COOLING THE STEAM POWER PLANT CONDENSER USING A VAPOR COMPRESSION REFRIGERATION SYSTEM

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ABSTRACT

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Steam power plants represent the largest segment of the world’s electricity production. With developing and foreseeable shortages of adequate water sources in the arid regions and increasing regulatory restrictions, alternate technologies are being sought for heat rejection. The U.S. Environmental Protection Agency has recently proposed that power plants that consume more than 7.6 x 10^6 L/day of water for cooling (equivalent plant capacity >250 MW) must consider alternate technologies to determine the best available technology for rejecting the waste heat. A steam condenser is an essential part of a steam power plant. Steam condensation occurs in a steam condenser using either wet cooling, dry cooling, or a combination of both. The use of wet cooling therefore results in a detrimental impact on the environment.

In the current proposed work an alternate method other than water or air, for cooling the steam power plant’s condenser will be investigated theoretically, numerically, and experimentally. The proposed method is a condenser-configuration using refrigerant in a closed-loop-cycle. The refrigeration will be a vapor compression cycle. The vapor compression refrigeration cycle system is a highly well-established technology forming the basis of many important industrial and agricultural and household applications.

The main goal of this proposed work is to test the feasibility and verify the proposed idea of using vapor compression refrigeration cycle as a condenser coolant, thereby replacing the environmentally polluting conventional water cooling and the low efficiency and costly air cooling methods. The current proposed project will also be able to compare the use of different refrigerants
on the basis of performance, cost, and environmental impact with the conventional water and air cooling systems.
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KEY TO SYMBOLS AND ABBREVIATIONS

Nomenclature

\( A_{\text{sur}} \) Total heat transfer area at a given control volume

\( A_{\text{mv}} \) Mean void area in the vapor space

\( A_{\text{vd}} \) Cross-sectional area of the vapor duct

\( C_{\text{pcl}} \) Specific heat of the coolant

\( C_{\text{vg}} \) Vapor shear correction term

\( C_r \) Parameter defined by equation (3.40)

\( \text{COP} \) Coefficient of performance

\( D \) Diffusion coefficient

\( d_{\text{out}} \) Outer tube diameter

\( d_{\text{in}} \) Inner tube diameter

\( e_{\text{in}} \) Exergy per unit mass entering the control volume

\( e_{\text{out}} \) Exergy per unit mass exiting the control volume

\( \dot{E}_{\text{in}} \) Flow of the exergy into the control volume

\( \dot{E}_{\text{out}} \) Flow of the exergy out of the control volume

\( \dot{E}_d \) Destruction of exergy

\( F \) Inundation correction factor

\( g \) Gravity

\( h_{\text{tn}} \) Convective heat transfer coefficient of the coolant
$h_{out}$  Convective heat transfer coefficient of the condensate

$h_a$  Convective heat transfer coefficient of the non-condensable

$h_{s, in}$  Enthalpy of the steam entering the condenser

$h_{s, out}$  Enthalpy of the condensate exiting the condenser

$h_{fg}$  Latent heat of vaporization

$h_{fg}^*$  Modified latent heat of vaporization

$h_0$  Specific enthalpy evaluated at dead state conditions

$n$  Row number of the tubes from the top of the tube bundle

$K_{cl}$  Thermal conductivity of the coolant

$K_{metal}$  Thermal conductivity of the tube material

$K_{cs}$  Thermal conductivity of the condensate

$L$  Length of the condenser tube

LMTD  Log Mean Temperature Difference

$m$  Mass

$\dot{m}_a$  Non-condensable gases mass flow rate

$\dot{m}_{cl}$  Coolant mass flow rate

$\dot{m}_{cs}$  Condensation flow rate

$\dot{m}_s$  Steam mass flow rate

$\dot{m}_v$  Mass velocity

$N$  Number of tubes inside the condenser

$Nu_{cl}$  Nusselt number of the coolant
\( Nu_{cs} \)  Nusselt number of the condensate

\( NTR \)  Number of tubes in each row

\( P_c \)  Condenser pressure

\( \Delta P_c \)  Steam exhaust resistance or pressure drop inside the condenser

\( P_s \)  Steam pressure

\( P_a \)  Air (non-condensable gases) pressure

\( P_{vac} \)  Vacuum pressure

\( P_{atm} \)  Atmospheric pressure

\( Pr_{cl} \)  Prandtl number of the coolant

\( P_t \)  Tube pitch

\( P_T \)  Power of turbine

\( \dot{Q} \)  Heat transfer rate within the condenser

\( \dot{Q}_s \)  Heat transfer rate from the steam to the coolant

\( \dot{Q}_{cl} \)  Coolant heat transfer rate

\( \dot{Q}_j \)  The rate of heat transfer across the control volume

\( Re_{cl} \)  Reynolds number of the coolant

\( Re_{mix} \)  Reynolds number of the mixture (steam/air)

\( Re_s \)  Reynolds number of the steam

\( R_{film,in} \)  Coolant film thermal resistance

\( R_{film,out} \)  Condensate film thermal resistance

\( R_{wall} \)  Wall thermal resistance
\( R_a \) Non-condensable gases thermal resistance

\( R_{tot} \) Overall thermal resistance

\( s \) Specific entropy

\( s_0 \) Specific entropy evaluated at dead state conditions

\( t \) Temperature

\( T \) Absolute temperature

\( T_h \) Hot stream temperature

\( T_c \) Cold stream temperature

\( T_{sat} \) Saturated temperature

\( T_{wall} \) Wall temperature

\( T_{film} \) Film (condensate) temperature

\( T_{cl,in} \) Inlet coolant temperature

\( T_{cl,out} \) Outlet coolant temperature

\( T_b \) Bulk Temperature

\( T_0 \) Dead state temperature

\( u \) Flow velocity

\( T_j \) Temperature on the boundary

\( U \) Overall heat transfer coefficient

\( V \) Coolant flow velocity

\( V_\infty \) Free stream velocity

\( w_{ts} \) Width of the tube sheet
\( \dot{W}_{c,v} \quad \text{Work done by steam} \)

**Greek Symbols**

\( \mu_{cl} \quad \text{Dynamic viscosity of the coolant} \)

\( \mu_{cs} \quad \text{Dynamic viscosity of the condensate} \)

\( \mu_{mix} \quad \text{Dynamic viscosity of the mixture vapor/air} \)

\( \rho_{cl} \quad \text{Density of the coolant} \)

\( \rho_{cs} \quad \text{Density of the condensate} \)

\( \rho_{mix} \quad \text{Density of the mixture} \)

\( \rho_{s} \quad \text{Density of the steam} \)

\( \dot{\sigma}_{c,v} \quad \text{The rate of entropy production} \)

\( \varepsilon \quad \text{Relative content of the gas/energetic efficiency} \)

\( \eta_{ex} \quad \text{Exegetic efficiency} \)

\( \eta_{m} \quad \text{Mechanical efficiency} \)

\( \eta_{th} \quad \text{Thermal efficiency} \)

**Subscripts**

A  Per unit area

a  Ambient

atm  Atmosphere

ac  Air cooled

b  Boiler

c  Condenser
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<tr>
<td>ACCS</td>
<td>Air cooled condenser system</td>
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<tr>
<td>LLSL-HX</td>
<td>Liquid- line suction heat exchanger</td>
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<td>SPP</td>
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<td>SPPC</td>
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<td>VCRC</td>
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1 Introduction to the Steam Power Plant

1.1 The Rankine Cycle and the Steam Power Plant

Steam power plants represent the largest segment of the world’s electricity production. With developing and foreseeable shortages of adequate water sources in the arid regions and increasing regulatory restrictions, alternate technologies are being sought for heat rejection. The U.S. Environmental Protection Agency has recently proposed that power plants that consume more than 7.6 x 10^6 L/day of water for cooling (equivalent plant capacity >250 MW) must consider alternate technologies to determine the best available technology for rejecting the waste heat. The steam condenser is an essential part of a steam power plant. Steam condensation occurs in a steam condenser using either wet cooling, dry cooling, or a combination of both. Use of wet cooling results in a detrimental impact on the environment, so that its implementation has been recently limited in the USA. Yet, industry is reluctant to adopt dry cooling technology as the process of choice due to higher initial costs and a slight loss of efficiency, especially at higher ambient air temperatures.

The steam power plant works based on a Rankine cycle, which is the thermal cycle that converts heat into work. Figure 1 shows a schematic view of a steam power plant configuration, and Figure 1.2 shows the T-S diagram for the corresponding Rankine cycle. Steam condensation occurs in the steam condenser by using either wet cooling, dry cooling or a combination of the two.

Use of wet cooling results in a detrimental impact on the environment, so that its implementation has been recently confined in the USA. In addition to increasing pollution, which comes mainly from steam power plants concentrated around rivers, the cooling is very costly and
gives rise to low thermal efficiency of the steam cycle. As a result, the need for finding new alternative methods for cooling the steam condenser has emerged.


