond}/T_{cr}) + 1}{T_{cond}^2} \Delta H_V \quad (1.1)$$

where s , T , C_p and H mean specific entropy, temperature, specific heat capacity at constant pressure and stagnation enthalpy respectively. n is given as 0.375 or 0.38 in [14].

The lowest dryness ratio, in other words vapor phase fraction in the working fluid, at the outlet of the turbine should be higher than 85% in order to avoid damage to blades. Wet fluids pose a risk in this issue because of the negative slope; hence, dry or isentropic fluids are more convenient to be used in ORC [15]. On the other hand, the working fluid should not be excessively dry, because there is superheated vapor at the outlet of the turbine for too dry situation and it needs more complex structure with regeneration equipment to prevent waste heat; thus, isentropic fluids accepted as the most suitable fluids for ORC in [2]. In spite of this, dry fluids are more appropriate for ORC owing to the lack of need on superheating vapor at the inlet of turbine with respect to [12]; moreover, superheated vapor causes to decrease the efficiency of the system for subcritical ORC according to [16]. In the light of these, both of dry and isentropic fluids are mostly investigated for ORC in the current study [6, 7].

Another classification for working fluids can be done with regard to atomic structure, and hydrocarbons (HCs), perfluorocarbons (PCFs), chlorofluorocarbons (CFCs), hydrofluorocarbons (HFCs), hydrochlorofluorocarbons (HCFCs), siloxanes, alcohols, ethers and some inorganics such as CO_2 are main types of working fluids. All these types have different thermodynamic and physical properties, for example, stability, flammability, toxicity, cost etc. [12]. HCFCs cannot be used anymore, since Montreal Protocol prohibits the usage of this group of working fluids due to the fact that they contribute the ozone layer depletion [8]. On the other hand, the ASHRAE safety categories are very helpful to indicate the flammability and toxicity of the fluids, and low level of flammability and toxicity are desired [5].

If thermodynamic and physical properties are individually and shortly examined for working fluids, firstly, vaporization latent heat is a significant property, and in the literature [17], Jacob number (Ja) is described as the ratio of sensible heat and vaporization latent heat. Thermal efficiency value increases with reducing Ja, but it influences exergy efficiency oppositely with respect to the thermal efficiency, and if Ja increases, exergy efficiency increases. Therefore, it can be said that there is a trade-off between the first and second law efficiencies [12]. Addition to this, the relation between the specific heat and the efficiency values can be interpreted according to Ja,

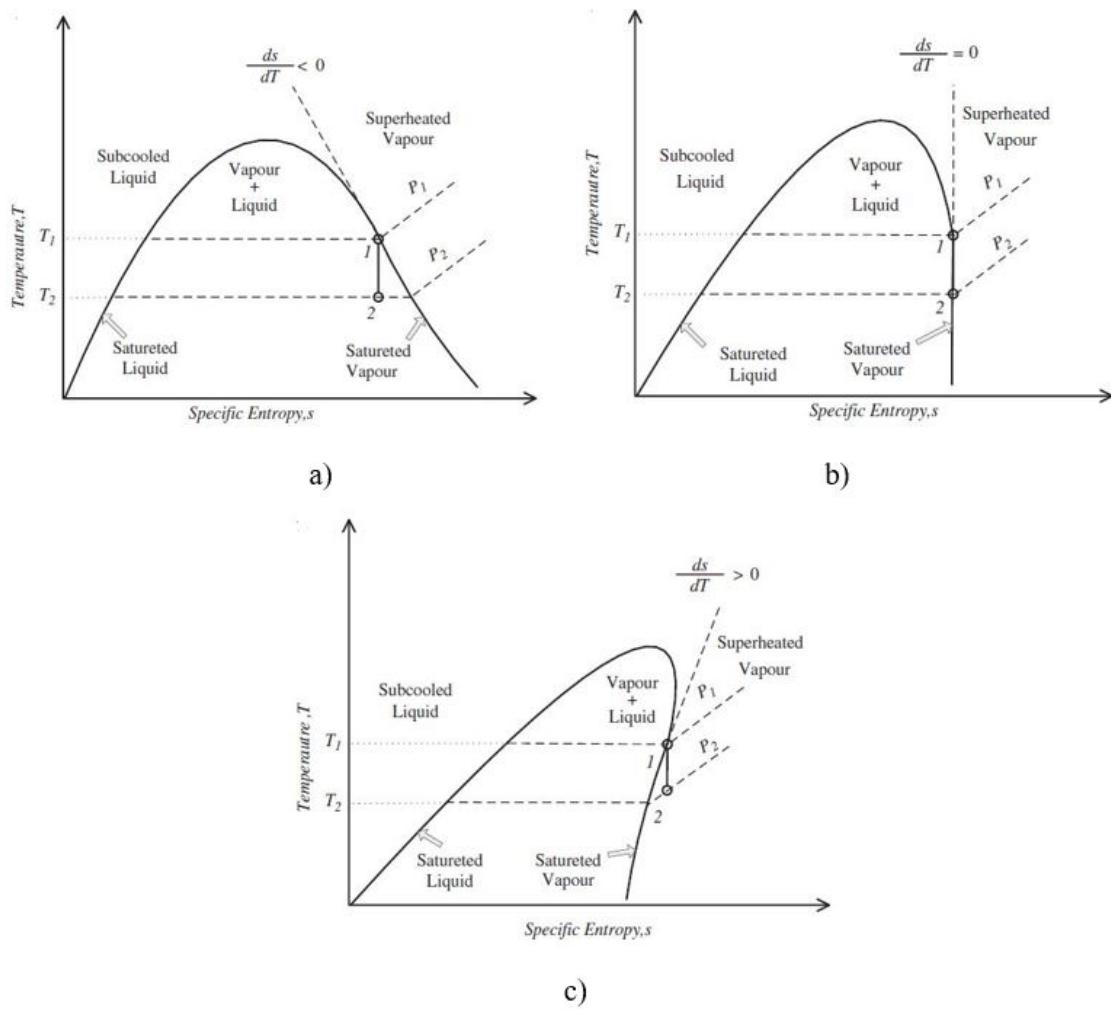


Figure 1.2. Vapor saturation curves for a) wet, b) isentropic, c) dry fluids [13].

because sensible heat depends on the specific heat linearly. Density is also an important feature, and it directly affects the cost of the system. Because, low density requires high volumetric flow rate, bigger heat exchangers and expander are needed. On the other hand, the pressure drop quantity in the evaporator increases with increasing volumetric flow rate [2]. The impact of the density on the turbine size is formulated as a size parameter by Macchi *et al.* [18]:

$$SP = \frac{\sqrt{\dot{V}_{t,out}}}{\sqrt[4]{\Delta H_{is}}} = \frac{\sqrt{\dot{m}_{wf}/\rho_{t,out}}}{\sqrt[4]{\Delta H_{is}}} \quad (1.2)$$

$$VFR = \frac{\dot{V}_{t,out}}{\dot{V}_{t,in}} = \frac{\rho_{t,in}}{\rho_{t,out}} \quad (1.3)$$

SP usually has a value from 0.02 m to 1 m for turbines used in subcritical ORC [19]. Isentropic volumetric flow ratio is also defined in the same study, and the turbine efficiency increases with reducing VFR [18], it was also used as an indicator in [6].

Critical temperature and pressure values are crucial properties as well, and low critical temperature or pressure are preferred, of course low critical temperature or pressure values are higher than the maximum values of the cycle for subcritical systems [5, 6]. However, there is an opposite claim in the literature, and some studies [12, 20, 21] advocate that higher critical temperature values positively influence the thermal efficiency. Molecular weight has both positive and negative sides for the cycle like Ja, and high molecular weight is desired to increase the turbine efficiency, but it gives rise to bigger areas for heat transfer. Therefore, the installation cost increases with molecular weight [22].

Rayegan *et al.* [23] introduced a technique to determine the maximum pressure level for dry fluids, and according to the study, evaporation pressure should not be higher than the points which have infinite slope in the T-s diagram to guarantee absence of the liquid phase in the turbine. The point is called inflection point. The lowest limit

of the condensation pressure is also specified as 5 kPa [23], but condensation pressure is suggested to be at least same with the atmospheric pressure so as to avoid the leakage into the working fluid [8].

Marcuccilli *et al.* [8] examined 58 different working fluids for ORC with a specific expander type, radial inflow turbine. Fluids are not just evaluated with respect to their thermodynamic efficiencies; safety, cost, feasibility conditions are considered with the legislations as well. Therefore, they suggested a performance factor to compare working fluids with regard to both thermal efficiency and turbine wheel size [8].

$$PF = \frac{\eta}{D^2} \quad (1.4)$$

Fluids were ranked with respect to performance factor (PF), and availability examination is done. Propane was found to be the most suitable fluid for the geothermal source at 423 K temperature [12].

Sauret *et al.* [2] investigated 5 working fluids for 423 K temperature of geothermal source again; they are R134a, R143a, R236fa, R245fa and nPentane. nPentane was considered since Ormat Technologies Inc. usually uses nPentane in the industry. In this study, PF was revised as follows [12]:

$$PF = \frac{\dot{W}_{net}}{D_{in}^2} \quad (1.5)$$

According to Eq. (1.5), R143a was first ranked with respect to the PF, in spite of that the highest net output work was obtained when R134a was used [5].

Guo *et al.* [6] examined 26 working fluids for low grade temperature resources (80-100°C), and R245fa is selected as the most appropriate one. Irreversibility values of each component were also calculated from exergy analysis, and evaporator is observed as the first exergy destruction source for ORC as mention above. Rodriguez *et al.* [16]

compared 15 different fluids via Aspen-HYSYS software in the range of 90-140°C, and the most convenient fluid was determined as propane (R290).

Finally, there is no an ideal working fluid for ORC applications as understood from the results in the literature, because the best efficiency value can be reached using different fluids altering the heat source temperature. Besides, the efficiency or net power output is not a unique criterion to select a fluid, because economic, environmental, safety and equipment restrictions have important effects on the working fluid selection as well [12].

1.1.2. Expander

There are different types of expanders; they are essentially classified into two groups: velocity type expanders (radial inflow turbine, axial turbine etc.) and volume type expanders (scroll, screw, reciprocating piston expander etc.) [12].

Scroll expander has constant volume ratio; thus, over expansion or under expansion problems can be observed, because the expansion ratio or volumetric flow rate may change with seasonal variations. On the other hand, scroll type expander is produced easily and has light weight. It can achieve high efficiency value and works with two phase flow without any damage. It works with low rotational speed, so there is no need of a gearbox to contact with the generator. However, the output power capacity is indeed low; thus, it is appropriate for experimental setups or very low capacity power plants. It is usually used to product power in range of 1-10 kW [12].

Screw expander is another volume type expander, and it can generate higher power outputs than scroll type. Screw expanders are already used in medium-scaled ORC plants used geothermal resource or waste heat recovery. They can tolerate two phase mixtures like scroll type and have high efficiency values for off design conditions. However, they are not appropriate to generate high power generation, since very high rotation speed is required, compared to the recommended speeds as the limitation of materials. Therefore, gear boxes are generally used to work them coherently with the

commercial generators, and it causes an increase in both installation and maintenance costs. It is generally preferred to generate power in range of 15-200 kW [12].

Turbines, in other words velocity type expanders, are already used for conventional Clasius-Rankine cycles, and they do not require a significant change to use in ORC plants. Axial turbines are more feasible under high flow rates with low pressure drops, and high flow rates violate the assumption that there is no pressure drop except for the expander in the cycle. Therefore, radial inflow turbines are more suitable to use in an ORC plant, and they have been already preferred in the commercial area [12], because it has good efficiency for low volumetric flow rate and high pressure ratio [5].

Radial inflow turbines (RIT) have high efficiency for seasonal variations thanks to the variable inlet vanes [9]. Marcuccilli *et al.* [8] particularly studied on radial inflow turbines, and the net output power were investigated under off design conditions. Small adjustments are adequate for different resources; thus, research and development cost can be recovered easily in a short time. Organic fluids are very dense, and radial inflow turbines are robust under the high load conditions. Therefore, this type of expander is a rational choice for ORC applications [5], and it is examined in this study.

Sauret *et al.* [5] roughly designed a RIT for a specific geothermal resource after the selection of the most appropriate working fluid. Non-dimensional specific speed used in the study is calculated from the following equation [24]:

$$n_s = \frac{\Omega \sqrt{\dot{V}_{t,out}}}{\sqrt[4]{\Delta H_{is}}} \quad (1.6)$$

According to Bao and Zhao [12], the specific speed should be between 0.3 and 0.9. The optimum range for the specific speed is advised to be 0.5-0.9 by Dixon [24]. Besides, the range of 0.4-0.8 is accepted as optimum value for specific speed in the textbook of Moustapha *et al.* [25]. Lastly, Aungier [26] also recommends optimal values between 0.45 and 0.75. The maximum rotation speed is determined as 24000 rpm which

correspond to commercial generators [5]. Then, the ratio of the tip speed to spouting velocity, U/c_0 , is used to optimize the efficiency of the turbine, and 0.7 is found to be the optimum ratio in ideal cases, in other words frictionless circumstance [5, 24]. Furthermore, this ratio should be between 0.68 and 0.71 for real cases to obtain the best efficiency [24]. Spouting velocity can be calculated as follows [24]:

$$c_o = \sqrt{2000\Delta H_{is}} \quad (1.7)$$

Marccuccilli *et al.* [9] studied on RIT in detail. U/c_0 and the specific speed were used to optimize the turbine sizes, and the best turbine isentropic efficiency range is found as 0.82-0.9. Another important result is that inlet turbine pressure should be close to the critical pressure of the working fluid to improve the net power output.

Mach number constraint for RIT was discussed by Quoilin [20], and if Mach number is too high, it results in drop in the efficiency. The pressure ratio is generally accepted as 4 in turbine to prevent choked flow [8]. Similarly, Guo *et al.* were fixed the ratio to 3.5 in their study [6]. Also, the pressure ratio can be increased up to 8 in practical applications; but, 4 is also suggested as the maximum value again by Sauret and Rowland [5].

1.1.3. Heat Exchanger

Heat exchangers influence cost, efficiency and net product of the cycle; thus, many researches have focused on these components. First of all, the biggest irreversibility source in ORC is evaporator; hence, there are many studies [1, 8, 16] on optimization of the evaporator to reduce exergy destruction in order to increase utilization of the energy. As mentioned above, pure substances are usually used in ORC. Therefore, constant vaporization temperature is a very significant point for the cycle, and pinch point analysis is applied to calculate extracted heat from the brine as high as possible [8]. For especially geothermal power plants, maximum heat addition is important since the main aim is to reach the highest net output power instead of highest efficiency

value. Pinch point analysis is widely used in the literature, and composite curves are the best way to indicate the analysis structure. Composite curve used by Rodriguez *et al.* [16] is indicated as an example in Figure 1.3.

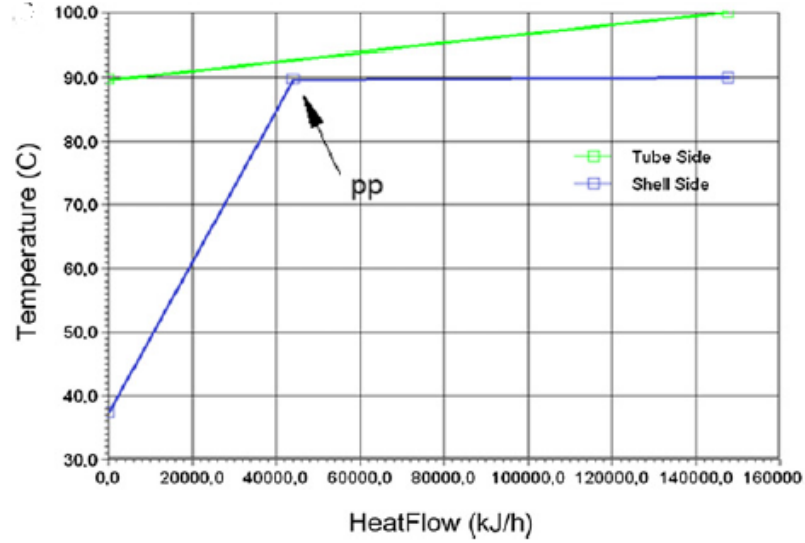


Figure 1.3. Composite curve example for pinch point analysis [16].

Two different studies [1, 16] emphasized that the irreversibility can be decreased by reducing pinch point temperature (PPTD) difference between two fluids; however, it has a disadvantage. The total heat transfer area should be increased with decreasing PPTD, and it increases the installation cost. Furthermore, other significant point on the pinch point analysis is that there is enough temperature difference to prevent the crossing of two curves [8]; thus, 3°C is accepted as minimum for the difference [8, 16].

Guo *et al.* [6] considered the total heat exchanger area, both of evaporator and condenser, as an economic indicator. The smaller total area is desired in the study, and the optimum working fluid was selected with respect to this condition.

1.1.4. Studies Related to Optimization

Many researchers concentrate on to enhance the system with respect to some design or performance parameters because it is already a mature technology. Therefore, there are a lot of researches in the literature to optimize ORC system, and a great number of studies were examined as follows.

Oyewunmi *et al.* [27] studied on thermodynamic optimization of recuperative subcritical ORC. The aim of the optimization process is to maximize net output power by using 30 different working fluids according to a certain case study (Hot and cold sources' properties are constant). The Interior Point algorithm which is developed to apply large scale non-linear problems was used to optimize the system in MATLAB, and they used NIST database to determine working fluid properties. Recuperator is required to obtain maximum net output power with respect to the results [27].

In another study, ratio of net output power to total heat exchanger area was used as the objective function to reach the optimum design parameters for different source conditions. The objective function provided an economic perspective to the study because the total heat exchanger area is the main installation cost reason. The results were also tested on a real ORC system that has a volume type expander since it is in lab-scale. Five decision variables are optimized, and they are turbine inlet pressure, mass flow rate of working fluid, degree of superheating, condensation pressure and mass flow rate of the brine. Lastly, 4 different system configurations were evaluated in the study [28].

Another economic-based optimization study on ORC was done by Erdeweghe *et al.* [29], and their study was included drilling costs for geothermal applications. The design of heat exchangers (evaporator and condenser) was optimized with respect to net present value. REFPROP (NIST) was used in Python with CasADi optimization algorithm. They determined optimum value of 13 design variables after the optimization, and some of these parameters were related to geometry of the heat exchanger. TEMA-E shell&tube type heat exchanger was used because it has better performance under

high brine pressure [29]. Same heat exchanger type was also optimized by Walraven *et al.* [30], and both studies were used the Bell-Delaware method to model the heat exchanger. Besides, working fluid passed in the shell side while geothermal oil flowed on the tube side [29, 30]. According to Walraven *et al.* [30], recuperator is necessary for high temperature resources.

Sabeti *et al.* [31] investigated the system with multi-objective optimization technique, and NSGA-II (Non-dominated Sorting Genetic Algorithm-II) was used in EES (Engineering Equation Solver) to obtain optimum values of 11 different decision variables: volumetric fraction of nanoparticles, pressure drop in ejector, mass flow rate and pressure at turbine inlet, area fraction of ejector, pressure and temperature at turbine outlet and PPTD at the evaporator. They examined the system with 4 different objective functions: thermal and exergy efficiencies, total heat exchanger area and product cost rate. Firstly, the results were gathered with single-objective optimization separately; then, multi-objective optimization outcomes were demonstrated as a Pareto figure due to the fact that there was a conflict among 4 objective functions [31].

Additionally, Massimiani *et al.* [32] optimized a small scale ORC using Artificial Neural Network (ANN) because the thermodynamic model of ORC has high degree of non-linearity. R1234yf was used as working fluid, and the main aim was to maximize the power production together with to minimize UA , the product of the total heat transfer coefficient and heat exchanger area in other words thermal conductance, and rotational speed of the expander. Therefore, there were 3 different objective functions in this study, and ANN was run separately for each of them with 6 decision variables using only one hidden layer. Then the results were compared to results obtained from a classical optimization toolbox in MATLAB, and more optimum outcomes were acquired in a shorter time thanks to ANN [32].

Su *et al.* [33] pointed out the effects of working fluids on performance of the system, and they optimized the system to find the best working fluid candidates. The net output power was chosen as objective function; the environmental and safety features were controlled after the optimization to select the fluid. MATLAB 2015b was used

for the numerical process. After all that, two novel working fluids were found the best choice: R245eb and R245cb. Additionally, analytic equations were given for almost entire thermodynamic properties of a fluid [33].

Liu *et al.* [34] analyzed the system with multi-objective programming which is produced using weighted summation method. They directly focused on binary cycles for geothermal resources. Four objective functions were used to generate multi-objective optimization for 4 different system configurations, and these functions were capitalized cost, net output power, and thermal and exergy efficiencies. The weights of all functions were determined to be 0.25. The decision variable of the optimization process was evaporation temperature since the system performance is highly sensible to this parameter. They tested their optimization process by using of “Rosenbrock’s banana”, and Microsoft Excel was used in order to program solver. R123 was found as a convenient fluid for simple cycle [34].

Marty *et al.* [35] applied Mixed Integer Non-Linear Programming (MINLP) algorithm in GAMS to optimize an ORC for a geothermal resources with determined properties such as mass flow rate and temperature of brine. R245fa was used as working fluid in the study. The main aim of the study was to reach maximum profit. All heat exchanger calculations were evaluated for shell&tube type. According to outcomes [35], the investment could be feasible regarding the electricity selling price as mentioned before [16].

Bina *et al.* [36] optimized 4 different configuration of ORC for a real power plant at Sabalan which is the unique geothermal power plant in Iran with a dry working fluid. The objective of the study was to acquire maximum energy and exergy level together with minimum total production cost. They applied variable metric searching method in EES. They used Shannon’s entropy method to determine the weight coefficients for multi-objective optimization. The sorting of the configuration was given at the end of the study and ORC with internal heat exchanger was the best solution for Sabalan. As an interesting outcome, basic ORC has better performance than regenerative one [36]. This claim is coherent with the results of Walraven *et al.* [30].

