

**MULTI-PERIOD OPTIMIZATION OF ORGANIC RANKINE
CYCLES (ORC) DESIGN UNDER WASTE HEAT LOAD
VARIABILITY**

BY

HANI MOHAMMED AL SAED

A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

CHEMICAL ENGINEERING

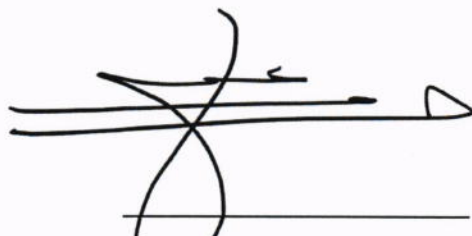
DECEMBER 2021

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN- 31261, SAUDI ARABIA

DEANSHIP OF GRADUATE STUDIES

This thesis, written by **Hani M. Al Saed** under the direction of his thesis advisor and approved by his thesis committee, has been presented and accepted by the Dean of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE IN CHEMICAL ENGINEERING**.



Dr. Hassan M. Baaqeel
Department Chairman



Dr. Suliman Saleh Al-Homidan
Dean of Graduate Studies

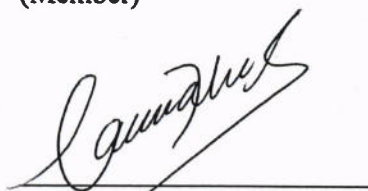
3/1/22
Date



Dr. Hassan M. Baaqeel
(Advisor)



Dr. Eid M. Al-Mutairi
(Member)



Dr. Umer Zahid
(Member)



© Hani M. Al Saed

2021

This work is dedicated to my beloved family

ACKNOWLEDGMENTS

I would like to express my deepest gratitude to my advisor, Dr. Hassan Baaqeel for his guidance and support. I also would like to thank my thesis committee members Dr. Eid Al-Mutairi and Dr. Umer Zahid for their valuable comments and suggestions.

I also would like to express my sincere gratitude to my mentor at work, Dr. Mahmoud Bahy Noureldin for his unlimited support and guidance.

I also place on record, my sense of gratitude to one and all, who directly or indirectly, have lent their hand in this work.

TABLE OF CONTENTS

ACKNOWLEDGMENTS	V
TABLE OF CONTENTS	VI
LIST OF TABLES.....	IX
LIST OF FIGURES.....	X
LIST OF ABBREVIATIONS.....	XI
ABSTRACT	XIII
ملخص الرسالة	XV
CHAPTER 1 INTRODUCTION.....	1
1.1 Industrial Waste Heat	1
1.2 Waste Heat Measures & Classifications	2
1.3 Industrial Waste Heat Sources	5
1.4 Waste Heat Recovery	6
1.4.1 Direct Heating	6
1.4.2 Waste Heat Upgrade	7
1.4.3 Waste Heat to Useful Work.....	8
1.5 Waste Heat Recovery Challenges.....	11
CHAPTER 2 ORC BACKGROUND.....	12
2.1 Working Principle	13
2.2 ORC Manufacturing	15
2.2.1 Ormat	15
2.2.2 Turboden	16

2.2.3	Atlas Copco.....	16
2.2.4	Exergy.....	17
2.2.5	General Electric	17
2.2.6	GMK	17
2.2.7	AQYLON	17
2.3	ORC Working Fluids.....	18
2.3.1	Working Fluid Classification.....	19
CHAPTER 3 ORC DESIGN OPTIMIZATION		22
3.1	Literature Review.....	22
3.2	Problem Definition.....	25
3.3	Problem Description	26
3.4	Mathematical Modeling.....	29
3.4.1	Evaporator Modeling.....	29
3.4.2	Condenser Modeling	38
3.4.3	Expander Modeling	40
3.4.4	Pump Modeling	41
3.4.5	ORC Off-design Modeling	42
3.4.6	Economic Modeling	45
3.4.7	Problem Formulation	48
3.4.8	Solution Approach.....	52
CHAPTER 4 CASE STUDY.....		54
4.1	Problem Background.....	54
4.2	Problem Description	56
4.3	Problem Constraints	59

4.4	Major Assumptions	61
4.5	Solution Approach	62
4.6	Results and Discussion.....	63
REFERENCES.....		71
VITAE		74

LIST OF TABLES

Table 1 Off-design efficiency correlation regression coefficients of the [21].....	44
Table 2 Turton et al. [25] Equipment correlation coefficients.....	46
Table 3 CEPCI for the used correlation reference year and study year.....	46
Table 4 20 years forecasted excess hot oil rate and wasted energy rates.	57
Table 5 Economic details used for this ORC design problem.....	58
Table 6 ORC process and equipment related information.....	58
Table 7 Optimization results for objective function	63
Table 8 Optimization results for the design variables	64
Table 9 Optimization results for the operating variables.....	65

LIST OF FIGURES

Figure 1 Carnot efficiency as a function of temperature	4
Figure 2 Waste heat quantity, quality, and work potential for US	5
Figure 3 Closed cycle heat pump.....	7
Figure 4 Thermo-compressor diagram.....	8
Figure 5 schematic diagram of a vapor absorption system.....	9
Figure 6 Organic Rankine Cycle (ORC) schematic diagram.....	10
Figure 7 Simple subcritical Organic Rankine Cycle schematic diagram	14
Figure 8 Typical ORC temperature-entropy diagram.....	15
Figure 9 ORC working fluids classification	20
Figure 10 Block flow diagram of CGT waste heat recovery and hot oil system.....	27
Figure 11 Evaporation process diagram	30
Figure 12 Typical ORC temperature-entropy diagram.....	30
Figure 13 Solution approach of the simultaneous design and off-design problem.....	53
Figure 14 Single CGT/sales gas unit attached to a hot oil waste heat recovery heater	55
Figure 15 Block flow diagram of CGT waste heat recovery and hot oil system.....	56
Figure 16 Solution approach of the simultaneous design and off-design problem.....	63
Figure 17 Optimum operating evaporation pressure for all the optimization periods.....	66
Figure 18 Evaporator operating approach temperature across operating time periods	67
Figure 19 Evaporator design capacity utilization across all operating time periods	67
Figure 20 Condenser design capacity utilization across all operating time periods	68
Figure 21 Condenser operating approach temperature across operating time periods	69
Figure 22 Effect of the ORC design capacity on the NPV objective function	70

LIST OF ABBREVIATIONS

$N_{u,t}$	Nusselt number for the tube side fluid
k_t	Thermal conductivity of the tube side fluid
d_i	Tube inside diameter [in]
J_C	Baffle window flow Correction factor
J_L	Baffle leakage effects Correction factor
J_B	Bundle bypass effects Correction factor
J_R	Correction factor for Laminar flow
J_S	Unequal baffle spacing Correction factor
D_b	Bundle diameter (outer-tube-limit diameter)
D_o	Tube outside diameter [in]
P_T	Tube pitch
h_{nb}	Heat transfer coefficient for nucleate pool boiling [Btu/h.ft ² .°F]
P_c	Fluid critical pressure [psia]
$\hat{q}$	Heat flux [Btu/h.ft ²]
F_p	Pressure correction factor
P_r	Reduced pressure
ρ_L	Density of condensate
ρ_V	Density of vapor
k_L	Thermal conductivity of condensate
μ_L	Viscosity of condensate
g	Gravitational acceleration
Γ^*	Modified condensate loading
W	Condensate rate

L	Length
n_t	Number of tubes in a bundle
$C_{p,V}$	Heat capacity of the vapor
T_V	Vapor temperature
T_{sat}	Saturation temperature
λ	Latent heat of condensation
$P_{Gen,j}$	Power generation per year
P_{tariff}	Power tariff
$S_{sb},$	Shell-to-baffle leakage area.
$S_{tb},$	Tube-to-baffle leakage area defined as total area of the gaps in one baffle.
$S_m,$	Minimum flow area in one baffle space at the center of the tube bundle.
$N_{ss},$	The number of pairs of sealing strips
$N_c,$	The number of tubes rows crossed between baffle tips

ABSTRACT

Full Name : [Hani Mohammed Al Saed]

Thesis Title : [Multi-Period Optimization of Organic Rankine Cycles (Orc) Design Under Waste Heat Load Variability]

Major Field : [Chemical Engineering]

Date of Degree : [December 2021]

The overwhelming availability of waste heat in industry indicates the inefficiencies current practices used in the industrial sector. Besides the extra money the facilities are paying to operate, it also adds an additional environmental impact. This impact is not just from excess CO₂ production but also from a steady source of high temperature air flowing into the atmosphere. Interests has been shown in low-grade waste heat recovery by Organic Rankine Cycle (ORC). ORC is similar to a typical Rankine Cycle but differs in terms of the type of working fluid. Which is here a low temperature boiling refrigerant.

In practice, operating facilities in Oil and Gas sector often operate on forecasted plans and schedules that impose different plant loads over the plant life cycle. Consequently, the waste heat generated within these facilities would be fluctuating along with plant loads. This kind of load changes results in difficulty for designing a proper ORC size for waste heat recovery. Under-sizing of the ORC will minimize the waste heat recovery efficiency, while over-sizing will make the ORC run at partial loads when the plant is running at lower loads. Operating the ORC at partial loads would lower the efficiency of the turbo-expander and the pump and therefore lower the efficiency of ORC cycle and minimize the power generation.

In order to maximize the waste heat to power conversion in addition to maximizing the economic benefit throughout the life cycle of the project, the optimum sizing of the ORC should be identified from the beginning using optimization techniques. In this work, a multiperiod simultaneous design and off-design optimization model is developed to identify the optimum ORC size subject to all process constraints.

A case study is analyzed and solved for compression gas facility that is driven by combustion gas turbines (CGT) and generate waste heat, flue gases, though the combustion exhaust.

ملخص الرسالة

الاسم الكامل: هاني محمد ال ساعد

عنوان الرسالة: تحسين متعدد المراحل لتصميم دورة رانكن العضوية تحت احمال متغيرة من الطاقة المهدرة

التخصص: الهندسة الكيميائية

تاريخ الدرجة العلمية: ديسمبر 2021

يبين التوافر الهائل للحرارة المهدرة في الصناعة أوجه القصور والممارسات المهدرة المستخدمة حالياً في القطاع الصناعي. هذه الطاقة المهدرة لا تكلف المرافق الصناعية فقط أموالاً إضافية لتشغيلها، بل تضيف أيضاً تأثيراً سلبياً على البيئة. هذا التأثير ليس فقط من إنتاج ثاني أكسيد الكربون الزائد ولكن أيضاً من إنتاج مصدر ثابت للهواء ذو درجة الحرارة العالية الذي يتدفق إلى الغلاف الجوي. وقد أظهرت الاهتمامات في استخدام الحرارة المهدرة منخفضة الدرجة بواسطة دورة رانكن العضوية. يشبه عمل هذه الدورة دورة رانكن التقليدية ولكن تختلف من حيث نوع السائل المستخدم. في الممارسة العملية، غالباً ما تعمل مرافق التشغيل في قطاع النفط والغاز على الخطط والجدول الزمنية المتوقعة التي تفرض أحمالاً مختلفة للمصنع على مدار دورة حياة المصنع. وبالتالي، فإن الحرارة المهدرة المتولدة داخل هذه المرافق سوف تزيد وتنقص مع زيادة ونقصان الأحمال للمصنع. هذا النوع من التغيرات في الحمل يؤدي إلى صعوبة لتصميم حجم دورة رانكن العضوية المناسب لاستعادة الحرارة المهدرة وتحويلها إلى طاقة كهربائية. تصميم حجم صغير من هذه التقنية سوف يقلل من كفاءة استرداد الحرارة المهدرة، في حين أن الإفراط في الحجم سيجعلها تعمل بأحمال جزئية عندما يعمل المصنع بأحمال أقل. تشغيل هذه التقنية في الأحمال الجزئية من شأنه أن يقلل من كفاءة محرك التوربينة والمضخة وبالتالي خفض كفاءة هذه التقنية وتقليل توليد الطاقة الكهربائية الخضراء.

من أجل الاستفادة القصوى من الحرارة المهدرة وتحويلها إلى الطاقة الكهربائية الخضراء بالإضافة إلى تعظيم الفائدة الاقتصادية طوال دورة حياة المشروع، يجب تحديد الحجم الأمثل للتقنية من البداية باستخدام تقنيات التحسين. في هذا العمل، تم تطوير تصميم متزامن متعدد الفترات ونموذج تحسين خارج التصميم لتحديد الحجم الأمثل لتقنية تحويل الحرارة المهدرة إلى الطاقة الكهربائية الخضراء مع مراعاة جميع القيود العملية.

يتم في هذا العمل أيضاً، تحليل دراسة حالة وحلها لمرفق غاز الضغط الذي يتم تشغيله بواسطة توربينات غاز الاحتراق وتوليد حرارة مهدرة مصاحبة مع غازات المداخن.

CHAPTER 1

INTRODUCTION

The well-known severe consequences of the global warming are deriving many environmental protection initiatives. CO₂ emission and other GHG emissions due to industrial expansion and advancement are presenting difficult challenges for all industrial sectors. Therefore, reducing the CO₂ emissions from these industrial sectors become first priority of all other tasks.

1.1 Industrial Waste Heat

Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use. The exact quantity of industrial waste heat is poorly quantified, but various studies have estimated that as much as 20- 50% of industrial energy consumption is ultimately discharged as waste heat [9]. While some waste heat losses from industrial processes are inevitable, facilities can reduce these losses by improving equipment efficiency or installing waste heat recovery technologies. Waste heat recovery entails capturing and reusing the waste heat in industrial processes for heating or for generating mechanical or electrical work.

For many cases, a such system can be improved to reduce the amount of energy wasted for its operation. A major example is a distillation column where high-grade energy is supplied at the reboiler and then rejected at lower grade in the overhead condenser. This amount of waste heat cannot be avoided; however, it can be converted into useful energy using the available technologies.

The most abundant form of this wasted heat in Oil & Gas industries lies within medium to low temperature range. The share volume of this low grade wasted heat makes this temperature range very important in energy recovery. There are many different methods and technologies that can be used to recover, store, transport and consume this energy. These different types of technologies allow the facility to utilize this purchased energy to offset costs elsewhere.

The overwhelming availability of waste heat in industry demonstrates the inefficiencies and the inefficient practices used currently in the industrial sector. The industrial energy consumption in the US is consist of one-third of the total energy consumption of the country [9]. In addition to the environmental impact, this wasted energy increases the industrial facility operating cost. This impact is not just from excess CO₂ production but also from a steady source of high temperature air flowing into the atmosphere.

1.2 Waste Heat Measures & Classifications

The quantity, or heat content, is a measure of how much energy is contained in a waste heat stream, while quality is a measure of the usefulness of the waste heat. Although the quantity of waste heat available is an important parameter, it is not alone an effective measure of waste heat recovery opportunity. It is also important to specify the waste heat quality; As determined by its temperature. The waste heat temperature is a key factor determining waste heat recovery feasibility. In order to enable heat transfer and recovery, it is necessary that the waste heat source temperature is higher than the heat sink temperature. Moreover, the magnitude of the temperature difference between the heat source and sink is an important determinant of waste heat's quality. The source and sink

temperature difference influence the rate at which heat is transferred per unit surface area of heat exchanger, and the maximum theoretical efficiency of converting thermal from of the heat source to another form of energy.

Carnot efficiency for waste heat streams in equation (1.1), is a measure of heat quality, simply the supply temperature of the waste heat stream. It is the theoretical maximum efficiency one can get when the heat engine is operating between two temperatures. In this case, the two temperatures are the waste heat supply temperature, heat source, (T_h) and the ambient temperature, heat sink, (T_c). From Carnot efficiency, the higher the supply temperature, the higher the efficiency.

$$\eta_{Carnot} = 1 - \frac{T_c}{T_h} \quad (1.1)$$

Multiplying Carnot efficiency by the available waste heat quantity would give a comprehensive measure of the waste heat capability to be converted to useful form of energy. Figure 1 illustrate the waste heat Carnot efficiency as a function of the temperature difference between heat source and heat sink [9]. For fixed ambient temperature, the higher the heat source temperature, the higher the Carnot efficiency and therefore the higher the work potential.