As shown in Figure 1.2, the Rankine cycle starts by heating water, which is the working fluid in the cycle, inside the heat supply “boiler”. Then the steam with high pressure and temperature will expand inside the turbine, which is the device that converts the energy of the
steam into work by rotating the turbine-generator shaft. Eventually, the rotation of the shaft will generate electric power in the generator. After the massive steam expansion, which happens inside the turbine, the steam will lose most of its energy, then it will exhaust into the condenser where it is condensed to water. The water, which is produced by the condenser, will be recirculated by using the high pressure pump into the boiler to gain heat again, and then it will be passed to the turbine. Figure 1.2 shows the schematic of a steam power plant and the T-S diagram of the Rankine cycle with one-stage turbine and one feed water heater. The ideal Rankine cycle does not involve any internal irreversibility and consists of the following four processes:

1-2 Isentropic compression in a pump
2-3 Constant pressure heat addition in a boiler
3-4 Isentropic expansion in a turbine
4-1 Constant pressure heat rejection in a condenser

Figure 1.2 Simple Rankine cycle of a steam power plant [2]
In the ideal steam power plant cycles, it is assumed that the connection between other components allow the working fluid to move between components by neglecting the intervening changes in state. Since the pump, boiler, turbine, and the condenser are considered steady-flow devices, we can analyze the processes that make up the Rankine cycle as the steady-flow process. In the turbine, the steam expands reversibly and adiabatically. By applying the First Law of Thermodynamic at steady-state condition we get

\[ w_{t,\text{out}} = h_{\text{inlet(3)}} - h_{\text{exit(4)}} \]  

(1.1)

Also, the power produced by the turbine to an external load (electrical generator) can be calculated by using the following formula:

\[ P_T = m_s w_{\text{turb, out}} \]  

(1.2)

By applying the steady flow First Law of Thermodynamic, the steam generator load can be obtained

\[ q_{\text{boiler,in}} = h_{\text{exit(3)}} - h_{\text{inlet(2)}} \]  

(1.3)

In the condenser where the heat rejection accurses, the phase usually changes according to the amount of deception heat. By applying the steady-state First Law of Thermodynamics on the fluids entering and leaving the condensers

\[ q_{\text{con, out}} = h_{\text{inlet(4)}} - h_{\text{exit(1)}} \]  

(1.4)

As the condensation takes place in the condenser, the temperature of steam gets lower and the coolant temperature rises

\[ m_s q_{\text{con}} = m_c C_{\text{water}} (T_{\text{out}} - T_{\text{in}}) \]  

(1.5)
Depending upon the steady-state First Law of Thermodynamic, the pump work and power can be calculated using the following equations:

\[ w_{\text{pump,in}} = h_{\text{exit}(2)} - h_{\text{inlet}(1)} \]  
\[ P_p = m_s w_p \]  

The net power produced by Rankine cycle is equal to the difference between the turbine power and pump power. The pump power is very small in compression to the turbine, and this is one of the others Rankine cycle advantages. The thermal efficiency of the Rankine cycle can be calculated by using the following equation:

\[ \eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} \]

1.1.1 Efficiency of the Steam Power Plant

Any small increase in thermal efficiency of a steam power plant can mean large savings from the fuel requirements. Therefore, many efforts are made to improve the efficiency of the cycle. The basic idea behind all the modifications to increase the thermal efficiency of a power cycle is the same:

- Increase the average temperature at which heat is transferred to the working fluid in the boiler, or
- Decrease the average temperature at which heat is rejected from the working fluid in the condenser.

That is, the average fluid temperature should be as high as possible during heat addition and as low as possible during heat rejection. There are three commonly practiced ways of improving the Rankine cycle thermal efficiency.
The first method is lowering the condenser pressure. Steam exists as a saturated mixture in the condenser at the saturation temperature corresponding to the pressure inside the condenser. Therefore, lowering the operating pressure of the condenser automatically lowers the temperature of the steam and thus, the temperature at which heat is rejected. The effect of lowering the condenser pressure on the Rankine cycle efficiency is shown in Figure 1.3.

![Figure 1.3 The effect of lowering the condenser pressure on the ideal Rankine cycle [2]](image)

The second method is increasing boiler temperature. The average temperature at which heat is added to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures.

The third method is to increases the boiler pressure. Another way of increasing the average temperature during the heat-addition process is to increase the operating pressure of the boiler, which automatically raises the temperature at which boiling takes place. This, in turn, raises the average temperature at which heat is transferred to the steam and thus raises the thermal efficiency of the cycle.
1.2 Refrigeration and Heat Pump

Refrigeration and heat pump are two majors application area of thermodynamics. The refrigerator is a cyclic device and use refrigerants as a working fluid. The main purpose of the refrigerator and the heat pump is transferring heat from low-temperature region to high-temperature one. However, we employ the refrigerant to maintain the refrigerated space at low temperature, and the heat pump is employed to maintain the heated space at high temperature.

The refrigerator and the heat pump is shown schematically in the following chapter.

Air conditioning field, manufacture of ice, and reservation of foods are common application of refrigeration. Each one of the previous applications need specific design to maintain the require temperature. Each application requires a different temperature for the refrigerated space, the determination of which is the first decision the engineers should make in the design of a vapor compression refrigeration system.

Energy Source:

As we understand from the second law of thermodynamic that in order for the refrigeration system to accomplished, the expense of work transfer from heat transfer from high-temperature reservoir or work reservoir are needed.

Energy Sink:

The second law of thermodynamics also states that for the refrigeration system to operate continuously, it needs to reject heat to an external reservoir.

Working Fluid and Cycle Selection:

Depending on the power cycle we can choose the working fluid. We usually need to look for the best combination between the working fluid and cycle that will result in consume the minimum power to produce the refrigeration.
Component Selection:

Compressor is one of the most common component in the refrigeration cycle. The reciprocating compressor and the centrifugal compressor are two general types of the compressor. The centrifugal compressor are best adapted to high specific volumes and low pressure.

1.3 The Reversed Carnot Cycle

The Carnot refrigeration cycle is consider the most efficient theoretical cycle for refrigeration system. The Carnot refrigeration cycle contains two reversible isothermal process and two isentropic process as the following figure shows

![Schematic and T-s diagram of reversed Carnot cycle](image)

Figure 1.4 Schematic and T-s diagram of reversed Carnot cycle [2]

Process 1-2: heat $Q_L$ is absorbed isothermally by refrigerant at low temperature $T_L$

Process 2-3: the working fluid is compressed isentropically to state 3.

Process 3-4: heat $Q_H$ is rejected isothermally by the refrigerant at high temperature $T_H$

Process 4-1: The refrigerant expands isentropically to state 1.
1.3.1 The Energy Analysis of Carnot Cycle

The all four processes of Carnot cycle are reversible process and the analysis for these processes are explained by the following equation.

The absorbed heat is:

\[ Q_L = T_L (s_2 - s_1) \]  \hspace{1cm} (1.9)

The rejected heat is:

\[ Q_H = T_H (s_3 - s_4) \]  \hspace{1cm} (1.10)

We also can calculate the coefficient COP of performance by using the following equations:

\[ COP = \frac{Q_L}{W_{net}} \]  \hspace{1cm} (1.11)

\[ COP = \frac{T_L}{T_H - T_L} \]  \hspace{1cm} (1.12)

1.4 The Ideal Vapor Compression Refrigeration Cycle

Even though the reversed Carnot cycle is the most efficient cycle for the refrigerant cycle, it is not the appropriate model for refrigerant one. The heat rejected isothermally in the condenser, and the isothermal heat absorbed by the evaporator could be done easily by fixing the temperature of the mixture since maintaining the same pressure on these two components.

However, we need a special design for the turbine and compressor components to handle the mixture of liquid- vapor phases. One way to eliminate this problem is replacing the turbine component with a throttling device and vaporizing the refrigerant completely entering the compressor component. The cycle with the new component is the ideal vapor- compression cycle.
refrigeration cycle. This cycle is usually used for air conditioning system, refrigerator and heat pump.

1.5 The Actual Vapor Compression Refrigeration Cycle

Fluid friction and heat transfer the most common causes of irreversibility are the difference between the ideal and actual vapor compression refrigeration cycle. In actual cycle, the very long connection between the evaporator and compressor will lead to decrease the pressure by heat transfer to the refrigerant from the surrounding and fluid friction. As a result, pressure drops and specific volume increase. Because of increasing the specific volume the requirement power input will increase.

Another difference between the ideal cycle and the actual cycle is the compression process. In the ideal case the refrigerant is compressed isentropically (reversible and adiabatic), however in the actual cycle the entropy increases by heat transfer and frictional effects.

In the ideal cycle, the refrigerant leaves the condenser in saturated liquid phase, but taking the fluid friction affects and heat transfer in account can change the leaving refrigerant phase. Also, in reality the refrigerant got subcooled before entering the expansion valve. Therefore, the enthalpy of refrigerant in refrigerant decrease and the absorption of heat increases.

The ordinary vapor- compression refrigeration system has several advantage such as maintenance- free, simple, reliable and inexpensive. For these reasons the simple vapor refrigeration system is widely used, and it is sufficient for most of refrigeration applications. While the simple vapor compression refrigeration system is not the appropriate one for large industrial and other applications. Therefore, some innovations were done to meet industrials requirements.
1.6 Cascade Refrigeration System

The temperatures that some industrial applications needed are too large comparing to those usually single vapor compression refrigeration involves. Therefore, low temperatures one of other requirements that the ordinary vapor refrigeration system cannot introduce it into large industrial applications. Cascade refrigeration cycle which consist of two or more refrigeration cycle is the one way can used to deal with this situation.

The figure shows the two stage cascade refrigeration system. In this system, the heat exchanger, which works as evaporator for the first cycle and as condenser for another one, connects the two cycle in the middle. The ratio of mass flow rate and coefficient of performance for cascade can be calculated using the following equations:

1.7 Multistage Compression Refrigeration System

The research managed some improvement on the cascade refrigeration system by replacing the heat exchanger with a flash chamber, which has better heat transfer characteristic, and called it the multistage compression refrigeration system.

In the multistage compression refrigeration system, the refrigerant liquid leaving the condenser passing to the flash chamber through the first expansion valve. Since the flash chamber has very good heat transfer characteristics, part of the liquid vaporize during that process. The saturated vapor mixes with the superheated refrigerant that comes from low pressure compressor. While the refrigerant liquid that leaves the flash chamber passes to evaporator during the second expansion valve.
1.8 Multipurpose Refrigeration System with a Single Compressor

When the applications need the refrigerants at different temperatures, the used system in this situation should contain two separate expansion valves and as well as the compressors. Such a system has two compressor will be uneconomical, some practical studies prefer to use a single compressor and route all the compression process.
2 The Objective of this Present Work and Literature Review

2.1. Problem Definition

In the current proposed work an alternate method (other than water or air) for cooling the steam power plant’s condenser will be investigated theoretically, numerically, and experimentally. The proposed method is a condenser-configuration using refrigerant in a closed-loop-cycle. The refrigeration will be a vapor compression cycle. The vapor compression refrigeration cycle system is a highly well-established technology forming the basis of many important industrial and agricultural and household applications. Amongst these the heat pumping (cooling and/or heating production), gas compression, air liquefaction, and separation and cryogenics. The power of industrial vapor compression units ranges from less than 1 kW to above 100MW. There is vast literature on the thermo-fluids-heat exchange analysis and principles of simple as well as modified vapor compression refrigeration cycles used in industry.

The main goal of this proposed work is to test the feasibility and verify the proposed idea of using a vapor compression refrigeration cycle as a condenser coolant, thereby replacing the environmentally polluting conventional water cooling and the low efficiency and costly air cooling methods. The current proposed project will also be able to compare the use of different refrigerants on the basis of performance, cost, and environmental impact with the conventional water and air cooling systems.