Addition to these, Dong *et al.* [37] investigated ORC with a design of radial inflow turbine as being different from mentioned before. Three components of the turbine are designed, and these are volute, nozzle and impeller. The effect of diffuser was ignored in this study. Turbine loss model was classified into five groups: Nozzle and impeller passage losses, leaving velocity loss, disk friction loss and inner leakage loss. The objection function was turbine efficiency, and Particle Swarm Optimization algorithm was used to maximize the function value in MATLAB. REFPROP (NIST) was used to acquire working fluid properties. Outlet temperature of the heat source fluid and six different turbine parameters were chosen as decision variables. R123 and R245fa were specified as suitable working fluids among 4 working fluids (others: R365mfc and pentane). According to the outcomes, flow coefficient and loading coefficient were not influenced by source temperature; on the other hand, it affected size and rotational speed of turbine. Besides, low reaction of degree was suggested as a result [37]. Another research about optimization an ORC with the design of radial inflow turbine was done by Bahadormanesh *et al.* [38]. They examined the system using 4 working fluids: R22, R245fa, R236fa and nPentane. Addition to turbine geometry variables, mass flow rate of working fluid, evaporation temperature and pressure were investigated as decision variables. Turbine was modeled in one dimension as 90° radial inflow turbine, in other words there was no swirl at the rotor exit, by using of Aungier's approach [26]. Thermal efficiency and PF in Eq. 1.5 were described as objection functions, and Firefly Algorithm (FA) was applied, and they advocated FA is more reliable rather than PSO. Furthermore, vibration and tension limitations of turbine were firstly investigated in an ORC study. Pareto frontiers were obtained at the end of the study, and a bit increase in the size of turbine provided higher efficiency with respect to the Pareto frontiers [38]. Furthermore, Kim *et al.* [39] preliminarily designed a radial inflow turbine for ORC using Aungier's process [26] again. They introduced a new approach to model preliminary design of the turbine, and there were no need to ideal gas assumption, performance charts and average state properties as novelties. The starting point of the new method was to begin with number of blades and find inlet relative angle from the blade number. Nozzle and volute were designed as well as rotor. According to the results, incidence angle and rotational speed of impeller had a

significant effect on the power output and efficiency of the turbine [39].

Jubori *et al.* [40] was also studied on optimization of an ORC together with radial inflow turbine. Addition to [37–39], Jubori *et al.* modeled the nozzle and rotor of the turbine in 3-D using ANSYS CFX; thus, more sensitive results could be obtained for the turbine in this study because turbulence case was also considered using Reynolds Average Numerical Simulation (RANS). Of course, mean-line design was firstly done in order to acquire initial values of 3-D modeling. Isopentane and R245fa were used as working fluid in the study, and power output and turbine isentropic efficiency were selected as objective functions. At the end of the study, the cycle works with higher thermal efficiency was reached. By the way, GA is used to optimize 6 decision variables [40].

Tarashri *et al.* [41] optimized an ORC system to work together with an internal combustion engine, and they used Taguchi method for optimization of 7 decision variables. Three of them was different than aforementioned studies: temperature at turbine inlet, degree of superheating, condensation temperature. R236fa, R245 and R124 were investigated regarding thermal efficiency and net power output as objective functions; the maximum thermal efficiency was obtained using R124. By the way, plate type heat exchanger (PHE) was used because better ratio of surface area to total volume [41]. Furthermore, Guillaume *et al.* [42] investigated ORC for an internal combustion engine with 4 different volume types expander: scroll, screw, piston and vane expander. PHE was used and total heat transfer area was calculated. The cost of working fluids used in the study (R245fa, ethanol, water) was evaluated according to their mass. Five decision variables were optimized via MATLAB optimization toolbox, and these are PPTD, evaporation pressure, expander rotational speed and degree of superheating [42].

Bernal-Lara *et al.* [43] explained thermodynamic and economic modeling of ORC in detail. Their main aim was to determine the ideal working fluid composition and working conditions for a given source. Total annual cost and thermal efficiency were used as objective functions, and there was a trade-off as mentioned before. The opti-

mization was applied using CONOPT solver in GAMS. Therefore, Pareto frontier was obtained as a plausible result because they did not use weighted multi-objective function. According to results, mixture working fluids provided higher efficiency values, but if cost of the cycle was considered, pure working fluids could be more feasible [43].

He *et al.* [44] concentrated on exergy loss distribution for a subcritical ORC using R245fa. Effects of heat source temperature, PPTD at evaporator, turbine isentropic efficiency and cooling water temperature on exergy destruction were investigated through sensitivity analysis. UA was used as an economic indicator for the evaporator, and the simulation of the system was carried out via EES. Additionally, quadratic optimization techniques were applied to obtain maximum value of net output power. The whole exergy distribution in the system was linearly dependent on source temperature and reduced by increasing expander isentropic efficiency [44].

Wang *et al.* [45] examined three ORC arrangements with 14 working fluids and optimized the systems with respect to the thermodynamic and economic performances in MATLAB together with REFPROP (NIST). PHE was used since it provides better performance values with more compact volume compared to shell&tube type. They directly calculated area of the heat exchanger instead of UA . NSGA-II was selected in order to optimize exergy efficiency and the cost per net power generation with 2 decision variables: evaporation and condensation temperatures. R141b had the best characteristics according to the results except safety factors [45].

Gimelli *et al.* [46] used thermal efficiency and total heat exchanger area as objective functions. The area was selected as an economic indicator because it dominates the cost of the overall system. Pareto frontier was obtained as a result of multi-objective process. There were five decision variables: recuperator efficiency, maximum and minimum pressures, degrees of superheating and subcooling. Constant overall heat transfer coefficients were accepted to simplify the modeling. MOGA-II (Multi-objective genetic algorithm II) was applied to optimize the system. According to the results, the total heat exchanger area of the system should be in the range from 446 to 1079 m² [46].

Feng *et al.* [47] optimized ORC using NSGA-II. The effect of zeotropic working fluids were also investigated, and R245fa, pentane and their mixtures were used. Exergy efficiency and levelized electricity cost (LEC) were selected as objection functions for five decision variables: mass fraction, temperature at the turbine inlet, condensation temperature, degree of superheating and PPTD. Thus, Pareto frontier was found as a result of the optimization. Technique for Order Preference by Similarity to Ideal Situation (TOPSIS) was used to specify an ideal solution from the Pareto solutions. The sensitivity of the objective functions were separately demonstrated for each decision variables. The mass fraction of R245fa inversely affected LEC, and the exergy efficiency had a maximum point regarding the mass fraction. Evaporator area was also calculated to precisely determine economic conditions [47].

Another optimization study for geothermal resources have different temperature values was done using R245fa as working fluid by Liu *et al.* [48]. The main focal point was PPTD because it is usually accepted constant in the literature and its effect on thermodynamic performance and output power. The optimal PPTD was found using quadratic optimization. UA , SP and VFR (Eq. 1.2 and 1.3) were also calculated. Sensitivity analyses of PPTD were exhibited using many thermodynamic parameters at the end of the study [48].

Yu *et al.* [49] improved a novel techno-economic optimization model for waste heat recovery ORC. First of all, they used Duran-Grossmann model in order to simulate heat exchanger and optimize, and temperatures should be fix in PPTD. The main objective was to observe the effect of variable heat source temperature. Then approximation was used to calculate LMTD instead of the classical approach like aforementioned researches. However, the heat transfer coefficient was assumed constant in the study, and more rigorous heat transfer calculations were determined as a future work. By the way, shell&tube heat exchanger was used as both evaporator and condenser. The objective function was constituted using only economic indicators: maintenance cost, equipment cost and profit. According to the results, the thermal efficiency of the system was approximately increased by 24.3% [49].

Braimakis *et al.* [50] elaborately examined ORC system for different source types. Four decision variables were used: maximum pressure level, condensation temperature, PPTD in evaporator and condenser. PHE was preferred in the study. Four different working fluids were also compared, and cyclopentane was found preferable choice for small systems, on the other hand, propane and R1234ze were suggested for larger systems. According to results, PPTD in evaporator was crucial for the performance of the system, but PPTD in condenser had a relatively unimportant influence [50].

Hayat *et al.* [51] investigated the comparison between basic and recuperative ORC using NSGA-II in MATLAB in order to utilize built-in program. Seven different working fluids were used for basic cycle, and these are R123, R1234ze, R21, R236ea, R601 and R245ca. On the other hand, only four working fluids were used in recuperative one. The properties of fluids were taken from REFPROP (NIST). The objective functions were selected as thermal efficiency and specific investment cost. Additionally, there are 4 decision variables: pressure at evaporator, degree of superheating and PPTD in evaporator and condenser. Pareto frontier solutions were obtained at the end of the study. According to results, R21, R601 and R245ca were plausible fluids for basic type of cycle, and using basic type was more logical rather than recuperative one due to higher investment cost; thus, the payback period was longer. By the way, they selected PHE due to better thermodynamic performances [51].

Furthermore, Zhang *et al.* [52] particularly concentrated on the comparison about different heat exchanger types for subcritical ORC. Plate type, shell&tube type and finned-tube type heat exchangers were compared, and their thermodynamic and economic parameters such as PPTD, evaporation pressure, payback period were examined. According to payback period, plate and shell&tube type exchangers were more preferable than finned-tube one [52].

Addition to Zhang *et al.* [52], another shell&tube and plate heat exchangers were compared by Walraven *et al.* [53]. They already examined shell&tube heat exchanger using Bell-Delaware method in a previous study [30]. By the way, both type heat exchangers are used for ORC applications in the literature. The optimization was

based on exergy efficiency; five and six decision variables were used for plate and shell&tube type respectively. Each heat exchanger was split into five pieces which have an equal heat input so as to provide accuracy of simulations. Simulations were run in CasADi with REFPROP (NIST). Besides, eight working fluids were used in different simulations to become sure from the results. According to outcomes, PHE had better efficiency compared to shell&tube one, but there was only one drawback. It was that the surface geometry of the both side is same, and this situation sometimes caused inefficient results [53].

Another research about design of a single stage radial inflow turbine was done by Da Lio *et al.* [54] using Aungier's suggestions [26]. MATLAB was used to develop preliminary model for the turbine, and R245fa was selected as working fluid. The real gas properties were obtained from REFPROP (NIST). They started to mean-line design with the assumption of 90° RIT. Loss correlations were also used to observe irreversibilities through the turbine. According to results, specific speed (Eq. 1.6) should be between 0.55 and 0.60 because loss effects and exit kinetic energy reduce the thermal efficiency for low and high specific speed values respectively [54].

Li *et al.* [55] studied on a new environmental working fluid, R1234ze(e), and the comparison between subcritical and transcritical ORC systems for 100-200°C of heat source. Temperature and pressure at turbine inlet were selected as decision variables, and the optimization process was performed to maximize net output power per mass flow rate of heat source fluid. Brine outlet temperature was also unlimited; it was evaluated using pinch point analyses. According to the results, subcritical cycle was more feasible for lower temperatures than 160°C, on the other hand, if the heat source has higher temperature value, transcritical cycle was more plausible. Additionally, R1234ze(e) had larger efficiency compared to classical working fluids such as R245fa and R600a. By the way, REFPROP was used to obtain real fluid properties [55].

Shao *et al.* [56] made an experimental study for a radial inflow turbine to work with ORC, and they used R123 as working fluids during experiments. The rotational speed of the rotor reached 53564 RPM as maximum value, and the maximum efficiency

was obtained with 34586 RPM. However, the heat source temperature could cause a change this RPM value as well [56]. Another experimental study related to ORC was done by Wang *et al.* [45], and they concentrated on condenser design. A segmented method was used to model and design the condenser due to the fact that this approach has a good accuracy compared to previous studies on condenser design in the literature. R245fa was selected as working fluid. Shell&tube heat exchanger was used, and there was only 4.1% difference in overall heat transfer coefficient value between the model and experiment at the end of the study [45].

Ghilvacs *et al.* [57] especially studied on heat exchanger design for ORC in an automobile together with screening working fluid. PHE was used to extract heat from the internal combustion engine. At the end of the study, a certain working fluid was determined, and necessary evaporator area was calculated evaluating heat transfer coefficients. Despite of that R141b provided the best net power output; R245fa was suggested because of the safety issues [57].

Another multi-objective research was done for a new combined cooling, heating and power system (CCHP) by Boyaghchi *et al.* [58]. Optimization process was applied with GA in EES using turbine inlet temperature and pressure, evaporation temperature, turbine back pressure and outlet temperature of heater as decision variables. This study also included effects of seasonal variations. There were three objection functions: total product cost rate, thermal and exergy efficiencies. The system was also optimized separately for each objective functions. According to results, the novel model was very successful in summer, but although it was achievable in winter, the improvement was not as high as in summer [58].

Long *et al.* [59] studied about screening working fluid with respect to the exergy efficiency for waste heat recovery ORC. The relationship between exergy efficiency and evaporation temperature was also investigated and they were aware of there was an optimal point according to sensitivity analysis. GA was used to determine maximum exergy efficiency. According to the result, fluids have low critical temperatures were more suitable for low grade temperature sources [59].

Another study on selection of working fluid was done for particularly geothermal sources by Al-Weshahi *et al.* [60]. 25 different working fluids were examined with respect to certain criteria: net output power, thermal efficiency, work done by pump, evaporation and condensation pressures and environmental issues with safety. According to results, R141b, R245ca, R600, R245fa, R717 and R123 provided better thermal efficiencies, but they could not meet the whole criteria; thus, they were not suggested in the study. R236ea, R227es and R236fa were selected with respect to the criteria.

Sauret *et al.* [61] maintained their study mentioned before [5] to develop a 3-D radial inflow turbine model. They already presented preliminary design of the turbine using RITAL, and then they simulated 3-D turbine using ANSYS CFX. R143a was used as working fluid, and the fluid properties were acquired using the Peng-Robinson. The rotor clearance was assumed as 3 mm and the approach in Moustapha *et al.* [25] was applied in the study for mean-line model [61].

Yang *et al.* [62] optimized an ORC to utilize waste heat from a marine internal combustion engine. Thermal efficiency and net output power per total cost of equipment, in other words net power output index, were selected as thermodynamic and economic performance indicators. The effects of turbine inlet and outlet pressure with cooling temperature on the indicators were investigated, and R245fa were found the best choice among R1234ze, R600 and R600a, although the highest thermal efficiency was obtained using R1234ze [62].

El-Emam *et al.* [63] studied on optimization of a new regenerative ORC for geothermal heat sources using EES, and heat exchanger area per net output power was selected as objective functions. By the way, shell&tube type made from titanium was preferred because they advocated that shell&tube type provides higher heat transfer coefficients with more compact sizes as interestingly and oppositely with respect to aforementioned studies. Additionally, optimum work conditions such as mass flow rates of brine and working fluid were examined for different source temperature values [63].

The selection of the working fluid was elaborately investigated for geothermal sources with different temperatures by Zhai *et al.* [64]. They optimized the system regarding net output power using evaporation temperature as a single decision variable for each working fluid and heat source temperature. R32, R134a and propylene were suggested for the range of 383-423 K at the end of the study [64].

Yang *et al.* [65] concentrated on economic optimization of ORC using geothermal energy as heat resource. Net output power index was used as objection function. Temperature at the turbine inlet and PPTD in both evaporator and condenser were also determined in the study to be decision variables. Besides, shell&tube heat exchanger was preferred. According to the outcomes, R600 is the most appropriate fluid for economic optimization rather than R1234ze, R600a, R1234yf and R209. Additionally, PPTD in evaporator was more important compared to condenser [65].

Imran *et al.* [66] examined the optimization of a PHE as evaporator in ORC in detail. NSGA-II was used in MATLAB to optimize the evaporator, and decision variables were selected as length and width of plate and spacing between plates. There were two objective functions, and both of them was related to evaporator: minimum cost of evaporator and minimum pressure drop through evaporator. Degree of superheating was taken as zero because they used R245fa and it is dry fluid; hence, calculation of single vapor phase segment in the evaporator was unnecessary. Both functions have opposite manner; thus, Pareto frontier was obtained at the end of the study. Furthermore, sensitivity analyses were done, and plate length has maximum influence on the objective functions. Lastly, pressure drop should be kept between 30 and 40 kPa in order to provide the highest net output power. By the way, according to their claim, there was no any study about particularly optimization of PHE before this research [66].

Maraver *et al.* [19] studied to suggest an optimization way for different energy sources such as geothermal, solar, waste heat for ORC. The best performance for different working fluids were detected according to exergy efficiencies. The working fluids were determined considering usage of the commercial brands; R134a, R245fa,

octamethyltrisiloxane, nPentane, Solkathern and Toluene were investigated. Direct search optimization method was used in EES using turbine inlet pressure and degree of superheating as decision variables. By the way, thermal conductance and size parameter (Eq. 1.2) were taken to be performance indicators for evaporator and turbine respectively. According to the results, increasing in degree of superheating decreased exergy efficiency [19].

Calise *et al.* [67] used shell&tube heat exchanger in an ORC system, and their main aim was to investigate the geometrical optimization of the exchanger. Additionally, off-design conditions were examined, and the importance of the mass flow rate of brine was emphasized so as to obtain high thermal efficiency. NTU was used to understand the overall performance of the heat exchanger, and EES was used to simulate and optimize the system. By the way, total cost was selected as objection function [67].

Wang *et al.* [68] elaborately investigated an ORC for waste heat recovery with optimization. They firstly made sensitivity analysis to determine the decision variables, and inlet pressure and temperature of turbine, PPTD and approach temperature difference in evaporator and condenser temperature difference were selected as decision variables. The objectives of the study were to maximize exergy efficiency and to minimize total capital cost. NSGA-II was used to optimize the system in MATLAB with REFPROP (NIST), and PHE was preferred. Therefore, Pareto frontier was obtained at the end of the study. Then, the optimal point in the frontier was also determined [68].

Kazemi *et al.* [69] compared 3 different ORC configurations and optimized the systems. By the way, one of them was a novel arrangement. Shell&tube heat exchanger was preferred and the heat transfer area was calculated using Kern's method. Exergy efficiency and specific investment cost were used as objective functions for thermodynamic and economic optimization separately. Then, multi-objective function was also applied integrating both of them via GA. Peng-Robinson equation of state was used to calculate real fluid properties for R123, and degree of superheating, PPTD in evaporator and evaporation and regenerator temperatures were taken as decision variables. According to results, the new configuration had better results both thermodynamically

and economically compared to basic ORC. Besides, the new system was investigated as an investment for 3 countries: Iran, U.S.A. and France [69].

Liu *et al.* [70] concentrated on zeotropic working fluid for geothermal ORC, and mixture of R600a and R601a was used. Optimization of the system was done for three different well temperatures, but the re-injection temperature was same and fix for each case. The zeotropic fluids provided better performance rather than pure working fluids regarding operation conditions according to the outcomes [70].

Astolfi *et al.* [71] examined ORC system for geothermal within integrated two papers, and they investigated only thermodynamic performance of the system in the first part. Additionally, economic analyses were added into the first model in the second part of the work. They used MATLAB and REFPROP (NIST) to construct a numerical ORC model, and turbine efficiency did not accepted as constant in contrast with most of the related studies. Rotational speed, average diameter and number of stages of turbine were adjusted as variable to estimate the turbine efficiency for each operation, however, interestingly they preferred axial turbine instead of radial inflow turbine because of the reliability of correlations. Besides, Lorentz efficiency was used as an ideal situation like Carnot efficiency, but it is dependent on geothermal source, ambient and re-injection temperatures. There were six decision variables, and they were optimized through two steps. Four of them firstly found in outer level, and then the rest of them were optimized in inner level. Well drilling cost was also considered for economic calculation, and it had a crucial influence according to the results [71].

Wang *et al.* [72] optimized evaporation and condensation temperatures, PPTD in evaporator and condenser for a subcritical waste heat recovery ORC using payback period as objection function. A country was actually determined before to begin the process because the electricity price alters regarding country. 5°C was accepted as a lower limit for PPTD, and air cooling fan was used in the simulation. Additionally, effect of working fluids was investigated for different temperature level, and R236a, R245fa and R113 were suggested to use for approximately 150°C , 200°C and 250°C of heat sources respectively [72].

Turbine inlet temperature and pressure, PPTD and approach temperature difference in evaporator were optimized with respect to the ratio of net output power and total heat transfer area for an ORC by Wang *et al.* [73]. PHE was preferred, and GA was used for optimization process with 30 generations. The simulation code was written in MATLAB and REFPROP (NIST) was used for fluid properties. Besides, three working fluids (isobutane, R123, R245fa) were tried in the simulation in order to detect the best one. According to results, isobutane provided the best performance for the ORC, and PPTD and approach temperature difference usually stayed around the top values of their ranges. Finally, requirement of multi-objective optimization was emphasized for ORC system in the study since whole system is very complicated as much as not to examine using single objective [73].

Li *et al.* [74] investigated a subcritical ORC for waste heat source from industrial boilers in view of economy. Evaporation temperature, PPTD in evaporator and condenser were examined with respect to the electricity production cost for different heat source temperatures using EES. Overall heat transfer coefficient was calculated using Kern's formula and the coefficient in working fluid side was accepted as very small compared to flue gas side. Only derivations were preferred to optimize the system instead of global search algorithms such as GA. Besides, nine possible working fluids for low grade temperature were tried for the system, and establishing a relation between critical temperatures of the fluids and their economic performances was particularly tried. PPTD in evaporator was found very important for the total irreversibility in the system because the biggest irreversibility source is evaporator in the ORC and it was directly changed with PPTD in the evaporator according to results [74].

An original study about determination of optimal heat source temperature was done by Liu *et al.* [75]. The optimal temperature of the heat source just depended on PPTD in vapor generator and evaporation pressure according to the study, and the analytical approach was tested and approved with a case study for 140°C heat source. The maximum pressure limit was determined as 30 bar in the study. The relation between optimal heat source temperature and PPTD in evaporator was indicated with an equation at the end of the study [75].

Zhu *et al.* [76] studied on optimization of ORC and analyzed the system with entropy, availability and entransy using R123 and nPentane as working fluids. Entropy generation, availability destruction, entransy loss and entransy dissipation were used to optimize the system. The entransy theory is actually based on electrical storage energy in a capacitor, and then it is used to produce energy. Therefore, it means the potential energy of heat in heat exchanger; it is improved to establish optimum thermal systems. Besides, the main purpose of the research was to obtain maximum output power. Three case studies were numerically done to determine the entransy, exergy and entropy theories' relations with the main aim in EES. According to results, entransy loss rate was just used as an indicator of maximum output power instead of others [76].