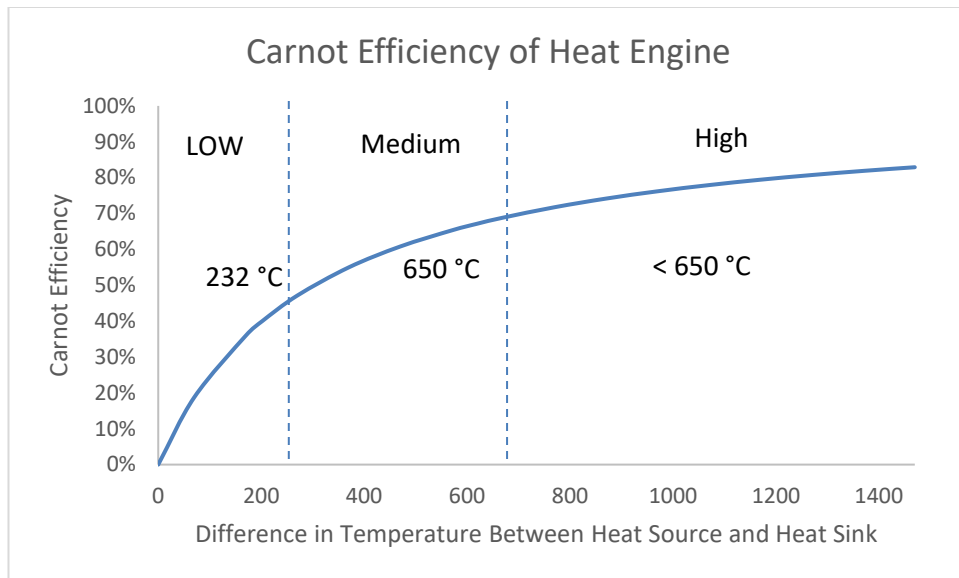


Figure 1 Carnot efficiency as a function of temperature

The waste heat can be classified in terms of quality as follows:

- Low grade waste heat, less than 232 °C
- Medium grade waste heat, between 232 °C and 650 °C
- High grade waste heat, greater than 650 °C

Low temperature waste heat is more dominant than medium and high temperature waste heat. Figure 2 below demonstrates relatively a typical waste heat quantity, quality, and work potential for US [9].

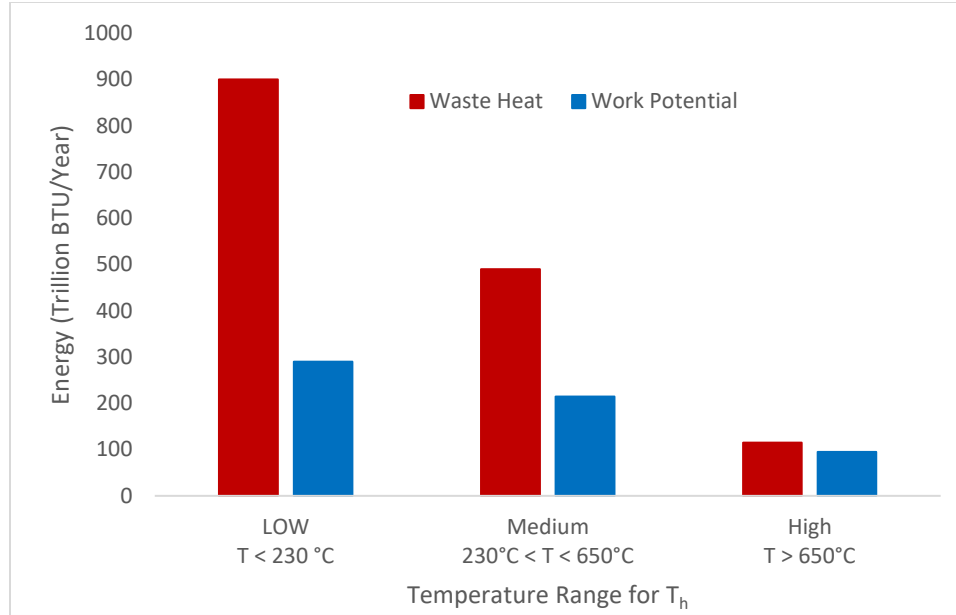


Figure 2 Waste heat quantity, quality, and work potential for US

Although low grade waste heat has a very low conversion efficiency, its huge quantity would still make it more attractive to be pursued. As illustrated in Figure 2, low grade waste heat has the highest work potential among the other classified waste heat. It's clear that from the demonstrated trend in Figure 2 that lower temperature waste heat within the classified low grade waste heat range ($<232^{\circ}\text{C}$) is expected to be more dominant.

1.3 Industrial Waste Heat Sources

Industrial waste heat is the heat generated in a process by way of fuel combustion or chemical reaction, and then “dumped” into the environment even though it could still be reused for some useful and economical purposes. Usually, this low-grade waste heat gets rejected to the environment through air and water coolers. The essential quality of heat is not the amount but rather its “value”. For that, Carnot Cycle efficiency is used as a measure of waste heat quality. The strategy of how to recover this heat depends in part on the temperature of the waste heat stream, heat duty, and economics involved.

Low grade waste heat is widely available in oil and Gas industries. Generally, this low-grade waste heat is rejected to the environment through air and water coolers. The list below provides some of the low-grade waste heat sources in typical industrial facilities:

- Poorly recovered flue gases from fired heaters and boilers
- Reactor effluent streams
- Excess low-pressure steam
- Hot product streams to storage
- Separation equipment overhead streams
- Compression inter-stage streams

1.4 Waste Heat Recovery

Waste heat can be recovered in various ways. Following are common methods for waste heat recovery.

- Direct heating source
- Upgraded by some technologies for use as heating source
- Converted to useful work

1.4.1 Direct Heating

Waste heat can be used as direct heating source to heat part of the process or to preheat the boiler feed water to save energy and reduce GHG emissions. Additional capital cost is required for new heat exchanger area to achieve this kind of waste heat recovery. Heat exchanger approach temperature is a very important factor in recovering the waste heat. The lower the approach temperature the larger the service area of the heat exchanger the higher the investment cost. High investment cost makes this option less attractive except

for the high-quality waste heat which leads to high approach temperature and lower investment cost.

1.4.2 Waste Heat Upgrade

Waste heat can be upgraded in terms of quality before being used as heating source. Vapor compression heat pump is one method used for upgrading the waste heat to a desirable temperature before using it as heat source. The heat pump convert the low-grade heat at the lower temperature to higher temperature heat by using a compressor. Petroleum refining and petrochemicals have been using heat pump in different applications.

Figure 3 shows a typical closed cycle heat pump.

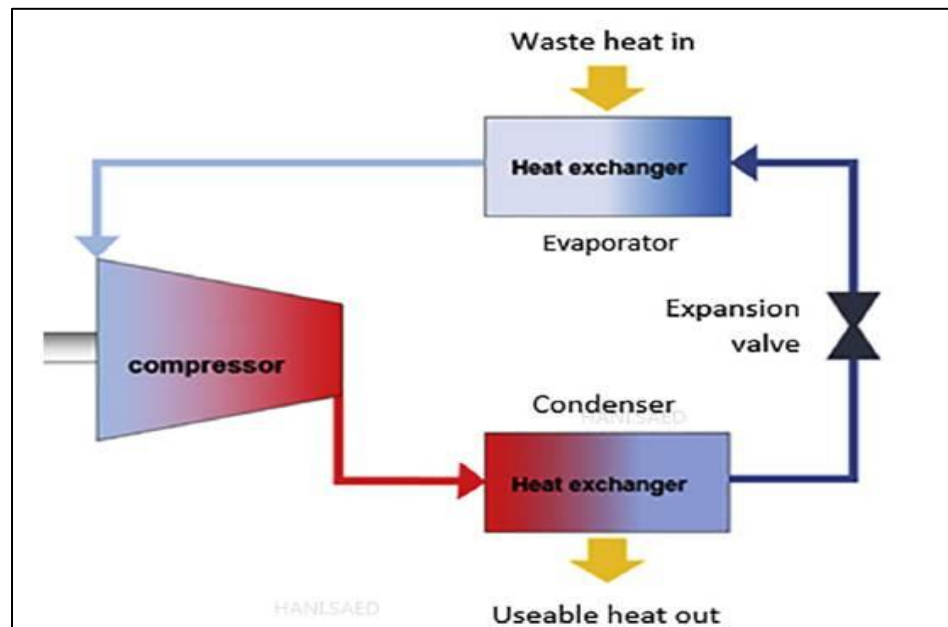


Figure 3 Closed cycle heat pump

The lower temperature heat source is transferred to the working fluid in the evaporator. While the electrical /mechanical energy is used in the compressor to raise the pressure of the vapor existing the evaporator. Higher grade heat with higher temperature is emitted from the condenser. Pressure of the vapor is decreased using a throttle valve to lower the

temperature and convert it to cooled liquid to transfer the heat to low grade heat source. The main problem with the using of the heat pump is that it uses expensive external electrical energy to convert the low-grade heat to higher grade heat. As a rule of thumb, one unit of high-grade electrical energy can generate 2-4 units of high-grade thermal energy.

Thermo-compressor is another method for upgrading waste heat in the form of wasted excess steam. To compress low or intermediate pressure waste steam into medium pressure steam Thermo-compressor uses high pressure steam. Figure 4 illustrate a thermo-compressor in which high pressure steam enters as a high velocity fluid and entrains the low-pressure steam from the suction. The resultant mixture is compressed and produced as a medium pressure steam. The major advantage of thermos-compressor is the higher reliability and less compression power consumption.

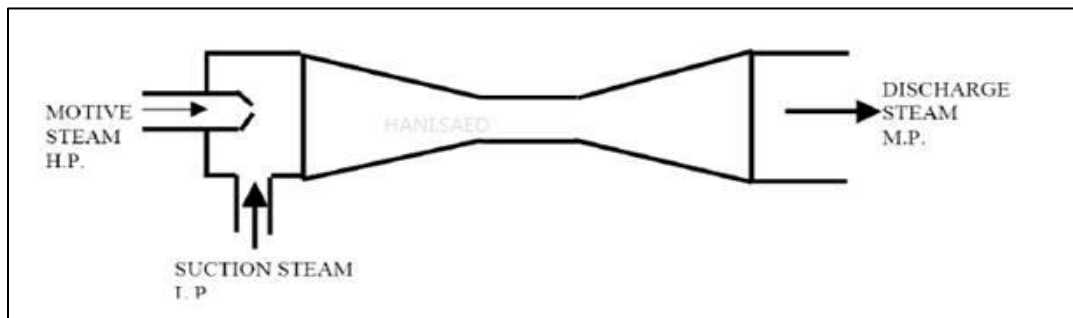


Figure 4 Thermo-compressor diagram

1.4.3 Waste Heat to Useful Work

Waste heat can be utilized in absorption refrigeration system to generate cooling medium to the minimize the energy used by a typical vapor compression system. The absorption refrigeration system differs fundamentally from vapor compression system only in the method of compressing the refrigerant. An absorber, generator and pump in the absorption

refrigerating system replace the compressor of a vapor compression system. Figure 5 shows the schematic diagram of a vapor absorption system.

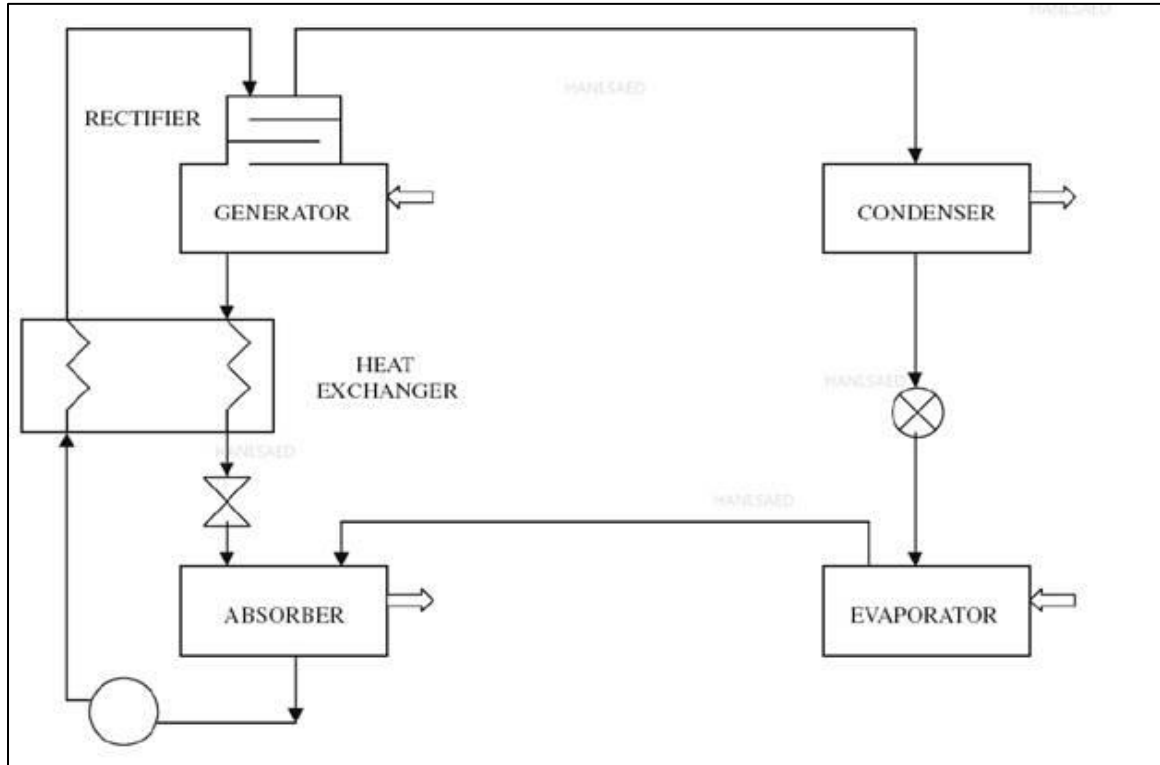


Figure 5 schematic diagram of a vapor absorption system

Waste heat can be also utilized in Organic Rankine Cycle (ORC) to generate electricity. The ORC applies the principle of the steam Rankine cycle, but uses organic working fluids with low boiling points, instead of steam, to recover heat from a lower temperature heat source. The simple Organic Rankine Cycle follows the scheme shown in Figure 5.

Process stream that contains source of heat, enters a heat exchanger(s) recovery system where heat is transferred to a buffer stream (Thermal Oil). The buffer stream that carries the heat, enters the system through the network of heat exchanger(s), in which heat is transferred to the working fluid and the buffer stream temperature returns to its original feed temperature. Typically, there are two stages of heat exchange: one occurs in a

preheater, where the temperature of the working fluid is raised to its bubble point, and the other in an evaporator, where the working fluid is vaporized. After isobaric heat addition, high-pressure vapor is expanded in the turbine which is connected to a power generator. The exhaust vapor of organic fluid from this process is superheated, which is a result of the characteristic retrograde shape of the working fluid saturation line. The superheated stream of exhaust gasses may be sent directly to the condenser, where it is cooled and condensed. However, if economically feasible, exhaust from the turbine may lead to another heat exchanger- regenerator which recovers part of the sensible heat of superheated vapor and transfers it to the working fluid stream entering the preheater. After leaving the condenser, the working fluid enters the pump, where its pressure is increased and returned directly, or through the regenerator, to the preheater, and this cycle will repeat.

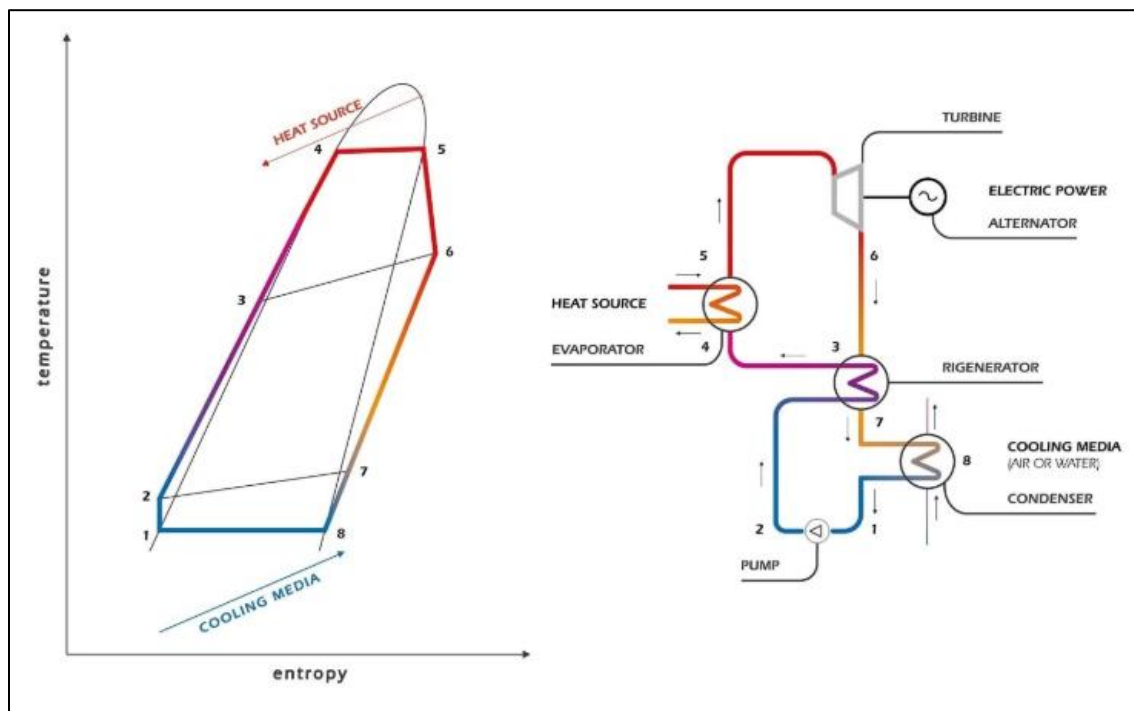


Figure 6 Organic Rankine Cycle (ORC) schematic diagram

1.5 Waste Heat Recovery Challenges

Costs of heat recovery equipment, auxiliary systems, and design services lead to long payback periods in certain applications. Material constraints and costs certain applications require advanced and more costly materials. Overall material costs per energy unit recovered increase as larger surface areas are required for more efficient, lower temperature heat recovery systems. Moreover, equipment costs favor large scale heat recovery systems and create challenges for small scale operations.