Hence, the objectives of the current proposal specifically are:

a) To develop a 1-D empirical-based theory for the thermal and fluid flow analyses, taking into consideration the operating conditions of the power plant steam cycle and environment

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b) To create programmable algorithms, based on the analyses, for predicting the refrigeration cycle’s performance under different operating conditions of the steam power plant cycle and environment and when using different refrigerants

c) To conduct experiments on a model refrigerant-cooled-condenser, then to verify, supplement, and complement the results obtained through the theoretical and numerical analysis and to demonstrate the feasibility of integrating the refrigeration cycles into the condenser of the steam power plant

d) To conduct cost analyses for the refrigerant cooled condenser

e) To be able to design and optimize a cost effective refrigerant-cooled-condenser unit for a specific steam power plant using the knowledge gained.

One of the very important components in any steam power plant is the condenser. It receives exhaust steam from turbine and condensates the vaporization by rejecting its heat to the cooling fluids that pass through the condenser tubes. In the condensing of steam, a vacuum is created. The vacuum reduces the backpressure on the turbine, and this reduction in backpressure increases the efficiency of the turbine.

There are two types of the steam power plant condensers: direct contact condensers and surface contact condensers. In the direct contact condensers, the exhaust steam from the turbine is cooled by mixing it with the coolant. Whereas, the cooling fluid passing through a sort of tubes in the surface contact condensers. According to the coolant perspective, the surface contact condensers (commonly used in the modern power plant) can be classified into wet cooled condensers, dry-cooled condensers, and hybrid condensers.

This chapter also provides some studies on steam power plant cooling, steam power plant condensers, and configurations dry cooling of steam power plant condensers that are drawn from
journals, books, conferences, academic theses, and workshop reports. Also, the most updated research on the factors that may affect the performance of dry-cooled condensers will be passed.

2.2 Studies on Steam Power Plant Condenser

In all power plants a large amount of heat has to be rejected by cooling via the condenser in order to sustain the thermodynamic cycle. Power plant cooling has been improved because it has a direct effect on the power plant efficiency. Conventional, power-plant, cooling-system recirculated, cooling water is sprayed over a horizontal tube bundle, while air is drawn over the bundle and steam in the turbine is condensed.

Wie et al. [7] designed and analyzed the variable working condition evaporative condenser for steam condensing of a steam feeding water pump for a 1000 MW air-cooled unit. The results showed that the condensing temperature decreased as the water flow decreased and the wind velocity increased; the water evaporation capacity increased as the wind velocity increased.

Parker and Treybal’s model (the first practical design used to evaluate the evaporative coolers) was employed to detect the relationship between mass and heat transfer at the fluid’s interfaces.

A new way to improve power plant efficiency with varying the condenser pressure was inducted by Vosough et al. [13]. They analyzed the energy and exergy for ideal Rankine cycle with reheat. The study showed that the condenser pressure played a prominent role in changing power plant exergy efficiency where the maximum energy losses mainly occurred in the condenser.

Najjar and Abubaker [11] investigated a new system for inlet air cooling called the indirect evaporative cooling system. A combination of a humidifier and vapor compression are included in this system. The results of this study revealed that this new system was able to cool down and
decrease the relative humidity in hot humid weather. Therefore, the IECS system played an essential role in increasing the power plant efficiency.

A numerical investigation of water spray for inlet air pre-cooling to enhance the performance in Natural Draft Dry Cooling Towers was performed by Alkheadhair et al. [12]. They generated a 3-D model numerical model to analyze the evaporation through a single nozzle. The model analyzed showed that spraying water into the inlet air can improve cooling by increasing the heat transfer rate. Also, the air velocity has an essential influence on droplet evaporation and transport. Moreover, the correct nozzle arrangement plays an important role to improve cooling.

In this section, a review for both cooling methods (wet-cooling condenser and dry-cooling condenser) will be accomplished.

2.2.1 Studies on the Wet Cooling of a Steam Power Plant Condenser

Several researches discussed the effect of cooling-water flow rates on the power plant’s costs and efficiencies. Anozie and Odejob [17] developed a theoretical study for simulation of a model for a thermal power plant with different rates of circulation condenser cooling water flow. The goals of this study was find the relationship between the flow rate and power plant efficiency, heat transfer area requirement, the operating cost of plant, and the fuel consumption. The study showed that minimizing the cooling-water flow rates reduces the heat transfer area and, therefore, the condenser size and annualized power plant’s capital coast decreasing. Also, the analysis of the thermal power plant showed that decreasing the cooling mass flow rate decreases the fuel consumption with a slight increase in the power plant efficiency.

Haseli et al. [18] introduced a theoretical study to evaluate the optimum temperatures of cooling water during condensation in a shell and tube through minimization of exergy destruction. The optimization results showed that (as the steam mass flow rates increases) the optimal inlet
cooling water temperature and exergy efficiency decrease, whereas exergy destruction increases. However, the results are higher for optimum values at higher condensation temperature in compression with the lower condensation temperature.

The condensation heat transfer for ammonia-water mixture in horizontal single pass shell and tube water cooled condenser was experimentally determined by Philpott and Deans [20]. The reason for adding ammonia is to enhance heat transfer rate in pure steam condensation. The results revealed that the condensate film disturbances increased the heat transfer through the film. However, the heat transfer enhancement was partially offset because of adding the thermal resistance of the vapor film, producing higher local condensation heat transfer coefficients of that predicted for pure steam only. The Marangoni effect (surface tension gradient along the condensation film) caused that enhancement and produced a disturbed, turbulent banded condensate film.

A modeling of steam condensation from the steam-air mixture in the inclined tubes of an air-cooled condenser was analyzed by Artemov et al. [19]. They stated that the value of the heat transfer coefficient controls the calculation of the effective coefficient of heat transfer.

A new equation to calculate the distribution temperature and velocity was derived by Dukler [25]. He calculated the condensing heat transfer coefficients; he also numerically calculated the liquid film thickness at the turbulent region with low Reynolds numbers.

Yousef et al. [26] investigated a compression analytical study using water and R-134a as cooling medium in the condenser of a steam power plant. The results showed that R-134a generates a higher condensation and heat transfer rate than water and, therefore, increases the cycle efficiency. In addition, using R-134a instead of water decreases the size of the condenser and increases its life time.
2.2.2 Studies on Dry/Air cooling of a Steam Power Plant Condenser

Because of the enhanced concerns about water use and water supply priorities, dry-cooling systems for thermal power plants are receiving increased consideration; even though the power plant with dry cooled condenser costs 10-15% more than a power plant with water cooled condenser [27]. Maulbetsch and DiFilippo [28] conducted a comparative study on the cost of wet cooling and dry-cooling on four different 500 MW gas-fired, combined-cycle power plants located in California. They found that the annual water consumption is reduced to 96% in dry cooling, but the plant cost is 5% to 15% higher than a wet-cooling power plant. Hassan et al. [29] provided studied the performance of the condenser in Al-Nassiriyah power plant in Iraq. He reported that the dry cooling of power plants can be used as an alternative to wet cooling since water conservation and environmental protection are critical issues. Also, Rebetez et al. [31] reported that due to water shortages in Europe in the summer periods of years 2003 and 2006, there was throttling in many steam power plants. Mideksa and Kallbekken [32] reported that using fresh water as a coolant in a condenser influences the electricity generation from a steam power plant due to the climate change in hot days and that makes the steam power plants incapable of producing the desired electrical power.

Direct air-cooled condenser units in power plants usually consist of finned tubes arranged in the form of a delta A-frame to drain condensate effectively, reduce distribution steam duct lengths, and minimize the required ground surface area. Conradie and Kroger [37, 44] reported a comparative study for two methods that can enhance the thermal performance of an air condenser: deluging the air-side surface of the air cooled condenser and cooling air entering the air-cooled condenser with adiabatic spray. Deluging the condenser with cooling water or spraying water into the inlet air can improve the rate of heat transfer.
Wen et al. [46] investigated a numerical simulation of flow and heat transfer of a direct air-cooled condenser cell in a power plant. This simulation described the mechanism of flow and heat transfer in the A-shaped frame condenser. The results showed that some flow phenomena such as backward flow and biased flow were gained through the coupled calculation.

A study of flow distribution from an air-cooled condenser module in a ~4000MW power plant is presented by Grimes et al. [49]. The results showed the existence of inhomogeneous distribution of cooling on the condenser fan due to the fan and heat exchanger interaction.

Hassan [50] conducted experimentally the effect of tube arrangement and condensate flow rate on a small, tube bundle in the presence of condensate inundation where a steam condenser simulation with air and artificial water is used. He tested two staggered tube arrangements having the same dimensions. The experimental results proved that the suggested tube arrangement has a less pressure drop coefficient than the conventional arrangement, but the suggested staggered tube arrangement has a less condensation rate than the conventional one.

Fischer and Ripley [51] reported a study of improving air-cooled condenser performance at the Yellowstone power plant by utilizing an innovative cleaning technology and a new finned tube. The results showed that using an effective cleaning technology and tracer gas inspection contributed to improving the air-cooled condenser.

2.3 Studies on Factors Affecting the Performance of Dry/Air Cooled Condenser

Maulbetsch and DiFilippo [43] conducted an adiabatic enhancement of air-cooled condenser power plants in California. They investigated test on different arrangements for various low-pressure nozzles. Also, they studied the effect of unevaporated droplets in the tube bundle. The results showed that the accumulation of unevaporated water droplets reduced the rate of heat transfer, and the nozzles tests showed that between 60% to 70% of the spry water is evaporated.
Esterhuyse and Kroger [36] conducted an experimental study to examine whether using electrostatic force can prevent or reduce droplets on the finned surface. They found that the wetting is reduced as the induction voltage is applied to the condenser.

Wen et al. [51] numerically investigated the influence of the ambient temperature on a direct air-cooled condenser with A-shaped frame. They found that the average inlet air velocity of the finned tube decreased as the ambient temperature increased. Owen and Kroger [44] reported that an increase in fan inlet temperatures above the reference temperature will result in a decrease in the ACC performance below its design value and cause subsequent reduction in turbine performance.