In addition to these, Rahbar *et al.* [77] focused on geometric parameters of a RIT with optimization of ORC system. They used GA in the study to optimize the system in order to maximize the turbine total-to-static efficiency in EES, and the inlet blade angle order to obtain more compact sizes with high specific work capacity. Losses models were applied addition to conservation of mass, momentum and energy to model impeller, volute and nozzle. Eight working fluids were also tested, and higher turbine isentropic efficiency were reached compared to previous reported studies. R152a provided the best performance with 7 degrees of superheating, but if superheated vapor is not desired, the best working fluid choice was determined as isobutane regarding the turbine efficiency results. By the way, they aimed to make an experiment for RIT as a future work [77]. Rahbar *et al.* [78] also made another similar research for small-scale RIT used in ORC. However, there were some differences compared to [77]. They used DIRECT algorithm to optimize the ratio of cycle net electricity generation to rotor size of turbine (similar to PF in Eq. 1.5) using six working fluids. According to the outcomes, isobutane provided the best value for the objection function [78]. Shortly, the main goal of both studies was to use dynamic turbine efficiency instead of constant one so as to simulate more accurately.

Andreasen *et al.* [79] concentrated on the working fluid selection for ORC; pure and mixture working fluids were compared using GA to maximize the net output power in MATLAB. The evaluation was predicated on crucial indicators of the system such

as thermal conductance, turbine size parameter and pressure levels for heat exchanger, expander. Temperature and pressure at inlet of the turbine and outlet temperature of the brine were selected as decision variables addition to the mole fraction of mixture. By the way, the study was the seldom example for using axial turbine as the expander. The mixture provided better performance values as a result of the study, and brine inlet temperature should be around half of the critical temperature of the working fluid to obtain the maximum net output power [79].

A multi-objective research about subcritical ORC was studied by Xiao *et al.* [80] due the fact that they claimed there was a lack of usage multi-objective approach in the literature. Total exergy destruction, total system cost and net output power were firstly used as single objective functions, and then they were collected in a multi-objective function using linear weighted function. Addition to these objections, evaporation and condensation temperatures were also optimized in the system as decision variables. Another argue in the paper was there was a shortage for adding economical effect of the expander, and the study meet the need as well as usage of multi-objective optimization. Total heat transfer coefficients for both evaporator and condenser were accepted as constant to evaluate the total heat transfer area. Both of pure and mixture working fluids were tested in the study, and zeotropic mixtures did not always provide higher efficiency compared to pure working fluids according to results [80].

Coşkun *et al.* [11] specifically focused on geothermal energy in Turkey, and resources in Simav, Kütahya were examined in the study to determine the best cycle type for maximum power production. Kalina cycle, double flash cycle, combined cycle and binary cycle were investigated optimizing turbine inlet pressure to reach the maximum net output power. Addition to this, economic indicators were considered, and drilling cost was especially taken into account because it dominates the installation cost for geothermal applications. According to results, Kalina cycle was the best choice for the geothermal area in view of net output power and economy, for example, the payback period of the Kalina cycle was calculated as 5.8 years. On the other hand, binary cycle could repay the investment in 9 years [11]. The same working group was made another very similar research for Simav region again, and the outcomes of both studies

supported each other [81].

Li *et al.* [82] made an experimental study to investigate the effect of the regenerator on the overall cycle efficiency. R123 was used as the working fluid, and its mass flow rate was examined during the experiments. A bypass system was added to the ORC so as to simulate turbine during start and stop procedure to protect the expander. 130°C of heat source was used, and 6 kW of power was generated. Additionally, the total energy efficiency increased by 1.83% thanks to the regenerator according to the outcome [82].

Ergün *et al.* [83] studied on a geothermal ORC in Tuzla, Çanakkale in aspects of exergy and exergoeconomy. The plant produced 7.5 MWe. They emphasized importance of using exergy to analyze thermal systems, and they advocated the first law of thermodynamic could not show the actual performance. Instead, the second law of thermodynamic was absolutely more appropriate to examine the potential of a thermal system. According to results, evaporator was biggest availability destruction reason, and turbine followed it [83].

Liu *et al.* [84] elaborately made sensitivity analysis of system parameters to understand their influences on the performance of ORC. Degree of superheating, PPTD in evaporator and condenser, evaporation temperature, efficiencies of expander and pump and working fluids were investigated as system parameters regarding net output power, total energy efficiency, size parameter in Eq. 1.2 and overall heat transfer capacity. Orthogonal design was applied to the system, and approximately three discrete values were tested for each system parameter. There were 18 trials in the study. By the way, RIT was used in system, and the most significant parameter was determined as evaporation temperature at the end of the study. Brine temperature did not influence the size of the turbine, but it affected the net output power; thus, SP was changed with the brine temperature according to the results. The outcomes of the research were crucial to determine decision variable for any thermodynamic and economic optimization of ORC [84].

Li *et al.* [85] proposed a novel ORC system which uses two stage evaporators to reduce the irreversibility of the system. The brine was taken into the system with two temperature and pressure levels; thus, the total irreversibility and net power output could be decreasing and increasing respectively. Objective function was determined as ratio of the net power generation to overall total conductance. R245fa was used as working fluid. The temperature of re-injection was determined to be higher than 70°C to eliminate fouling effect. According to the results, the thermal efficiency constantly increased with evaporation temperature; on the other hand, the exergy efficiency had a pick point, in other words maximum value, if its movement was examined regarding the evaporation temperature. Addition to these, main irreversibility sources in ORC were evaporator and turbine [85].

Pang *et al.* [86] experimentally studied to select ideal composition of R245fa and R123 for an ORC. Shoa *et al.* [56] and Li *et al.* [82] also used only R123 in their experiments; on the other hand, Wang *et al.* [45] used merely R245fa as working fluid in the experiments. Therefore, this study carried a step further the experimental analysis. Two different fractions and pure fluids were tested for two different heat source temperatures (110°C and 120°C) in experiments to obtain maximum net output power. Degree of superheating at the inlet of the expander and the mass flow rate of the working fluid were investigated to reach the objection. According to results, R245fa solely had the best choice for 110°C of heat source; on the other hand, one of the tested zeotropic mixtures provided the highest power generation. Additionally, approximately 0.15 kg/s of working fluid mass flow rate were determined as optimum for each kind of working fluid whatever the temperature of the heat source [86].

Kanoglu *et al.* [87] made another research similar to previous study [81], and they investigated exergy destruction sources and exergy efficiencies of the each component separately using real power plant data. According to the shared real data, isobutane was used as working fluid, and air cooling system was preferred in ORC system. By the way, the temperature of the brine was 160°C and 90°C at inlet and outlet respectively. Then, the relation between temperature and pressure at turbine inlet, condensation pressure and the first and second law efficiencies, net output power and re-injection

temperature were demonstrated in sensitivity diagrams as outcomes of the study [87].

Feng *et al.* [88] numerically studied on the comparison of using pure and mixture fluids as working fluid. Two objective functions were used to evaluate the system; hence, multi-objective optimization was applied, and the second law efficiency and levelized electricity cost were selected as the functions. Furthermore, the comparison of mixture and pure working fluids were indicated using R245fa and R227ea. NSGA-II was applied in MATLAB with REFPROP (NIST) to optimize degree of superheating, PPTD in evaporator and evaporation and condensation temperatures. Pareto frontier was obtained at the end of the study. Additionally, an ideal point was determined on the Pareto frontier using TOPSIS like the previous study of Feng *et al.* [47]. According to the results, mixture did not always provide high exergy efficiency rather than a pure fluid like the study of Xiao *et al.* [80]. Besides, availability efficiency increased by increasing evaporation temperature and decreasing condensation temperature. Lastly, an analytic relationship between two objective functions was established applying curve fitting in Pareto diagram [88].

Ventura *et al.* [89] used Python to create a new model for preliminary design of RIT. Brute-force algorithm was applied to search whole domain. Loss models were also added into Baines' model, and all study was came to the fruition in one-dimension. According to the model, flow and head (stage loading) coefficients were entered into the iterative code with rotational speed [89].

Quoilin *et al.* [90] directly investigated ORC thermo-economically using EES. Five different working fluids were used for the ORC to reach the optimum case; PHE and volumetric expander were used in the system. Thermal conductivity of plates was not considered to calculate overall heat transfer coefficient. By the way, R1234yf was particularly selected instead of R134a since R134a has high global warming potential. PPTD in evaporator and condenser were directly inserted. According to the results, n-Butane is the best fluid in aspect of both economic and thermodynamic [90].

Wei *et al.* [91] focused on the effect of environmental changes on the net output power of a waste heat recovery ORC system. R245fa was used as working fluid and the properties were calculated using REFPROP (NIST). The numerical simulations were done in Medica/Dymola. Exergy destruction was examined for each component separately, and a relation between exhaust gas mass flow rate and evaporator or turbine exergy destruction. As a result, the destructions of two components oppositely moved with respect to the mass flow rate [91].

Liu *et al.* [92] studied on determining optimal evaporation temperature for waste heat recovery ORC regarding the second law efficiency as an objective function. The effects of PPTD in evaporator, critical temperatures of working fluids and temperature of the heat source on optimal evaporation temperature were also investigated using 28 different working fluids. The optimization process was done using quadratic approximation method in EES because there is a unique maxima of the exergy efficiency changing evaporation temperature. Only saturated vapor enters into the expander was assumed, and hence, the inlet temperature of the turbine and evaporation temperature was accepted as same. According to the results, PPTD in evaporator should be around 5 K, and the evaporation temperature should be kept at under by 18 ± 5 K of the critical temperature. Therefore, the fluids have lower critical temperatures than 345 K were not convenient to use in subcritical ORCs [92].

Mago *et al.* [93] essentially concentrated on exergy destruction and its sources in an ORC system. R113 was used as working fluid. Exergy wheel diagram was used in order to perfectly depict the exergy destruction and exergy movement throughout the system. According to the outcomes, thermal efficiency and total exergy loss increased and decreased respectively increasing evaporation pressure. Additionally, evaporator and turbine were the first and second sources of availability destruction like the study of Ergün *et al.* [83].

Shengjun *et al.* [94] studied on the comparison the performance of subcritical and transcritical ORC system for 90°C of geothermal heat source. Additionally, the systems were simulated using different working fluids in MATLAB with REFPROP (NIST).

Optimization was achieved by means of multi-dimensional optimization technique using five objection functions: thermal, exergy and recovery efficiencies, the ratio of heat transfer surface area to produced power and LCOE. Single objective optimization was preferred separately for each objective function instead of multi-objective optimization. Recovery efficiency was defined as net output power per maximum producible power, and the maximum producible power was calculated using Carnot's approach. Shell&tube heat exchanger made by carbon steel was used in the study, and the area of the heat exchanger was used to observe total cost of the system. Furthermore, quantity of saving petroleum usage and preventing carbon dioxide emission by virtue of using geothermal energy to generate electricity were evaluated at the end of the study. According to results, R123 and R125 were suggested for subcritical and transcritical ORC systems respectively and subcritical system had a better thermal efficiency value than transcritical one for the certain source and sink properties in the study [94].

Pezzuolo *et al.* [95] investigated the optimum ORC plant arrangement with screening working fluid. Energetic, exergetic and economic performances were evaluated using GA Toolbox in MATLAB with REFPROP (NIST) to find the optimum configuration. Oil outlet temperature, evaporation pressure, expander inlet temperature and recuperator efficiency were selected as decision variables. Addition to these, PPTD in evaporator, condenser and recuperator was optimized. According to results, recuperative ORC was chosen as the best one, and Toluene provided better performance rather than other working fluids for all plant configurations [95].

Cayer *et al.* [96] firstly tried to optimize six performance indicators at the same time using maximum temperature and pressure as decision variables, and these were thermal and exergy efficiencies, net output power, total thermal conductance, total system cost and surface of heat exchangers. However, they concluded the trial with the claim that six independent indicators cannot be optimized at the same time, and it is impossible. Thus, multipoint optimization techniques were required for the optimization problem of ORC. EES used as software, and it has some present tools for optimization like Golden Search Optimization. However, Golden Search optimization was used to solve one degree of freedom problems; therefore, Powel Direct Search Method

was used to optimize the system with more than one degree of freedom. According to the results, the ideal working fluids were altered regarding the selected objective function or functions [96].

Fiaschi *et al.* [97] emphasized the importance of the preliminary design of a RIT for screening working fluid because both of them has a significant effect on each other. REFPROP (NIST) was used to determine the properties of working fluids because ideal gas assumption can not use for the expander calculation in ORC although the perfect gas model is suggested to use in RIT in Dixon's book [24], but the crucial difference from Dixon is that the fluid is around very near region to critical point. Therefore, the ideal gas assumption cannot be applied to model the turbine for ORC. The main aim was designed a 90° RIT. The numerical model was constructed in EES. According to results, the rotational speed was found in range of 32000-55000 rpm, and specific speed was calculated from 0.074 to 0.1 [97].

Main objective of the present study is to perform thermodynamic analyses of ORCs having three different heat source temperatures (90, 120 and 150°C) with economic perspective via multi-objective optimization. This extensive study employs 28 different working fluids. This thesis is essentially divided into two parts. The first part involves analysis and optimization of ORC that includes the turbine design and the thermal conductance of the heat exchangers. A preliminary RIT design has been developed with a detailed loss model and dynamic turbine stage efficiency which increases the accuracy when compared to models that use constant turbine efficiency. Effects of cycle parameters and turbine design are studied. The objective functions are selected as total thermal conductance per net power output and PF in the first part. Simultaneous use of these objective functions together with economic considerations result in optimized performance. Specific speed, condensation pressure, pressure ratio at the turbine stage, the radius ratio of nozzle inlet and outlet, pinch point temperature difference (PPTD) in the evaporator and degree of superheating (SHEAT) at the turbine inlet are used as decision parameters. Pareto frontier is obtained because there is a conflict between variations of two objective functions with respect to the decision variables. For each heat source and working fluid pair, TOPSIS is used to determine

an ideal one among Pareto solutions. TOPSIS solutions yielding maximum net power outputs is selected as a group of ideal working fluids for each heat source.

In the second part of the study, a detailed heat transfer model for PHE is included addition to RIT model that includes loss models. Specific speed, condensation pressure, expander pressure ratio, the radius ratio of the nozzle inlet and outlet, pinch point temperature difference in the evaporator (PPTDevap), degree of superheating at the turbine inlet (SHEAT), the length, the width of the plates and the spacing between the plates in PHE are the decision variables. Pareto frontiers are obtained as the outcome of the optimization. TOPSIS method is used to select a single ideal solution from the Pareto frontier for each working fluid and heat source temperature. Sorting of the ideal solutions with respect to their net power outputs, a group of ideal working fluids with the highest power output are determined for each heat source temperature similar to the first part. This methodology for working fluid screening is based on ORC installation cost indicators which is appropriate for many different type of heat sources except the geothermal because the drilling cost may dominate the ORC installation cost [16].

2. ORC SYSTEM WITH PRELIMINARY RADIAL INFLOW TURBINE DESIGN

A simple ORC has four components: evaporator, expander, condenser and pump. The ORC system is shown together with the T-s diagram of R245fa in Figure 2.1. The working fluid is vaporized in the evaporator and the vapor has high enthalpy and pressure at the turbine inlet. Some of this enthalpy is converted into power by the turbine and the pressure decreases whilst the vapor expands. Low pressure vapor condenses in the condenser and the fluid is pressurized in the pump before it enters the evaporator, and the cycle is completed [6].

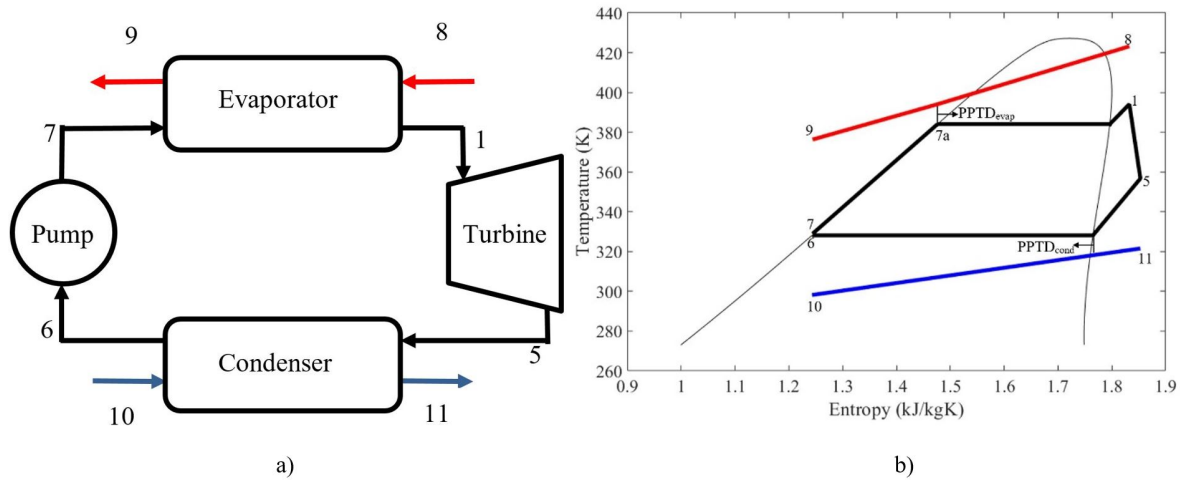


Figure 2.1. a) ORC system, b) T-s diagram with R245fa.

2.1. Thermodynamic Analysis

The assumptions are listed below:

- There is no pressure drop in pipes, in the evaporator and in the condenser,
- change in potential and kinetic energies are ignored except for the kinetic energy change in the turbine,

- all components operate at steady-state,
- heat losses are neglected, in other words adiabatic assumption is made,
- flow is one dimensional and entirely fully developed,
- fouling influences are neglected,
- saturated liquid is obtained at the condenser outlet.

Energy balance for the evaporator:

$$\dot{m}_{hw}(h_8 - h_9) = \dot{m}_{wf}(h_1 - h_7) \quad (2.1)$$

Energy balance for the condenser:

$$\dot{m}_{cw}(h_{11} - h_{10}) = \dot{m}_{wf}(h_5 - h_6) \quad (2.2)$$

PPTD value is evaluated for both the evaporator and the condenser, and assuming the pinch point is located on the saturated liquid line (point 7a in Figure 2.1), the formulations for the evaporator are given below (the same process is applied for the condenser) [98]:

$$\dot{m}_{hw}(h_8 - h_{pp}) = \dot{m}_{wf}(h_1 - h_{sat}) \quad (2.3)$$

$$\dot{m}_{hw}(h_{pp} - h_9) = \dot{m}_{wf}(h_{sat} - h_7) \quad (2.4)$$

Pinch point can also be located at 7 in Figure 2.1 [99,100], and the location of the pinch point is checked in the study.

Thermal conductance, UA , is evaluated using logarithmic mean temperature difference (LMTD) as [44, 47]:

$$LMTD = \frac{(T_{hot,in} - T_{cold,out}) - (T_{hot,out} - T_{cold,in})}{\ln\left(\frac{T_{hot,in} - T_{cold,out}}{T_{hot,out} - T_{cold,in}}\right)} \quad (2.5)$$

$$UA = \frac{\dot{Q}}{LMTD} \quad (2.6)$$

Eq. 2.5 and 2.6 are applied to the evaporator and the condenser, and the total of the UA values for the evaporator and the condenser is $(UA)_{tot}$. In many studies such as [6, 19, 44, 79], thermal conductance is used as an indicator for the heat exchanger size. Among turbine, pump and heat exchangers in the cycle, the dominant installation cost is due to the heat exchanger. The installation cost of heat exchanger increases with the size. Therefore, thermal conductance is used as an economic indicator in the first part of this study.

The power input to the pump is calculated as:

$$\dot{W}_p = \dot{m}_{wf}(h_7 - h_6). \quad (2.7)$$

The power generated by the turbine is calculated as:

$$\dot{W}_t = \dot{m}_{wf}(h_1 - h_5). \quad (2.8)$$

The cycle thermal efficiency is calculated as follows:

$$\eta_{thermal} = \frac{\dot{W}_t - \dot{W}_p}{\dot{m}_{hw}(h_8 - h_9)} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}. \quad (2.9)$$

2.2. Limitations for Real ORC Applications

Operating conditions and limitations are significant for durability of components. For example, there should be no liquid in the turbine and vapor in the pump. In the optimization process, the inlet conditions to these units should be selected accordingly [12]. Additionally, to ensure safety 44 bar is suggested as the maximum pressure for engineering applications [9]. The condensation pressure is the lowest pressure in the cycle, and the minimum value of the condensation pressure is assumed to be 1.5 bar to prevent leakage from the ambient [8]. Maximum rotational speed is specified as 24000 rpm for the turbine [5]. In order to avoid choked flow which is the source of major loss of efficiency the Mach number at the outlet is kept below unity [20], the maximum value of Mach number at turbine impeller inlet is chosen as 1.5 [101]. The lowest value of the outlet temperature is set to 70 °C because temperatures under 70 °C can violate the assumption about the fouling resistance for some ORC applications [35, 98, 102]. The last crucial point is about heat exchangers, and a lower PPTD value is desired to decrease the irreversibility. However, the value should be high enough to provide heat transfer sturdily; 3 °C is determined as constant PPTD and lower limit of PPTD for the condenser and evaporator respectively [8].

Calculations made in RIT design may have some errors as a result of use of models that are based on gas turbines which operate on air. In the absence of experimental data for turbines operating with organic fluids, this is an approach used in the literature [37, 54, 77, 78, 101].

2.3. Mean-line RIT Design

Generally half of the time that is spent to design a turbine is allocated to the preliminary design process [26], and it is a very important step in the model of a turbine. It also provides a basis for the 3-D analysis. Axial turbine researches are very common in the literature [38], but RIT studies are relatively less. Although, properties of working fluids have critical impact on turbine parameters, a great number of resources of RIT is based on ideal gas assumption. Especially for the cycle states which are

close to the critical point, ideal gas assumption distorts the design from reality [97]. Therefore, real gas properties are used in the present study using REFPROP v9.1 [103] in MATLAB R2017b.