Heat transfer rates for low temperature differences between the waste heat source and heat sink, in recovery equipment, lead to reduced heat transfer rates and require larger surface areas. Moreover, gaseous components can condense as hot streams cool in the recovery equipment, leading to potential corrosive and fouling conditions. The additional cost of materials that can withstand corrosive environments often prevents low temperature recovery.

Furthermore, many facilities have limited physical space in which to access waste heat streams. This is often making waste heat recovery a difficult task. Besides, some gaseous waste heat streams are discharged near atmospheric pressure limiting the ability to transport them to and through recovery equipment without additional energy input. In many cases this will make the waste heat recover impractical.

CHAPTER 2

ORC BACKGROUND

Gas and steam cycles are well-known for their use in large-scale stationary power production. Open-cycle gas turbines (often combined with steam cycles) are widely used, especially when low-cost natural gas is available, while the steam cycle is the preferred solution when the characteristics of the energy source, usually coal or nuclear, requires a closed cycle power plant. However, there are a variety of energy sources for which neither the gas cycle nor the steam cycle can provide technically and / or economically viable power generation solutions. When the temperature and/or thermal power available for the power source is limited, it is attractive to use different types of prime movers, usually known by the acronym ORC (Organic Rankine Cycle).

In 1935 Professor Luigi D'Amelio, was the first to use an organic fluid in actual turbine system at the university of Naples [10]. He was also the earliest to examine the benefits of higher molecular mass fluids in reducing turbine rotational speed and number of stages rather than focusing just on equilibrium vapor pressure.



2.1 Working Principle

The Organic Rankine Cycle (ORC) applies the principle of the steam Rankine cycle, but uses organic working fluid with low boiling point, instead of steam, to recover heat from a lower temperature heat source. The simple subcritical Organic Rankine Cycle follows the scheme shown in Figure 7.

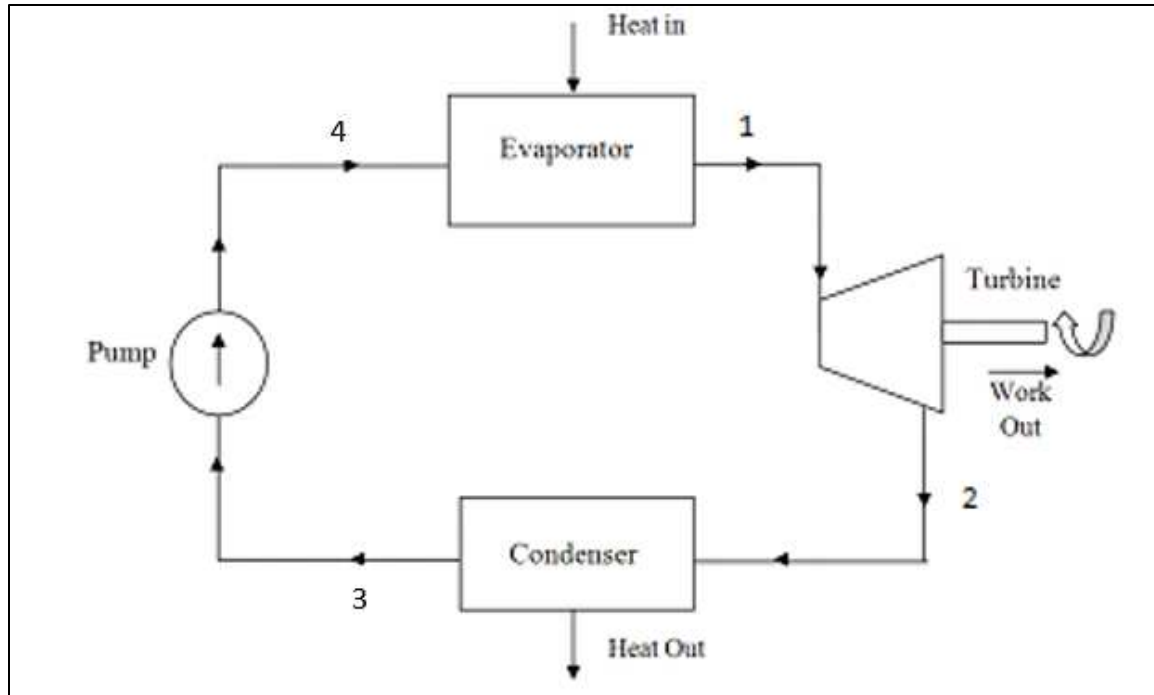


Figure 7 Simple subcritical Organic Rankine Cycle schematic diagram

Waste heat stream enters the evaporator directly or a heat exchanger(s) recovery system where heat is transferred to a buffer stream (Thermal Oil). In the second case, the buffer stream that carries the heat enters the system through the evaporator, in which heat is transferred to the working fluid and the buffer stream temperature returns to its original feed temperature. In both cases the working fluid evaporation occurs in two stages (5-1). The first stage is the preheating where the working fluid is temperature is raised up to its bubble point at the operating pressure. In the second stage the saturated liquid is evaporated at given operating pressure. The generated high pressure organic vapor is then expanded in turbo-expander (1-2) that is connected to a power generator. The exhaust vapor from the turbo-expander is in step 2 is a superheated vapor, which a result of the characteristic retrograde shape of the working fluid saturation line. Figure 8 illustrates the temperature-entropy diagram for a typical ORC.

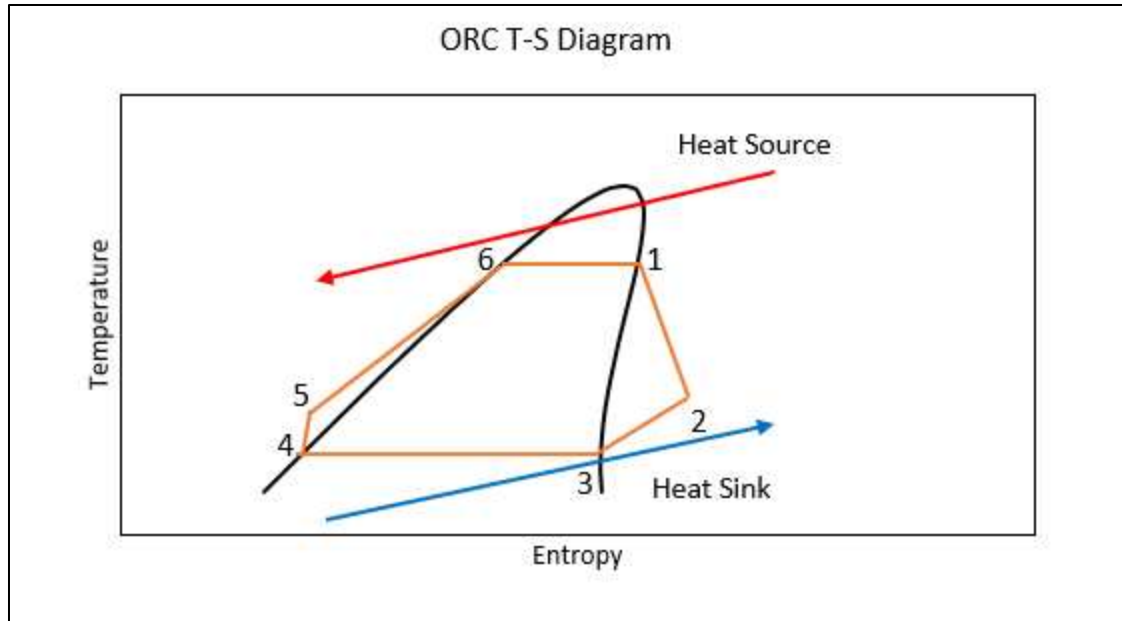


Figure 8 Typical ORC temperature-entropy diagram

The superheated stream of the exhaust gases is condensed in the condenser (2-4). However, in some cases, if economically feasible, the exhaust for the turbo-expander stream is integrated with subcooled fluid to the evaporator, stream 5, to transfer some of its heat to stream 5 and contributes to preheating process before it gets condensed. After leaving the condenser, the working fluid enters the pump to increase its pressure back to the evaporation pressure and the cycle will repeat.

2.2 ORC Manufacturing

ORC technology has gained a lot attention from many industries around the world. In the section some of the major ORC manufacturing company will be presented.

2.2.1 Ormat

The company was founded in 1964 by Lucien and Dita Bronicki with the help of National Preparedness Leadership Initiative (NPLI) to implement Lucien's work and to practice the

patented novel cycle. The company is a leader in power business utilizing renewables. This company designs, builds, owns and operates recovered energy-based power plants and geothermal energy primarily using the ORC of their design and manufacturing. Its original technology is covered by about 100 US patents.

2.2.2 Turboden

Turboden was created in 1980 by Mario Gaia with the goal of studying, engineering, producing, and maintaining ORC turbogenerators [11]. Turboden is a privately held corporation. United Technology Corporation bought 51 percent of Turboden's stock in 2009. The UTC quota was purchased by Mitsubishi Heavy Industries in 2013. The management is still in the hands of the Italian quota holders. Turboden, is similar to Ormat, a big renewable energy player that builds its own turbines.

2.2.3 Atlas Copco

Atlas Copco Group is an international group combined of companies located in Sweden. They focused on compressor design and other industrial equipment. Turbines manufacturing is one of their product. Therefore, they manufacture ORC units as of 2012. They have manufactured a 22.5 MW for geothermal energy and 2 MW using n-butane and 134a as a working fluid respectively. Moreover, they have built a 3.6 MW and two 22.5 MW plants utilizing isobutane in 2015[8].

2.2.4 Exergy

an Italian company firm that specializes in radial outflow turbines. Claudio Spadacini started the firm as a power plant providing engineering services in the late 1990s, and after 2005, it became an ORC power plant supplier.

2.2.5 General Electric

GE developed a 3 kW ORC In the 1970s for mobile homes. currently GE has two ORC product lines. In 2010 GE Energy Infrastructure purchased a subsidiary of Calnetix Inc. to construct the Clean Cycle 125 kW ORC units. The cycle is used to recover exhaust heat from GE Jenbacher gas engines and different other applications. As of beginning 2011, more than 100 ORC systems have been constructed [8]. GE Power system purchased Roto flow In 2000 [13], which is a producer of radial expanders that are used in industry process and power plants using geothermal. GE today has the ORegen ORC line of merchandise for waste heat recovery [5].

2.2.6 GMK

GMK is based in Germany founded by the University of Rostock in. from 2000, around 30 ORC units have been manufactured with a capacity of up to 2 MW. Most units have a capacity of 40 kW and use a superior expander that directly drives the 50 Hz electrical generator.

2.2.7 AQYLON

AQYLON is a global leader in the design and manufacturing of Organic Rankine Cycle (ORC) solutions, which transform heat into electricity and thermal power from renewable sources (biomass, geothermal, solar) or from waste heat created by industrial processes,

engines or gas turbines. Founded in 2009 in Paris by Pierre Convert and Guillaume Jeangros, both alumni from the leading French research institute Ecole Polytechnique, AQYLON represents a new vision for ORC technology integrating innovative fluids, implementing new architectures and offering cycle-accurate design simulation. Beyond the typical equipment manufacturer business model AQYLON becomes an Energy Service Company (ESCO), selling cost-saving electricity solutions to industrial clients.

2.3 ORC Working Fluids

The main task in designing ORC system is the selection of appropriate working fluid for the operating conditions of the waste heat source. This task is very crucial in attaining optimal operating condition for power generation and maintain small capital investment and safe operation. The right working fluid allows for an appropriate volumetric flow rate, which is essential for optimal turbine sizing for all power levels [14], while also determining the right type and capacity of the turbine, pumps, heat exchangers and regenerator in advanced cycles. Xi et al. [26] predicted that hydrocarbons (HCs) are generally the best choice as working fluids for ORCs due to their high conversion efficiency based on data from experiments and environmental factors, low cost and ability to great environmental compatibility, especially alkanes with high molecular weight, are always stable and their explosive and flammability properties can be prevented by the addition of flame retardants. The choice of the working fluid is mainly influenced by the heat source and cooling medium temperatures and general cycle performance. The following are the important considerations for selecting a working fluid for ORC application summarized by Chintala et al. [7]. These includes:

- Capability to dissolve in lubricant oils
- Ecological and Safety factors
- Easiness to handling
- Plentiful availability, and
- Economic factors.

2.3.1 Working Fluid Classification

ORC working fluids have been classified in recent literature as wet, isentropic, or dry. Dry fluid is an organic fluid with high molecular mass and showing a positive slope of saturated vapor line on the temperature-entropy (T-S) diagram as shown in Figure 9. Wet fluid is an organic fluid with low molecular mass that shows a negative slope on the T-S diagram as illustrated in Figure 9. Isentropic fluids are normally have medium molecular mass that show infinite or almost vertical line on the T-S diagram as illustrated in Figure 9. The dryness of the fluid is measured by the slope $m = dT/dS$ and thus the fluid is dry if $m > 0$, wet fluid if $m < 0$, and isentropic fluid if $m = 0$.

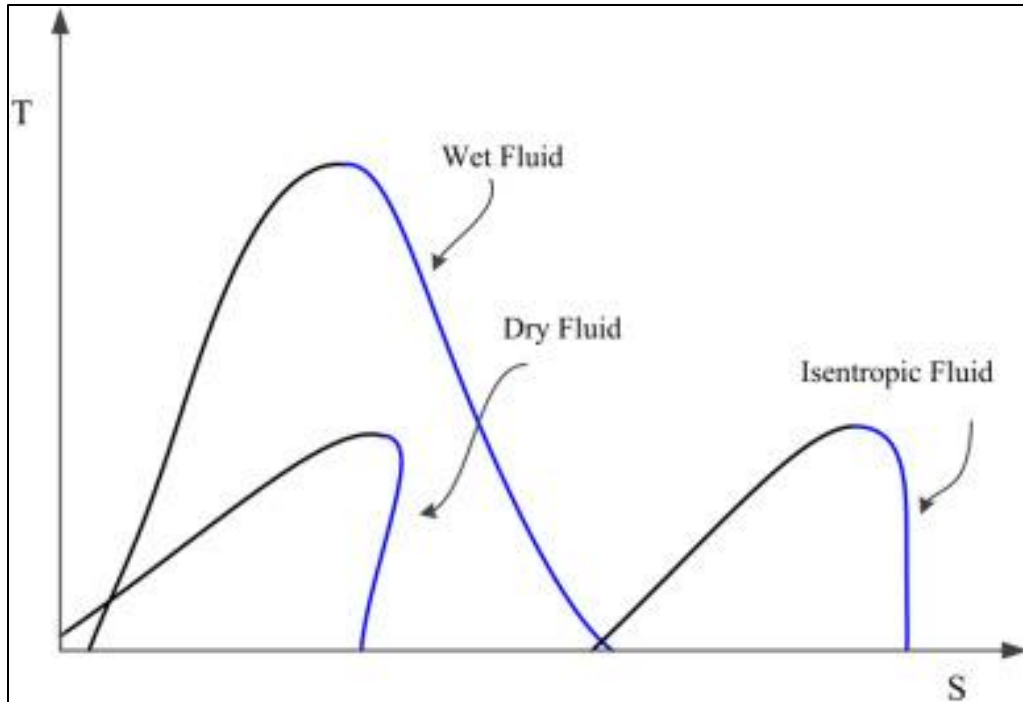


Figure 9 ORC working fluids classification

Wet fluids require superheating due to their high vaporization enthalpies that can be attributed to the presence of hydrogen bonds in some fluids like water, ammonia, and ethanol. Therefore, wet fluids are not good choice for ORC application.