Maulbetsch et al [43] presented a study on the effects of wind on air-cooled condensers performance for power plants where it considered one of the significant challenges associated with air-cooled condensers design and performance. They tested an air-cooled condenser under different wind conditions to determine its operation and performance; then they compared their field data with the results of CFD modeling. The results showed that air recirculation can help in increasing the inlet temperature by 3 to 6°F, and the hot air recirculation has a lesser effect on the fan performance. They also found that the cross-flow over the fan inlet plan leads to a reduction in the flow in the affected cells. These effects increased the turbine exhaust pressure more than the ideal performance curves values.

A reliable numerical method for evaluating the performance of an air-cooled condenser was investigated by Owen and Kroger [47]. They presented a correlation between numerical results and test data of the air-cooled condenser at the El Dorado Power Plant under wind effect. They showed that the increase of the wind speed can lead to irregular speed flow at the inlet of fan; and, hence, the air-cooled condenser performance reduced. Also, Rooyen and Kroger [45]
numerically investigated in a study that modeled the thermal-flow of air through and about air-cooled steam condenser. The model analyzed showed that the flow distortions and low-pressure region at the upstream edge fan reduced the air-cooled condenser as the wind speed increased. However, the volumetric effectiveness of certain downstream fans increased as the wind speed increased.

He et al. [55] proposed a numerical model of an air-cooled power plant to provide the mechanism of flow through an air-cooled steam condenser using a User Defined Function (UDF). The results showed that the wind speed played an essential role in changing the pressure distribution, increasing the wind speed leads to an increase in the stable back pressure and, hence, the rate of heat transfer reduces.

Computational fluid dynamics is used by Owen and Kroger [56] to investigate the effect of porous wind screens on an air-cooled steam condenser’s performance under windy conditions at the El Dorado Power Plant in United States. They found that the installation of a wind screen below the air-cooled condenser fan platform, in a cross-type arrangement, showed that the performance of a fan upstream improved due to the stagnation effect of the screen on flow.

Yang et al. [51] reported that ambient winds may deteriorate the thermo-flow performances of air-cooled condensers, so it is of use to measure against the adverse impacts of winds upon the air-cooled condensers in a power plant. They proposed a computational model of a new trapezoidal array of air-cooled condensers at various wind speeds and directions. A comparative CFD simulation for trapezoidal array and rectangular was provided. The results showed that the inverted flow that happened due to high wind speed disappeared in the trapezoidal array of the air-cooled condenser; and, hence, the inlet cooling air temperature decreased as heat rejection increased.
O’Donovan et al. [49] performed a series of tests on condensers under vacuum conditions where they concentrated on the steam-side characterization of a Modular air-cooled condenser. The results from the vacuum measurements indicated that the fan has a direct influence on the condenser conditions and, hence, affect turbine work and the power plant efficiency. For a given steam, mass-flow rate, the condenser pressure and temperature decrease as the fan’s rotational speed increases.

Ma et al. [53] established a numerical simulation and thermal calculation model of the dry cooling tower to study the effects of ambient temperature and crosswind on thermos-flow performance of the tower under the energy balance of the indirect dry-cooling system. They found that the temperature of the outlet water of the tower is changed linearly with the ambient temperature, whereas it changed nonlinearly with crosswind speed.

A new theoretical model to predict the part-load thermal performance of natural draft dry cooling towers (NDDCT) was investigated by Ma et al. [50]. The model is appropriate to calculate the heat rejection of the tower at different ambient temperatures but unsuitable under crosswind velocities higher than critical wind velocity. Based on the observed results, it can be concluded that at a high temperature difference, the tower has a larger resistance to crosswind, and there was a slight decrease in cooling efficiency.

He et al. [55] investigated a numerical simulation with Computational Fluid Dynamics (CFD) for hot air recirculation in an air-cooled condenser under various ambient conditions. Wind speed and wind direction were the two mechanism of hot air that were simulated. They found that the air recirculation increased with wind speed. Also, the rate of heat transfer decreased; and, hence, the hot air recirculation has a adverse influence on the air-cooled steam condenser performance.
2.4 Studies on Vapor Compression Refrigeration Systems

As the previous studies proved, the refrigerants have much lower temperatures and much higher heat transfer rates than air and water, so the refrigerants are optimized instead of water/air as a coolant for steam power plant condenser.

2.4.1 Studies on Exergy Analysis of Vapor Compression Refrigeration Systems

There are many factors that can lead to an increase in the exergy efficiency and minimize the exergy losses. Increasing the reference temperature and reducing the temperature difference between condensing and evaporating temperature can improve the exergy efficiency [54].

An exergy analysis for a simple vapor compression regeneration system was reported by Kumar et al. [57]. They found that the compressor has the most exergy loss compared with other components; however, using multistage compression may help to reduce the exergy losses.

Based on exergy analysis, Yumrutas et al. [58] introduced a computational model to realize the effects of the evaporating and condensing temperatures on the exergy losses, the pressure losses, and the coefficient of performance (COP) of a vapor compression refrigeration cycle. This study indicated that the lowest pressure with air flow and the largest one with air flow takes place in the evaporator due to the increasing evaporating temperature. Also, the results showed that the exergy losses decrease with the evaporator and increase in the condenser with an increase of evaporator temperatures.

Arora and Kaushik developed a theoretical exergy analysis of a vapor compression refrigeration system (VCR) with R404A, R502, and R507A [56]. The results indicated that R502 shows the maximum COP and energetic efficiency among all the refrigerants according to condenser temperature. However, the COP and energetic efficiency of both R507A and R404A can be proved by the subcooling of the condensed liquid refrigerant.
Kabul et al. [59] introduced a study for energy and exergy analysis of a vapor compression refrigeration system with an internal heat exchanger using R600a. The results showed that the value of COP, the efficiency ratio, and the energetic efficiency increase with the increase of the evaporator temperature; whereas, the irreversibility decreases. Also, the results showed that the increase in condenser temperature can decrease the exergetic efficiency, the efficiency ratio, and the value of COP. However, the total irreversibility rate increases with an increase in the condenser temperature.

2.4.2 Studies on Refrigerant Heat Transfer

More researchers have started studying the concept of SPPC cooling by a refrigeration system to investigate this cooling method’s performance in the steam power plant in different ambient conditions. Traditional chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) are commonly used in the development and commercial application of hydrofluorocarbon (HFC) refrigerants. One of the reasons why researchers have substituted the HCFC-22 with the CFC in recent years is its chemical structure. The CFC refrigerant has a chlorine that combines with ozone in the atmosphere. Therefore, the alternative refrigerants whom researchers prefer to use in the existing facilities should not only have low ozone depletion potential (ODP) but also should have low global warming potential (GWP), be less flammable, be safe, be reliable, and finally be economical [60].

Recently researchers have replaced R-22 by HFC-410A because of its appropriate thermodynamics of heat transfer and environment friendliness. HFC- 410A, which is a mixture of HFC-32 and HFC-125, has been used widely in air conditioning system applications and in high operating pressure equipment. In a domestic refrigeration and mobile air conditioning system
HFC-134a has become the alternative for CFC-12 in chillers and heat pumps. Whereas HCFC-22 has been replaced by HFC-407C; which is mixture of HFC-32, HFC-134a, and HFC-125 [62].

Many researchers have studied the heat transfer and flow characteristics of refrigerants. An experimental study of heat transfer and the pressure gradient of hydrocarbon refrigerants R-600aR-290, R-1270, and HCFC, R-22 inside a horizontal double-pipe, heat exchanger during a condensation and evaporating process is presented by Lee et al. [61]. The results show that the heat transfer coefficient of R-22 is lower than the other hydrocarbon refrigerants. Also, increasing the mass flux usually leads to an increase in the average heat transfer coefficient. Ebisu and Torikoshi [64] presented a theoretical study of heat transfer characteristics of R-22, R-410A and R-407C in a 6.4 mm diameter tube, while Wang et al. [65] discussed the pressure gradient for the same refrigerants in small diameter tubes. The results showed that R-410A has a higher heat transfer coefficient than other refrigerants, otherwise R-410A has a lower pressure gradient than others. In addition, a theoretical study of the heat transfer coefficient and pressure drop for R-13a, R-410A, and R-22 in micro and smooth tubes is reported by Christoffersen et al. [66]. The results show that heat transfer rate in micro fin tubes is higher than the heat transfer over smooth tubes. However, in the smooth tubes, R-410A has a higher heat transfer coefficient and lower pressure drop than R-22.

Researchers noted some differences between R-410A and R-22. R-410A has higher pressure than R-22 in both the suction and discharge section of the refrigeration system but has lower temperature in the same situation. Another difference is that R-410A needs small diameter tubing and then a high heat transfer coefficient with lower refrigerant, side-heat transfer exchange surface area compared with R-22 [67].
Several experimental studies for refrigerants pressure gradient and condensation heat transfer were introduced by researchers in past years. One of studies discussed the condensation heat transfer of R-22, R-125, R-410A, R-134a, and R-125 by Cavallini et al. [68]. The study implied that R-134a has the lowest heat transfer coefficient and R-410A has the highest heat transfer coefficient. Also, they did another study [69] to measure the pressure gradient and heat transfer of R-125, R-134a, R-32, R-22, R-410A, and R-236ea inside a smooth tube. The result revealed that the mass velocity plays an important role in heat transfer amount, where the heat transfer coefficient increases as the mass velocity increases.
3 The Condenser in the Steam Power Plant

3.1 Introduction

Power plant condensers have been improved since their use as a power plant component. The performance of the condenser has been a main focus of interest for designers to develop because the performance of the condenser directly affects the power plant efficiency. There were many areas in which the performance of the condenser was enhanced. One of these areas is the type of the condenser. Power plant condensers can be classified into two major types: direct contact condensers and, surface contact condensers. In the direct contact condensers the steam, which is exhausted from the turbine, is cooled by mixing it directly with the cooling fluid. On the other hand, in the surface contact condensers the steam and the cooling fluid are not mixed directly, there is a solid surface, such as tubes, which separate the steam and the cooling fluid.

3.2 Surface Contact Condensers

A surface contact condenser is commonly used in the modern steam power plant. This type of power plant condenser is further classified based on the type of cooling fluid water cooled condenser, air cooled condenser, and hybrid wet-dry condenser.