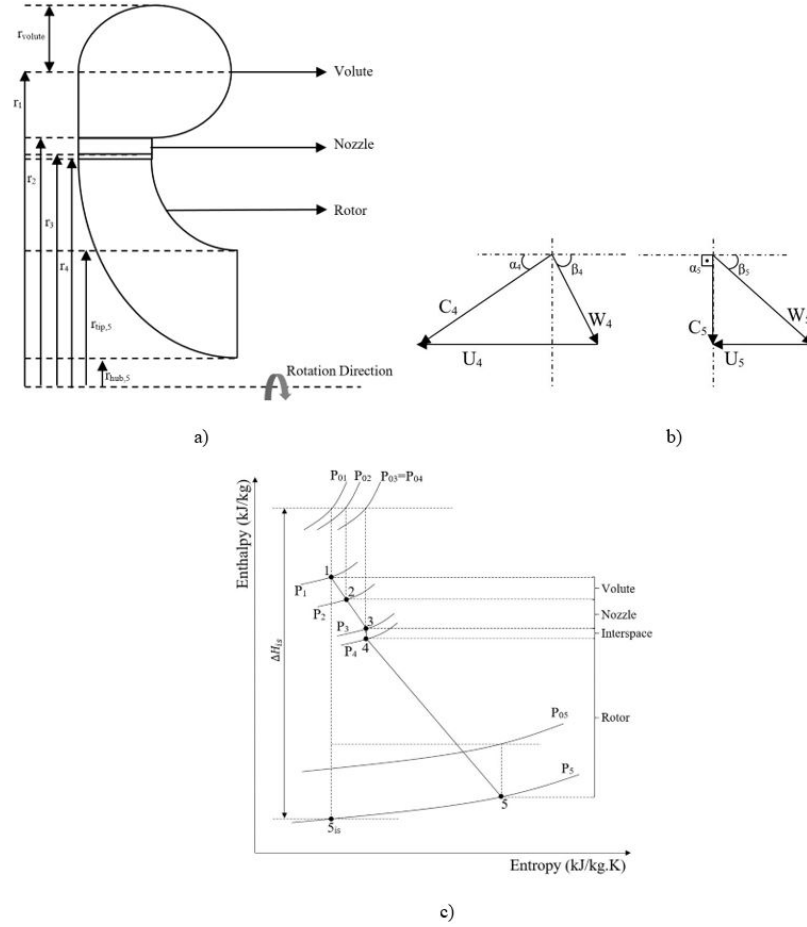


Figure 2.2. a) RIT geometry and its components, b) Velocity triangles at the rotor inlet and outlet, c) The Mollier Diagram.

In this study, three main parts: volute, nozzle and rotor are modeled, and diffuser is ignored to decrease the computational cost as it is done many studies in the literature [38, 54, 77, 78, 101]. These parts, example velocity triangles at the rotor inlet and outlet and the Mollier diagram are plotted in Figure 2.2. Loss models explained later are used with mass, momentum and energy equations and the turbine stage efficiency is calculated iteratively. The convergence tolerance is determined as 10^{-6} in the study for the turbine stage efficiency. The kinetic energy is taken into account in

the turbine analysis due to high velocity values; thus, stagnation enthalpies, pressures and temperatures are evaluated as well. The general flow of the design procedure is demonstrated in Figure 2.3.

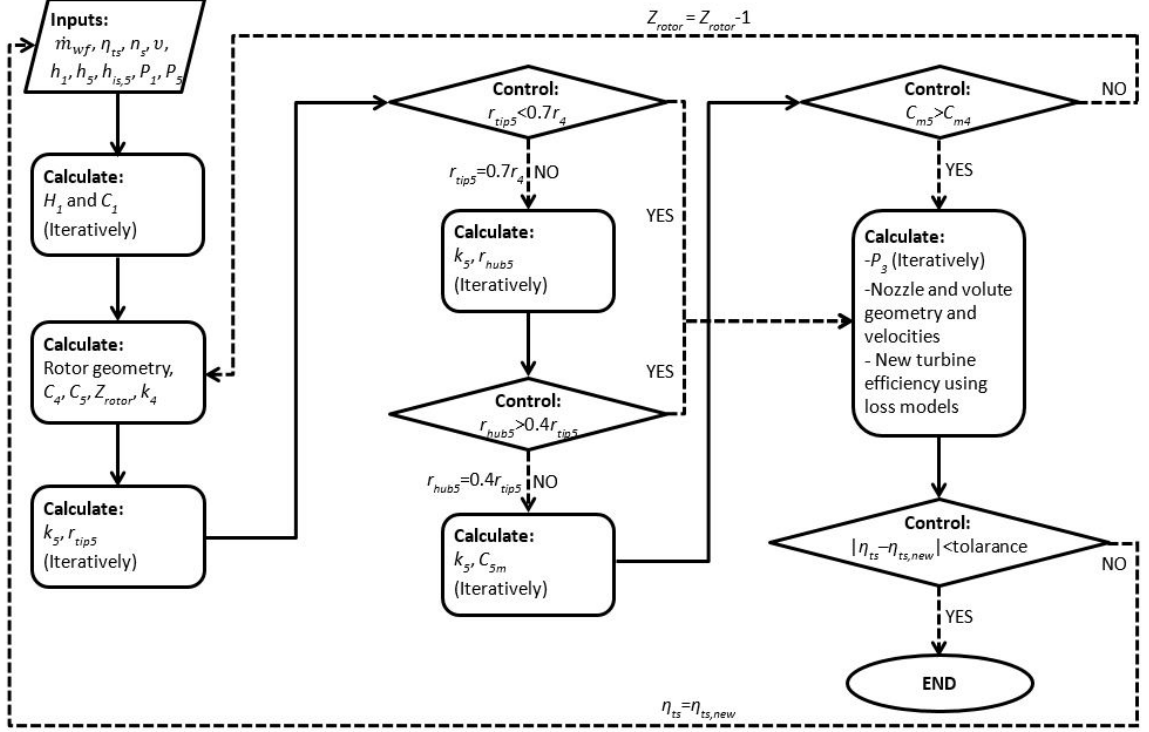


Figure 2.3. Flow chart of the RIT design procedure.

Euler's turbine equation is used to evaluate power generated by the turbine [24].

$$\dot{W}_t = \dot{m}_{wf}(C_{\theta 4}U_4 - C_{\theta 5}U_5) \quad (2.10)$$

Actual enthalpy difference between the turbine inlet and outlet is related with the velocities as follows:

$$\Delta h_{act} = C_{\theta 4}U_4 - C_{\theta 5}U_5 \quad (2.11)$$

The relation between the turbine power and velocities can be derived by combining Eq. 2.10 with geometric relations in Figure 2.2 or with conservation of rothalpy [24].

$$\dot{W}_t = \dot{m}_{wf} \left[\frac{U_4^2}{2} - \frac{U_5^2}{2} + \frac{C_4^2}{2} - \frac{C_5^2}{2} - \left(\frac{W_4^2}{2} - \frac{W_5^2}{2} \right) \right] \quad (2.12)$$

The blade tip speed at the rotor inlet, U_4 , is always bigger than the speed at the outlet, U_5 , for RITs. In Eq. 2.12, $(U_4 - U_5)$ is always positive indicating a contribution to turbine power generation.

Total-to-total efficiency is used for multi-stage turbine models because the kinetic energy of the fluid at the exit stage is utilized in the next stage. However, since the RIT has a single stage in the present study, the total-to-static efficiency is preferred. The total-to-static efficiency is calculated as follows [24]:

$$\eta_{ts} = \frac{H_1 - H_5}{H_1 - h_{is,5}}. \quad (2.13)$$

Total-to-static velocity ratio, v , [26] is

$$v = \frac{U_4}{c_0} \quad (2.14)$$

where c_0 is spouting velocity, and it is previously defined in Eq. 1.7. It is specifically calculated for this RIT model as follows [24]:

$$c_0 = \sqrt{2\Delta H_{is}} = \sqrt{2(H_1 - h_{is,5})}. \quad (2.15)$$

The formulation of non-dimensional specific speed, n_s , is given before in Eq. 1.6, and it is modified this turbine model as [24]:

$$n_s = \frac{\Omega \sqrt{\dot{m}_{wf} / \rho_{05}}}{\sqrt[4]{\Delta H_{is}^3}}. \quad (2.16)$$

Aungier [26] proposed a simple relation between v and n_s as follows:

$$v = 0.737 n_s^{0.2}. \quad (2.17)$$

2.3.1. Rotor Design

The first step in the turbine modeling is the rotor design [25, 26]. The mass flow rate of the working fluid, the isentropic static enthalpy difference through the turbine stage, the initial guess for the turbine stage efficiency and static state properties at the turbine inlet and outlet such as pressure, temperature, density and entropy are obtained from the general cycle model and are used as the inputs as shown in Figure 2.3. Swirl is set to zero at the turbine outlet, $C_{\theta 5} = 0$, to minimize the exit loss, and the 90° RIT is modelled in the present study like many studies such as [26, 38, 54] in the literature. Furthermore, stagnation enthalpy is conserved in volute and nozzle stages thanks to the adiabatic assumption [26, 54, 77, 78, 97].

First, stagnation enthalpy and pressure are iteratively calculated with the velocity at the turbine inlet. Then, rotational speed is found using Eq. 2.16, and U_4 is evaluated using Eq. 2.14, 2.15 and 2.17. The inlet radius of the rotor, r_4 , is calculated as follows:

$$r_4 = \frac{U_4}{\Omega} \quad (2.18)$$

The tangential component of the absolute flow velocity, $C_{\theta 4}$, at the rotor inlet is calculated using Eq. 2.10, 2.13 and 2.14 with the assumptions of equal stagnation enthalpy from the volute to the rotor inlet, $H_1 = H_4$ [26] and no swirl, $C_{\theta 5} = 0$ [54].

$$C_{\theta 4} = \frac{U_4 \eta_{ts}}{2v^2} \quad (2.19)$$

Absolute flow angel at the rotor inlet, α_4 , is found from the following relation [26]:

$$\alpha_4 = 10.8 + 14.2n_s^2. \quad (2.20)$$

The meridional component of the absolute flow velocity at the rotor inlet, C_{m4} , is calculated using the geometry with Eq. 2.19 and 2.20. Additionally, stagnation pressure at the rotor inlet, P_{04} , is evaluated as in [26].

$$P_{04} = P_{01} - \frac{\rho_{01} \Delta H_{is} (1 - \eta_{ts})}{4} \quad (2.21)$$

With the known stagnation enthalpy and pressure values s_4 is found. The static state properties are calculated from s_4 and h_4 since h_4 is obtained from the stagnation enthalpy and the absolute velocity.

Following Glassman [104], number of rotor blades, Z_{rotor} , is directly calculated using the absolute flow angle.

$$Z_{rotor} = \frac{\pi}{30} (110 - (90 - \alpha_4)) \tan(90 - \alpha_4) \quad (2.22)$$

The blade thickness at the rotor alters from the inlet to the outlet an thickness values are obtained from the following relations:

$$t_4 = 0.04r_4, \quad (2.23)$$

$$t_5 = 0.02r_4. \quad (2.24)$$

Blockage factor at the rotor inlet, k_4 , can be calculated; hence, the blade width at the inlet, L_4 , is found from continuity.

$$k_4 = 1 - \frac{t_4 Z_{rotor}}{2\pi r_4} \quad (2.25)$$

$$L_4 = \frac{\dot{m}_{wf}}{2\pi r_4 C_{m4} \rho_4 k_4} \quad (2.26)$$

The rotor outlet hub radius, r_{hub5} , and absolute velocity, C_5 , are defined as [26]:

$$r_{hub5} = 0.185r_4, \quad (2.27)$$

$$C_5 = C_{m5} = C_{m4} \left[1 + 5 \left(\frac{L_4}{r_4} \right)^2 \right]. \quad (2.28)$$

The values which are obtained from Eq. 2.27 and 2.28 can be changed according to the constraints indicated in Figure 2.3 [54,105]. Outlet tip radius, r_{tip5} , is calculated together with the blockage factor at the outlet, k_5 , using equation given below till k_5 converges.

$$r_{tip5} = \sqrt{\frac{\dot{m}_{wf}}{2\pi C_{m5}\rho_5 k_5} + r_{hub5}^2} \quad (2.29)$$

$$r_5 = 0.5(r_{tip5} + r_{hub5}) \quad (2.30)$$

$$W_{\theta5} = -\Omega r_5 \quad (2.31)$$

$$\beta_5 = \left| \arctan\left(\frac{C_{m5}}{W_{\theta5}}\right) \right| \quad (2.32)$$

$$k_5 = 1 - \frac{t_5 Z_{rotor}}{2\pi r_5 \sin\beta_5} \quad (2.33)$$

The result of Eq. 2.33 is used in Eq. 2.29 during the iterative process [54]. If $r_{tip5} > 0.7r_4$, $r_{tip5} = 0.7r_4$ is assumed [39, 54] and r_{hub5} is calculated with continuity; thus, Eq. 2.27 lost its validness. The same logic is applied for the iterative process. If $r_{hub5} < 0.4r_{tip5}$, $r_{hub5} = 0.4r_{tip5}$ is assumed [54] and C_5 is calculated with continuity again; thus, Eq. 2.28 also lost its validness. The last control step for the rotor outlet is done by testing the condition of $C_5 > C_{m4}$ [26] and if it is not satisfied, the whole rotor outlet process is remade with a missing blade. The process is easily seen in Figure 2.3. The final geometric information about the rotor is the axial length, L_{ax} , and it is related to width of the outlet, L_5 , [26]:

$$L_5 = r_{tip5} - r_{hub5}, \quad (2.34)$$

$$L_{ax} = 1.5L_5. \quad (2.35)$$

Lastly, choked flow is not desired as mentioned before, and hence, Mach number, Ma , is controlled at the impeller inlet and outlet. However, absolute fluid velocity is higher than relative one at the inlet; on the other hand, the opposite condition is valid for the outlet. Therefore, Eq. 2.36 and 2.37 are used to check Ma [24]:

$$Ma_4 = \frac{C_4}{\alpha_4}, \quad (2.36)$$

$$Ma_5 = \frac{W_5}{\alpha_5}. \quad (2.37)$$

As aforementioned, $Ma_5 < 1$ should be satisfied to prevent choked flow at the turbine outlet. Additionally, $Ma_4 < 1.5$ is determined as a restriction for the rotor inlet coherent with [101].

2.3.2. Nozzle and Volute Design

The nozzle outlet radius, r_3 , is found from a relation which is based on the rotor inlet geometric parameters [26].

$$r_3 = r_4 + 2L_4 \sin \alpha_4 \quad (2.38)$$

Tangential component of the absolute flow velocity at the nozzle outlet, $C_{\theta 3}$, is evaluated from conservation of angular momentum as shown below [54]:

$$C_{\theta 3} = C_{\theta 4} \frac{r_4}{r_3}. \quad (2.39)$$

Then, the interspace between the nozzle and the rotor is modeled as ideal; thus, there is no loss and irreversibility. Therefore, entropy and stagnation pressure at the

nozzle outlet are known, and the meridional component of the absolute flow velocity, C_{m3} , is iteratively calculated using continuity. The blockage factor, k_3 , is assumed to be 0.9 and the width of the nozzle is assumed as constant and equal to the rotor inlet width, L_4 [78]. The nozzle inlet radius, r_2 , is obtained from a given ratio, r_{23} , between r_2 and r_3 [78, 97]:

$$r_2 = r_{23}r_3. \quad (2.40)$$

The volute cross-section is assumed as circular as in [78], and the absolute flow velocity at the volute inlet, C_1 , is known as mentioned before. Therefore, the volute inlet area can be calculated from continuity and r_1 is found; thus, r_{max} is obtained as follows:

$$A_1 = \frac{\dot{m}_{wf}}{\rho_1 C_1}, \quad (2.41)$$

$$r_{volute} = \sqrt{\frac{A_1}{\pi}}, \quad (2.42)$$

$$r_1 = r_2 + r_{volute}, \quad (2.43)$$

$$r_{max} = r_1 + r_{volute}. \quad (2.44)$$

According to [25, 26], C_1 has only tangential component, and $C_{\theta 2}$ is found using conservation of angular momentum. Additionally, C_{m2} can be evaluated iteratively using continuity again:

$$C_{\theta 2} = C_1 \frac{r_1}{r_2}. \quad (2.45)$$

2.3.3. Loss Models and Turbine Efficiency Calculation

Eight different loss sources are considered; some loss models are empirical [25]. They are used in iterative evaluation of turbine stage efficiency.

The incidence loss is caused by the incidence angle, and it is very small when the inlet flow angle is around 90° [25, 26, 89, 106]. In this study, radial element blades are used for which the loss is equal to tangential kinetic energy as written below:

$$\Delta h_{incidence} = \frac{W_{\theta 4}^2}{2}. \quad (2.46)$$

Some of fluid may leak to the back side of the impeller, and it may stay between the impeller and the back plate of shroud. This situation results in a friction torque and a windage loss [25]. It sometimes called as disk friction loss and calculated as follows [40, 77]:

$$\Delta h_{windage} = k_f \frac{\bar{\rho}_{rotor} U_4^3 r_4^2}{4\dot{m}_{wf}}. \quad (2.47)$$

where

$$k_f = \frac{3.7 \left(\frac{\varepsilon_b}{r_4} \right)^{0.1}}{\sqrt{Re_{rotor}}} \text{ for } Re_{rotor} < 10^5, \quad (2.48)$$

$$k_f = \frac{0.102 \left(\frac{\varepsilon_b}{r_4} \right)^{0.1}}{Re_{rotor}^{0.2}} \text{ for } Re_{rotor} > 10^5, \quad (2.49)$$

$\bar{\rho}_{rotor}$ is the average of ρ_4 and ρ_5 , the densities at the inlet and the outlet respectively, and ε_b is the clearance between the rotor and the back plate. In the study, ε_b is taken from [38] as 0.0003 m. Additionally, Reynolds number is calculated for the rotor as

shown below [26]:

$$Re_{rotor} = \frac{\left(\frac{W_{tip5} + W_{hub5}}{2}\right) \rho_5 L_5}{\mu_5}. \quad (2.50)$$

The friction between fluid and the rotor surface causes a skin friction loss, and it is evaluated as follows [25, 89, 106]:

$$\Delta h_{skin} = f_{rotor} \frac{l_{h,rotor}}{D_{h,rotor}} \left[\frac{W_4 + \left(\frac{W_{tip5} + W_{hub5}}{2}\right)}{2} \right], \quad (2.51)$$

where

$$l_{h,rotor} = \frac{\pi}{2} \sqrt{\frac{\left(r_4 - r_{tip5} + \frac{L_4}{2}\right)^2 + \left(\frac{L_5}{2}\right)^2}{2}}, \quad (2.52)$$

$$D_{h,rotor} = \frac{1}{2} \left[\frac{4\pi r_4 L_4}{2\pi r_4 + Z_{rotor} L_4} + \frac{2\pi (r_{tip5}^2 - r_{hub5}^2)}{L_5 (\pi + Z_{rotor})} \right], \quad (2.53)$$

$l_{h,rotor}$ and $D_{h,rotor}$ are hydraulic length and diameter of the rotor, respectively. $l_{h,rotor}$ is also called surface length [89]. Another important parameter in Eq. 2.51 is the friction factor of the rotor, f_{rotor} , and it is found as follows:

$$f_{rotor} = f_c \left[Re_{rotor} \left(\frac{r_4}{r_c} \right)^2 \right]^{0.05}, \quad (2.54)$$

$$f_c = f \left(1 + 0.075 Re_{rotor}^{0.25} \sqrt{\frac{D_{h,rotor}}{2r_c}} \right), \quad (2.55)$$

r_c represents the mean curvature radius. f is the Fanning friction factor [89], $f = 16/Re_{rotor}$ for laminar flows and it can be found from the Moody Chart for turbulent flows [106]. However, Churchill [107] suggested only one equation to obtain f for all Reynolds numbers to be used especially in numerical calculations. Accordingly, in this study, f is calculated as:

$$f = 2 \left\{ \left(\frac{8}{Re_{rotor}} \right)^{12} + \left[\left(2.457 \ln \left(\frac{1}{(7/Re_{rotor})^{0.9} + 0.27\epsilon/L_5} \right) \right)^{16} + \left(\frac{37530}{Re_{rotor}} \right)^{16} \right]^{-1.5} \right\}^{1/12} \quad (2.56)$$

where ϵ is effective roughness and it is equal to 0.0002 m in the present study [78, 89].

The fluid which does not follow the main flow disturbs the flow and causes some secondary loss. Secondary loss always occurs through the rotor even if ideal inlet conditions exist [106]. Secondary loss is found as [106]:

$$\Delta h_{secondary} = \frac{2C_4^2 r_4}{Z_{rotor} r_c}. \quad (2.57)$$

Furthermore, the kinetic energy at the turbine outlet is considered as a loss in single stage turbines, and it is named as exit loss [89, 106]:

$$\Delta h_{exit} = \frac{C_5^2}{2}. \quad (2.58)$$

Some fluid will always leak from the clearance between the rotor and the shroud causing a pressure loss, and decreasing the efficiency. This kind of loss is called tip clearance loss, and it is obtained as follows [25, 78, 89]:

$$\Delta h_{clearance} = \frac{U_4^3 Z_{rotor}}{8\pi} \left(0.4\epsilon_r C_{ax} + 0.75\epsilon_{ax} C_r - 0.3\sqrt{\epsilon_{ax}\epsilon_r C_{ax} C_r} \right) \quad (2.59)$$

where

$$C_{ax} = \frac{1 - \left(\frac{r_{tip5}}{r_4} \right)}{C_{m4}L_4}, \quad (2.60)$$

$$C_r = \left(\frac{r_{tip5}}{r_4} \right) \frac{L_{ax} - L_4}{C_{m5}r_5L_5} \quad (2.61)$$

ε_r and ε_{ax} are radial and axial clearances respectively, and they are calculated following [78] as follows:

$$\varepsilon_r = \varepsilon_{ax} = 0.04L_5. \quad (2.62)$$

The nozzle loss is calculated as follows [77]:

$$\Delta h_{nozzle} = 4f_{nozzle}\overline{C}_{nozzle}^2 \frac{l_{h,nozzle}}{D_{h,nozzle}}, \quad (2.63)$$

where

$$l_{h,nozzle} = r_2 - r_3, \quad (2.64)$$

$$D_{h,nozzle} = \frac{1}{2} \left[\frac{8\pi r_2 L_4 (C_{m2}/C_2)}{4\pi r_2 + (4\pi L_4 r_2/\sigma)} + \frac{8\pi r_3 L_4 (C_{m3}/C_3)}{4\pi r_3 + (4\pi L_4 r_3/\sigma)} \right], \quad (2.65)$$

σ is the nozzle solidity, and it is equal to 1.35 [104]. $\overline{C}_{nozzle}$ represents the average absolute velocity at the inlet, C_2 , and at the outlet, C_3 . f_{nozzle} is calculated using Eq. 2.56 where L_4 is used instead of L_5 ; Reynolds number is calculated as follows [26]:

$$Re_{nozzle} = \frac{\rho_3 C_3 L_4}{\mu_3}. \quad (2.66)$$

The volute loss is found by multiplying a pressure loss coefficient, 0.1 [78], with kinetic energy at the nozzle inlet:

$$\Delta h_{volute} = 0.1 \frac{C_2^2}{2}. \quad (2.67)$$

In the calculations, the loss models are used iteratively to estimate the turbine stage efficiency. Total enthalpy reduction is calculated as follows:

$$\begin{aligned} \Delta h_{loss} = \Delta h_{incidence} + \Delta h_{windage} + \Delta h_{skin} + \Delta h_{secondary} + \\ \Delta h_{exit} + \Delta h_{clearance} + \Delta h_{nozzle} + \Delta h_{volute}. \end{aligned} \quad (2.68)$$

The stage efficiency is estimated using the following equation

$$\eta_{ts,estimation} = \frac{\Delta h_{act}}{\Delta h_{act} + \Delta h_{loss}}. \quad (2.69)$$

2.4. Validation

Two different models are developed and used together in this study. One of them is for the ORC and the pinch analysis; the other one is for the RIT. The results are checked with the ones available in the literature, and both models are validated separately. The results of the ORC model are compared to a case given in Kaşka's work [98], and the mean-line RIT model is validated with two cases given by Da Lio *et al.* [54]. The comparisons are tabulated in Tables A.1, A.2 and A.3 for the ORC and RIT models respectively. The relative errors show that both models are reliable in representing physical ORC and the RIT systems.