The isentropic and dry fluid are common for subcritical ORC. They can be expanded from the saturation condition without forming liquid droplet at the turbine exhaust. This makes them suitable for ORC application. Extra superheating at the inlet of turbine would make the outlet vapor more superheated and would require a larger condenser. Instead, the generated superheat vapor at the outlet of the turbine without any superheating at the inlet can be used in a regenerator to pre-heat, to some extent, the subcooled liquid to the evaporator and save some service area in the evaporator and improve the overall ORC efficiency in some application.

Since the efficiency of the ORC is highly dependent on the selected dry working fluid and they are for a fixed working state in addition the life of the turbine [2], a process of optimization of the probable working fluids is required to select the appropriate working fluid. fluid simultaneously with ORC Architecture and the turbine selection. [4], therefore, identified 13 basic guidelines for the selection of working fluids, which are shown below:

- Fluid Thermodynamic performance.
- Positive/isentropic saturation vapor curve
- Fluid with high vapor density
- Fluid with low viscosity
- Fluid with high conductivity
- Fluid with reasonable evaporating pressure
- Positive condensation pressure
- Great stability temperature
- Suitable melting point
- Great safety level

CHAPTER 3

ORC DESIGN OPTIMIZATION

3.1 Literature Review

ORC design, mathematical modeling, and optimization have been an interesting topic for many researchers. With the current stringent environment regulation, it become a viable option to recover waste heat from different industries and generate a green power. The power generated from ORC technology is a carbon free comparing to the conventional way of power generation. Furthermore, to pursue ORC optimization problems, a well-defined mathematical model is required. Alireza Javanshir et al. has developed simple mathematical models for subcritical, supercritical, and trans critical ORCs [3]. These models become readily available for any future optimization work.

Many researchers have invested in optimizing operation of the ORC unit by studying the off-design behavior of the unit. Astolfi M. et al. has presented a techno-economical analysis of a geothermal ORC with novel “Emeritus” heat rejection units in hot climates [16]. They included the part-load operation of the ORC for the whole plant optimization.

In the past few years, using the ORC technology to recover waste heat from Gas Turbines have gained a lot of attention. This move was justified by the potential economical added value as well as the energy efficiency improvement that’s associated with green power generation and the CO₂ emission reduction from the conventional power generation.

Design and off-design analysis of the ORC system coupled with Gas turbine was investigated by Alberto Benato et. al. [1]. They invested in optimizing the design of the ORC itself by selecting the best-fit working fluid that will ensure the maximum power

generation, hence maximum efficiency, during design and off-design capacity operation. Moreover, optimizing the off-design operation of ORC engines with different heat exchanger architectures in waste heat recovery application was investigated by Chatzopoulou and his team. They concluded that ORC with plate heat exchanger can slightly generate more power for similar heat source conditions [18].

Many recent studies exist for maximizing the efficiency of the ORC system. One example is the work of Steven Lecompte et al. They developed a thermo-economic optimization using multi-objective technique for subcritical and transcritical ORC cycles for waste heat recovery [23]. In addition, Techno-economic analysis of ORC in gas compression stations taking into account actual operating conditions was investigated by M.Bianchi et. al. They have considered two ORC configurations, direct and indirect, taking into account the effect of the gas compression cycle on the ORC [17]. They came up with the optimum ORC design configuration for a gas compression facility located in North America.

Multiperiod optimization is a technique used to solve the optimization problem over the whole planning horizon. It can be used either for operation or design problems. A clear example to illustrate this technique is the optimization of a boiler steam system that required to be operated on different process demand due to seasonal effect. Guohui Chen et al. have published a Multiperiod Optimal Operational Planning of Boiler Steam System using Line-up Competition Algorithm [12]. Similar technique can be undertaken to formulate an ORC multiperiod optimization problem.

Furthermore, single period optimization of ORC powered by gas turbine was investigated by Sonia L. Gomez-Alaez et al. and a performance model was developed [22]. They came

up with an optimum design and optimum part-load operation of an ORC system during seasonal changes, however they considered a small gas compression variation utilizing annual simulation technique.

ORC sizing multiperiod optimization problem, considering the existing technologies, has been overlooked in the academic research. On the other hand, the development of new efficient ORC designs is being intensively pursued in the academic research. Moreover, operation optimization of the ORC has been also investigated in the academic and industrial research.

Most of available ORC optimization for multiperiod are either design optimization or operation optimization. The design optimization is performed to meet the objective of the optimum design by identifying the independent design parameters such as operating pressures, heat exchangers minimum approach temperatures, and equipment sizing. On the other hand, operation optimization is performed after the design parameters have been already fixed to identify the optimum operating parameters to maximize the ORC net power generation. Xianglong Luo et al. in their recent publication have proposed a method for simultaneous design and off-design ORC optimization [27]. In their work, they have conventional compared two step optimizations, design then off-design for given periods, and simultaneous design and off-design ORC optimization. They have shown and proved the superiority of the simultaneous design and off-design ORC optimization method. In their work the changing parameter over the time periods was the sink temperature and they treated the waste heat source as fixed overall the periods. In fact, in the industry if the waste heat comes from a stack of a combustion engine, which solely dependent on production demand, the main changing variable across the optimization periods would be the waste

heat itself. Moreover, the sink temperature would be a secondary variable that depends on the season change.

This opens an opportunity to pursue practical industrial ORC sizing optimization problem using multiperiod technique.

3.2 Problem Definition

Many industrial facilities are using combustion turbine/engine as driver to generate mechanical power to run their heavy rotating machineries. These kinds of drivers generate a waste heat source with their exhaust gases being at high temperature. In many cases, the generated waste heat is utilized in a waste heat recovery system to generate high-pressure steam or to heat a thermal oil system. A common practice is to collect the flue gases in Hot Oil Recovery Heater (HORH). A portion of hot oil is used for process heating demand within the plant while the remaining is planned to be dissipated in ORC to generate power to improve the overall plant energy efficiency. In the current practice, the remaining portion of the hot oil, other than the one used for internal process heating, is cooled down in a set of air coolers to meet the hot oil return temperature. This kind of waste heat recovery set up is usual designed to handle all range of predicted forecasted facility production rate. Consequently, at some point of the forecasted period, the combustion drivers would be running at partial loads. Thus, the waste heat source quantity and quality would vary accordingly with plant loads. Practically, in almost all industries the temperature of the waste heat is controlled by hot oil recovery system by absorbing any small temperature variation due to the plant load variation. It's important to avoid hot oil temperature change to process heating to avoid any process upset. Therefore, the hot oil system usual equipped with air coolers to control the hot oil return temperature.

Its desirable to come up with an economically optimum ORC size that can handle the wasted hot oil for the available forecasted operating demands. The forecasted demand can be translated into different hot oil loads at multiperiod with different periods' lengths. The hot oil supplied at fixed temperature to meet the process heating demand and must be returned at certain return temperature.

3.3 Problem Description

Many industrial facilities contain rotating equipment such as compressors which are driven by combustion gas turbines (CGTs). The waste heat from the exhaust of the CGT is collected through a hot oil circuit in HORHs. The facility will operate according to given forecasted demand plan. The amount of the produced hot oil from the HORHs depends on the forecasted facility load. The hot oil will always be produced at fixed temperature to meet the process heating demand. Part of the produced hot oil will be used for the process heating demand while the other part will be available for the ORC unit. Figure 10, present the block flow diagram of the CGT waste heat recovery and hot oil system.

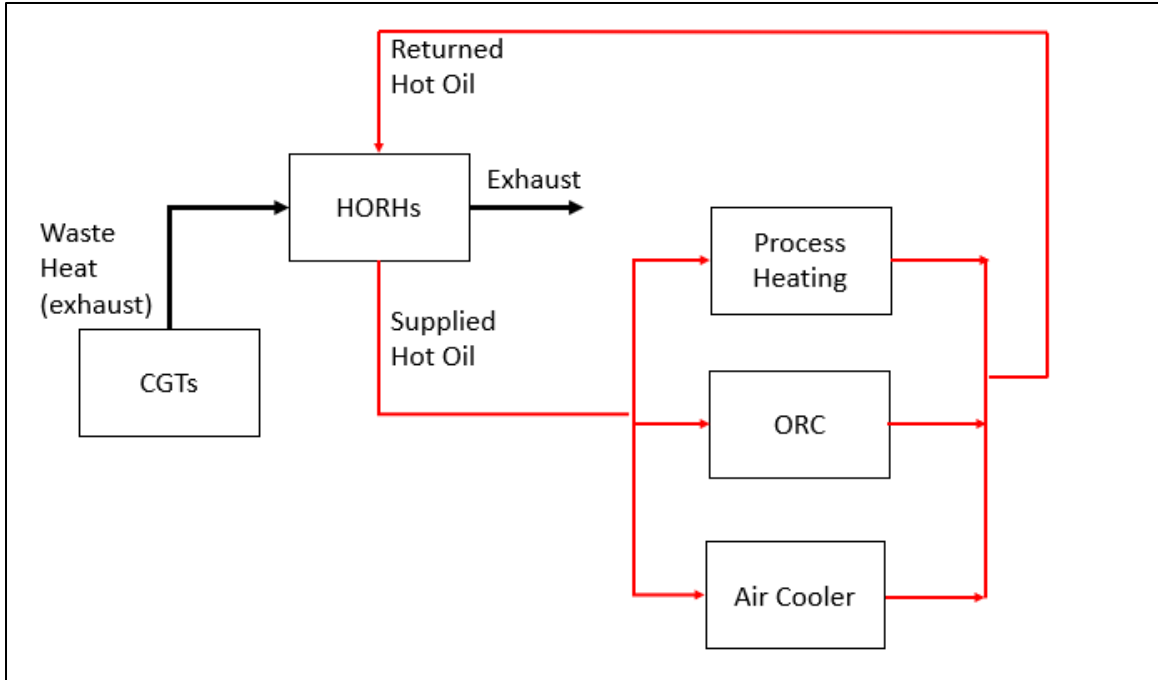


Figure 10 Block flow diagram of CGT waste heat recovery and hot oil system

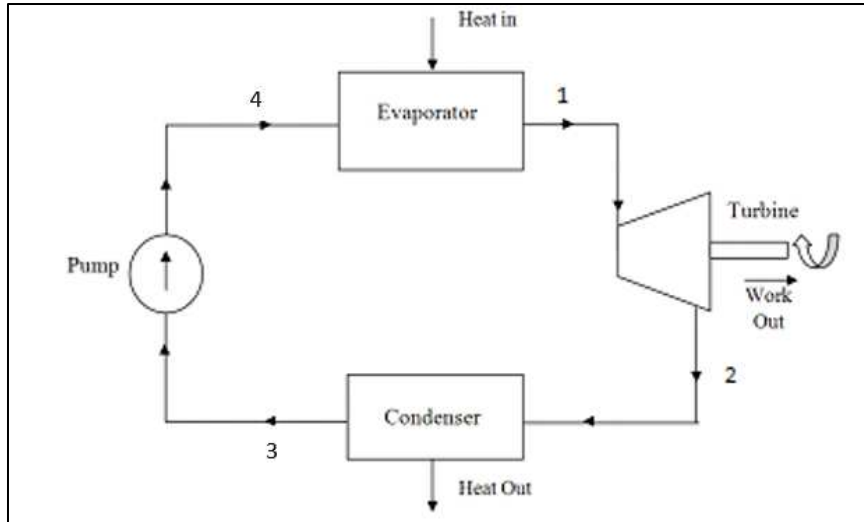
The forecasted demand can be translated into different hot oil production rate. Since the supplied temperature and return temperatures are fixed, each hot oil production rate can be converted into hot oil heat load. The ORC will be operating at fixed hot oil inlet and outlet temperatures. Depending on the size selection of the ORC, it will operate at some period on full load and at some other period at partial load. At some point of operation, the hot oil duty may become more than what the ORC can dissipate, in other words waste heat is more than the design capacity. In that case, part of the extra heat load can be diverted to the existing set of air coolers to dissipate the heat and maintain the desirable hot oil return temperature. This kind of setup will have additional degree of freedom for the ORC sizing optimization problem.

It worth mentioning that this kind of optimization problem can be generalized to solve wide range of ORC multiperiod design optimization problem. In this case, the problem is

extended to include operation optimization in addition to the design problem. Moreover, the operation and design problem can be solved simultaneously to ensure solution optimality. Therefore, the problem in hand becomes a simultaneous multiperiod design and operation optimization problem.

The designing of an ORC that utilizes excess hot oil in a CGT-HORH system is a capacity planning problem that may be stated as follows:

Given a planning horizon of N_t years, $INTERVALS = (t|t = 1, 2, 3, \dots, N_t)$ represents a set of the horizon discretization into annual counterparts. The hot oil flow rate available for ORC at any given interval is denoted by $\dot{M}_{HO,t}$ while the hot oil supplied (T_{WHS}) and returned (T_{WHR}) temperatures are assumed constant. Moreover, a design capacity basis for the system, denoted by C_{des} , can be defined as the heat duty of the hot oil stream used as basis for the design of the ORC system.



The ORC consists of an evaporator, expander, condenser, and pump with the following decision variables:

- ORC design Capacity basis (C_{des})
- Design evaporation pressure ($P_{Evap,des}$)
- Design Condensation pressure ($P_{cond,des}$)
- Working fluid circulation rate ($\dot{m}_{wf,des}$)
- Evaporator design approach temperature ($\Delta T_{app,Evap,des}$)
- Condenser design approach temperature ($\Delta T_{app,cond,des}$)
- Evaporator operating pressure ($P_{Evap,op}$)
- Condenser operating pressure ($P_{cond,op}$)

It is desired to design the ORC with design capacity to yield the minimum net present value (NPV over the planning horizon.

3.4 Mathematical Modeling

The following component of the ORC will be modeled alone then, all the components will be interconnected in a single mathematical model to solve for the optimization decision variables.

- Evaporator
- Condenser
- Expander
- Pump

3.4.1 Evaporator Modeling

The evaporator absorbs the heat from the waste heat source to generate high pressure vapor. In this work Shell & Tube was selected for this kind of large industrial application. For the subcritical ORC application, the evaporator was modeled by splitting the service area into

two parts. The first part is for preheating the working fluid to its bubble point temperature while the second part is for evaporating the working fluid at the bubble point temperature as shown in the diagram below.

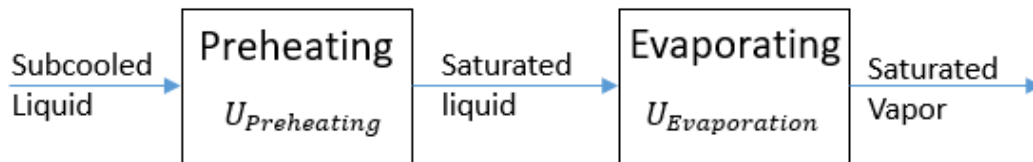


Figure 11 Evaporation process diagram

A kettle type shell & tube heat exchanger is designed for this application where the working fluid is placed in the shell side while the thermal oil placed in the tube side. The evaporator is modeled as two heat exchangers, preheater and evaporation, to have more accurate results for heat exchanger service area calculation.

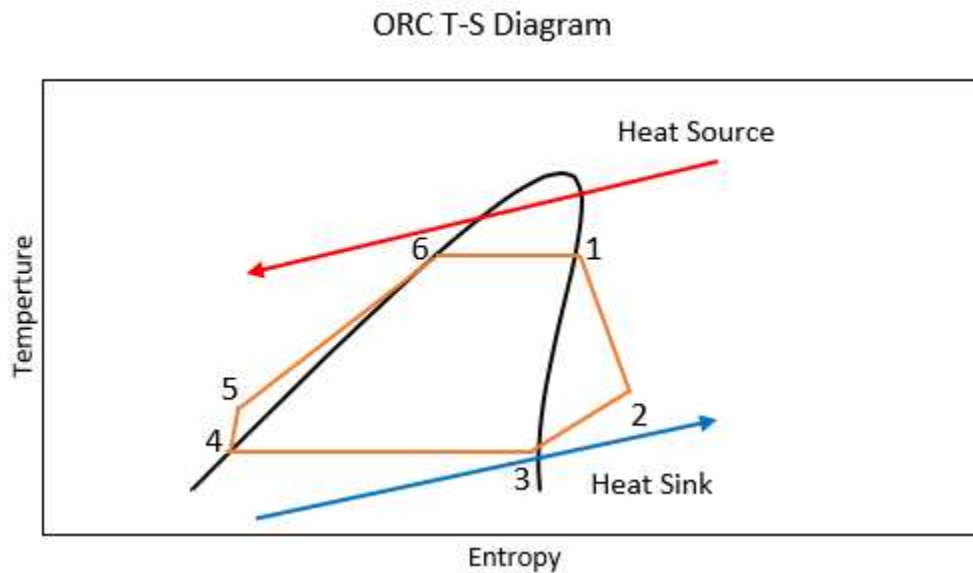


Figure 12 Typical ORC temperature-entropy diagram

Figure 12 Typical ORC temperature-entropy diagram shows working fluid single phase heat transfer taking place between point 5 and 6 while the phase change taking place

between point 6 and 1. The overall heat transfer coefficient is calculated for both single phase heat transfer and phase changer heat transfer as described below.