3.2.1 Water Cooled Condenser

The water cooled condenser is widely used. It is mainly a shell-and-tube type in which the steam flows in the shell side and the water flows in the tube side. The cooling water can be drawn from a source or what is called a low temperature reservoir, such as a sea, a lake, or a river. After it cools the condenser, the water will return back to the source; this type of cooling system is called once-through cooling system. The cooling water, moreover, can be recirculated by using cooling tower systems. In the cooling tower, the cooling water is circulated in a closed cycle. The hot
cooling water, which comes from the condenser, will be cooled down by evaporation in huge towers, and remove the heat to the surrounding. Then it will be returned back to the condenser again. This system is known as a recirculating wet cooling system. The shell-and-tube condenser can further be divided into a horizontal type and a vertical type.

3.2.1.1 Shell-and-Tube Condenser: Horizontal Type

The tubes inside this type of condenser are arranged horizontally and supported by baffles. The baffles are usually put with cut vertical to let the steam to flow from side to side to reach every point inside the condenser. As shell-tube condenser has two types: first, a single-pass or E-type where the cooling water inters the condenser from one side and exits from the other side, Fig. 3.1 shows the E-type condenser.

![Figure 3.1 Single-pass and double-pass condenser [3]](image)

The second type is double-pass condenser where the cooling water inters and exits the condenser from one side as shown on Fig. 3.2 In the double-pass condenser, the cooling water is circulated inside the condenser to increase the performance of the condenser
3.2.1.2 Shell-and-Tube Condenser: Vertical Type

In the vertical type, the tubes of the condenser are arranged vertically and supported by baffles. The vapor can flow inside or outside the tubes and the cooling water as well. The vertical type is further classified into two types: down-flow where the vapor enters from the top side of the condenser, and the condensate drains down to the bottom of the condenser through the use of gravity and vapor shear effects, as is shown on the Fig. 3.3. The second type is up-flow condenser. In this type, vapor enters the condenser from the bottom side and flows upward inside the tubes, while the condensate drains down the tubes through the effect gravity, Fig. 3.4 shows schematic of the condenser.

3.3 Theoretical Basis to Guide Experiment

To help design the optimum experimental platform for this project, the required guiding theoretical basis has to be developed first.

In the Rankine cycle, it is essential to have a low temperature reservoir to reject some of the heat that is gained in the high temperature reservoir in order to complete the thermodynamic cycle. Basically, the high temperature reservoir is the boiler, and the low temperature reservoir is the condenser. In the condenser, heat transfer occurs between the hot steam and the coolant. The condenser resaves “dead” steam from the final stage of the steam turbine. The steam will release its latent heat condensation in order to change its phase to liquid. The condenser is a necessary
Component in the steam power plant because it directly affects the thermal efficiency of the cycle. Thus, keeping the performance of the condenser as high as possible is a key factor to gain high power plant efficiency. In order to predict and improve the performance of the condenser, heat transfer, energy, and exergy analyses have to be carried out.

3.4 Heat Transfer Analysis for Power Plant Condensers

The heat is transferred from the hot steam, which flows over the exterior surface of tubes (shell-side of the condenser), to the coolant, which flows interior the tubes of the condenser (tube-side of the condenser). The result from this process is that the steam is condensed by removing its latent heat of condensation and the coolant is heated; Fig. 3.5 shows a schematic of the cooling process. The phase change of the steam causes a dramatic decrease in the specific volume of the steam;
consequently the steam will become liquid (condensate). Then the condensate will flow down through effect of gravity into the lower part of the condenser, in the so called hot-well, all condensate is collected there and pumped again into the cycle. The heat that is removed by the coolant is discharged to the surrounding, low temperature reservoir such as oceans, lakes, or rivers. With an assumption of a constant overall heat transfer coefficient, a circulating coolant mass flow rate, and a total heat transfer area, the only factor that affects the heat transfer inside the condenser is the logarithm mean temperature difference (LMTD).

There are six different thermal resistances that affect the heat transfer rate inside the condenser. The thermal resistances are water film resistance exterior and interior the tubes, wall material resistance, non-condensable gases resistance, and fouling resistance that exists on the exterior and interior tube’s surfaces. The fouling is made of inorganic deposits, biofouling, or a layer of corrosion. In this analysis, we assume that the tubes are clean, so there is no fouling on the tubes. The overall thermal resistances can be calculated by the sum all the above resistances.

![Figure 3.3 Schematic of energy balance for condenser](image-url)
3.4.1 The Governing Equations for the Heat Transfer Analysis

3.4.1.1 Heat Transfer inside the Condenser

Inside the condenser, the steam is in a saturated or a mixture (steam/liquid) phase. That means the temperature of the steam entering the condenser and the temperature of the condensate leaving the condenser are at a saturated temperature ($T_{\text{sat}}$) value. The saturated temperature is a direct function of the pressure of the steam. The pressure inside the condenser drops slightly due to flow resistance. As a consequence of this pressure drop, the saturated temperature will also decrease as will be shown later. The steam mass flow rate ($m_{s\text{.in}}$) enters the condenser is mixed with the non-condensable gases ($m_{a\text{.in}}$) because the condenser never receives pure steam from the turbine. As the mixture (steam/ non-condensable gases) moves inside the condenser the mass of the steam is decreased due to the phase change of the steam from gas to liquid, and the mass of non-condensable gases will increase due to the air leakage and the chemical reaction; however, in this analysis the assumption of constant mass of the non-condensable gases is applied. Moreover, as illustrated in Fig. 3.5 the inlet coolant temperature is ($T_{\text{cl\text{.in}}}$), the outlet coolant temperature is ($T_{\text{cl\text{.out}}}$) and the coolant mass flow rate is($\dot{m}_{\text{cl}}$).

In order to develop an expression that represents the heat transfer within the condenser, we will consider an elemental tube disk as in Fig. 3.6 where ($dQ$) is the differential heat flux through the element, ($dA$) is the differential area, ($\Delta T$) is the temperature difference across the partial element, and ($U$) is the overall heat transfer coefficient, thus the heat flux can be expressed by equation (3.1)
\[ dQ = U \Delta T d \]  \hspace{1cm} (3.1)

where

\[ \Delta T = T_h - T_c \]

On the other hand the heat transfer can be defined as

\[ dQ = - m_h C_{ph} \, dT_h = m_c C_{pc} \, dT_c \]  \hspace{1cm} (3.2)

From equation (4.2), we can get

\[ dT_h - dT_c = - dQ \left( \frac{1}{m_h C_{ph}} + \frac{1}{m_c C_{pc}} \right) \]  \hspace{1cm} (3.3)

Substituting equation (4.1) into (4.3)

\[ \frac{d(T_h - T_c)}{(T_h - T_c)} = - U \left( \frac{1}{m_h C_{ph}} + \frac{1}{m_c C_{pc}} \right) \, dA \]  \hspace{1cm} (3.4)

By integrating both sides between the ends of the heat exchanger 1-2, with a consideration that the specific heat (\( C_{ph} \)) and (\( C_{pc} \)) are constant.
\[
\ln \left( \frac{(T_{h2} - T_{c2})}{(T_{h1} - T_{c1})} \right) = -UA \left( \frac{1}{m_h C_{ph}} + \frac{1}{m_c C_{pc}} \right) \tag{3.5}
\]

Also integrating equation (4.2) between 1-2

\[
Q = -m_h C_{ph} (T_{h2} - T_{h1}) = m_c C_{pc} (T_{c2} - T_{c1}) \tag{3.6}
\]

Solving for \((m_h C_{ph})\) and \((m_c C_{pc})\) on equation (4.6)

\[
m_h C_{ph} = \frac{Q}{(T_{h1} - T_{h2})}
\]

\[
m_c C_{pc} = \frac{Q}{(T_{c2} - T_{c1})}
\]

Substituting these equations into equation (4.5)

\[
Q = UA \left( \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln \left( \frac{(T_{h2} - T_{c2})}{(T_{h1} - T_{c1})} \right)} \right) \tag{3.7}
\]

The second term on the R.H.S is called Log Mean Temperature Difference \((LMTD)\).

\[
LMTD = \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln \left( \frac{(T_{h2} - T_{c2})}{(T_{h1} - T_{c1})} \right)} \tag{3.8}
\]

This expression is valid for a counter-flow heat exchanger or for a heat exchanger where the temperature of one of the flows is constant during the condensation process which is the case in the power plant condensers. Consider the following:

Since the change between \(T_{sat \text{ in}}\) and \(T_{sat \text{ out}}\) is very small, we assume that \(T_{sat \text{ in}} = T_{sat \text{ out}} = T_{sat}\) in this case.

\[
T_{h1} = T_{h2} = T_{sat}
\]

\[
T_{c1} = T_{c\text{lin}}
\]

\[
T_{c2} = T_{c\text{out}}
\]

Thus equation (4.8) becomes
\[
\text{LMTD} = \frac{(T_{\text{sat}} - T_{\text{cl.out}}) - (T_{\text{sat}} - T_{\text{cl.in}})}{\ln \left( \frac{(T_{\text{sat}} - T_{\text{cl.out}})}{(T_{\text{sat}} - T_{\text{cl.in}})} \right)} = \frac{(T_{\text{cl.out}} - T_{\text{cl.in}})}{\ln \left( \frac{(T_{\text{sat}} - T_{\text{cl.out}})}{(T_{\text{sat}} - T_{\text{cl.in}})} \right)}
\] (3.9)

We can write equation (4.7) as

\[
\dot{Q} = U A_{\text{sur}} \text{ LMTD}
\] (3.10)

The (LMTD) is an important parameter to design the condenser or to predict the performance of the condenser. It relates the heat transfer within the condenser with quantities such as saturated steam temperature, inlet and outlet coolant temperature, overall heat transfer coefficient, and total heat transfer surface area. If the desire is to calculate the condenser geometry, the designer should have an idea about the (LMTD), as the total surface area can be determinant from equation (3.11).

\[
A_{\text{sur}} = \frac{\dot{Q}}{U \text{ LMTD}}
\] (3.11)

By knowing the total heat transfer area, the geometric parameters of the condenser and the number of tubes can be estimated by equation (3.12). These parameters are needed and must be optimized to suit the design criteria.

\[
A_{\text{sur}} = \pi d_{\text{out}} L N
\] (3.12)

### 3.4.1.2 Overall Heat Transfer Coefficient

The thermal resistances inside the condenser are difficult to estimate due to the geometry of the tubes and the resistances being represented by the solid and liquid. As mentioned previously, in the condenser there are six thermal resistances as illustrated in Fig. 3.7 Two are liquid resistances by water film inside and outside the wall of the tubes; these resistances will be determined by calculating the convective heat transfer coefficient for both water films. The non-condensable gases resistance is one of the thermal resistances inside the condenser that should be accounted for. The other three are solid resistances: wall material resistance and two fouling resistances. Fouling resistances are difficult to determine because they do not have uniform geometry and there
is no specific thermal conductive for the fouling material; so they are estimated from the experiments [72, 73]. In our calculation, we assume that the tube surfaces are clean so there are no fouling resistances taking place.