2.5. Working Fluid Screening

In this study, 28 different working fluids (HCs, HFCs and HFOs) which have zero ozone depletion potential (ODP) are used. Their properties are tabulated in Tables 2.1, 2.2 and 2.3 with their global warming potential (GWP) and ASHRAE classes respectively for dry, isentropic and wet fluids. In the later sections of the thesis, for each heat source, ten working fluids are selected from TOPSIS that yield maximum net power output. Then, out of these ten, the ORC analysis is performed in detail using the first four fluids.

2.6. Optimization

In this section, a multi-objective optimization is used to design an ORC with minimum thermal conductance per net power output and maximum performance factor (PF). Genetic algorithm (GA) is a stochastic global optimization technique, and it is based on biological evolution theory. GA is frequently applied to solve non-linear, multi-dimensional complex optimization problems; thus, it is a very convenient optimization technique for many engineering problems including ORC modelling [77, 101, 108].

NSGA-II is chosen in this study since it allows for multi-objective optimization. The main idea in this algorithm is to find non-dominated solutions, in other words Pareto solutions, which minimize the objective functions. Elitist approach is applied in NSGA-II which has some advantages over other optimization algorithms. Firstly, it is efficient and requires a small CPU time [45]. Secondly, it is more successful to find the best solution globally compared to traditional optimization techniques [108]. Lastly, initial guesses do not have an important effect on the final solution [77]. The similarity between NSGA-II and the biology can be explained with chromosomes and genes. Each decision parameter is like a gene, and their vectors for each different solution can be seen as a chromosome. There is a built-in function for NSGA-II in MATLAB R2017b. The goal of the built-in function is to minimize the objective functions. It is used for ORC applications in many studies [45, 51, 66, 68, 109].

Table 2.1. Dry working fluid list with thier properties.

Fluid	Molecular Weight (kg/kmol)	T_{cr} ($^{\circ}\text{C}$)	P_{cr} (kPa)	T_{bp} ($^{\circ}\text{C}$)	GWP (100 Years)	Safety Data
R1233zd(e)	130.5	166	3570	18	7	A1
R227ea	170	102	2926	-16	3220	A1
R236ea	152	139	3410	6	1200	A2
R236fa	152	125	3200	-2	9810	A1
R245ca	134	174	3925	25	726	n.a.
R245fa	134	153	3610	15	950	B1
Benzene	78	289	4906	80	~ 20	n.a.
Butane	58	152	3796	0	~ 20	A3
Cyclohexane	84	280	4075	81	~ 20	n.a.
Heptane	100	267	2736	98	~ 20	n.a.
Hexane	86	235	3034	69	~ 20	n.a.
Isobutane	58	135	3640	-12	~ 20	A3
Isohexane	86	225	3040	60	~ 20	n.a.
Isopentane	72	187	3378	28	~ 20	A3
Neopentane	72	161	3196	10	~ 20	n.a.
Octane	114	296	2497	126	~ 20	n.a.
Pentane	72	196	3360	36	~ 20	A3
Toluene	92	319	4126	111	~ 20	n.a.

Table 2.2. Isentropic working fluid list with thier properties.

Fluid	Molecular Weight (kg/kmol)	T_{cr} ($^{\circ}\text{C}$)	P_{cr} (kPa)	T_{bp} ($^{\circ}\text{C}$)	GWP (100 Years)	Safety Data
R1234yf	114	95	3382	-29	<4.4	A2
R1234ze(e)	114	110	3636	-19	6	A2
R134a	102	101	4060	-26	1430	A1
R365mfc	148	187	3266	40	850	n.a.

Table 2.3. Wet working fluid list with thier properties.

Fluid	Molecular Weight (kg/kmol)	T_{cr} ($^{\circ}\text{C}$)	P_{cr} (kPa)	T_{bp} ($^{\circ}\text{C}$)	GWP (100 Years)	Safety Data
R143a	84	73	3800	-47	4470	A2
R152a	66	112	4450	-24	124	A2
R32	52	78	5782	-52	716	A2
Propane	44	97	4251	-42	~ 20	A3
Propylen	42	89	4555	-47	~ 20	A3
Propyne	40	129	5626	-25	~ 20	n.a.

In multi-objective function optimization, a group of Pareto solutions is obtained instead of only one. Therefore, use of a decision-making technique such as TOPSIS is needed to select a solution from the group. Many researchers applied TOPSIS for multi-objective ORC studies such as [88, 109, 110]. Among Pareto solutions for each objective, the solution with the minimum value of the objective function is defined as the ideal solution, the solution with the maximum value of the objective function is defined as the non-ideal solution for that objective. In TOPSIS calculations, the main aim is to find the solution which is the nearest to the ideal and the most far from the non-ideal one simultaneously. First of all, objective functions are normalized to prevent biased final solution due to the order difference between the objective functions [111]. Then, the ideal and non-ideal (nadir) solutions are found for each objective function separately. Euclidean distances are calculated as follows [88, 110]:

$$ED_{ideal} = \sqrt{(Obj_{n1,ideal} - Obj_{n1})^2 + (Obj_{n2,ideal} - Obj_{n2})^2}, \quad (2.70)$$

$$ED_{nadir} = \sqrt{(Obj_{n1,nadir} - Obj_{n1})^2 + (Obj_{n2,nadir} - Obj_{n2})^2}, \quad (2.71)$$

$$Y = \frac{ED_{nadir}}{ED_{nadir} + ED_{ideal}} \quad (2.72)$$

Eq. 2.70, 2.71 and 2.72 are applied to all Pareto solutions, and the solution that has maximum Y is selected as the TOPSIS solution.

2.6.1. Application of NSGA-II to ORC

The coupled code (ORC and RIT) is run with NSGA-II with the purpose to minimize two objective functions simultaneously:

$$Obj_1 = \frac{(UA)_{tot}}{\dot{W}_{net}}, \quad (2.73)$$

$$Obj_2 = \frac{D^2}{\dot{W}_{net}} 10^3 = \frac{4r_4^2}{\dot{W}_{net}} 10^3. \quad (2.74)$$

The first objective function represents the total heat exchanger area in the ORC; thus, it is used to examine the biggest installation cost reason. On the other hand, the second one is related to PF, it is the reciprocal of the PF because the built-in optimization function minimizes the objectives. 10^3 multiplier is used to provide a similar order for both objective functions. According to [5, 38], PF can be used to examine the system in economic perspective because larger value of PF means more power from a smaller turbine. However, higher PF sometimes increases the cost for particularly high temperature applications due to more complexity during manufacturing process. Each objective function is separately used in studies in the literature. An important contribution of this work is the use of both of these objective functions simultaneously. As stated in [85], evaporator and turbine are the biggest irreversibility sources in the system, and it is suggested that both components should be considered together to investigate an ORC [80].

Table 2.4. Decision parameters and their boundaries.

	n_s	P_5 (kPa)	P_1/P_5	r_2/r_3	$PPTD_{evap}$ (°C)	SHEAT (°C)
Lower Boundary	0.45	$P_{min} = \max\{150; P_{sat} \text{ at } 30 \text{ °C}\}$	1.5	1.2	3	0
Upper Boundary	0.75	$P_{max} = \min\{4400; 0.9P_{cr}\}$	P_{max}/P_{min}	1.5	20	15

In the optimization, there are six decision variables, and these are listed with their boundaries in Table 2.4. Input parameters are given in Table 2.5. The general flow of NSGA-II is shown in Figure 2.4. All simulations are performed using Intel® i7-3770 processor. Typical run time is about four hours. Firstly, the first population

is randomly constituted, and then parents are selected according to their value of the objective functions. The selection probability of an individual becomes higher with low fitness value. After the selection, a new generation is produced changing genes between individuals, in other words cross-over procedure is applied to obtain a new generation from the previous one. Additionally, mutations are applied during the optimization procedure to prevent the convergence to local optimal solutions and to scan whole decision parameter domain. Fractions or types of operators are determined before the beginning of the procedure [108]. Pareto frontier is obtained at the end of the optimization; thus, many optimal solutions are found, and the number of the solutions depend on the Pareto fraction of NSGA-II [51]. The control operators and their values are tabulated in Table 2.6. More information on NSGA-II can be found in [112].

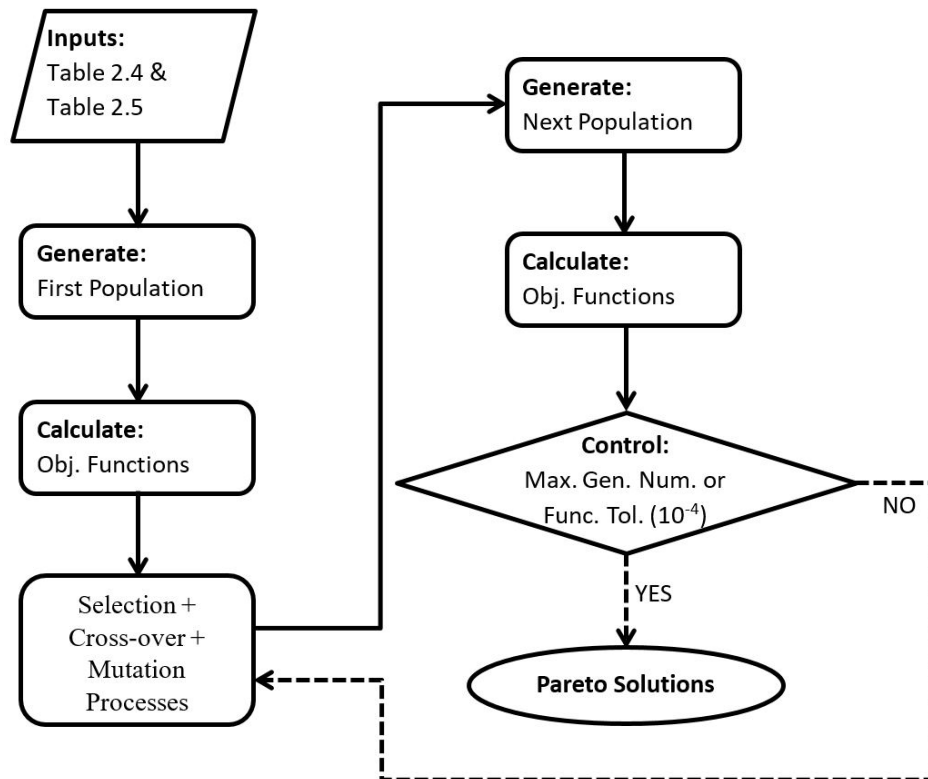


Figure 2.4. Flow chart of the implementation of NSGA-II and whole system code.

The heat supplied to the cycle is directly proportional to the mass flow rate, $\dot{m}_{hw}$, and it has an effect on results because there is a non-linear relation between the mass

Table 2.5. Constant parameters and initial estimations.

Parameter	Value	Reference
$\dot{m}_{hw}$	10 kg/s	-
η_{ts}	$0.87 - 1.07(n_s - 0.55)^2 - 0.5(n_s - 0.55)^3$	[26]
η_{pump}	0.75	-
T_8	90 / 120 / 150 °C	-
T_{10}	25 °C	-
$PPTD_{cond}$	3 °C	-
ε_b	0.0003 m	[38]
ϵ	0.0002 m	[26]
σ	1.35	[104]

flow rate and the second objective function. There is a linear relation between the mass flow rate and the rotor inlet diameter square. The net power output is affected by the mass flow rate, but the influence of mass flow rate on net power output is different than its influence on rotor area. The ratio of rotor inlet diameter square and net power output is non-linear, and hence the relation between the second objective function and the mass flow rate is non-linear. Following studies on low-grade ORC systems such as [37, 98, 101, 108], the mass flow rate has been selected as 10 kg/s.

Table 2.6. Control operators of NSGA-II.

Operators	Value or Type
Pareto Fraction	0.35
Cross-over Fraction	0.8
Maximum Generation Number	1200
Population Size	200
The First Generation Size	400
Mutation Function	Adaptive Feasible

2.7. Results and Discussion

The optimization is carried out for 28 different working fluids (Tables 2.1, 2.2 and 2.3) using three different heat source temperatures (90, 120 and 150 °C). The working fluids are ranked according to net power outputs obtained from TOPSIS solutions, and the first ten of them are tabulated in Table 2.7. The results are similar to literature. For example, R227ea is an appropriate candidate for around 90 °C source temperature in [113], and R227ea is away from to be an optimal choice with source temperature. As another example, isobutane is suitable to use for approximately 150 °C source temperature according to [113], and isobutane becomes more favorable choice with source temperature in Table 2.7.

Table 2.7. The first 10 working fluids regarding the net power outputs obtained from TOPSIS solutions ($\dot{m}_{wf} = 10$ kg/s).

90 °C		120 °C		150 °C	
Fluid	$\dot{W}_{net}$ (kW)	Fluid	$\dot{W}_{net}$ (kW)	Fluid	$\dot{W}_{net}$ (kW)
R1234yf	49.2	R1234ze(e)	151.8	Isobutane	269.4
R1234ze(e)	45.6	Isobutane	149.8	R152a	266.3
R227ea	43.6	Neopentane	147.2	R134a	263.7
R134a	40.9	R134a	146.9	Butane	257.3
R236ea	40.2	R1234yf	144.1	R1234ze(e)	256.8
R143a	40.0	Butane	122.0	Neopentane	223.7
Neopentane	37.9	R227ea	116.7	Isopentane	223.0
R236fa	37.8	R152a	114.5	R227ea	209.6
Isobutane	35.8	R236fa	111.3	Propane	204.7
Butane	35.7	Propane	105.9	Propyne	199.2

In Table 2.7, R1234ze(e) is the most favorable one among recent working fluids (R1234yf, R1234ze(e), R1233zd(e)) because it is always in the first five fluids for all heat

source temperatures. On the other hand, the worst candidate seems to be R1233zd(e) because net power outputs of its TOPSIS solutions are 24.1, 96.1 and 147.5 kW for 90 °C, 120 °C and 150 °C, respectively. However, R1233zd(e) has an advantage in flammability when compared to other fourth generation working fluids as shown in Table 2.1.

The net power output is increasing with the heat source temperature as expected for any working fluid. There is an ideal working fluid or a group of ideal working fluids for each source temperature which gives the highest net power output. Additionally, safety and environmental criteria should be considered; for example, R227ea can be preferred as the working fluid for 90 °C instead of R1234yf despite of the lower net power output value because R227ea is A1 in the ASHRAE classification, while R1234yf is A2.

Pareto frontiers are plotted in Figures 2.5, 2.6 and 2.7 for the first four working fluids, and TOPSIS solutions are indicated on the plots. Considering three different source temperatures, there are 12 Pareto frontiers for 8 different working fluids. R134a ranks among the first four fluids for all three temperatures which is also suggested in the literature for geothermal sources with a temperature range of 110-150 °C [64]. If one of the objective functions is considered more important, the solution which will give the lowest value for that objective function can be selected. However, the choice of working fluid may be different depending on which objective function is used in the selection. For instance, R227ea yields the lowest value for the first objective function and R1234yf gives the lowest value for the second objective function for 90 °C. This stimulates the authors for future work on optimal solutions that can be obtained using mixtures. In the literature, some works on use of mixtures in ORCs have already been done in recent years like [114]. If the thermal conductance, net power output and safety criteria are considered in equal importance, then the selection can be done among the TOPSIS solutions as explained before. R227ea is found to be an appropriate choice for 90 °C which was also suggested by Weshahi *et al.* [60] for low-grade (70 °C) geothermal applications.

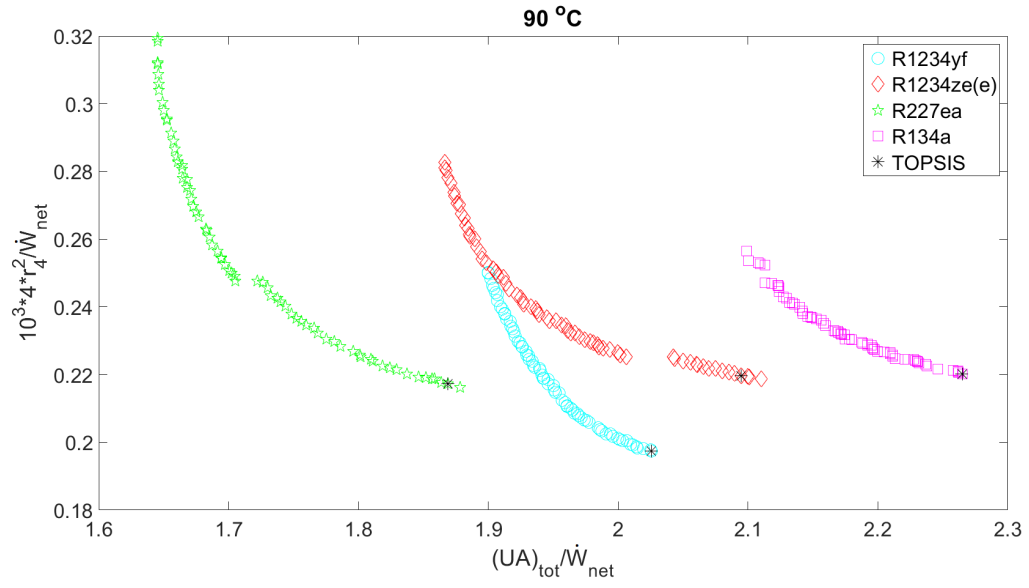


Figure 2.5. Pareto frontiers and TOPSIS solutions using heat source temperature of 90 °C.

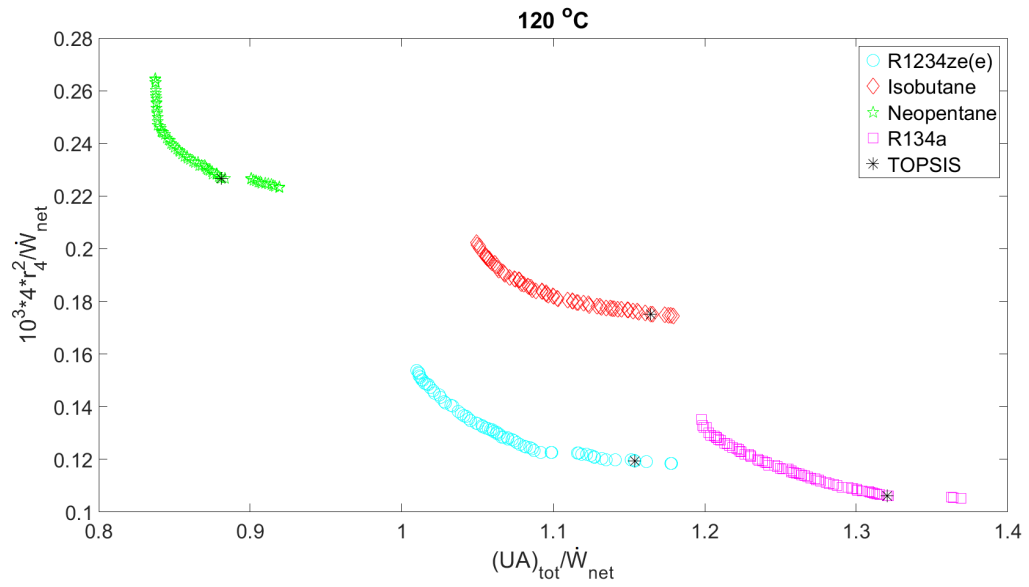


Figure 2.6. Pareto frontiers and TOPSIS solutions using heat source temperature of 120 °C.

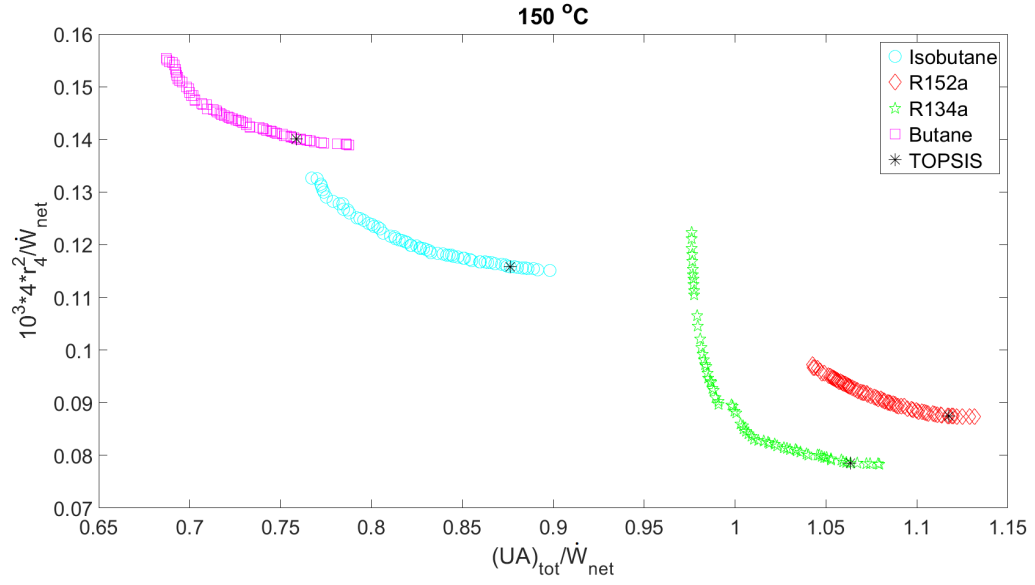


Figure 2.7. Pareto frontiers and TOPSIS solutions using heat source temperature of 150 °C.

Table 2.8. Optimized decision variables of the ideal solutions for heat source temperature of 90 °C.