Energy balance for the preheater and the evaporator are illustrated by equation (1) and (2) at any given time period t respectively.

$$Q_{ph,t} = \dot{M}_{wf,t}(h_{6,t} - h_{5,t}) = \dot{M}_{wf,t}C_{p,wf,t}(T_6 - T_5) \quad (3.1)$$

$$Q_{evp,t} = \dot{M}_{wf,t}(h_{1,t} - h_{6,t}) = \dot{M}_{wf,t} \gamma_{wf,t} \quad (3.2)$$

$$Q_{evp,total,t} = Q_{ph,t} + Q_{evp,t} \quad (3.3)$$

The heat exchanger design equations for preheater and evaporator are given by equation (6) and (7) respectively at any given time period t .

$$Q_{ph} = U_{evp} A_{evp} F \Delta T_{LMTD,ph} \quad (3.4)$$

$$Q_{evp} = U_{ph} A_{ph} F \Delta T_{LMTD,evp} \quad (3.5)$$

The design total evaporator duty is given below:

$$Q_{evp,total,des} = C_{des} = U_{des} A_{des} F \Delta T_{LMTD,des} \quad (3.6)$$

$$Q_{evp,total,des} \geq Q_{evp,total,t} \quad (3.7)$$

Design overall heat transfer coefficient is calculated based on the tube outside diameter, using the following expression at any given time period t :

$$U_{des} = \left[(1/h_i + R_{fi})(D_o/D_i) + \frac{D_o \ln(D_o/D_i)}{2 k_{tube}} + R_{fo} + \frac{1}{h_o} \right]^{-1} \quad (3.8)$$

ΔT_{LMTD} , is a logarithmic average of the temperature difference between the hot and cold streams at each end, A and B, and calculated at any given time period t as follow

$$\Delta T_{LMTD} = \frac{\Delta T_A - \Delta T_B}{\ln(\Delta T_A / \Delta T_B)} \quad (3.9)$$

F, is a correction factor to allow for the deviation from perfect counter-current flow in the heat exchanger.

The heat transfer coefficient is calculated for both single phase heat transfer and phase change heat transfer as described below.

Tube side Single Phase Heat Transfer

For the inner flow in circular tube heat transfer coefficient, h_{in} , Petukhov and Kirillov correlation [15] is used for turbulent flow for the thermal oil, given by equation (10). The correlation is within 5 to 6 % error according to author for the range of $10^4 < Re < 5 \times 10^6$ and $0.5, Pr < 200$.

$$N_{u,t} = \frac{\left(\frac{f}{2}\right) Re Pr}{1.07 + 12.7 \left(\frac{f}{2}\right)^{\frac{1}{2}} (Pr^{\frac{2}{3}} - 1)} \quad (3.10)$$

Where f is given by equation (9),

$$f = (1.58 \ln Re - 3.28)^{-2} \quad (3.11)$$

Equation (10) and (11) are used for both preheater and condenser for inner circular tubes flow heat transfer coefficient.

$$h_i = N_{u,t} \frac{k_t}{d_i} \quad (3.12)$$

Where,

$N_{u,t}$ is the Nusselt number for the tube side fluid

k_t , is the thermal conductivity of the tube side fluid

d_i , is the tube inside diameter

Shell side Single Phase Heat Transfer

Shell side single phase heat transfer coefficient is calculated by Bell Delaware Method [20] using the following equations for preheater shell flow.

The tube bank heat transfer coefficient, h_o , is given by the following equations.

$$h_o = h_{ideal}(J_C J_L J_B J_R J_S) \quad (3.13)$$

Where,

h_{ideal} , is the ideal heat transfer for ideal tube bank multiplied by correction factors to account for deviation from idealities in a baffled heat exchanger.

$$h_{ideal} = j C_p G_s P_r^{-2/3} \left(\frac{\mu_s}{\mu_{s,w}} \right)^{0.14} \quad (3.14)$$

$$j = a_1 \left(\frac{1.33}{P_T/D_o} \right)^a (Re)^{a_2} \quad (3.15)$$

$$a = \frac{a_3}{1 + 0.14 (Re)^{a_4}} \quad (3.16)$$

Where a_1 , a_2 , and a_3 are correlation constants based on layout angle and Reynolds number and available in [20].

Correction factor for baffle window flow (J_C)

The factor is correlated using a single parameter, F_c , the fraction of tubes in cross flow between the baffle tips as shown in equation (15).

$$J_C = 0.55 + 0.72F_c \quad (3.17)$$

Correction factor for baffle leakage (J_L)

$$J_L = 0.44(1 - r_s) + [1 - 0.44(1 - r_s)]\exp(-2.2r_l) \quad (3.18)$$

The correction factors J_L explain the effects of the tube-to-baffle and shell-to-baffle streams leakage on heat transfer and correlated based on the following area ratios.

$$r_s = \frac{S_{sb}}{S_{sb} + S_{tb}} \quad (3.19)$$

$$r_l = \frac{S_{sb} + S_{tb}}{S_m} \quad (3.20)$$

Correction factor for bundle bypass effects (J_B)

$$J_B = \exp \left[-C_J \left(\frac{S_{sb}}{S_m} \right) (1 - \sqrt[3]{2r_{ss}}) \right] \quad \text{for } r_{ss} < 0.5 \quad (3.21)$$

$$J_B = 1.0 \quad \text{for } r_{ss} \geq 0.5$$

Where,

$$C_J = 1.35 \text{ for } \text{Re} < 100$$

$$= 1.25 \text{ for } \text{Re} \geq 100$$

The correction factors J_B express bypass flow on heat transfer effect of the bundle and depends on the ratio of bypass flow area to cross-flow area, and the number of pairs of sealing strips.

$$r_{ss} = N_{ss}/N_c \quad (3.22)$$

Correction factor for laminar flow (J_R)

The correction factor J_R adjusts for the fact that in laminar flow the heat transfer coefficient declines with downstream distance, which in the present context is interpreted as the number of crossed tube rows.

$$J_R = (10/N_{ct})^{0.18} \quad \text{for } Re \leq 20 \quad (3.23)$$

$$J_R = 1.0 \quad \text{for } Re \geq 100$$

For $20 < Re < 100$, J_R is calculated by linear interpolation between the above values.

The correlating parameter is the total number, N_{ct} , of tube rows crossed in the entire exchanger, i.e., in the inlet, outlet, and central baffle spaces, and in the baffle windows. The detailed calculation of the N_{ct} , is given in [20].

Correction factor for unequal baffle spacing (J_S)

The correction factors J_S depend on the inlet baffle spacing, B_{in} , the outlet baffle spacing, B_{out} , the central baffle spacing, B , and the number of baffles, n_b . They are calculated from the following equations.

$$J_S = \frac{(n_b-1)+(B_{in}/B)^{(1-n)}+(B_{out}/B)^{(1-n)}}{(n_b-1)+(B_{in}/B)+(B_{out}/B)} \quad (3.24)$$

Where,

$$n = 0.2 \text{ for } Re \geq 100$$

$$= 1.0 \text{ for } Re < 100$$

For $B_{in} = B_{out} = B$ equation (22) gives $J_S = 1.0$.

Shell side Phase Change Heat Transfer

Kettle reboiler include boiling on the external surfaces of the tubes installed in horizontal bundles. Heat transfer coefficient for boiling on tube bundle involve convection circulation effect along with nucleate pool boiling phenomena. The average boiling heat transfer coefficient, h_b , can be approximated using Palen method [19] as shown below.

$$h_b = h_{nb}F_b + h_{nc} \quad (3.25)$$

where h_{nc} is a heat-transfer coefficient for liquid-phase natural convection and F_b is a factor that accounts for the effect of the thermosyphon-type circulation in the tube bundle. The bundle convection factor is correlated in term of bundle geometry by the following expression [19].

$$F_b = 1.0 + 0.1 \left[\frac{0.785D_b}{C_1(P_T/D_o)^2 D_o} - 1.0 \right]^{0.75} \quad (3.26)$$

Where,

D_b , bundle diameter (outer-tube-limit diameter)

D_o , tube OD

P_T , tube pitch

$C_1=1.0$ for square and rotated square layouts

=0.866 for triangular layouts

The coefficient h_{nc} can be roughly approximated as Palen suggested [19] as 250 W/m². K (44 Btu/h.ft². °F) for hydrocarbons and 1000 W/m².K (176 Btu/h.ft². °F) for water and aqueous solutions.

Nucleate pool boiling phenomena have been extensively studied and many heat transfer correlations have been projected for this regime. For this work the Mostinski correlation is used which is in accordance with the principle of corresponding states and expressed as follows [20].

$$h_{nb} = 0.00622 P_c^{0.69} \hat{q}^{0.7} F_p \quad (3.27)$$

Where,

h_{nb} , heat transfer coefficient for nucleate pool boiling [Btu/h.ft². °F]

P_c , fluid critical pressure [psia]

$\hat{q}$, heat flux [Btu/h.ft²]

F_p , pressure correction factor

The pressure correction factor is expressed by the following equation

$$F_p = 1.8 P_r^{0.17} + 4 P_r^{1.2} + 10 P_r^{10} \quad (3.28)$$

Where,

$P_r = P/P_c$, reduced pressure

3.4.2 Condenser Modeling

Condenser is used to condense the superheat vapor from the expander outlet at low pressure. The condensation pressure is selected based on the available colling medium temperature to make sure that proper condensation will take place. Since the inlet vapor to the condenser is superheated, the condenser must be capable of providing precooling duty as well as condensation duty.

Energy balance for the condenser is illustrated by equation (29) at any given time period.

$$Q_{Cond,t} = \dot{M}_{wf,t}(h_{4,t} - h_{2,t}) \quad (3.29)$$

$$Q_{Cond,t} = \dot{M}_{wf,t}C_{p,wf,t}(T_2 - T_3) + \dot{M}_{wf,t} \gamma_{wf,3,t} \quad (3.30)$$

The heat exchanger design equation for condenser is given by the following equation at any given time period t.

$$Q_{cond} = U_{cond} A_{cond} F \Delta T_{LMTD,cond} \quad (3.31)$$

Design overall heat transfer coefficient is calculated based on the tube outside diameter, using the following expression at any given time period t:

$$U_{ds} = \left[(1/h_i + R_{fi})(D_o/D_i) + \frac{D_o \ln(D_o/D_i)}{2 k_{tube}} + R_{fo} + \frac{1}{h_o} \right]^{-1} \quad (3.32)$$

In this application a shell and tube heat exchanger is used to provide precooling and condensation for the superheated vapor. The condensation will take place in the shell side on a horizontal tubes bundle. For a circular tube bundle used in shell-and-tube condenser

Kern [14] proposed the following modified Nusselt's equation for estimating average heat transfer coefficient.

$$h_o = 1.52 \left[\frac{k_L^3 \rho_L (\rho_L - \rho_V) g}{4 \mu_L \Gamma^*} \right]^{1/3} \quad (3.33)$$

$$\Gamma^* = \frac{W}{Ln_t^{2/3}} \quad (3.34)$$

Where,

ρ_L , Density of condensate

ρ_V , Density of vapor

k_L , Thermal conductivity of condensate

μ_L , Viscosity of condensate

g , Gravitational acceleration

Γ^* , Modified condensate loading

W , Condensate rate

L , Length

n_t , Number of tubes in a bundle

To account for the superheating of the inlet vapor, the following modification to equation (28) is applied.

$$\frac{h}{h_{Nu}} = \left[1 + \frac{C_{P,V}(T_V - T_{sat})}{\lambda} \right]^{1/4} \quad (3.35)$$

Where,

h_{Nu} , is the heat transfer coefficient calculated from equation (33)

For tube side heat transfer coefficient, the previously described tube side single phase heat transfer coefficient is used, equation (10) to (12).

3.4.3 Expander Modeling

The expander is the key component of an ORC which can be designed depending on the size of the plant and the working fluid properties. The main function of the expander is to expand the saturated high pressure organic vapor to the condensation pressure. The work of expanded vapor is then recovered and converted to electrical power using electrical generator attached to the expander.

The power generated $\dot{W}_{T,t}$ in any time period t from the vapor expansion to the desired outlet pressure is calculated using the following equations.

$$\dot{W}_{T,t} = h_{2,t} - h_{1,t} \quad (3.36)$$

Equation (36) calculate the turbine power consumption as the actual enthalpies difference at inlet and outlet conditions.

$$\dot{W}_{T,t} = (h_{2,is,t} - h_{1,t}) * \eta_{T,t} \quad (3.37)$$

Equation (37) calculate the turbine power consumption as the difference between outlet isentropic enthalpy and inlet actual enthalpy divided by pump isentropic efficiency.

The expander isentropic efficiency, η_T , plays a very important role in off-design performance as the load varies from the design point.

The generated power from the expander is calculated at each time period. The expander outlet temperature is also estimated using the enthalpy and pressure difference. Predicting

turbine outlet temperature is important to properly estimate the condenser area requirement. Because the electrical generator has high efficiency and small variation relative to the expander, the generator efficiency is neglected and expected not to have a major impact on the optimization result.

3.4.4 Pump Modeling

A typical centrifugal pump is required to provide the desired working fluid evaporating pressure and establish circulation flow rate.

ORC pumps are usually variable speed multistage centrifugal pumps and their design is relatively common because of their wide use in both the petrochemicals and refinery fields. Depending on the cycle configuration, fluid selection and cycle design parameters, the pump might show a consumption which is relatively small compare to the turbine gross power output. Therefore, the pump efficiency will have a minimal effect on the ORC efficiency during off-design operation. The effect is even more negligible for large size industrial application.

The power requirement $\dot{W}_{P,t}$ in time period t to pump the working fluid to the desired discharge pressure is calculated as follows.

$$\dot{W}_{P,t} = h_{p,out,t} - h_{p,in,t} \quad (3.1)$$

Equation 1 calculate the pump power consumption as the actual enthalpies difference at inlet and outlet conditions.

$$\dot{W}_{P,t} = \frac{h_{p,is,out,t} - h_{p,in,t}}{\eta_{P,t}} \quad (3.2)$$

Equation 2 calculate the pump power consumption as the difference between outlet isentropic enthalpy and inlet actual enthalpy divided by pump isentropic efficiency.

$$\dot{W}_{P,t} = \frac{(P_{out,t} - P_{in,t}) \dot{M}_{wf,t}}{\rho_{wf,t} \eta_{P,t}} \quad (3.3)$$

Equation 3 is another representation for pump power consumption using pump inlet and outlet pressure difference, mass flow rate and density of the working fluid along with pump isentropic efficiency.

The pump power consumption is calculated at each time period with fixed pump efficiency. The pump outlet temperature is also estimated from the pumping effect using the enthalpy and pressure difference. Predicting pump out let temperature is important to properly estimate the preheating area requirement.

3.4.5 ORC Off-design Modeling

ORC off-design model is very crucial in order to assess the ORC performance along the life cycle at different waste heat load. During ORC off-design operation evaporation pressure, temperatures, and working fluid mass flow rate can change to accommodate the waste heat load change. These variables change is mainly effecting the following parameter:

- Preheater overall heat transfer coefficient
- Evaporator overall heat transfer coefficient
- Condenser overall heat transfer coefficient
- Expansion turbine efficiency

- Pump efficiency
- Pressure drop

In this study, pump efficiency change will be neglected due to the relatively small impact on the ORC overall efficiency.

The off-design performance of the heat exchangers (preheater, evaporator, and condenser) is predicted using the following relation at any given time period.

$$U_t = U_{des} \left(\frac{\dot{m}_{wf,t}}{\dot{m}_{wf,des}} \right)^n \quad (3.38)$$

Where the exponent n is given below,

$n = 0.63$ for the preheater

$= 0.56$ for the evaporator

$= 0.57$ for the condenser

These exponent values were obtained for each heat exchanger from calculating the overall heat transfer coefficient at different flow rates and then correlating the results using equation (38) to solve for the exponent n .