![Figure 3.5 Thermal resistance on the condenser tube](image)

After calculating each single resistance, the overall thermal resistance is the sum of all resistances illustrated above. By taking the outer surface area as reference overall thermal resistance is as in equation (3.13).

\[
R_{\text{tot}} = R_{\text{film,in}} \frac{d_{\text{out}}}{d_{\text{in}}} + R_{\text{wall}} + R_{\text{film,out}} + R_a
\]  

(3.13)

Thus the overall heat transfer coefficient (U) can be determined by

\[
U = \frac{1}{R_{\text{tot}}}
\]  

(3.14)
3.4.1.3 Coolant Film Thermal Resistance

The coolant film resistance is a thermal resistance that is formed on the interior surface of the tubes. When the coolant flows inside the tube, it creates a thin boundary layer on the tube wall. This layer will build a thermal resistance in the direction of the heat flow. The coolant film resistance is a function of flow velocity, temperature, density, specific heat, tube diameter, and viscosity [72]. It is known from convective heat transfer analysis that in order to compute the fluid thermal resistance, the convective heat transfer coefficient must be determined. There are many empirical correlations that estimate the convective heat transfer coefficient. One of these correlations is by Ozisik [72] who derived it from Nusselt. Ozisik found from experimental data that convective heat transfer coefficient is proportional to the diameter to length ratio of the tube \( \left( \frac{d_{in}}{L} \right) \). He developed an empirical correlation that relates Nusselt number to Reynolds number, Prandtl number, and the diameter to length ratio [73]. This correlation estimates the interior thermal resistance of the circular tube by obtaining convective heat transfer coefficient from equation (3.16).

\[
\begin{align*}
\text{Nu}_{cl} &= \frac{h_{in} d_{in}}{k_{cl}} = 0.036 \text{Re}_{cl}^{0.8} \text{Pr}_{cl}^{1/3} \left( \frac{L}{d_{in}} \right)^{-0.054} \quad (3.15) \\

h_{in} &= 0.036 \frac{k_{cl}}{d_{in}} \text{Re}_{cl}^{0.8} \text{Pr}_{cl}^{1/3} \left( \frac{L}{d_{in}} \right)^{-0.054} \quad (3.16) \\

R_{film.in} &= \frac{1}{h_{in}} \\

R_{film.in} &= \left[ 0.036 \frac{k_{cl}}{d_{in}} \text{Re}_{cl}^{0.8} \text{Pr}_{cl}^{1/3} \left( \frac{L}{d_{in}} \right)^{-0.054} \right]^{-1} \quad (3.17)
\end{align*}
\]

This relation is valid in the range of \( 10 < \frac{L}{d_{in}} < 400 \)

where
\[ Pr_{cl} = \frac{\mu_{cl} \cdot C_{pcl}}{K_{cl}} \]  
\[ Re_{cl} = \frac{\rho_{cl} \cdot d_{in} \cdot V}{\mu_{cl}} \]  
\[ V = \frac{m_{cl}}{\rho_{cl} \cdot N \cdot \frac{d_{in}}{4} \cdot \pi} \]

All properties of the coolant are evaluated at bulk temperature \( T_b \), which is the average between the inlet coolant temperature and the outlet coolant temperature [72].

\[ T_b = \frac{\left( T_{cl,\text{in}} + T_{cl,\text{out}} \right)}{2} \]

### 3.4.1.4 Tube Wall Thermal Resistance

The second thermal resistance is the resistance of the material of the tube or the tube wall thermal resistance. This resistance is accurately and easily obtained because it is a direct function of the thermal conductivity of the material of the tube. The thermal conductivity of various materials are known and well documented. The wall resistance is a small amount compared to the fluid resistances because the tube thickness is small and the thermal conductivity of the metal is high. The relation that governs the wall resistance can be found in different conductive heat transfer textbooks [74].

\[ R_{\text{wall}} = \frac{d_{out}}{2 \cdot K_{\text{metal}}} \ln \left( \frac{d_{out}}{d_{in}} \right) \]

### 3.4.1.5 Condensate Film Thermal Resistance

The third thermal resistance that exists inside the condenser. When the hot steam contacts the cold exterior surface of the tubes, the steam temperature will drop to the wall temperature. Because of the phase change, a thin film of water (condensate) forms the outer the tube surface; this film creates a thermal resistance for the heat to transfer from the steam to the coolant, Fig. 3.8 shows
the condensate film on a single horizontal tube surface. To calculate the condensate heat transfer coefficient for a single horizontal smooth tube, Nusselt approximated a correlation for a laminar flow equation (3.23) [75]. This relation estimates the Nusselt number for a single horizontal tube; and then from the Nusselt number, the convective heat transfer coefficient can be calculated.

Within the condenser, the condensate moves by effect of gravity to the lower tube banks. Since the flow path is too short to form turbulence flow, the flow inside the condenser is always laminar. The thickness of the film is the key parameter to estimate the thermal resistance. The film thickness is varied from one tube to another because in the horizontal tube condenser, the tubes are arranged in banks so when the condensate falls off one tube, it will fall onto the tube below. As a result, in the lower tubes the average condensate of the film thickness is increased. This extra thickness will decrease the heat transfer coefficient as we go down in the condenser bundle. This phenomenon is known as the effect of condensate inundation.

There have been many theories that studied the inundation effect in the surface condensers. McNaught and Cotchin [74] developed a simple correction factor that accounts for the inundation effects; the correction factor is based on the diameter and pitch of the tube bundle. However, in this study another correction factor that is represented by Davidson and Rowe [74] will be used. In Davidson and Rowe’s relation, the row number of the tube is the main parameter. The row number of the tube is counted from the top of the condenser to the bottom. Equation (4.25) shows the correction factor of the inundation effect \( F \) as represented by Davidson and Rowe [74].

Vapor shear is another phenomenon that should be considered when calculating the condensate thermal resistance. “There is a small amount of data for the influence of vapor shear in the tube bundles” [75]. Berman and Tumanov [75] recommended a correlation that accounts for
vapor shear in the condenser, equation (4.26). In this analysis both inundation and vapor shear will combine their effects in one condensate thermal resistance relation as in equation (4.27) [75].

\[
\text{Figure 3.6 Laminar film condensation on a horizontal tube [5]}
\]

Nusselt number correlation for a single horizontal tube [73]

\[
\text{Nu}_{cs} = \frac{h_{out} d_{out}}{K_{cs}} = 0.728 \frac{d_{out}}{K_{cs}} \left[ \frac{\rho_{cs} (\rho_{cs} - \rho_{s}) g h^*_{fg} K_{cs}}{\mu_{cs} (T_{film} - T_{wall})} \right]^{\frac{1}{4}} \tag{3.23}
\]

Where

\( \rho_{cs}, \mu_{cs}, K_{cs} \): Density, viscosity, and thermal conductivity of the condensate around the exterior tube surface are evaluated at film temperature. The later discussion will show how to calculate film and wall temperature.

A modified latent heat of condensation \((h^*_{fg})\) is used instead of the latent heat of condensation \((h_{fg})\); which is presented in equation (3.23) to account for the cooling of the liquid below the saturation temperature [75].
\[ h_{fg}^* = h_{fg} + 0.68 \ C_{pcs} (T_{sat} - T_{wall}) \]  \hspace{1cm} (3.24)

The correction factor of the inundation effect \( F \) for \( (N)^{th} \) tubes is in equation (3.25) [73]

\[ F = \frac{3}{n} - (n - 1) \frac{3}{n} \]  \hspace{1cm} (3.25)

The vapor shear correlation or correction term as presented by [75] is as in (3.26).

\[ C_{v,g} = 1 + 0.0095 \ \text{Re}_{mix}^{11.8 / Nu_{cs}} \]  \hspace{1cm} (3.26)

The correlation for the condensate thermal resistance that is combined with both inundation and vapor shear effects for \( (N)^{th} \) tubes is presented as follows. From equations (3.23), (3.24), (3.25), and (3.26)

\[ R_{film\text{.out}} = \left[ \frac{K_{cs}}{d_{out}} \ \text{Nu}_{cs} \left( 1 + 0.0095 \ \text{Re}_{mix}^{11.8 / Nu_{cs}} \right) F \right]^{-1} \]  \hspace{1cm} (3.27)

Where

\[ \text{Re}_{mix} = \rho_{mix} d_{out} V_{\infty} \]  \hspace{1cm} (3.28)

Free stream velocity \( (V_{\infty}) \) depends on the location inside the condenser where \( (V_{\infty}) \) in the first row is different from \( (V_{\infty}) \) in the other rows [75].

a- \( (V_{\infty}) \) in the first row

\[ V_{\infty(n=1)} = \frac{\dot{m}_s + \dot{m}_a}{\rho_s A_{vd}} \]  \hspace{1cm} (3.29)

where

\[ A_{vd} = w_{ts} L \]

b- \( (V_{\infty}) \) in the other rows

\[ V_{\infty(n+1)} = \frac{(m_s + m_a) - \sum_{i=1}^{n} \dot{m}_{cs,i}}{\rho_{mix} A_{mv}} \]  \hspace{1cm} (3.30)

where
$$A_{mv} = L (NTR - 1) \left( P_t - \frac{\pi d_{out}^2}{2\sqrt{3} P_t} \right)$$

### 3.4.1.6 Non-Condensable Gases Thermal Resistance

Non-condensable gases commonly exist in numerous heat exchanger applications. In the case of the power plant condenser, a small fraction of gases, which is dissolved in the feed-water will reach the condenser with the dead steam [76]. Thus, the condenser never receives pure steam from the turbine. The other source of non-condensable gases is that the condenser being under the vacuum, which lets the gases to leak into the condenser from the atmosphere. And also, gases such as oxygen and hydrogen can resulted from water decomposition because of the thermal or chemical reaction inside the condenser [77].