Fluids	R1234yf	R1234ze(e)	R227ea	R134a
n_s	0.56	0.60	0.67	0.56
P_5 (kPa)	904.9	741.3	592.6	961.7
P_1/P_5	2.27	2.20	2.67	1.94
r_{23}	1.26	1.26	1.23	1.24
PPTD (°C)	6.24	4.95	6.13	10.10
SHEAT (°C)	9.80	5.86	7.12	14.01

Table 2.9. Optimized decision variables of the ideal solutions for heat source temperature of 120 °C.

Fluids	R1234ze(e)	Isobutane	Neopentane	R134a
n_s	0.61	0.55	0.56	0.56
P_5 (kPa)	700.6	481.7	238.9	1050.2
P_1/P_5	3.45	2.72	3.88	3.05
r_{23}	1.27	1.21	1.24	1.22
PPTD (°C)	3.94	6.84	3.39	3.43
SHEAT (°C)	14.08	7.66	6.68	14.98

Table 2.10. Optimized decision variables of the ideal solutions for heat source temperature of 150 °C.

Fluids	Isobutane	R152a	R134a	Butane
n_s	0.55	0.53	0.63	0.54
P_5 (kPa)	617.4	1060.1	986.8	423.0
P_1/P_5	3.25	3.04	3.70	3.55
r_{23}	1.21	1.22	1.22	1.22
PPTD (°C)	3.72	4.58	15.67	5.66
SHEAT (°C)	5.34	14.53	14.98	7.36

Specific speed, condensation pressure, pressure ratio at the turbine, radius ratio at the nozzle inlet and outlet, PPTD in the evaporator and degree of superheating are tabulated in Tables 2.8, 2.9 and 2.10 for the TOPSIS solutions given in Figures 2.5, 2.6 and 2.7. Specific speed, the turbine pressure ratio and radius ratio at the nozzle inlet and outlet do not vary significantly for different working fluids. Tables 2.8, 2.9 and 2.10 shows that for TOPSIS solutions, specific speed varies in a narrow range of 0.54-0.56 for all HCs (isobutane, neopentane and butane). If all Pareto solutions are considered, specific speed ranges are 0.49-0.55, 0.48-0.56, 0.49-0.53 and 0.46-0.64 for isobutane, butane, R152a and R134a respectively at 150 °C. The similarity of specific speed ranges of working fluids is actually anticipated in the literature [24–26] since the specific speed limits are caused by the loss effects at low speeds and reduction of efficiency especially due to exit loss for high specific speeds [54]. The turbine pressure ratio is also quite unchanged with the working fluids. Its value never reaches four to prevent the choked flow at the turbine outlet, as suggested in [5], not only in TOPSIS solutions but in all the Pareto solutions calculated. Therefore, the turbine pressure ratio upper limit can be set to four to decrease computational cost. The radius ratio of the nozzle inlet and outlet is also not strongly affected by the working fluids, and maximum ratio is observed as 1.33 for R1234ze(e) at 120 °C. Thus, its upper limit can be reduced to 1.35 from 1.5 to reduce computational cost.

In Tables 2.8, 2.9 and 2.10, it is seen that PPTD, degree of superheating and condensation pressure vary significantly with the working fluid. PPTD values in the evaporator are 3.7 °C and 15.7 °C for isobutane and R134a respectively at 150 °C. The degrees of superheating are 5.3 °C and 14.5 °C respectively for isobutane and R152a at 150 °C. Isobutane is a dry fluid and R152a is a wet fluid as shown in Tables 2.1 and 2.3; thus, R152a needs more superheating at the turbine inlet to prevent formation of droplets in the turbine. A similar result is highlighted by Rahbar *et al.* [77]. When all Pareto solutions are examined, the minimum and maximum degree of superheating are 4.0 and 10.4 °C respectively for isobutane and 14.5 and 15.0 °C for R152a at 150 °C. Considering all fluids in Tables 2.8, 2.9 and 2.10, degrees of superheating of R134a and R152a have the highest values.

Table 2.11. Results of optimization for the ideal solutions for heat source temperature of 90 °C.

Fluids	R1234yf	R1234ze(e)	R227ea	R134a
$\dot{W}_t$ (kW)	56.7	50.5	49.0	45.3
$\dot{W}_{net}$ (kW)	49.2	45.5	43.6	40.9
$\eta_{thermal}$	5.87	5.52	5.90	4.94
η_{ts}	72.7	71.5	68.9	73.0
h_1 (kJ/kg)	413.8	432.3	374.6	446.5
h_4 (kJ/kg)	408.9	427.7	371.1	441.9
C_4 (m/s)	101.0	98.5	85.9	98.9
W_4 (m/s)	32.3	35.0	37.7	31.4
U_4 (m/s)	115.7	117.1	110.1	112.9
C_{m5} (m/s)	41.9	45.2	48.9	40.9
Ma_5	0.66	0.67	0.78	0.59
$(UA)_{tot}$ (kW/°C)	99.7	95.4	81.6	92.5
r_1 (mm)	92.4	96.1	92.7	89.7
r_2 (mm)	67.7	69.4	66.6	64.6
r_3 (mm)	53.8	55.2	54.1	52.3
r_4 (mm)	49.3	50.0	48.7	47.4
r_{tip5} (mm)	34.5	35.0	34.1	33.2
r_{hub5} (mm)	13.8	14.0	13.6	13.3
L_4 (mm)	8.5	9.5	9.2	9.3
L_5 (mm)	20.7	21.0	20.4	19.9
Z_{rotor}	14	13	13	14
Ω (rad/s)	2348.1	2342.1	2260.3	2380.1
T_9 (°C)	70.0	70.4	72.4	70.3
$\dot{m}_{wf}$ (kg/s)	5.11	4.64	5.51	4.30

Table 2.12. Results of optimization for the ideal solutions for heat source temperature of 120 °C.

Fluids	R1234ze(e)	Isobutane	Neopentane	R134a
$\dot{W}_t$ (kW)	171.7	160.8	154.1	172.2
$\dot{W}_{net}$ (kW)	151.8	149.8	147.2	146.9
$\eta_{thermal}$	7.84	7.32	8.63	7.48
η_{ts}	70.7	73.3	72.6	72.6
h_1 (kJ/kg)	450.8	674.1	452.5	455.6
h_4 (kJ/kg)	443.5	660.4	436.9	448.0
C_4 (m/s)	123.7	167.8	179.3	125.3
W_4 (m/s)	46.0	51.1	57.6	39.5
U_4 (m/s)	149.7	188.7	205.8	143.0
C_{m5} (m/s)	59.1	66.0	74.5	50.9
Ma_5	0.82	0.71	0.83	0.75
$(UA)_{tot}$ (kW/°C)	175.1	174.4	129.6	193.7
r_1 (mm)	124.7	146.9	162.8	110.4
r_2 (mm)	92.6	105.8	120.6	81.6
r_3 (mm)	72.8	87.5	97.6	67.1
r_4 (mm)	67.3	81.0	91.3	62.4
r_{tip5} (mm)	47.1	56.7	63.9	43.7
r_{hub5} (mm)	18.8	22.7	25.6	17.5
L_4 (mm)	9.9	12.5	11.9	8.9
L_5 (mm)	28.3	34.0	38.4	26.2
Z_{rotor}	13	14	14	14
Ω (rad/s)	2223.4	2329.3	2253.2	2291.0
T_9 (°C)	74.1	71.4	79.6	73.4
$\dot{m}_{wf}$ (kg/s)	9.76	5.32	4.37	10.08

Table 2.13. Results of optimization for the ideal solutions for heat source temperature of 150 °C.

Fluids	Isobutane	R152a	R134a	Butane
$\dot{W}_t$ (kW)	300.0	306.1	314.8	275.3
$\dot{W}_{net}$ (kW)	269.4	266.3	263.7	257.3
$\eta_{thermal}$	8.24	8.01	8.00	8.68
η_{ts}	73.0	73.2	69.8	73.0
h_1 (kJ/kg)	692.8	575.2	456.6	739.1
h_4 (kJ/kg)	677.2	563.1	448.8	721.1
C_4 (m/s)	179.0	158.3	126.9	192.3
W_4 (m/s)	55.2	46.3	50.8	58.5
U_4 (m/s)	202.4	175.9	157.6	216.5
C_{m5} (m/s)	71.0	58.7	65.4	75.0
Ma_5	0.77	0.73	0.85	0.79
$(UA)_{tot}$ (kW/°C)	235.9	297.4	280.4	195.1
r_1 (mm)	156.9	135.3	129.0	168.5
r_2 (mm)	114.9	100.0	94.7	124.1
r_3 (mm)	94.6	81.6	77.9	101.3
r_4 (mm)	88.3	76.3	72.0	94.9
r_{tip5} (mm)	61.8	53.4	50.4	66.4
r_{hub5} (mm)	24.7	21.4	20.1	26.6
L_4 (mm)	12.1	10.4	10.5	12.4
L_5 (mm)	37.1	32.1	30.2	39.9
Z_{rotor}	14	14	13	14
Ω (rad/s)	2291.6	2304.5	2190.5	2280.3
T_9 (°C)	72.8	71.5	72.2	80.1
$\dot{m}_{wf}$ (kg/s)	8.65	11.50	16.58	6.91

Tables 2.11, 2.12 and 2.13 tabulates the optimized system parameters calculated for TOPSIS solutions shown in Figures 2.5, 2.6 and 2.7. Thermal conductance increases when the heat source temperature increases as shown by the results of R134a present in all heat source temperatures. For all Pareto solutions calculated for R134a, the ranges of thermal conductance are 74.7-92.6, 133.4-200.6 and 245.0-285.2 kW/°C at 90, 120 and 150 °C respectively. The first objective is proportional to thermal conductance and inversely proportional to net power output. Therefore, increase of thermal conductance not necessarily leads to an increase of the objective function. R134a results in Tables 2.11, 2.12 and 2.13 also shows that thermal efficiency increases with the source temperature. The ranges of thermal efficiency for all Pareto solutions of R134a are 4.94-5.41 %, 7.12-7.72 % and 7.93-8.58 % at 90, 120 and 150 °C respectively. As expected, $\eta_{thermal}$ is very low and around 5.50 % for all four fluids at 90 °C and increases with the source temperature. Many studies in the literature such as [37,38,62] used $\eta_{thermal}$ as an objective function to maximize which may lead to solution with small power output or larger thermal conductance. If $\eta_{thermal}$ was used as objective function in the present study, the ideal solutions would be changed. Thus, selection of the objective functions in the optimization for ORC applications is a very important step. Both objective functions selected are essentially related to the net power output and installation cost. Economic indicators of the installation are changed with respect to the ORC application. According to [16], for example, the biggest economic cost for the installation of a geothermal power plant is the well cost; thus, objective functions should be coherent with the purpose of the system and constraints. Furthermore, as mentioned before, source fluid evaporator exit temperature lower limit is set to 70 °C and its flow rate is 10 kg/s in this study. Maximum heat input available for each source is obtained when the outlet temperature is 70 °C, and the values are 839, 2106 and 3387 kW at 90, 120 and 150 °C respectively. One of the objective functions to maximize can be selected as net power output per maximum heat input for a geothermal power plant coherently the purpose of the plant.

One way of increasing of turbine output is to increase the turbine size and the mass flow rate through the turbine as shown in Tables 2.11, 2.12 and 2.13. The results given for R134a in Tables 2.11, 2.12 and 2.13 indicate that as heat source

temperature increases turbine power output and turbine size increase and rotational speed decreases. However, the relation between the turbine power output and turbine size may significantly be affected by the working fluid properties. For example, at 120 °C, the turbine power output is calculated as 171.7 kW for R1234ze(e) and as 160.8 kW for isobutane with 67.3 and 81.0 mm rotor diameters respectively. For R1234ze(e), more power output is produced with a smaller turbine. Molecular weights of R1234ze(e) and isobutane are 114.0 and 58.1 kg/kmol respectively, and according to [64] and [12] molecular weight along with critical temperature, boiling temperature, molecular complexity etc. may be the reason for the increase of power output for the smaller turbine. As seen in Tables 2.11, 2.12 and 2.13, HCs (isobutane, neopentane and butane) have lower flow rates than HFCs (R227ea, R134a and R152a) and HFOs (R1234yf, R1234ze(e)), and mass flowrates increase with increase in source temperatures.

PPTD and thermal conductance are inversely proportional [115]. However, if one compares PPTD for different fluids the situation may be different. Table 2.13 shows that at 150 °C, when isobutane and R134a are used thermal conductances are 235.9 and 280.4 kW/°C respectively. Table 2.10 shows that at the same temperature, when isobutane and R134a are used, PPTD values are 3.7 and 15.7 °C respectively. So, the values of PPTD and thermal conductance of R134a are both higher than those of isobutane. Differences in thermal properties such as thermal conductivity and specific heat may cause this situation.

Lastly, the results in Tables 2.11, 2.12 and 2.13 show that turbine efficiency is in the range of 0.69 and 0.73 for all the TOPSIS solutions. Although to obtain more accurate results a dynamic turbine efficiency in the calculations is desirable, the results indicate that the error, associated with assuming a constant efficiency as done in many research studies in the literature, will be limited to a low value.

3. ORC SYSTEM WITH DESIGN OF EVAPORATOR AND RADIAL INFLOW TURBINE

In this chapter, there are some differences in thermodynamic analysis of the ORC. The maximum pressure of the system, in other words the evaporation pressure, is set to 40 bar instead of 44 bar because this limit is suggested as maximum pressure for plate heat exchangers (PHEs) [116]. Additionally, pressure drop in the evaporator is considered in this chapter.

Radial inflow turbine is designed using the model given in the previous chapter. Same working fluid list is also used in this chapter. The main novelty is evaporator is designed as a PHE, and the length, width and the space between plates are optimized in this chapter. Optimization mechanism is same with the previous chapter, and NSGA-II is used for multi-objective optimization. However, new three decision variables on the evaporator are added. Evaporator area is used as in the objective function about the heat exchangers instead of total thermal conductance as well. Objective functions and additional decision variables and their boundaries will be explained below.

3.1. Evaporator Design

Herringbone, in other words chevron type PHE is chosen in this study [51, 53, 66, 116, 117]. Chevron type corrugations increase the level of turbulence despite of low Reynolds number [118] leading to a higher heat transfer rate rather than shell and tube heat exchangers for the same volume exchanger, but this turbulence also results in high pressure drop quantities at the same time [117]. Therefore, pressure drop is checked for each working condition, and it is restricted by five percent of the fluid pressure at the evaporator inlet like in [51]. The geometry of the PHE is illustrated in Figure 3.1. Chevron angle, β , is chosen to be 60° to obtain minimum evaporator cost [119].

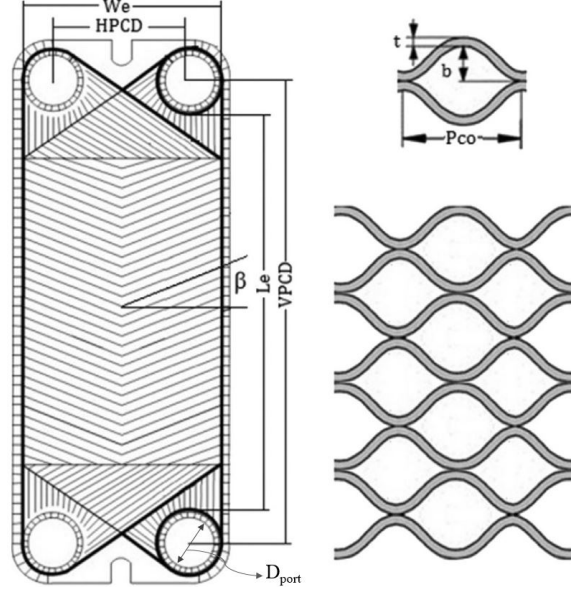


Figure 3.1. Configuration of the herringbone PHE [66].

According to the figure, the effective length, L_e , and width, W_e , are evaluated as follows:

$$L_e = VPCD - D_{port}, \quad (3.1)$$

$$W_e = HPCD - D_{port}. \quad (3.2)$$

where D_{port} is the port diameter, and it is assumed to be 0.2 m like in [51]. The enlargement factor, ϕ , is defined to consider the effect of the corrugations on the plate area [117].

$$\phi = \frac{1}{6} \left[1 + \sqrt{1 + (b\pi/P_{co})^2} + 4\sqrt{1 + \frac{(b\pi/P_{co})^2}{2}} \right] \quad (3.3)$$

Therefore, the plate area and the total heat transfer area are calculated as follows:

$$A_p = \phi L_e W_e, \quad (3.4)$$

$$A_{tot} = (N_p - 2)A_p. \quad (3.5)$$

where N_p represents the number of the plates. Hydraulic diameter, D_h , is also found by the definition as [117]:

$$D_h = \frac{4 \times \text{Channel flow area}}{\text{Wetted perimeter}} \cong \frac{2b}{\phi} \text{ for } b \ll W_e. \quad (3.6)$$

Number of channels, N_{ch} , is obtained depending on N_p as follows:

$$N_{ch,hw} = \frac{N_p}{2} ; N_{ch,wf} = \frac{N_p - 2}{2} \text{ for even } N_p, \quad (3.7)$$

$$N_{ch,hw} = N_{ch,wf} = \frac{N_p - 1}{2} \text{ for odd } N_p. \quad (3.8)$$

The logarithmic mean temperature difference (LMTD) is used in the present study to model the counter-current flow heat exchange mechanism. The calculation LMTD and thermal conductance are given in Eq. 2.5 and 2.6. Heat transfer area is evaluated after overall heat transfer coefficient, U , is obtained as follows:

$$\frac{1}{U} = \frac{1}{\alpha_{wf}} + \frac{t_p}{k_p} + \frac{1}{\alpha_{hw}}. \quad (3.9)$$

where α and k_p symbolize convective heat transfer coefficient and thermal conductivity of plate, respectively. The second term in the right-hand side in Eq. 3.9 means the thermal resistance of the plate, and this term is ignored in the study because the

common used materials in ORC heat exchangers has high thermal conductivity [120]. Many studies [90,120,121] made this assumption to simplify the heat exchanger model in the literature as well.

The iterative process is initiated assuming N_p , and a guess for the total heat transfer is calculated using Eq. 3.5. LMTD method is also used to find the heat transfer area using appropriate heat transfer correlations to obtain overall heat transfer coefficients. Then, equality of both estimations is checked. If the equality is not satisfied, the iterative process maintains.

The evaporator is divided into three main regions as indicated in Figure 3.2: subcooled, phase change and superheated. Fluid properties are calculated using average of the bulk temperatures at the inlet and outlet of the regions [68] except the phase change region and its reason will be explained at the next sections in the manuscript. Additionally, the heat source fluid is simulated as water like many studies [51,53,116,122].

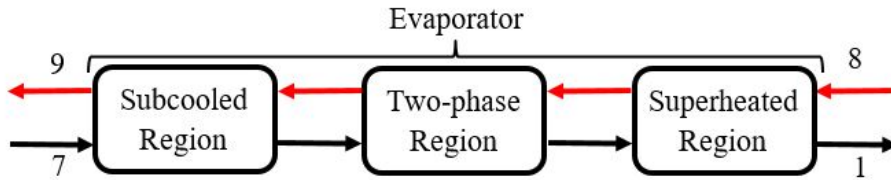


Figure 3.2. Evaporator regions.

3.1.1. Single Phase Regions

There is no phase change in both working fluid and heat source sides in the subcooled and superheated regions, and Chisholm and Wanniarachchi correlation is used to calculate the Nusselt number of both sides for the single phase regions [123]. The correlation was also employed in many studies [47,51,52,66,68,121] about PHEs

in ORCs.

$$Nu_{sp} = \frac{D_h \alpha}{k_f} = 0.724 \left(\frac{6\beta}{\pi} \right)^{0.646} Re^{0.583} Pr^{0.33} \quad (3.10)$$

where k is the thermal conductivity of the fluid and β is expressed in rad. Reynolds and Prandtl numbers are calculated as follows:

$$Re = \frac{GD_h}{\mu}, \quad (3.11)$$

$$Pr = \frac{C_p \mu}{k_f}. \quad (3.12)$$

where G , μ and C_p are mass velocity, dynamic viscosity and specific heat capacity, respectively. G means channel mass flow rate per channel cross-sectional area [66].

$$G = \frac{\dot{m}}{N_{ch} b W_e} \quad (3.13)$$

After the calculation of convective heat transfer coefficient of working fluid and heat source sides, the overall heat transfer coefficient is found using Eq. 3.9, and the required heat transfer area for the region is found with the help of LMTD method.

In the subcooled and superheated regions, only frictional pressure drop is considered [51, 53, 66],

$$\Delta P_{sp} = \frac{4f_{sp} N_{ch} G^2 L_r}{2\rho D_h} \quad (3.14)$$

where L_r represents the length along the flow direction of the region. The fanning friction factor for the single phase region, f_{sp} , is given as follows [51, 66]:

$$f_{sp} = \frac{0.572}{Re^{0.217}} \text{ when } Re > 550. \quad (3.15)$$

3.1.2. Two Phase Region

The phase of the heat source fluid does not change through evaporator; hence, Chisholm and Wanniarachchi correlation is used for the hot water side in the two phase region. However, the working fluid is vaporized in this region, and the fluid properties significantly alter with the quality during evaporation period. Therefore, LMTD method cannot be employed using the average of the bulk temperatures at the inlet and outlet of the region. Instead of that, the region is divided into small parts, and the number of parts are determined as minimum number so that constant fluid properties can be assumed. Each part has same enthalpy change, and the properties can be calculated knowing the enthalpy and pressure values. Han, Lee and Kim correlation [116] is employed to calculate the convective heat transfer coefficient of the working fluid side.

$$Nu_{tp,i} = \frac{D_h \alpha}{k} = Ge_1 Re_{eq,i}^{Ge_2} Bp_{eq,i}^{0.3} Pr_l^{0.4}, \quad (3.16)$$

where i is used to symbolize each small parts and

$$Ge_1 = 2.81 \left(\frac{P_{co}}{D_h} \right)^{-0.041} \left(\frac{\pi}{2} - \beta \right)^{-2.83}, \quad (3.17)$$

$$Ge_2 = 0.746 \left(\frac{P_{co}}{D_h} \right)^{-0.082} \left(\frac{\pi}{2} - \beta \right)^{0.61}. \quad (3.18)$$

Prandtl number is evaluated using saturated liquid properties [53, 118]. P_{co} is corrugation pitch (see Figure 3.1), and equivalent Reynolds and Boiling numbers are evaluated as follows:

$$Re_{eq,i} = \frac{G_{eq,i} D_h}{\mu_l}, \quad (3.19)$$

$$Bo_{eq,i} = \frac{q_i''}{G_{eq,i} i_{lg}}. \quad (3.20)$$

q'' , i_{lg} and G_{eq} are heat flux, enthalpy of vaporization and equivalent mass flux, respectively. The equivalent mass flux is found as:

$$Ge_{eq,i} = G \left[1 - x_i + x_i \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right]. \quad (3.21)$$

where x means the quality. The saturated liquid thermal conductivity is employed to calculate Pr in Eq. 3.16 and obtain from Nu_{tp} [118].