The cycle pressure drop, ΔP_{ds} , including main heat exchangers are assumed at the design rated point and estimated at off-design operation by the following simple relation.

$$\Delta P_t = \Delta P_{des} \left(\frac{\dot{m}_{wf,t}}{\dot{m}_{wf,des}} \right)^2 \quad (3.39)$$

The off-design expansion turbine efficiency is predicted using the following correlation proposed by [21].

$$\frac{\eta_t}{\eta_D} = a + b \left(\frac{\Delta h}{\Delta h_D} \right) + c \left(\frac{\Delta h}{\Delta h_D} \right)^2 + d \left(\frac{\dot{V}_{out}}{\dot{V}_{out,D}} \right) + e \left(\frac{\dot{V}_{out}}{\dot{V}_{out,D}} \right)^2 + f \left(\frac{\Delta h}{\Delta h_D} \right) \left(\frac{\dot{V}_{out}}{\dot{V}_{out,D}} \right) \quad (3.40)$$

Where a, b, c, d, e, and f are regression coefficients given in Table 1.

Table 1 Off-design efficiency correlation regression coefficients of the [21]

Coefficients	Number of stages		
	1	2	3
a	0.245	0.418	0.548
b	1.632	1.066	0.937
c	-1.940	-0.568	-0.609
d	0.033	0.057	-0.056
e	-1.085	0.000	-0.043
f	2.112	0.035	0.228

The above off-design efficiency correlation can be further simplified for the purpose of the simultaneous design and operation optimization. Under the assumption of the fixed evaporation pressure control during off-design operation or even small evaporation pressure change, along with the constant outlet pressure of the expander due to fixed cooling water supply temperature, the specific enthalpy change factor in equation (40) can be neglected. Therefore, equation (40) can be further simplified to the following correlation.

$$\frac{\eta_t}{\eta_D} = A + B \left(\frac{\dot{V}_{out}}{\dot{V}_{out,D}} \right) + C \left(\frac{\dot{V}_{out}}{\dot{V}_{out,D}} \right)^2 \quad (3.41)$$

Where, correlation coefficients are given by the following relation

$$A = a + b + c \quad (3.42)$$

$$B = d + f \quad (3.43)$$

$$C = e \quad (3.44)$$

3.4.6 Economic Modeling

The economic model is based on the estimated purchased cost of each individual equipment including evaporator, condenser, expander, generator, pump, piping, and instrumentation. The purchased cost of each individual component is escalated using the Chemical Engineering Plant Cost Index (CEPCI).

The purchased cost of the heat exchanges, pump, and turbine are calculated using the correlations of Turton et al. [25], as shown below.

$$\log_{10} C_p^0 = K_1 + K_2 \log_{10}(A) + K_3 [\log_{10}(A)]^2 \quad (3.45)$$

C_p^0 , is the base cost at ambient pressure using carbon steel, while A is the capacity/size of the equipment, service area for the heat exchanger and power for pump and turbine. The equipment correlation coefficients K_1 , K_2 , and K_3 for CEPCI of the year 2001 are given in Table 2.

Table 2 Turton et al. [25] Equipment correlation coefficients

Equipment	Correlation coefficients		
	K1	K2	K3
Heat Exchangers	4.1884	-0.2503	0.1974
Pump	3.3892	0.0536	0.1538
Turbine	2.7051	1.4398	-0.1776

The base cost for the generator is calculated from Toffolo et al. correlation [24] based on the power generated, P and CEPCI for the year 1993.

$$C_p^0 = 1,850,000 \left(\frac{P}{11,800} \right)^{0.94} \quad (3.46)$$

The estimated base cost form both correlations are modified from the base CEPCI to the CEPCI for the year 2020 according to the following table.

Table 3 CEPCI for the used correlation reference year and study year

Year	CEPCI
1993	359.2
2001	394
2020	596.2

The modified base cost can be obtained from the calculated base cost at the correlation reference year using the following equation.

$$C_{p,2020}^0 = C_{p,Ref}^0 \left(\frac{CEPCI_{2020}}{CEPCI_{Ref}} \right) \quad (3.47)$$

After obtaining the modified base cost for ambient pressure and carbon steel as a material of construction, the bare module cost, C_{BM} , can then be calculated by multiplying the modified base cost by the bare module cost factor, F_{BM} , which depends on the operating pressure and specific material of construction as illustrated below [25].

$$C_{BM} = C_{p,2020}^0 F_{BM} \quad (3.48)$$

$$F_{BM} = B_1 + B_2 F_M F_p \quad (3.49)$$

Where B_1 , B_2 , F_M , and F_p are factors related to specific equipment operating pressure and material of construction and available in [25].

The total bare module cost is calculated by adding all the individual equipment bare module cost multiplied by some factor to account for instrumentation and piping cost as shown below. In this study a 3% of the total equipment cost are used as a factor for piping and instrumentation cost.

$$C_{TM} = 1.03 \sum_i^n C_{BM,i} = CAPEX \quad (3.50)$$

Project Net Present Value (NPV)

The NPV calculation consist of the three following components:

- Capital investment (CAPEX)
- Operating cost (OPEX)
- Revenue (Rev)

The ORC capital investment is calculated as illustrated above in USD, while the operating cost is estimated as 5% of the estimated capital investment. The capital investment is

assumed to be incurred in the beginning of the year and the operating cost is incurred at the end of each year. Revenue is also generated at the end of each year. Therefore, revenue and operating cost must be converted to present worth to compare it with capital investment at some discount rate (i) and known project life cycle (n).

$$NPV_{Project} = Rev_{tot,PV} - OPEX_{tot,PV} - CAPEX \quad (3.51)$$

$$Rev_{tot,PV} = \sum_j^n \left[\left(\frac{1}{(1+i)^j} \right) \left(P_{Gen,j} \times \frac{Operating\ Hours}{Year} \times P_{tariff} \right) \right] \quad (3.52)$$

Where,

$j = 1 \text{ to } n$

$P_{Gen,j}$, is the power generation per year

P_{tariff} , is the power tariff

The total operating power is converted to present worth using the following equation.

$$OPEX_{tot,PV} = \left(\frac{(1+i)^n - 1}{(1+i)^n} \right) (CAPEX \times 0.05) \quad (3.53)$$

Where CAPEX can be estimated using equation (50).

3.4.7 Problem Formulation

The problem is formulated to includes both design and off-design performance behavior of a subcritical ORC to recover the waste heat from the supplied hot oil. The objective function is minimizing the NPV of the ORC installation. This require obtaining the optimum process design variables, PDV , and then obtain the optimum process operation variables, POV , in all discretized time periods.

Objective Function:

$$\max NPV_{Project} (PDV_{ORC}, POV_i) \quad (i \in t) \quad (3.54)$$

The process design variables (PDV) include the following decision variables:

- ORC design Capacity (C_{des})
- Design evaporation pressure ($P_{Evap,des}$)
- Design Condensation pressure ($P_{cond,des}$)
- Working fluid circulation rate ($\dot{m}_{wf,des}$)
- Evaporator design approach temperature ($\Delta T_{app,Evap,des}$)
- Condenser design approach temperature ($\Delta T_{app,cond,des}$)

All of the process design variables will be determined first and then be fixed across all time periods. These parameters will have direct effect in the ORC CAPEX, OPEX, and Revenue. On the other hand, process operation variables (POV) will have direct effect in the Revenue only. The process operation variables include the following decision variables:

- Evaporator operating pressure ($P_{Evap,op}$)
- Condenser operating pressure ($P_{cond,op}$)
- Working fluid circulation rate ($\dot{m}_{wf,op}$)
- Evaporator design approach temperature ($\Delta T_{app,Evap,op}$)
- Condenser design approach temperature ($\Delta T_{app,cond,op}$)

Constraints:

Design capacity constraint

The design capacity depends on the forecasted waste heat availability. Thus, the design capacity should not exceed the maximum value of the waste heat availability. Also, it should not be less than the minimum value of the waste heat availability.

$$WH_{av,min} \leq C_{Des} \leq WH_{av,max} \quad (3.55)$$

Pressure constraints

Because we are dealing with subcritical ORC, the design and operating evaporation pressure should not exceed the critical pressure of the working fluid component in use. Moreover, the evaporation pressure should be high enough to run the expander and generate work. This condition is to limit the search space to a reasonable viable value.

$$P_{Ev,lim} \leq P_{Ev,des} \leq P_C \quad (3.56)$$

$$P_{Ev,lim} \leq P_{Ev,op,i} \leq P_C \quad (i \in t) \quad (3.57)$$

The design and operating condensation pressure correspond to condensation temperature that should be greater than the cooling water supply temperature for the condensation to take place. In addition, the condensation pressure should be low enough to run the expander and generate work. Similarly, this condition is to limit the search space to a reasonable viable value.

$$P_{Dew,CwT} < P_{cond,des} \leq P_{cond,lim} \quad (3.58)$$

$$P_{Dew,CwT} < P_{cond,op,i} \leq P_{cond,lim} \quad (i \in t) \quad (3.59)$$

$P_{Dew,CwT}$ is the working fluid dew point pressure at the cooling water supply temperature

Approach temperature constraints

Minimum approach temperature for both evaporator and condenser at design conditions must be confined within a reasonable acceptable industrial design limit to avoid any thermodynamic laws violation at operating condition and provide reasonable heat exchanger size at design stage. In addition, an upper limit is used for the design case using the industrial design practice and heuristics to achieve proper design.

$$\Delta T_{min,limit} \leq \Delta T_{app,Evp,des} \leq \Delta T_{max,limit} \quad (3.60)$$

$$\Delta T_{min,limit} \leq \Delta T_{app,cond,des} \leq \Delta T_{max,limit} \quad (3.61)$$

Heat exchangers capacity limit constraints

Heat exchangers capacity limit constraints are applicable for both evaporator and condenser at operating conditions for each time periods. It is there to make sure that during change of other operating parameters such as pressure and mass flow rate, the design service area of both evaporator and condenser are not exceeded at any time period.

$$A_{Evp,des} \geq A_{Evp,i} \quad (i \in t) \quad (3.62)$$

$$A_{cond,des} \geq A_{cond,i} \quad (i \in t) \quad (3.63)$$

Where $A_{Evp,des}$ and $A_{Evp,i}$ include the service area of both preheating and evaporation segments as described in the evaporator modelling section.

3.4.8 Solution Approach

The mathematical design of this problem includes many process design variables including many sizing parameters for the ORC equipment. However, all of the process design variable will be dictated by some main decision variables, where other process variables can be calculated as function of those decision variables.

The equipment size such as evaporator, condenser, pump, expander, and generator will be determined based on the selected optimum ORC design Capacity (C_{des}) for all possible operation conditions along the life cycle. In other words, select optimum quantity of the produced hot oil from the forecast as a design basis for the ORC. The equipment will be sized based on the selected optimum design basis and will be fixed along the life cycle of the ORC project. Consequently, the designed ORC will be running at the design conditions in all of the discretized time periods where the produced hot oil is equal to or greater than the selected optimum design basis. On the other hand, the design ORC will be running at off-design conditions, partial loads, at any discretized time period where the produced hot oil is less than selected optimum design basis. The design and operating process variables will be solved simultaneously in order to find the optimum objective function while satisfying all the process constraints. Figure 13 gives an overall solution approach for the simultaneous design and off-design problem.

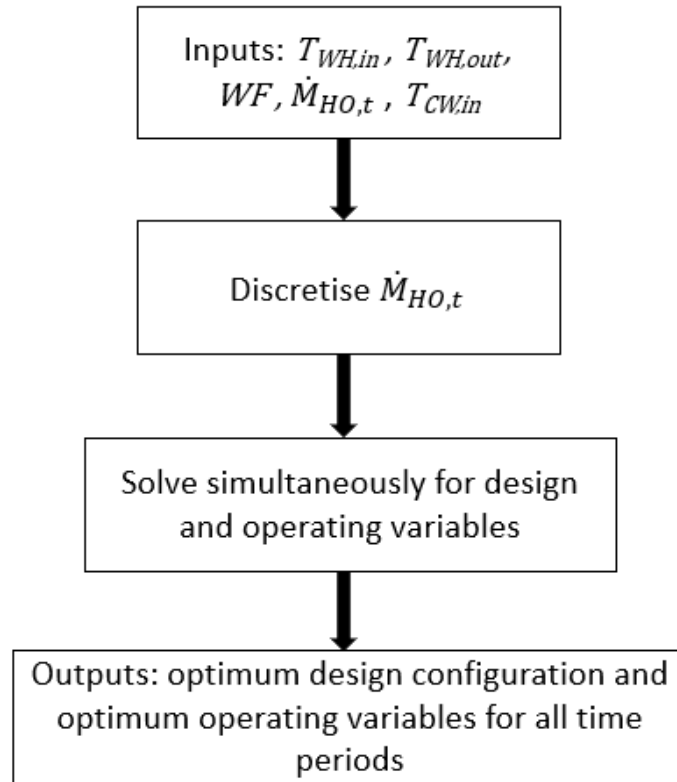


Figure 13 Solution approach of the simultaneous design and off-design problem

CHAPTER 4

CASE STUDY

In this chapter an industrial design waste heat recovery problem will be formulated and solved using the proposed formulation and optimization techniques. The case study will cover the problem details and any major assumptions to arrive to the final solution.

4.1 Problem Background

In practice, operating facilities in Oil and Gas sector often operate on forecasted plans and schedules that impose different plant loads over the plant life cycle. Consequently, the waste heat generated within these facilities would be fluctuating along with plant loads. This kind of waste heat availability changes have triggered a problem in designing a proper ORC size for waste heat recovery. Under-sizing of the ORC will minimize the waste heat recovery efficiency, while over-sizing will make the ORC run at partial loads when the plant is running at lower loads. Operating the ORC at partial loads would lower the efficiency of the turbo-expander and therefore lower the efficiency of ORC cycle and minimize the power generation as well as the ORC utilization. In order to maximize the economic benefit throughout the life cycle of the project, the optimum sizing of the ORC should be identified from the beginning using optimization techniques.

In this case study, an optimization model is developed to identify the optimum ORC size that utilize the waste heat from multiple combustion gas turbines (CGTs) used to run sales gas compressors subject to all process constraints and energy price. The compressed sales gas quantity is varied in yearly basis based on planned sales gas demand forecast. Part of the waste heat is collected through waste heat hot oil recovery heaters (HORH) from each

CGT exhaust, while the remaining is diverted to a bypass stack directed to the atmosphere. A portion of hot oil is used for internal process heating demand and the remaining is currently dissipated in a set of air coolers. The plane is to use the wasted heat in the air coolers in an ORC to generate power to improve the overall plant energy efficiency. Figure 14, illustrates the configuration of a single CGT/sales gas unit attached to a hot oil waste heat recovery heater while, Figure 15 present the block flow diagram of the CGT waste heat recovery and hot oil system.