The non-condensable gases have two main effects inside the condenser; first of all, it will raise the operating pressure of the condenser if not being vented. The condenser pressure ($P_c$) is the sum of steam partial pressure ($P_s$) and the gases partial pressure($P_a$), equation (3. 31). As more gases leak or form inside the condenser, the relative content of the gases ($\varepsilon$) in the mixture (steam/gases) will increase; as result the partial pressure of the gases will increase. Equation (3. 33) shows the relation. This increase pressure of the gases will block the steam to the flow into the condenser from the turbine; and, thus, the steam will not completely expand in the turbine. As a result the turbine output will decrease and the plant efficiency will decrease as well.

The second effect of the non-condensable gases is that the gases will blanket the outer surface of the tubes, which will cause a thermal resistance ($R_a$)for the heat to transfer from the hot steam to the coolant [77]. Marto [75] found an “empirical expression for the mass transfer coefficient in a tube bundle downward flow of a steam-gas mixture”. This correlation can be an equivalent to non-condensable gases heat transfer coefficient, which will add another thermal resistance inside the condenser, equation (3. 35) [75].
Even a small friction of gases inside the condenser will cause a significant heat resistance. Fig. 3.9 by Collier and Thome [76] shows the relation between the mass fraction of the gases and the heat transfer with the presence of gases and the heat transfer with pure steam. The chart compares the heat transfer for both quiescent vapor and forced vapor conditions. It shows that for the quiescent vapor, even a small fraction of gases will have a dramatic impact on the heat transfer. On the other hand for the forced vapor, the impact is a little bit less. In this study the calculations are based on the region that is very close to zero. The mass fraction of the gases ($\varepsilon$) is assumed to be constant throughout the condenser and equals to (0.00012), adapted from a previous study [73].

\[
P_c = P_s + P_a \quad (3.31)
\]

\[
\varepsilon = \frac{\dot{m}_a}{\dot{m}_a} \quad (3.32)
\]

From the gas law with consideration that the volumes and temperature are the same, one can get

\[
\frac{P_a}{P_s} = 0.622 \varepsilon \quad (3.33)
\]

From (3.31) and (3.33)

\[
P_s = \frac{P_c}{1 + 0.622 \varepsilon} \quad (3.34)
\]
sent of steam in the presence of air [67]

Non-condensable gases heat transfer coefficient as represented by [75]

\[
h_a = \frac{a D}{d_{out}} \frac{1}{Re_s^{\frac{1}{2}}} \left( \frac{P_c}{P_c - P_s} \right)^b P_c^{\frac{1}{3}} \left( \frac{\rho_s L}{T_{sat}} \right)^{\frac{2}{3}} \frac{1}{(T_{sat} - T_{film})^{\frac{1}{3}}} \]  \tag{3.35}

Thus, the thermal resistance is:

\[
R_a = \frac{1}{h_a} = \left[ a \frac{D}{d_{out}} \frac{1}{Re_s^{\frac{1}{2}}} \left( \frac{P_c}{P_c - P_s} \right)^b P_c^{\frac{1}{3}} \left( \frac{\rho_s L}{T_{sat}} \right)^{\frac{2}{3}} \frac{1}{(T_{sat} - T_{film})^{\frac{1}{3}}} \right]^{-1} \tag{3.36}

Where:

(D) is the diffusion coefficient [75]
\[ D = \frac{0.926}{P_c} \left( \frac{T^{2.5}}{T + 245} \right) \] (3.37)

where

\[ T = \frac{(T_{\text{sat}} + T_{\text{film}})}{2} \text{ in (K), and } P_c \text{ in (Pa)} \] (3.38)

(a) and (b) in equations (4.37) and (4.38) are constants and they depend on \( \text{Re}_s \):

\[ a = 0.52 \quad \text{and} \quad b = 0.7 \quad \text{for } \text{Re}_s \leq 350 \]

\[ a = 0.82 \quad \text{and} \quad b = 0.6 \quad \text{for } \text{Re}_s > 350 \]

Now we reach to the point where we can calculate all thermal resistances that are occurred inside the surface condenser, and therefore, we can determine the overall heat transfer coefficient by using equation (3.14).

Condenser pressure \( (P_c) \):

The condenser pressure is not constant inside the condenser as we always assume; it is slightly lowered. It decreases because of increasing of the resistance to the flow of the steam [13]. As the steam moves across the rows of the condenser, the pressure is slightly lowered. To calculate the amount of pressure in each row, consider relation (3.41) [75]. According to [78], “the pressure drop from the inlet to the exit of the condenser is called the steam exhaust resistance of the condenser” equation (3.40).

\[ P_{c(n+1)} = P_{c(n)} - \left[ \left( 0.2 \cdot C_r \cdot \frac{\bar{m}_v^2}{\rho_s} \right)_n - \left( \frac{\bar{m}_v^2}{\rho_s} \right)_n - \left( \frac{\bar{m}_v^2}{\rho_s} \right)_{n+1} \right] \] (3.39)

where

\[ C_r = \frac{A_{vd}}{A_{mv}} \]

\[ \bar{m}_v = \rho_s V_\infty \]

The steam exhaust resistance is
\[ \Delta P_c = P_c - P_c^e \]  

(3. 40)

There is another relation which is useful to assess the condenser performance. This relation is the vacuum pressure inside the condenser [9]. It is a relation between the condenser pressure and the atmospheric pressure.

\[ P_{\text{vac}} = \left( \frac{P_{\text{atm}} - P_c}{P_{\text{atm}}} \right) \times 750.061 \]  

(3.41)

where:

\( P_{\text{atm}} \), \( P_c \) are in (bar), and \( P_{\text{vac}} \) is in (mmHg)

Wall and film temperature \( (T_{\text{wall}}, T_{\text{film}}) \):

The wall temperature and the outer film temperature are unknown. The best method to calculate these temperatures is the iterative approach [78]. The steps of the iteration are described below.

\[ T_{\text{film}} = T_{\text{sat}} - \frac{3}{4} \left( T_{\text{sat}} - T_{\text{wall}} \right) \]  

(3. 42)

1- Assume initial film temperature.

2- Evaluate all properties of the condensate at the assumed film temperature.

3- Use (4.46) to find the initial wall temperature.

4- Use the properties of the condensate to find the condensate heat transfer coefficient \( (h_{\text{out}}) \) and the overall heat transfer coefficient \( (U) \) as described previously.

5- Calculate the wall temperature by using the heat transfer equation as follows.

\[ h_{\text{out}} A_{\text{sur}} (T_{\text{sat}} - T_{\text{wall}}) = A_{\text{sur}} U (T_{\text{sat}} - T_b) \]

Thus

\[ T_{\text{wall}} = T_{\text{sat}} - \frac{A_{\text{sur}} U}{h_{\text{out}} A_{\text{sur}}} (T_{\text{sat}} - T_b) \]  

(3.43)
6- Use the new wall temperature to calculate the film temperature using equation (4.43).

7- Compare the calculated new film temperature with the one from the initial step if they are not the same repeat the calculation till the iteration converges.

3.4.2 Relations for the Heat Transfer and Energy Analysis

1. Latent heat transfer rate is governed by [73].

   \[ \dot{Q}_s = \dot{m}_c s \dot{h}_{fg} \]  

   (3.44)

2. Heat transfer to the coolant is calculated by the energy balance of the in-flow and out-flow.

   \[ \dot{Q}_{cl} = \dot{m}_{cl} C_{pcl} (T_{cl,\text{out}} - T_{cl,\text{in}}) \]  

   (3.45)

If no heat losses to the environment is assumed, we can relate equations (3.10), (3.44) and (4.45).

   \[ \dot{Q}_s = \dot{Q}_{cl} = \dot{Q} \]  

   (3.46)
3. From (3, 9), (3, 10), and (3, 45), \((T_{cl,\text{out}})\) can be calculated by knowing the saturated temperature of the steam, the inlet temperature of the coolant, overall heat transfer coefficient, the given heat surface area, and the coolant mass flow rate [75].

\[
m_{cl} c_{pcl} (T_{cl,\text{out}} - T_{cl,\text{in}}) = U A_{\text{sur}} \left( \frac{T_{cl,\text{out}} - T_{cl,\text{in}}}{\ln \left( \frac{T_{sat} - T_{cl,\text{out}}}{T_{sat} - T_{cl,\text{in}}} \right)} \right)
\]

Thus

\[
T_{cl,\text{out}} = T_{sat} - \frac{(T_{sat} - T_{cl,\text{in}})}{\exp \left( \frac{U A_{\text{sur}}}{m_{cl} c_{pcl}} \right)}
\]

(3. 47)

4. Condensation mass flow rate can be calculated using equation (4. 47) and the heat balance analysis.

\[
m_{cs} = \frac{U A_{\text{sur}} (T_{sat} - T_b)}{h_{fg}}
\]

(3. 48)

3.5 Exergy Analysis of Steam Power Plant Condensers

Literatures define exergy as the property that quantifies potential for use, or the energy that is available to be used [4]. More precisely, exergy is “the maximum theoretical work obtainable from an overall system consisting of a system and the environment as the system comes into the dead state-equilibrium with the environment”[4]. Exergy, in contact to energy, is destroyed by irreversibilities such as friction and heat transfer, and also it is transferred to and from the systems [4]. Exergy destruction within the systems represented losses by the irreversibilities; therefore when the exergy destruction is reduced in the systems, the use of energy will be increased. The main purpose of exergy analysis is to identify the locations where the most exergy destructions occur and rank order them for significance [4]. By doing this analysis, it will be easier to find a
suitable solution to reduce the losses and improve the efficiency of the systems. In this part, exergy analysis will be carried out to evaluate the condenser performance.