Knowing the convective heat transfer coefficient of both sides, the overall heat transfer coefficient is obtained for each small part to be similar to Eq. 3.9; thus, the required heat transfer area is calculated using LMTD method for the parts separately. The areas of each part are then summed to find the total heat transfer area of the two phase region.

In the two phases region along L_e , there is a pressure drop as a result of friction, inlet and outlet flow ports, elevation change and acceleration of the working fluid. The pressure drop due to friction is calculated as follows [116]:

$$\Delta P_{friction,i} = \frac{f_{tp,i} N_{ch} G_{eq}^2 L_{r,i}}{\rho_l D_h}, \quad (3.22)$$

$$f_{tp,i} = Ge_3 Re_{eq,i}^{Ge_4}, \quad (3.23)$$

$$Ge_3 = 64710 \left(\frac{P_{co}}{D_h} \right)^{-5.27} \left(\frac{\pi}{2} - \beta \right)^{-3.03}, \quad (3.24)$$

$$Ge_4 = -1.314 \left(\frac{P_{co}}{D_h} \right)^{-0.62} \left(\frac{\pi}{2} - \beta \right)^{-0.47}. \quad (3.25)$$

The pressure drop due to elevation change is obtained as follows:

$$\Delta P_{ele,i} = \rho_{mean} g L_{r,i}, \quad (3.26)$$

$$\rho_{mean} = \left(\frac{x}{\rho_g} + \frac{1-x}{\rho_l} \right)^{-1}. \quad (3.27)$$

The port pressure drop is found as:

$$\Delta P_{port} = 1.4 \frac{G_{port}^2}{2\rho_{mean}}, \quad (3.28)$$

where G_{port} is port mass velocity and evaluated as follows:

$$G_{port} = \frac{\dot{m}_{eq}}{\pi D_{port}^2 / 4}, \quad (3.29)$$

$$\dot{m}_{eq} = \dot{m}_{wf} \left[1 - x + x \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right]. \quad (3.30)$$

The pressure drop due to the acceleration is evaluated as:

$$\Delta P_{acc} = \left(\frac{G_{eq}^2 x}{\rho_l - \rho_g} \right)_{inlet} - \left(\frac{G_{eq}^2 x}{\rho_l - \rho_g} \right)_{exit}. \quad (3.31)$$

The total pressure drop is restricted to be less than five percent of the evaporator inlet pressure [51].

$$\Delta P_{evap} = \Delta P_{sp} + \Delta P_{friction} + \Delta P_{ele} + \Delta P_{port} + \Delta P_{acc} \quad (3.32)$$

Following [52], the pump power for the compensation is given as:

$$\dot{W}_{p,compensation} = \Delta P_{evap} \dot{V}_{wf}. \quad (3.33)$$

where $\dot{V}_{wf}$ is the volumetric flow rate of the working fluid at the pump inlet. $\dot{W}_{net}$ in Eq. 2.9 is evaluated with this additional required power in this chapter.

$$\dot{W}_{net} = \dot{W}_t - (\dot{W}_p + \dot{W}_{compensation}) \quad (3.34)$$

The outcomes of this chapter are compared to results of Hayat *et al.* [51] addition to validation of previous chapter, and tabulated in Table A.4.

3.2. Radial Inflow Turbine

The design procedure of the RIT is same within the first chapter except a novel failure check. In this chapter, modeled RIT is checked according to structural and vibrational criteria to obtain a feasible turbine design. The centrifugal force can damage the rotor; the stress is calculated as follows [25, 89]:

$$\sigma_{rr} = 0.3\rho_m U_4^2. \quad (3.35)$$

where ρ_m is the material density and 0.3 is a geometric factor [25]. The material used is INCONEL IN718 [38, 89]. For a feasible design, the stress has to be below ninety percent of the yield stress [89].

In the vibration analysis, the first bending mode is checked. The design is considered to be feasible if the angular velocity of the rotor is four times higher than the first bending mode. The first bending mode is calculated using the following equation [25, 89]:

$$\omega_{n1} = \frac{6.94}{2\pi L_4^2} \sqrt{\frac{Et_5^3}{12\rho_m(1-\nu^2)}}. \quad (3.36)$$

where E and ν are elastic modulus and Poission's ratio respectively.

3.3. Optimization

A multi-objective optimization procedure is employed in this chapter to minimize the ratio of the evaporator area to net output power and to maximize PF; NSGA-II is used again as optimization algorithm. There are three new decision variables: the length, width and spacing among plates. Therefore, there are totally nine decision variables. Maximum generation number given in Table 2.6 differs from the previous chapter, and it is accepted to be 1800 due to the increase in the decision variable number.

New decision variables and decision variables which have differences for their boundaries compared to the previous chapter are tabulated in Table 3.1.

Table 3.1. Evaporator and changed decision variables.

	P_5 (kPa)	r_{23}	L_e (m)	W_e (m)	b (m)
Lower Boundary	$P_{min}=\max\{150; P_{sat} \text{ at } 30^\circ\text{C}\}$	1.2	0.3	0.15	0.002
Upper Boundary	$P_{max}=\min\{4000; 0.9P_{cr}\}$	1.35	1.25	0.55	0.005

TOPSIS is used as the decision-making method similar to the previous chapter. Eq. 2.74 is used without any change. On the other hand, Eq. 2.73 is altered in this

chapter as follows:

$$Obj_1 = \frac{A_{evap}}{\dot{W}_{net}}. \quad (3.37)$$

Eq. 3.37 is the ratio of the total evaporator heat transfer area to net output power, and it also represents the cost of the evaporator since the cost depends linearly on with the area [66]. The constant parameters of the evaporator used in this chapter are tabulated in Table 3.2.

Table 3.2. Constant parameters of evaporator.

Parameter	Values	Reference
β	60°	[51, 66]
P_{co}	0.008 m	[66]
D_{port}	0.2 m	[51]

3.4. Results and Discussion

28 different working fluids (tabulated in Tables 2.1, 2.2 and 2.3) are used for 90, 120 and 150 °C heat source temperatures. For each source temperature, 10 fluids which give the highest net power output of TOPSIS solution are tabulated in Table 3.3 similar to the previous chapter. Butane is at the top of the list for 120 and 150 °C, and butane is also suggested by Zhang *et al.* [52] for 125 °C. R152a is offered as a suitable working fluid choice for 150 °C by Bao and Zhao, and Tocci *et al.* [12, 113]; in Table 3.3, R152a yields a better performance with increase in temperature. R152a is also reported as a good working fluid for a simple ORC at 150 °C in the study of Hayat *et al.* [51].

Among the fourth generation working fluids investigated, R1233zd(e), R1234yf, R1234ze(e), R1234ze(e) appears as one of the ideal working fluids for three heat source

Table 3.3. Working fluids with respect to the net power outputs of TOPSIS solutions.

90 °C		120 °C		150 °C	
Fluid	$\dot{W}_{net}$ (kW)	Fluid	$\dot{W}_{net}$ (kW)	Fluid	$\dot{W}_{net}$ (kW)
R1234yf	35.2	Butane	128.3	Butane	231.3
R245fa	34.7	R134a	118.5	R152a	227.6
R143a	34.2	R1234yf	115.9	R134a	217.2
R134a	33.4	R1234ze(e)	112.1	Isobutane	200.3
R236fa	30.8	Neopentane	104.8	R1234ze(e)	190.0
R227ea	29.4	R152a	95.6	Isopentane	170.8
R1234ze(e)	28.2	Isopentane	94.7	Propylen	165.3
R245ca	26.8	Propane	89.4	Propane	164.8
R152a	26.5	Isobutane	87.1	R227ea	157.2
R236ea	26.0	R227ea	85.1	Propyne	153.8

temperatures. R1234yf outperforms R1234ze(e) for 90 and 120 °C but does not show up at all in the case of 150 °C. Net power output is not the only parameter for the selection of the working fluid; safety and environmental concerns are to be taken into account. Although R1233zd(e) has the worst performance among all three, because R1233zd(e) is not flammable (ASHRAE class A1) it may become a useful working fluid candidate. Similarly, R134a can be selected instead of R1234yf for 90 °C despite its lower net power output since its ASHRAE class is A1 while R1234yf's ASHRAE class is A2. As a result of the determination of a group of working fluids is more reasonable than identifying a single ideal working fluid with the consideration of net power output. Safety and environmental requirements may require another fluid be selected from the determined group.

In Figures 3.3, 3.4 and 3.5 for saving space Pareto frontiers of only the first four working fluids are plotted for each source temperature like in the previous chapter. All Pareto solutions are checked for structural and vibrational limitations; all of them are

in safe region when the first bending mode and the meridional stress are calculated as aforementioned. The TOPSIS solutions are also marked on the corresponding Pareto frontier in Figures 3.3, 3.4 and 3.5. In Figures 3.3, 3.4 and 3.5, although there are totally 12 Pareto frontiers, there are 8 different working fluids (some of them are repeated). R134a is in the first four for all source temperatures. In the study of Zhai *et al.* [64], R134a has similarly been suggested for geothermal sources between 110 and 150 °C. The use of TOPSIS method to select the ideal solution among Pareto frontiers inherently assumes that neither objective function is dominant. In an ORC application, if one of the objective functions is dominant, the selection of working fluid may change. For example for 120 °C, the lowest evaporator heat transfer area per net output power is obtained for R1234yf, and the lowest reciprocal PF is calculated with R134a. If the first objective function is more important, R1234yf may be a better choice; but if the second objective function is more important, R134a may be the choice. Zeotropic fluids may yield better performance, which should be the topic of a future study. For example, there is a recent work by Magdalena *et al.* [114] that focuses on zeotropic fluids.

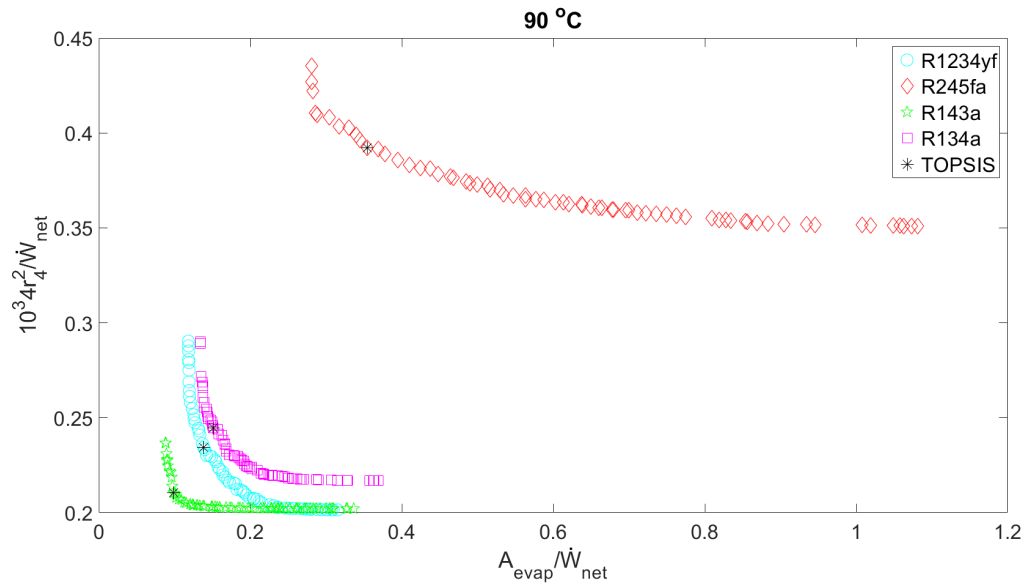


Figure 3.3. Pareto frontiers and TOPSIS solutions using heat source temperature of 90 °C.

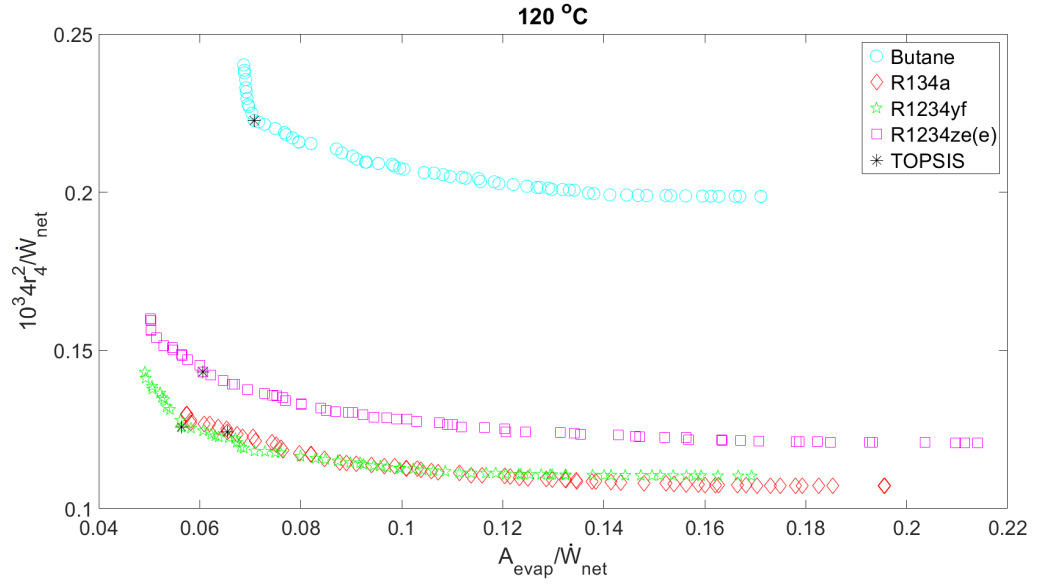


Figure 3.4. Pareto frontiers and TOPSIS solutions using heat source temperature of 120 °C.

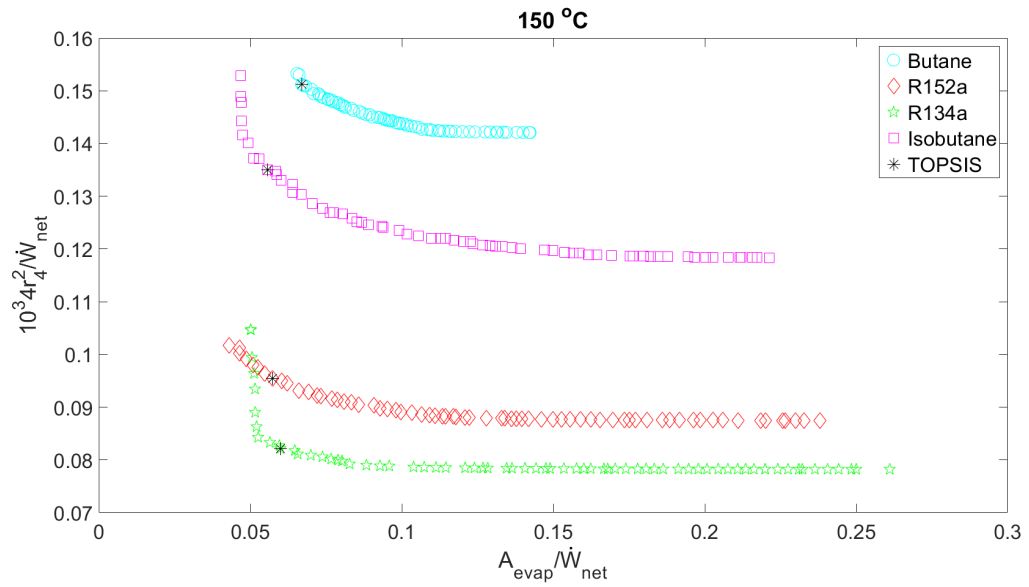


Figure 3.5. Pareto frontiers and TOPSIS solutions using heat source temperature of 150 °C.

Table 3.4. Optimized decision variables of the TOPSIS solutions for heat source temperature of 90 °C.

Fluids	R1234yf	R245fa	R143a	R134a
n_s	0.60	0.68	0.54	0.56
P_5 (kPa)	905.8	231.9	1691.7	964.9
P_1/P_5	2.01	2.68	1.75	1.88
r_{23}	1.25	1.22	1.21	1.26
PPTD (°C)	13.21	8.06	14.85	13.29
SHEAT (°C)	4.57	7.75	7.97	7.00
L_e (m)	1.21	1.21	0.87	1.21
W_e (m)	0.40	0.44	0.42	0.39
b (m)	0.003	0.005	0.003	0.003

Table 3.5. Optimized decision variables of the TOPSIS solutions for heat source temperature of 120 °C.

Fluids	Butane	R134a	R1234yf	R1234ze(e)
n_s	0.58	0.58	0.62	0.60
P_5 (kPa)	327.8	908.1	862.8	660.8
P_1/P_5	2.81	2.89	3.12	3.32
r_{23}	1.26	1.26	1.27	1.23
PPTD (°C)	13.12	14.77	16.52	15.10
SHEAT (°C)	0.86	8.76	7.15	4.12
L_e (m)	1.23	1.20	1.18	1.20
W_e (m)	0.48	0.40	0.49	0.40
b (m)	0.004	0.004	0.004	0.004

Table 3.6. Optimized decision variables of the TOPSIS solutions for heat source temperature of 150 °C.

Fluids	Butane	R152a	R134a	Isobutane
n_s	0.56	0.52	0.65	0.53
P_5 (kPa)	393.6	1050.2	1133.3	651.3
P_1/P_5	3.37	2.99	3.21	2.94
r_{23}	1.23	1.25	1.22	1.31
PPTD (°C)	13.22	11.71	17.57	14.36
SHEAT (°C)	6.47	9.80	12.34	2.04
L_e (m)	0.85	1.12	0.98	1.22
W_e (m)	0.42	0.50	0.51	0.49
b (m)	0.004	0.003	0.004	0.004

Optimized decision variables of TOPSIS solutions are listed in Tables 3.4, 3.5 and 3.6. As seen, specific speed, n_s , pressure ratio, P_1/P_5 , at the turbine stage and the radius ratio, r_{23} , of the nozzle inlet to the outlet do not significantly change with the working fluid. Independent behavior of specific speed is also noted in the literature [24–26]. There are two HCs (butane, isobutane) in Tables 3.4, 3.5 and 3.6, and their specific speeds, for instance, are in the range of 0.53-0.58. All Pareto solutions at 120 °C show that specific speed is limited within a narrow range; it varies in the ranges of 0.54-0.61, 0.55-0.64, 0.55-0.66, 0.53-0.64 for butane, R134a, R1234yf and R1234ze(e) respectively. At low specific speeds, loss effects become dominant and decrease the efficiency; at high specific speeds exit loss particularly causes a reduction in the efficiency [54]. Another decision variable independent from screening working fluid is the pressure ratio at the turbine stage, and its upper limit is especially restricted to prevent choked flow at the turbine outlet. While its maximum value is 3.37 in TOPSIS solutions tabulated in Tables 3.4, 3.5 and 3.6, the value reaches a maximum value of 3.55 for R1234ze(e) at 120 °C among all Pareto solutions as demonstrated in Figure 3.6. Additionally, the radius ratio of the nozzle inlet to the outlet does not change

much with working fluid. The maximum ratio, r_{23} , of 1.33 is obtained for isobutane at 150 °C among all Pareto solutions in Figures 3.3, 3.4 and 3.5.

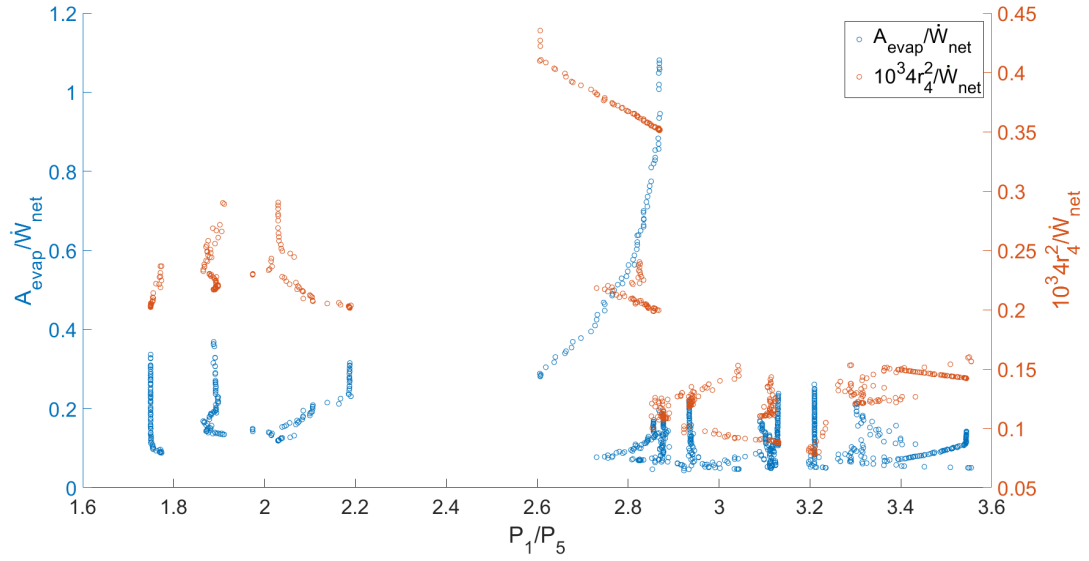


Figure 3.6. Pressure ratios at the turbine stage for all Pareto solutions in Figures 3.3, 3.4 and 3.5.

On the other hand, the working fluid has an important effect PPTD; PPTD in the evaporator may significantly change with respect to selected working fluid. For instance, a PPTD of 8.06 °C is obtained when R245fa is used at 90 °C, but it is 14.85 when R143a is used as tabulated in Table 3.4. Maximum and minimum PPTD values in all Pareto solutions in Figures 3.3, 3.4 and 3.5 are 18.86 °C and 3.66 °C for R134a at 150 °C and R245fa at 90 °C respectively. The degree of superheating also strongly depends on the working fluid and temperature. For example, isobutane and R152a have degrees of superheating of 2.04 °C and 9.80 °C at 150 °C respectively; R152a is a wet fluid and isobutane is dry which is also emphasized in the study of Rahbar *et al.* [77]. The degrees of superheating of butane are 0.86 and 6.47 °C for source temperature of 120 °C and 150 °C respectively. Condensation pressure, P_5 , also significantly varies with working fluid.