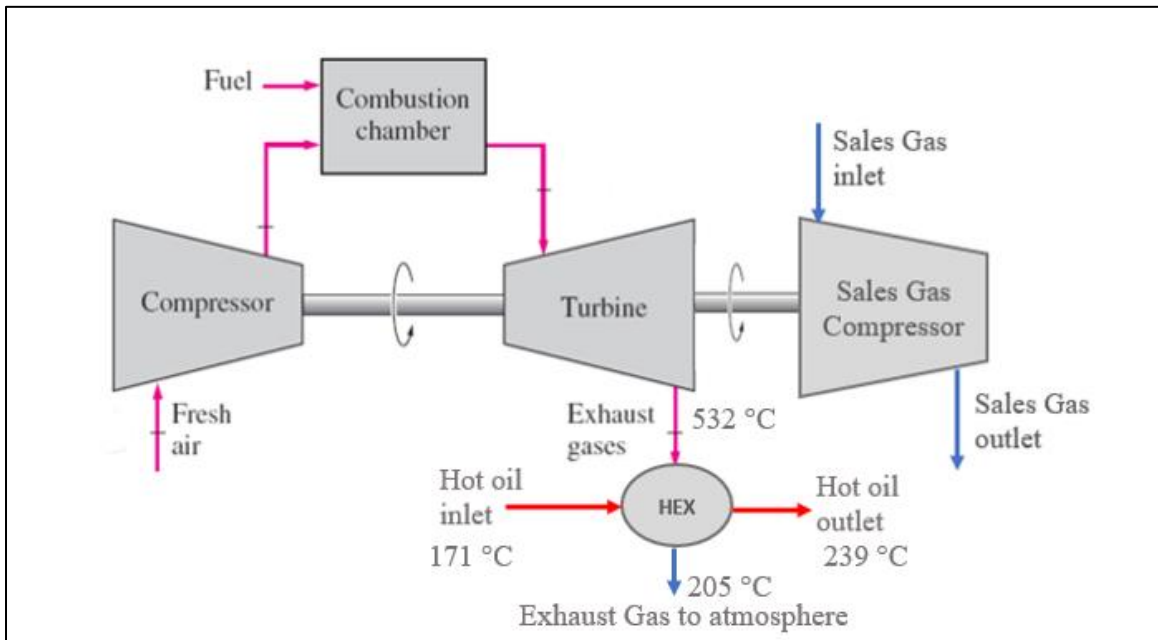


Figure 14 Single CGT/sales gas unit attached to a hot oil waste heat recovery heater

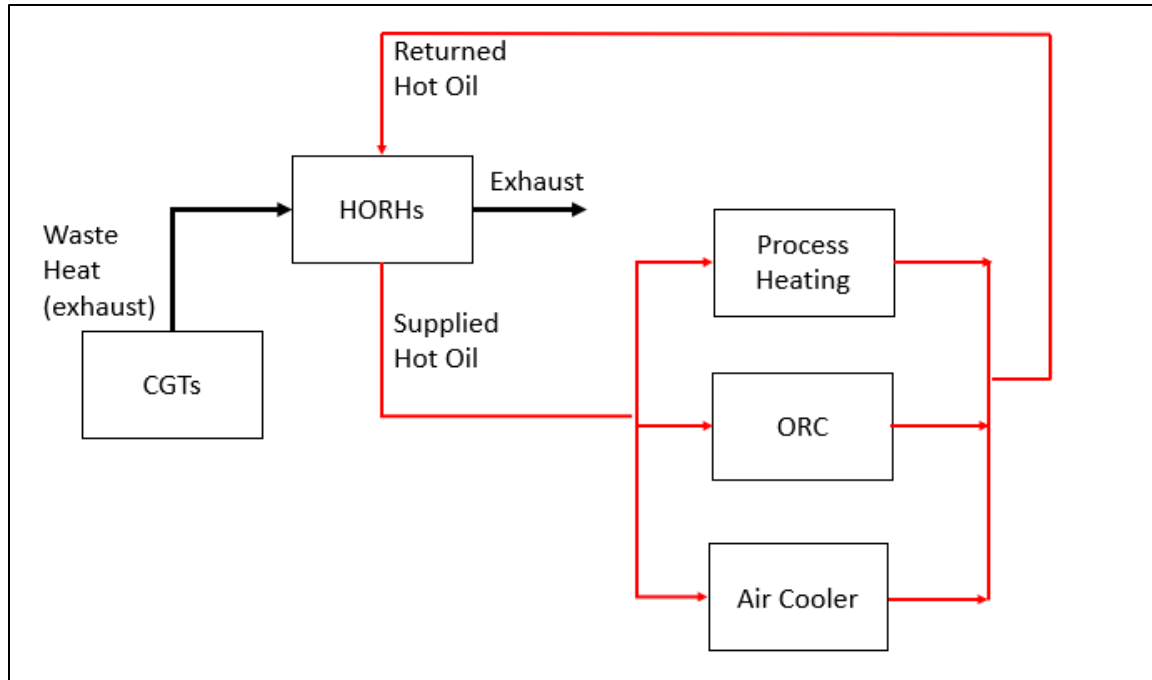


Figure 15 Block flow diagram of CGT waste heat recovery and hot oil system

4.2 Problem Description

A sales gas compression facility that uses combustion Gas Turbines (CGTs) to drive their sales gas compressors to meet pipeline pressure requirement. The plant runs based on known 20 years forecasted sales gas demand. The CGTs produce a medium quality waste heat from the exhaust gases. The medium quality waste heat is collected in Hot Oil Recovery Heater (HORH). A portion of hot oil is used for process heating demand within the plant while the remaining is planned to be dissipated in ORC to generate power to improve the overall plant energy efficiency. Currently the remaining portion of the hot oil, other than the one used for internal process heating, is cooled down in a set of air coolers to meet the hot oil return temperature.

Its desirable to come up with an economically optimum ORC size that can handle the wasted hot oil at the available forecasted operating demands. The forecasted demand can be translated into different hot oil production rate at multiperiod with different periods'

lengths. The hot oil supplied at fixed temperature, controlled by stack by-pass, to meet the process heating demand and must be returned at certain return temperature. The hot oil produced at temperature of 238.9°C and must be returned to the HORH at a temperature of 156.8°C. Table 4, presents the 20 years forecasted extra hot oil production rate and the correspondent wasted energy rates at fixed supply and returned temperatures.

Table 4 20 years forecasted excess hot oil rate and wasted energy rates.

Forecasted Year	Extra Hot Oil Production Rate (kg/s)	Wasted Energy Rate (MW)
Year 1	203.7	40
Year 2	203.7	40
Year 3	203.7	40
Year 4	203.7	40
Year 5	254.6	50
Year 6	254.6	50
Year 7	254.6	50
Year 8	280.1	55
Year 9	280.1	55
Year 10	280.1	55
Year 11	280.1	55
Year 12	305.6	60
Year 13	305.6	60

Year 14	331.0	65
Year 15	356.5	70
Year 16	356.5	70
Year 17	407.4	80
Year 18	407.4	80
Year 19	407.4	80
Year 20	407.4	80

The economic details used for this ORC design problem including electricity tariff, yearly discount rate, OPEX, and operating hours per year are given in Table 5.

Table 5 Economic details used for this ORC design problem

Description	Unit	Value
Electricity tariff	\$/MWh	0.13
Yearly discount rate	%	6
OPEX	%	5% of the CAPEX
Operating hours per year	Hours	8400

ORC process and equipment related information is given in Table 6.

Table 6 ORC process and equipment related information

Description	Unit	Value
Cooling water supply temperature	°C	30

Cooling water return temperature	°C	40
Hot oil production temperature	°C	238.9
Hot oil return temperature	°C	156.8
ORC turbine isentropic design efficiency	%	75
ORC pump design efficiency	%	75
Used working fluid	-	Cyclopentane

Given the above forecasted waste heat rates and economic factors, its desired to design a waste heat recovery ORC that would grantee the maximum Net Present Value (NPV) for a project life cycle of 20 years. This requires obtaining the optimum process design variables (*PDV*) first and fix it across all discretized time periods, then obtain the optimum process operation variables (*POV*) for all discretized time periods.

4.3 Problem Constraints

1. Design capacity constraint (C_{Des}):

The design capacity depends on the forecasted waste heat availability. Thus, the design capacity should not exceed the maximum value of the waste heat availability. Also, it should not be less than the minimum value of the waste heat availability. Thus,

$$40 \leq C_{Des} \leq 80 \quad (4.1)$$

2. Pressure constraints:

Evaporation pressure should not exceed the critical pressure of the working fluid, cyclopentane, and should be high enough to run the expander and generate useful work. In this specific problem the working fluid correspondent dew point pressure at the cooling water supply temperature is less than the atmospheric pressure. Thus, the minimum condensation pressure is limited to the atmospheric pressure to avoid vacuum at the condenser. In addition, the condensation pressure should be low enough to run the expander and generate useful work.

$$15 \leq P_{Ev,des} \leq 45 \quad (4.2)$$

$$15 \leq P_{Ev,op,i} \leq 45 \quad (i \in t) \quad (4.3)$$

$$1.01325 \leq P_{cond,des} \leq 4 \quad (4.4)$$

$$1.01325 \leq P_{cond,op,i} \leq 4 \quad (i \in t) \quad (4.5)$$

3. Approach temperature constraints:

Minimum approach temperature for both evaporator and condenser at design conditions must be confined within a reasonable acceptable industrial design limit to avoid any thermodynamic laws violation at operating condition and provide reasonable heat exchanger size at design stage.

$$3 \leq \Delta T_{app,E,des} \leq 25 \quad (4.6)$$

$$3 \leq \Delta T_{app,c,des} \leq 25 \quad (4.7)$$

4. Evaporator and Condenser Operating Capacity constraints:

The evaporator and condenser capacity constraints applicable at operating conditions for each time periods after obtaining the optimum design service areas for both evaporator and condenser. It is there to make sure that during change of other operating parameters such as pressure and mass flow rate, the design service area of both evaporator and condenser are not exceeded at any time period.

$$A_{Ev,des} \geq A_{Ev,i} \quad (i \in t) \quad (4.8)$$

$$A_{cond,des} \geq A_{cond,i} \quad (i \in t) \quad (4.9)$$

4.4 Major Assumptions

With the intention of solving such a wide industrial problem, many related assumptions need to be highlighted and discussed thoroughly in order to get the essence of the problem and arrive at acceptable reasonable solution. The following are the list of the major assumptions pertaining to the described multiperiod simultaneous design and off-design optimization problem:

- Hot oil is supplied at fixed temperature and must be returned at fixed temperature.

This optimization problem is quoted from real industrial problem where the hot oil supply to the ORC system is only part of the total produced hot oil and the rest of it is used in process heat. Therefore, temperature variations are not recommended to avoid any process upset. However, the load of the hot oil, flow rate, can change based on the plant load.

- Cooling water is supplied to the ORC at fixed temperature.
- Design capacity of the ORC including all the individual equipment capacity, cannot be exceeded at any time period. It means if the produced hot oil from the waste heat to the ORC is more than the selected design capacity, which is a decision variable, the extra hot oil should be handled by air cooling system to maintain the hot oil return temperature as shown in Figure 15.
- Turbo-expander efficiency is varying during off-design operation however, pump efficiency is considered to run at fixed efficiency. This is a valid assumption since the power consumption for the pump is much less than the power produced for the turbine and therefore pump efficiency variation would have minimal or almost no effect on optimization result.

4.5 Solution Approach

The mathematical design of this problem includes many process design variables including many sizing parameters for the ORC equipment. However, all of the process design variable will be dictated by some main decision variables, where other process variables can be calculated as function of those decision variables.

The equipment size such as evaporator, condenser, pump, expander, and generator will be determined based on the selected optimum ORC design Capacity (C_{des}) for all possible operation conditions along the life cycle. In other words, select optimum quantity of the produced hot oil from the forecast as a design basis for the ORC. The equipment will be sized based on the selected optimum design basis and will be fixed along the life cycle of the ORC project. Consequently, the designed ORC will be running at the design conditions in all of the discretized time periods where the produced hot oil is equal to or greater than

the selected optimum design basis. On the other hand, the design ORC will be running at off-design conditions, partial loads, at any discretized time period where the produced hot oil is less than selected optimum design basis. The design and operating process variables will be solved simultaneously in order to find the optimum objective function while satisfying all the process constraints Figure 16 gives an overall solution approach for the simultaneous design and off-design problem.

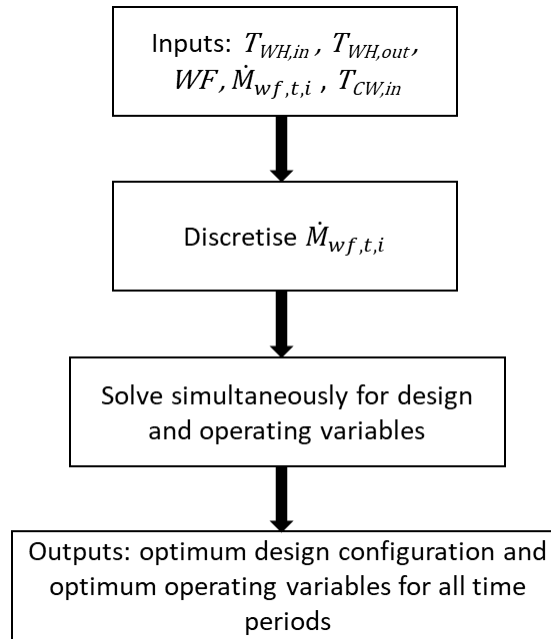


Figure 16 Solution approach of the simultaneous design and off-design problem

4.6 Results and Discussion

The problem was solved simultaneously for design and operating variables using the described solution technique for the given objective function.

Table 7 Optimization results for objective function

Description	Value	Unit
Present Value of Electricity Sales	107.12	MM\$

Present Value of OPEX	5.75	MM\$
Total CAPEX	10.02	MM\$
NPV	91.35	MM\$

Table 7 shows the optimization results for the objective function. Table 8 shows the optimization results for the obtained design variables while, Table 9 shows the optimization results for the operating variables for all the discretized time periods. Although, the ORC is running at partial load at the time period T_3 , the ORC has slightly higher overall efficiency than the design point. This is because that the turbine efficiency at this point, with a load of approximately 92 %, has an efficiency of only 0.8% less than the design point of 75%. In summary, the rate of reduction in supplied waste heat is higher than the rate of reduction in turbine efficiency in addition to having extra capacity in the evaporator to increase the evaporation pressure have contributed to the slight improvement in the overall ORC efficiency.

Table 8 Optimization results for the design variables

Design Variables	Value	Unit
Design Capacity (C_{Des})	60	MW
Design Evaporation Pressure ($P_{Evap,des}$)	31.28	bar
Design Condensation pressure ($P_{cond,des}$)	1.013	bar
Working fluid circulation rate ($\dot{m}_{wf,des}$)	100.3	Kg/s
Evaporator design approach temperature ($\Delta T_{app,Evp,des}$)	4.58	°C

Condenser design approach temperature ($\Delta T_{app,cond,des}$)	6.01	°C
ORC Power Net Generation	10.30	MW
ORC Overall Efficiency	17.17	%

Table 9 Optimization results for the operating variables

		Time Period						
Operating Variables		T ₁	T ₂	T ₃	T ₄	T ₅	T ₆	T ₇
Available Waste Heat	MW	40	50	55	60	65	70	80
Evaporation Pressure	bar	32.54	31.99	31.75	31.53	31.53	31.53	31.53
Evaporator Approach Temperature	°C	2.72	3.51	3.89	4.25	4.25	4.25	4.25
Condensation Pressure	bar	1.013	1.013	1.013	1.013	1.013	1.013	1.013
Condenser Approach Temperature	°C	8.94	7.59	6.83	6.01	6.01	6.01	6.01
Working fluid circulation rate	Kg/s	67.8	84.2	92.3	100.3	100.3	100.3	100.3
Turbine Isentropic Efficiency	%	66.9	73.0	74.4	75.0	75.0	75.0	75.0
ORC Power Net Generation	MW	6.24	8.47	9.45	10.3	10.3	10.3	10.3
ORC Overall Efficiency	%	15.59	16.94	17.19	17.17	17.17	17.17	17.17

The optimum operating evaporation pressure for all the optimization periods are presented in Figure 17. The ORC will be running at higher evaporation pressure at all the times where the forecasted waste heat is less than the selected optimum design capacity. This behavior is expected to increase the power generation as much as possible during the ORC partial load operation while not violating the evaporator design capacity limit.

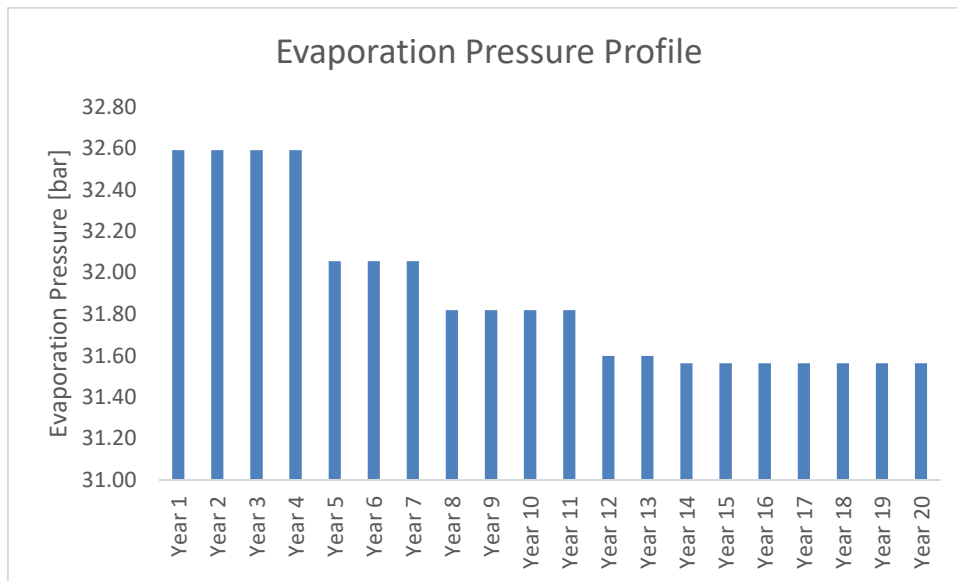


Figure 17 Optimum operating evaporation pressure for all the optimization periods

A decrease in the operating evaporator approach temperature with decrease in available waste heat was observed, as shown in Figure 18. This kind of trend is mainly to increase the evaporator design capacity utilization at all time periods to eventually increase the power generation. Figure 19 shows the evaporator design capacity utilization in all time periods. It is obvious that the optimizer would bush the evaporator to its design limit

to utilize all of the available service area. This was achieved by increasing the operating pressure and therefore decreasing the evaporator approach temperature.