Length, width and plate spacing of the PHE evaporator are the last three decision variables. In the optimized solutions, length and width of the evaporator change usually within a narrow range. 66 percent of the lengths indicated in Figure 3.7 is in a range of 1.1-1.3 m; thus, if a constant plate length is to be assumed in a study, the length can be easily taken as 1.2 m. In the literature, the use of 1.25 m is seen [51, 52] which is in accordance with the present study. Besides, 71 percent of the widths shown in Figure 3.8 is between 0.4 and 0.5; thus, the use of 0.45 m can be suggested. Some studies [51, 52, 124] assumed the width as 0.55 m, whereas in one study [66], it was accepted to be 0.35 m. Another decision variable from the geometry of the evaporator is the plate spacing, and 55 percent of plate spacing values of all Pareto solutions is in the range of 0.003-0.004 m. In Tables 3.4, 3.5 and 3.6, plate spacing of TOPSIS solutions are either 0.003 m or 0.004 m except one of them. Imran *et al.* [66] used the spacing as 0.003 m. Therefore, $b=0.0035$ m is proposed if it is going to be used as a fixed design parameter.

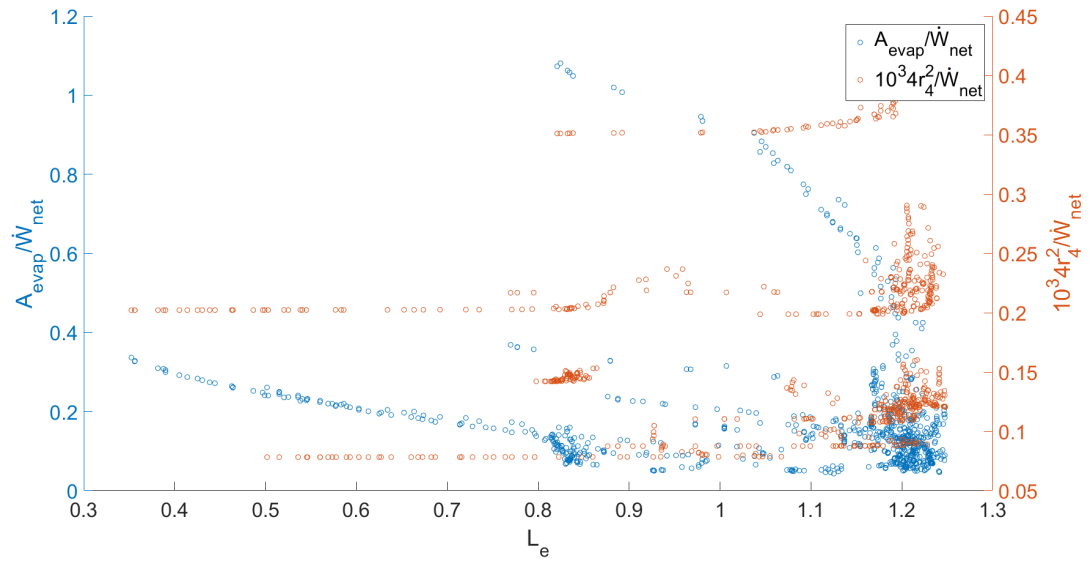


Figure 3.7. Length of the evaporator for all Pareto solutions given in Figures 3.3, 3.4 and 3.5.

Optimized system parameters of the TOPSIS solutions are tabulated in Tables 3.7, 3.8 and 3.9. In some ORC studies, thermal conductance is used as an indicator

Table 3.7. Optimized results of the TOPSIS solutions for heat source temperature of 90 °C.

Fluids	R1234yf	R245fa	R143a	R134a
$\dot{W}_t$ (kW)	40.9	35.9	45.3	37.5
$\dot{W}_{net}$ (kW)	35.2	34.7	34.2	33.4
$\eta_{thermal}$	5.01	5.77	4.26	4.70
η_{ts}	71.27	68.76	73.25	72.88
h_1 (kJ/kg)	404.2	464.3	417.8	437.0
h_4 (kJ/kg)	400.5	459.3	414.0	432.9
C_4 (m/s)	88.9	102.6	90.3	94.2
W_4 (m/s)	32.3	46.2	27.2	30.0
U_4 (m/s)	106.4	132.6	101.3	107.6
C_{m5} (m/s)	41.7	60.4	35.0	39.1
Ma_5	0.64	0.79	0.55	0.58
$(UA)_{evap}$ (kW/°C)	25.4	27.2	31.9	27.3
$(UA)_{tot}$ (kW/°C)	85.9	61.1	115.3	85.3
A_{evap} (m ²)	4.9	12.3	3.4	5.1
r_1 (m)	87.7	113.4	78.3	86.8
r_2 (m)	63.0	80.1	56.6	62.9
r_3 (m)	50.4	65.4	46.7	50.0
r_4 (m)	45.4	58.3	42.4	45.2
r_{tip5} (m)	31.8	40.8	29.7	31.6
r_{hub5} (m)	12.7	16.3	11.9	12.7
L_4 (m)	9.2	11.8	8.2	9.0
Z_{rotor}	13	13	14	14
N_p	10	16	10	10
ΔP_{evap}	90.9	29.1	143.8	71.0
Ω	2345	2272	2386	2381
T_9	73.1	75.7	70.4	72.9
$\dot{m}_{wf}$	4.6	2.8	5.3	3.9

Table 3.8. Optimized results of the TOPSIS solutions for heat source temperature of 120 °C.

Fluids	Butane	R134a	R1234yf	R1234ze(e)
$\dot{W}_t$ (kW)	134.8	136.8	140.0	127.2
$\dot{W}_{net}$ (kW)	128.3	118.5	115.9	112.1
$\eta_{thermal}$	7.22	7.07	7.16	7.63
η_{ts}	72.17	71.55	70.25	71.13
h_1 (kJ/kg)	693.3	444.0	413.5	434.7
h_4 (kJ/kg)	679.8	437.3	407.6	427.9
C_4 (m/s)	166.8	117.7	110.3	118.5
W_4 (m/s)	55.7	40.4	42.6	42.5
U_4 (m/s)	194.0	138.6	135.2	141.5
C_{m5} (m/s)	71.5	51.6	54.8	54.4
Ma5	0.73	0.75	0.80	0.80
$(UA)_{evap}$ (kW/°C)	41.6	46.0	44.2	37.2
$(UA)_{tot}$ (kW/°C)	167.4	180.3	169.2	152.4
A_{evap} (m ²)	9.1	7.8	6.5	6.8
r_1 (m)	160.4	112.6	112.6	114.1
r_2 (m)	115.9	83.0	83.3	84.3
r_3 (m)	91.8	65.8	65.6	68.4
r_4 (m)	84.5	60.7	60.4	63.3
r_{tip5} (m)	59.2	42.5	42.3	44.3
r_{hub5} (m)	23.7	17.0	16.9	17.7
L_4 (m)	13.6	9.5	9.4	9.3
Z_{rotor}	13	13	13	13
N_p	12	14	10	12
ΔP_{evap}	44.6	125.5	132.7	108.9
Ω	2296	2283	2240	2235
T_9	77.7	79.9	81.2	84.9
$\dot{m}_{wf}$	4.4	8.8	9.9	8.0

Table 3.9. Optimized results of the TOPSIS solutions for heat source temperature of 150 °C.

Fluids	Butane	R152a	R134a	Isobutane
$\dot{W}_t$ (kW)	246.7	264.5	268.7	225.1
$\dot{W}_{net}$ (kW)	231.3	227.6	217.2	200.3
$\eta_{thermal}$	8.36	7.74	7.11	7.70
η_{ts}	72.78	73.17	69.10	73.15
h_1 (kJ/kg)	728.9	564.6	451.9	681.1
h_4 (kJ/kg)	712.1	553.2	445.4	666.9
C_4 (m/s)	186.2	153.3	116.2	170.5
W_4 (m/s)	59.4	44.6	49.6	50.3
U_4 (m/s)	212.9	170.1	147.5	190.1
C_{m5} (m/s)	77.0	56.4	64.0	64.0
Ma_5	0.79	0.72	0.83	0.73
$(UA)_{evap}$ (kW/°C)	59.8	76.3	85.6	59.1
$(UA)_{tot}$ (kW/°C)	183.7	299.1	258.4	181.5
A_{evap} (m ²)	15.5	13.0	13.0	11.2
r_1 (m)	168.8	132.7	123.2	154.5
r_2 (m)	123.1	98.7	89.2	115.3
r_3 (m)	100.4	78.8	73.1	88.1
r_4 (m)	93.5	73.7	66.8	82.2
r_{tip5} (m)	65.5	51.6	46.8	57.6
r_{hub5} (m)	26.2	20.6	18.7	23.0
L_4 (m)	13.1	10.1	10.9	11.5
Z_{rotor}	14	14	13	14
N_p	33	20	20	16
ΔP_{evap}	64.6	151.7	172.7	95.1
Ω	2277	2307	2209	2311
T_9	84.6	80.0	77.1	88.3
$\dot{m}_{wf}$	6.5	10.6	16.6	7.3

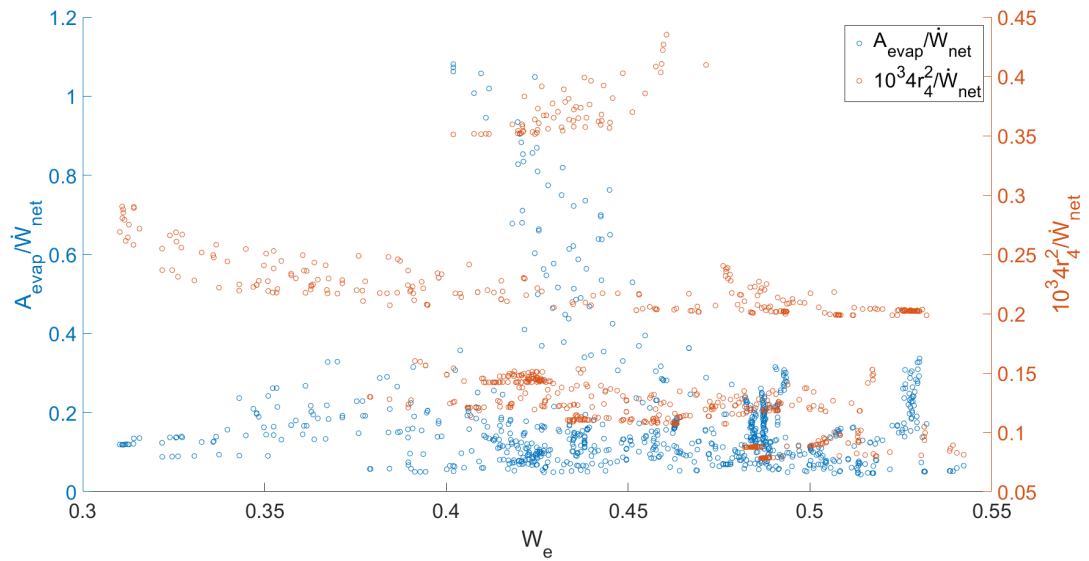


Figure 3.8. Width of the evaporator for all Pareto solutions given in Figures 3.3, 3.4 and 3.5.

of the heat exchanger area. For the same working fluid, the thermal conductance increases with the heat source temperature. For example, for R134a, an increase in the evaporator and total thermal conductance are seen. A similar trend is also observed for the evaporator heat transfer area. Therefore, the thermal conductance can be employed as an indicator of the heat transfer area when used for the same working fluid. However, using thermal conductance to compare heat transfer areas for different working fluids results in inaccuracies due to difference in thermal properties of fluids. For instance, while the evaporator thermal conductance and the heat transfer area of R1234yf at 90 °C are 25.4 kW/°C and 4.9 m² respectively, same parameters are found to be 31.9 kW/°C and 3.4 m² when R143a is used as the working fluid. Therefore, the thermal conductance is not always applicable for the comparison of heat transfer areas of different working fluids even for same heat source temperatures. Although the evaporator heat transfer area is proportional to the first objective function, the value of $A_{evap}/\dot{W}_{net}$ decreases with heat source temperatures due to the fact that the objective function is inversely proportional to the net output power. For example, for R134a, the values are 0.15, 0.07 and 0.06 for 90, 120 and 150 °C respectively from Tables 3.7, 3.8 and 3.9 despite an increase in the heat transfer area.

Thermal efficiency increases with the source temperature which is also shown by Bekiloğlu *et al.* [125]. Thermal efficiency ranges of 4.56-4.88, 6.81-7.24 and 7.01-7.59 are also obtained for all Pareto solutions of R134a for 90, 120 and 150 °C respectively. The thermal efficiency was used as an objective function in some studies [37, 38, 62]. If it were accepted as one of the objective functions in the present study, the optimization results would probably be different and thermal efficiency would probably be higher. However, it would simultaneously cause lower net power output or higher evaporator heat transfer area. Therefore, the selection of the objective functions has an importance in an optimization study, and net power output is determined as essential evaluation parameter together with the installation cost in this study. Indicators used for installation cost should be changed according to the ORC application. For example, the indicators in this manuscript are dominated by drilling cost for a geothermal application [16].

According to Tables 3.7, 3.8 and 3.9, an increase in the turbine rotor size and mass flow rate of the working fluid provide more turbine power output for the same working fluid. The outcomes which are obtained using R134a show that the turbine power output, turbine size and the mass flow rate increase with heat source temperature, on the other hand, the angular velocity of the rotor decreases. However, the turbine size and the mass flow rate are not the only parameters that have an effect on the turbine power output. Working fluid characteristics also influence the output power. For instance, powers of 140.0 kW and 134.8 kW are generated when R1234yf and butane are used respectively at 120 °C, but the radii of the rotor are 60.4 mm and 84.5 mm respectively. Therefore, the usage of R1234yf provides more turbine power with a smaller rotor size. Comparison of R1234yf and R134a at 120 °C is another example; the rotor sizes are almost the same, they are 60.4 mm and 60.7 mm for R1234yf and R134a respectively. A 136.8 kW power is produced with R134a despite a bit bigger rotor size. R1234yf, R134a and butane have molecular weights of 114, 102 and 58 kg/kmol respectively, and molecular weight with other properties such as critical temperature, boiling temperature and likewise caused the circumstances according to [12, 64]. HCs (butane and isobutane) usually have lower mass flow rate when compared to HFCs (R134a and R152a) and HFOs (R1234yf and R1234ze(e)) for source temperatures of

120 and 150 °C.

According to Li *et al.* [115], there is an inverse relation between PPTD and the heat transfer area. Similar outcomes are obtained in this chapter (Tables 3.4, 3.5, 3.6, 3.7, 3.8 and 3.9). For example, bigger heat transfer area is required when butane is used as the working fluid at 150 °C instead of isobutane, while PPTD becomes higher for isobutane. On the other hand, if different working fluids are employed as done in the present study, the inverse relation between PPTD and the heat transfer area cannot be always observed. For example, same heat transfer areas are needed when R152a and R134 are used at 150 °C where PPTD values are 11.71 and 17.57 °C.

4. SUMMARY AND CONCLUSION

There are two main parts in this study, and one of them is about optimization of ORC system with detailed mean-line RIT model. A PHE evaporator model is added in the other part as well, and three geometric parameters of the evaporator (length, width of plates and spacing between plates) are optimized in this part. Multi-objective optimization is conducted in both parts using NSGA-II algorithm. The optimization is carried out using 28 different working fluids for three different heat sources, 90, 120 and 150 °C.

In the first part, objective functions are total thermal conductance per net power output and inverse of the performance factor. The optimization algorithm minimize these functions. According to the results, the working fluid properties have a significant effect on cycle parameters and turbine geometries. R1234yf, R1234ze(e) and isobutane provide the best net power outputs for 90, 120 and 150 °C respectively. Additionally, an ideal working fluid group is suggested for each source temperature, and R134a is present in all ideal groups. However, when R134a is used, a higher degree of superheating is needed to avoid droplet formation in the turbine. Safety and environmental issues should also be considered in the working fluid screening process; for example, R134a is good candidate in the ASHRAE classification, but it may be hazardous to the environment with its high GWP value.

Optimized geometric parameters such as specific speed, radius ratio at the nozzle inlet and outlet and the turbine pressure ratio do not vary significantly with heat source temperature or working fluid type in contrast to PPTD in the evaporator and the degree of superheating that change significantly. The specific speed is generally in the range of 0.50-0.55 for all cases studied. Additionally, the turbine pressure ratio is always lower than four due to unchoked flow constraint at the turbine outlet.

In the second part, evaporator area per net power output is used as an objective function instead of the total thermal conductance per net power output. According to

the results, The highest net power output is obtained for R1234yf at source temperature of 90 °C, and butane provides the best net power outputs for both source temperatures of 120 and 150 °C. R134a is a prominent working fluid because it is in all suggested groups. Environmental and safety issues are also significant for ORC applications, and thus they should be considered in the selection process. For instance, according to ASHRAE classification, butane has high flammability; therefore, another working fluid with a lower net power output can be preferred over butane.

Optimized geometric parameters of the evaporator are generally in a narrow range independent of the working fluid and the source temperature. The suggested values for the length and width of plates and the spacing between plates are 1.2, 0.35 and 0.0035 m respectively. Manner of the other decision variables is similar to the first part. Furthermore, parameters such as specific speed, pressure ratio through the turbine and radius ratio at the nozzle inlet and outlet behave similar to evaporator parameters and do not change significantly with cycle conditions. On the other hand, evaporator PPTD and degree of superheating strongly depend upon working fluid.

An important point that should be highlighted is the selection of the objective functions. The objective functions used in this study are suitable for many ORC applications such as solar thermal, waste heat and biomass applications because turbine and evaporator dominate installation cost. However, reasons of main cost can vary with applications. For example, the drilling cost is dominant in geothermal power plants; thus, the objective functions should cover the cost of drilling in the optimization process.

The results indicate that, although a group of working fluids is promising, one working fluid does not come out to be dominant; a mixture may provide better results, and it is planned as a future work.

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APPENDIX A: VALIDATION

Table A.1. Comparison of the present ORC model including PHE with Kaşka [98].

Parameter	[98]	Present ORC Model	Relative Error (%)
T_9 (°C)	86.2	86.6	0.46
h_9 (kJ/kg)	361.2	362.7	0.42
$\dot{m}_{wf}$ (kg/s)	11.1	11.0	0.90
T_1 (°C)	92.9	93.0	0.11
h_1 (kJ/kg)	469.9	470.1	0.04
h_5 (kJ/kg)	446.2	446.2	0.00
T_6 (°C)	34.9	34.8	0.29
h_6 (kJ/kg)	245.6	245.5	0.04
h_7 (kJ/kg)	246.4	246.6	0.08
$\dot{W}_t$ (kW)	262.2	262.5	0.11
$\dot{Q}_{in}$ (kW)	2479	2455.1	0.96
$\eta_{thermal}$	0.10	0.10	0.00
$\dot{m}_{cw}$ (kg/s)	104	103.4	0.58

Table A.2. Validation of the RIT model comparing with the results of Dai *et al.* [108]-Scenario 1.

Parameter	[108]	Present Study	Error (%)
r_4 (m)	0.246	0.247	0.41
Z_{rotor}	23	23	0.00
k_4	0.850	0.854	0.47
L_4 (m)	0.032	0.032	0.00
r_{tip5} (m)	0.170	0.169	0.59
r_{hub5} (m)	0.068	0.068	0.00
L_{ax} (m)	0.153	0.152	0.65
L_3 (m)	0.032	0.032	0.00
L_2 (m)	0.032	0.032	0.00
r_3 (m)	0.260	0.262	0.77
r_2 (m)	0.280	0.287	2.50
r_1 (m)	0.377	0.390	3.45
H_1 (kJ/kg)	453.500	453.421	0.02
P_{01} (kPa)	462.600	461.539	0.23
C_{m2} (m/s)	15.800	16.092	1.85
P_{03} (kPa)	452.300	451.874	0.09
C_3 (m/s)	117.700	118.351	0.55
Ma_3	0.860	0.865	0.58
P_3 (kPa)	311.400	309.664	0.56
C_4 (m/s)	124.600	126.234	1.31
Ω (rad/s)	469.000	466.015	0.64
H_5 (kJ/kg)	441.000	439.314	0.38
P_{05} (kPa)	221.500	221.303	0.09
P_5 (kPa)	197.600	197.600	0.00
SP (m)	0.120	0.122	1.67
VR	2.287	2.289	0.08
η_{ts}	0.889	0.901	1.35

Table A.3. Validation of the RIT model comparing with the results of Dai *et al.* [108]-Scenario 2.

Parameter	[108]	Present Study	Error (%)
r_4 (m)	0.194	0.196	1.03
Z_{rotor}	23	23	0.00
k_4	0.850	0.854	0.47
L_4 (m)	0.015	0.015	0.00
r_{tip5} (m)	0.133	0.137	3.01
r_{hub5} (m)	0.054	0.055	1.85
L_{ax} (m)	0.118	0.124	5.08
L_3 (m)	0.015	0.015	0.00
L_2 (m)	0.015	0.015	0.00
r_3 (m)	0.201	0.203	1.00
r_2 (m)	0.220	0.222	0.91
r_1 (m)	0.277	0.279	0.72
H_1 (kJ/kg)	486.900	486.917	0.00
P_{01} (kPa)	1569.800	1571.488	0.11
C_{m2} (m/s)	12.400	12.385	0.12
P_{03} (kPa)	1456.000	1459.005	0.21
C_3 (m/s)	185.900	185.785	0.06
Ma_3	1.360	1.350	0.74
P_3 (kPa)	582.900	584.798	0.33
C_4 (m/s)	191.400	193.714	1.21
Ω (rad/s)	942.000	930.337	1.24
H_5 (kJ/kg)	456.600	452.580	0.88
P_{05} (kPa)	269.600	256.134	4.99
P_5 (kPa)	197.600	197.600	0.00
SP (m)	0.100	0.099	1.00
VR	8.700	8.590	1.26
η_{ts}	0.869	0.872	0.35

Table A.4. Comparison of the efficiencies of the present model and those of Hayat *et al.* [51].

Organic Fluid	[51]	Present ORC Model	Relative Error (%)
R152a	0.09	0.09	0.00
R236ea	0.11	0.11	0.00
R245ca	0.12	0.12	0.00
R1234ze	0.09	0.09	0.00
Pentane	0.12	0.12	0.00