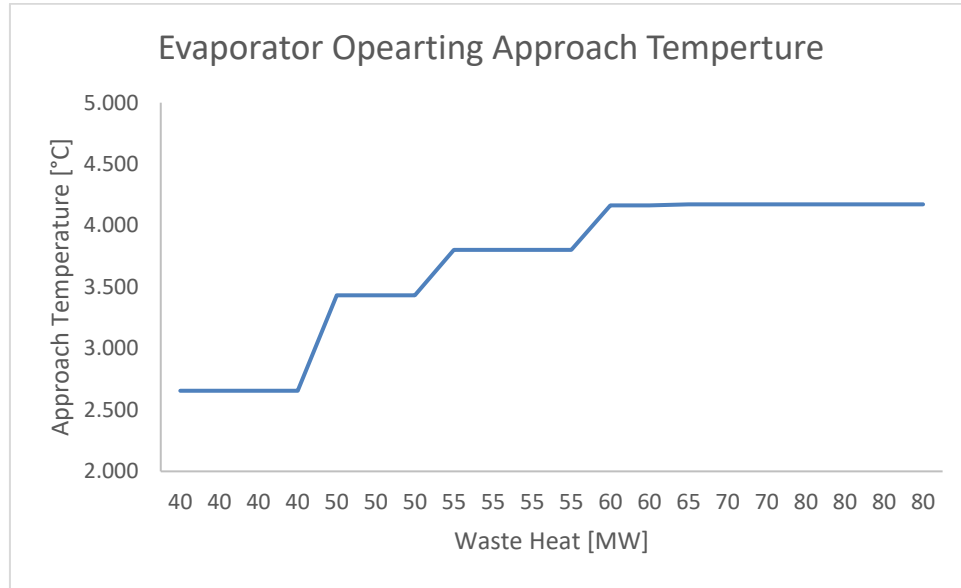


Figure 18 Evaporator operating approach temperature across operating time periods

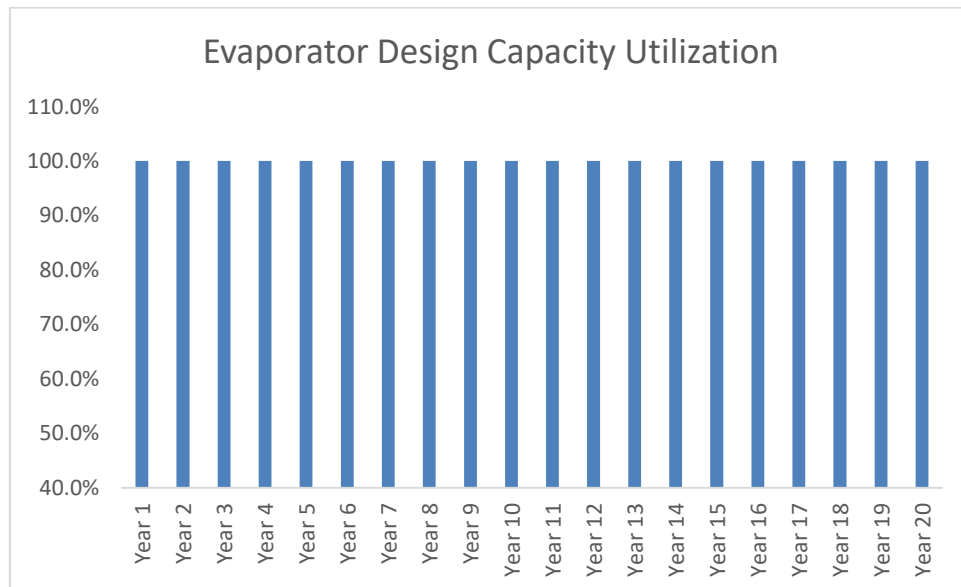


Figure 19 Evaporator design capacity utilization across all operating time periods

On the other hand, the design capacity of the condenser was not fully utilized across all the time periods as illustrated in Figure 20. Full utilization was observed during the time periods where the available waste heat is at or greater than design capacity of the ORC. This behavior can be also observed from the increasing trend of the condenser approach temperature with reduction in ORC load, in other word, reduction in waste heat availability as demonstrated in Figure 21.

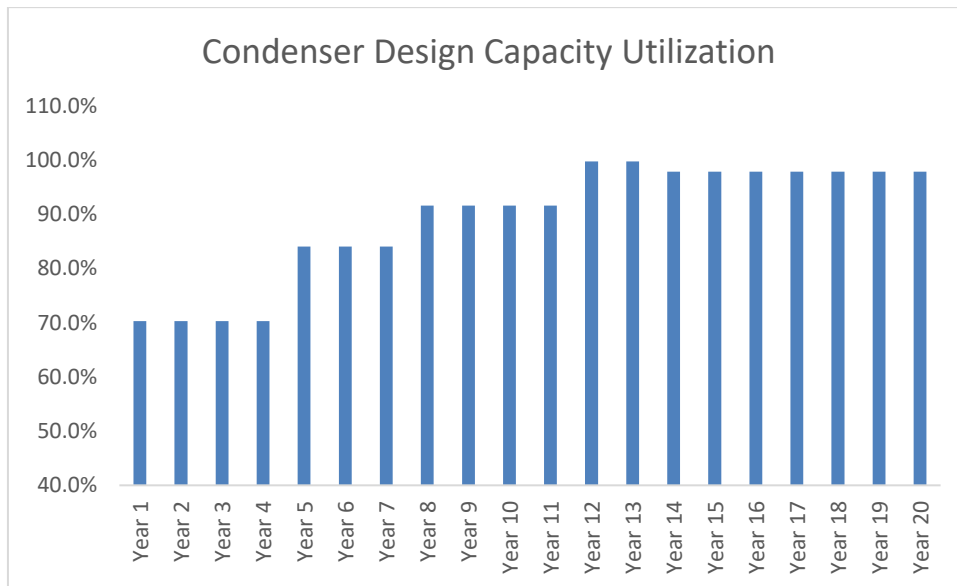


Figure 20 Condenser design capacity utilization across all operating time periods

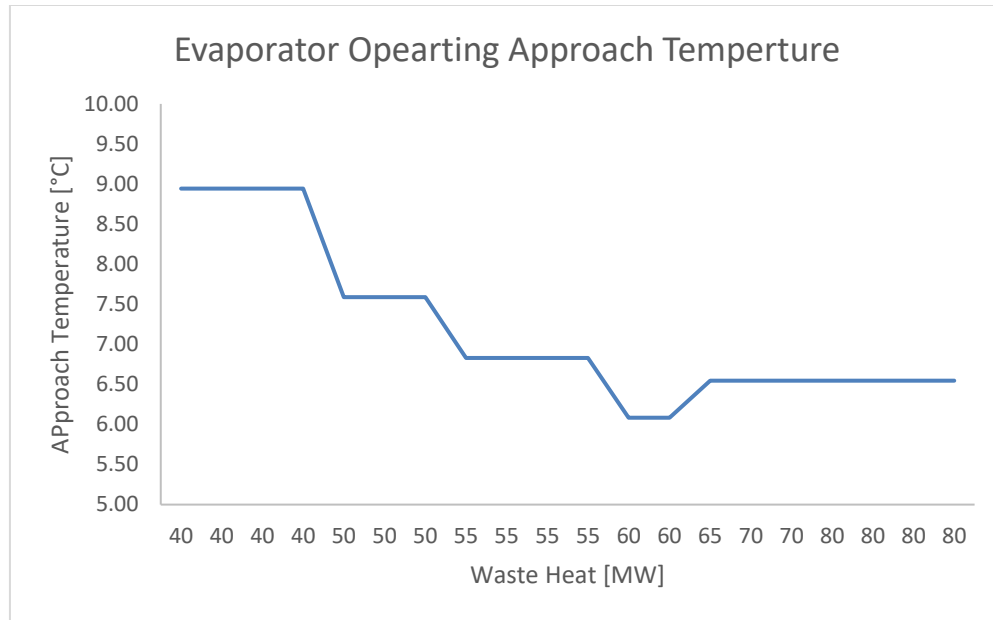


Figure 21 Condenser operating approach temperature across operating time periods

The condenser would have been utilized more if the low condensation pressure limit was allowed to go below atmospheric pressure. However, would reach to another limit which is the cooling water temperature and mass flow rate. In summary, the evaporator operating parameters plays a bigger role in the ORC simultaneous design and off-design optimization than the condenser.

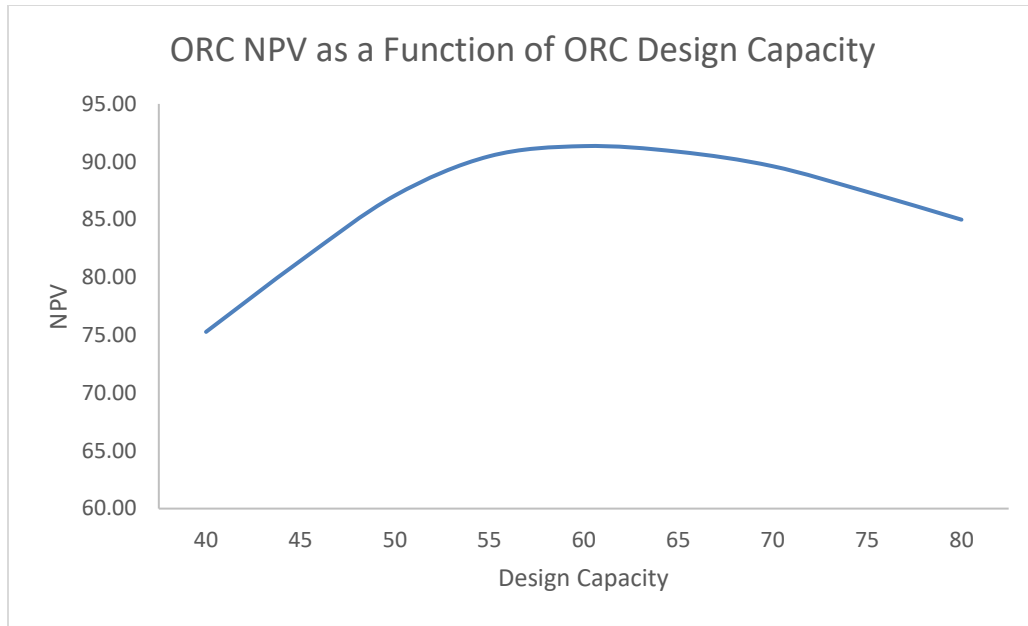


Figure 22 Effect of the ORC design capacity on the NPV objective function

ORC design capacity factor was one of the major decision variables for the given objective function. It dictates the size of the ORC, more specifically the generator size which in turn determine the maximum potential power generation and capital investment. Figure 22 shows the effect of the ORC design capacity on the NPV objective function. Varying the design capacity decision variable from the minimum to the maximum acceptable design capacity for this given problem, while maintaining optimum design and off-design evaporation pressure, would change the objective function by a range of 32%. This shows the importance of the selection of the right ORC design capacity from the beginning.

Determining the size of the ORC for such a complex problem is not straight forward and can't be solved by intuition. The outlined optimization technique need to be conducted to determine the optimum ORC size that would make best investment option.

References

- [1] Alberto Benato, A. S. (2017). *Design and Off-Design Analysis of an ORC Coupled with a Micro-Gas Turbine*. Italy: Energy Procedia.
- [2] Algieri, A., & Morrone, P. (2015). Energy analysis of organic rankine cycles for biomass applications. *Thermal Science*, 193-205.
- [3] Alireza Javanshir, N. S. (2017). *Thermodynamic Analysis of ORC and Its Application for Waste Heat Recovery*. USA: Sustainability.
- [4] Amicabile, S., Lee, J., & Kum, D. (2015). A comprehensive design methodology of organic Rankine cycles for the waste heat recovery of automotive heavy-duty diesel engines. *Applied Thermal Engineering*, 574-585.
- [5] Burrato, A. (2013). ORegen to waste heat recovery: development and application. *2nd International Seminar on ORC Power System* (pp. 7-8). Rotterdam (NL): SAME ORC2013.
- [6] Chena, G., Shib, B., & Yanb, L. (2011). *Multi-period Optimal Operational Planning of Boiler Steam System Based on Line-up Competition Algorithm*. 2nd International Conference on Advances in Energy Engineering.
- [7] Chintala , V., Kumar, S., & Pandey, J. (2017). A technical review on waste heat recovery from compression ignition engines using organic Rankine cycle. *Renewable and Sustainable Energy Reviews*, 81, 493-509.
- [8] Colonna, P., Casati, E., Trapp, C., Mathijissen, T., Larjola, J., Turunen-Saaresti, T., & Unsitalo, A. (2015). *Journal of Engineering for Gas Turbines and Power*, 137.
- [9] Energy, U. D. (2008). *Waste Heat Recovery: Technology and Opportunities in U.S. Industry*. BCS, Incorporated.
- [10] Ennio Macchi, M. A. (2017). *Organic Rankine Cycle (ORC) Power Systems Technologies and Applications*. Woodhead Publishing.
- [11] Gaia M. (2011). 30 years of ORC development In: Keynote Lecture at the ORC 2011. *1st International Seminar on ORC Power System* . Delft (NL).
- [12] Guohui Chen, B. S. (2011). *Multi-period Optimal Operational Planning of Boiler Steam System Based on Line-up Competition Algorithm* . China: Energy Procedia.

- [13] Holm, J. (1981). The application of turboexpanders for energy consevation. *Transaction of the IMechE*.
- [14] Kern DQ. (1950). *Process Heat Transfer*. New York: McGraw-Hill.
- [15] Kutateladze S.S. (1963). *Fundamentals of Heat Transfer*. London: Edward Arnold.
- [16] M. Astolfi, L. N. (2019). *Techno-economical optimization of geothermal ORC with novel "Emeritus" heat rejection unit in hot climate* . Itly: Renewable Energy.
- [17] M. Bianchi, L. A. (2017). *Techno-Economic Analysis of ORC in Gas Compression Stations Taking Into Account Actual Operating Conditions*. Italy: Energy Procedia .
- [18] Maria Anna Chatzopoulou, S. L. (2018). *Off-design operation of ORC engines with different heat exchanger architectures in waste heat recovery applications*. Energy Procedia.
- [19] Palen JW. (1988). *Shell-and-tube reboilers. In: Heat Exchanger Design Handbook, Vol. 3*. New York: Hemisphere Publishing Corp.
- [20] Robert W. Serth, T. G. (2007). *Process Heat Transfer - Principles and Applications*. Kingsville, Texas, USA: Elsevier.
- [21] Roberto Pili, N. S. (2019). EFFICIENCY CORRELATIONS FOR OFF-DESIGN PERFORMANCE PREDICTION OF ORC AXIAL-FLOW TURBINES. *5th International Seminar on ORC Power System* (pp. Paper ID: 70, Page 1). Munchen, Germany: 5th International Seminar on ORC Power System.
- [22] Sonia L. Gomez-Alaez, V. B. (2017). *Off-design study of a waste heat recovery ORC module in gas pipelines recompression station* . Italy: Energy Pocedia.
- [23] Steven Lecompte, S. L. (2015). *Multi-Objective Thermo-Economic Optimization Strategy for ORCs Applied to Subcritical and Transcritical Cycles for Waste Heat Recovery*. Belgium: Enetgies.
- [24] Toffolo A., L. A. (2014). A multi-criteria approach for the optimal slection of working fluid and design parameters in Organic Rankine Cycle system. *Appl. Energy* , 121,219-232.
- [25] Turton R., B. R. (2013). *Analysis. Synthesis, and Design of Chemical Processes, 4th ed* . Upper Saddle River, NJ, USA: Pearson Education International.

- [26] Xi, X., Zhou, Y., Guo, C., Yang, L., & Du, X. (2015). Characteristics of Organic Rankine Cycles with Zeotropic Mixture for Heat Recovery of Exhaust Gas of Boiler. *Energy Procedia*, 1093-1101.
- [27] Xianglong Luo, Y. W. (2020). *Simultaneous design and off-design operation optimization of a waste*. Elsevier.

Vitae

Name	Hani Mohammed Al Saed
Nationality	Saudi Arabian
Date of Birth	4/4/1982
Email	hanialsaed@gamil.com
Address	Saudi Arabia-Dhahran 3036-34515
Academic Background	B.Sc. Chemical Engineering, University of Ottawa, Ottawa, Canada. MS Chemical Engineering, KFUPM, Dhahran, Saudi Arabia
Research Interest	Process and energy systems optimization