### 3.5.1 Governing Equations

For steady state flow, the balances of mass, entropy, and exergy within the control volume shown in figure 3.11 are as follows [77]

#### I. Mass balance:

\[
\sum_j m_{\text{in}}^j = \sum_j m_{\text{out}}^j \quad (3.49)
\]

![Figure 3.9 Exergy and energy balance](image)

#### II. Entropy balance:

\[
\sum_j \frac{\dot{Q}_j}{T_j} + \sum_{\text{in}} \dot{m}_{\text{in}} s_{\text{in}} - \sum_{\text{out}} \dot{m}_{\text{out}} s_{\text{out}} + \dot{\sigma}_{\text{cv}} = 0 \quad (3.50)
\]
where

\[ \sum_j \frac{\dot{Q}_j}{T_j} \]: Entropy transferred by heat

\[ T_j \]: Temperature on the boundary where the heat transfer occurs

\[ \sum \dot{m} s \]: The rate of entropy transfer due to mass flow

**III. Exergy balance:**

\[ \sum_j \left( 1 - \frac{T_0}{T_j} \right) \dot{Q}_j - \dot{W}_{c,v} + \sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} - E_d = 0 \]  

(3.51)

The equation (3.51) can be written

\[ \sum_j \dot{E}_{q_j} - \dot{W}_{c,v} + \dot{E}_{in} - \dot{E}_{out} - \dot{E}_d = 0 \]  

(3.52)

Where:

\[ \dot{E}_{q_j} = \left( 1 - \frac{T_0}{T_j} \right) \dot{Q}_j \]: The rate of exergy transfer accompanying heat.

\[ \dot{E}_{in} = \sum_{in} \dot{m}_{in} e_{in} \]  

\[ \dot{E}_{out} = \sum_{out} \dot{m}_{out} e_{out} \]  

Specific flow of the exergy (J)

where

\[ e = h - h_o - T_o(s - s_o) + \frac{V^2}{2} + g Z \]
(h) and (s) are specific enthalpy and entropy evaluated at the inlets and exits, \((h_o)\) and \((s_o)\) are specific enthalpy and entropy evaluated at dead state conditions \((T_o)\) and \((P_o)\), \(\left(\frac{v^2}{2}\right)\) is the kinetic energy, and \((g Z)\) is the potential energy.

\[E'_d = T_o \dot{\sigma}_{c,v}\] : Destruction of exergy within the control volume due to irreversibility.

**Notice:**

In our analysis of the power plant condenser, the following assumptions are made, which are reasonable assumptions for a thermal system like condensers:

1- Steady state flow.

2- No heat transfer to the surrounding that implies \((\dot{Q}_j = 0)\).

3- No work done within the control volume that means \((\dot{W}_{c,v} = 0)\).

4- The effects of motion and gravity are negligible \((\frac{v^2}{2} = g Z = 0)\).

5- Dead-state conditions \((T_o = 0.01 K\) and \(P_o = 1 bar)\).

Applying the above assumptions to equations\((3.50),(3.51)\), we get

\[
\sum_{in} \dot{m}_{in} s_{in} - \sum_{out} \dot{m}_{out} s_{out} + \dot{\sigma}_{cv} = 0 \quad (3,53)
\]

\[
\sum_{in} \dot{m}_{in} e_{in} - \sum_{out} \dot{m}_{out} e_{out} - \dot{E}_d = 0 \quad (3,54)
\]

\[e = h - h_o - T_o(s - s_o)\]

**3.5.2 Energetic and Exergy Efficiency for the Power Plant Condenser**

Energetic efficiency is the parameter that assesses the energy utilization in the system [4]. The efficiency takes many different forms depend on the type of the thermal system which is applied for. For the surface contact heat exchanger like the condenser, we can develop an expression for
the energetic efficiency as follows [4]. Applying exergy balance to figure 3.11 and using equation (4.54), we get

\[
(m_{cl} e_{cl, in} - m_s e_{s, in}) - (m_{cl} e_{cl, out} - m_s e_{s, out}) - \dot{E}_d = 0
\]

\[
\dot{E}_{in} = (m_{cl} e_{cl, in} + m_s e_{s, in})
\]

\[
\dot{E}_{out} = (m_{cl} e_{cl, out} + m_s e_{s, out})
\]

\[
\dot{E}_d = \dot{E}_{in} - \dot{E}_{out}
\]

Rearranging the first equation

\[
m_{cl}(e_{cl, in} - e_{cl, out}) + m_s(e_{s, in} - e_{s, out}) - \dot{E}_d = 0
\]

\[
m_{cl}(e_{cl, in} - e_{cl, out}) = m_s(e_{s, out} - e_{s, in}) + \dot{E}_d
\]

Thus, the exegetic efficiency of the condenser is

\[
\varepsilon = \frac{m_{cl}(e_{cl, in} - e_{cl, out})}{m_s(e_{s, out} - e_{s, in})}
\]

Moreover, some references defend another expression for the efficiency using the exergy analysis. They call it exergy efficiency, and it has the following form as it is represented on references [77] and [80].

\[
\eta_{ex} = 1 - \frac{\dot{E}_d}{\dot{E}_{in}}
\]
4 Experimental Studies to Verify the Concept of a Refrigerant Cooled Condenser

4.1 Introduction

The MSU Turbomachinery Lab has shown an analytical and experimental study in previous research [26, 100] where the vapor compression refrigeration (VCRS) can be a possible method to directly cool the steam condenser of a steam power plant (SPP). This result was based on 1-D fluid flow and thermal and experimental analyses. But to be capable of predicting the performance of the SPP in more realistic form, detailed fluid flow and thermodynamic analyses of the SPP need to be conducted in more depth.

In the present thesis, programmable algorithms and codes will be created based on advanced thermal and fluid flow. Based on these programs and simulation, performance of the SPP configuration will be predicted. In the previous experimental work, a 10 kW model of refrigerant-cooled-condenser was tested. The experiments carried out showed the need for a more detailed study when using a refrigeration system to cool a steam power plant condenser. To be able to find improved solutions to these problems, additional experiments will be carried out in the present work. The experiments to be performed will also help verify, supplement, and complement the results obtained through the theoretical analysis and demonstrate the feasibility of integrating the refrigeration based cycles into the condenser of the SPP.

Steam power plants (SPPs) are the major providers of electrical energy in most areas of the world. One of the main components of the SPP is the condenser. A steam power plant condenser (SPPC) is typically a shell and tube, in which the steam leaving the turbine is converted into a liquid by the transfer of latent heat to the coolant, which is in most cases water. Steam power plants are one of the largest users of water, often requiring construction near large water sources.
The reliance on natural water sources leads to a variation in power plant efficiency due to seasonal changes in ambient conditions. Moreover, both the intake of fresh water and release of exhaust hot water from the power plant can be detrimental to the survival of organisms in the aquatic environment, particularly fish and crustaceans. In addition, cooling water is the major source of wastewater generated by most thermal power plants. A 500-MW facility generates about 3800 m$^3$ per day of wastewater, with about 70% of this wastewater coming from cooling tower blow down [87]. This wastewater can alter the chemical composition and temperature of the water body into which it is released, which can affect fish and other aquatic organisms, animals, and the local habitat. Another adverse impact of re-circulating cooling is the effects of the visual plume of vapor emitted from the cooling tower. Such plumes represent a vision disturbance and in cold conditions, some tower designs allow ice to form, which may coat the ground or nearby surfaces. Another possible problem is carryover, where salt and other contaminants may be present in the water droplets.

Changes in climate have led to water shortages in several areas of the world, which have impacted electrical power production [79, 104]. Throttling of power output of numerous power plants due to water shortages in Europe was required during the summer months of the years 2003 and 2006 due to high water temperature caused by a hot and dry summer [105]. Recently, new restrictions on the use of fresh water have been proposed. The combination of environmental impact and government regulations have motivated a search for alternative cooling systems by means other than water for SPPC [43].

The design and operation of wet-cooled SPPC has received greater research attention. Several studies have concluded that the operating pressure of the condenser and inlet cooling temperature are important parameters that affect output power, power potential, and thermal and
exergy efficiency of the cycle [48, 13]. The Effect of a steam–ammonia mixture on steam condensation heat transfer in a horizontal shell and tube condenser was reported in Ref. [111]. Blending ammonia with steam was found to disturb the condensate film, which enhanced the condensation heat transfer within 14–34%.

Dry cooling, other coolants, and energy storage systems for SPPC have recently received increased attention as alternatives to wet systems. Dry-cooling systems have been used in thermal power plants in many sites, even though electric power from dry-cooled power plants currently costs 10–15% more than power from wet-cooled plants [103]. Dry cooling has been the subject of some research, which has focused on improved heat exchanger geometries for finned tube bundles in air-cooled condensers [108, 109], enhancement of the performance of air-cooled condensers with the use of limited water [22, 110], the use of an evaporative condenser [21], and using double wet and dry condensers where the heat from the wet condenser is dissipated into a cold-storage container [95]. Some of the investigators of this proposal performed a theoretical study, which determined that R-134a is a superior coolant to water in SPPC [26]. They then made a comparative study between using R-410A, R-407C, R-22, and R-134a as cooling mediums to select the best efficient refrigerants for SPPC [24]. Based on these two analytical studies, R-410A was determined to be the best refrigerant for cooling SPPC.

The main goals of this present work are to study in more depth the feasibility of using VCRS for cooling SPPC. There are three options to study this goal. Integration of the refrigerant system (RS) into SPP can be brought about by using the cold refrigerant liquid of the RS to directly cool the steam in the SPPC (Figure 5.1). This configuration of combined RS and SPP requires replacement of the existing SPPCs with RS. This is very costly and can be avoided while improving the existing SPPCs performance through implementing one of the of two other
configurations shown in Figures 5.2 and 5.3. In these configurations, the temperatures of the cooling air of dry cooled and cooling water of wet cooled SPPCs are lowered by using VCRS before admitted to the SPPC. It is to be mentioned here that the RSs shown in Figs 5.3 are all VCRSs.

Figure 4.1 Configuration I. steam condenser cooled directly by a vapor compression cycle

Water/steam  Refrigerant
Figure 4.2 Configuration II. steam condenser cooled with air and vapor compression cycle
____ Water/steam
____ Refrigerant

Figure 4.3 Configuration III. steam condenser cooled with water and vapor compression cycle
____ Water/steam
____ Refrigerant
The undesirable environmental impacts of using surface and ground water for cooling the steam power plants encourage researchers around the world to find cooling method for these plants. Also, the steam power plants that use water for cooling their condensers must be built around the water bodies. In addition to the previous reasons, many countries found restrictions on using water as a coolant for steam power plant. Therefore, some technologies have been found for cooling the steam power plants. The present work will show the experimental test rig construction and the effect of using refrigerant R-410A as a coolant in steam power plant using the vapor compression refrigeration system. In this experiment, the refrigerant will not be used to cool the condenser directly, the R-410A will be used to cooling the water in an intermediate heat exchanger for two reasons. The first reason is to avoid the thermal stress that can be found due to the temperature difference between the steam and R-410A. The second reason is to avoid the excessive static pressure that may be found in order to decrease the pressure difference between the steam and R-410A. For experimental the set up, there are two heat exchangers available with different properties. The first one has a design pressure of 10.2 bar for both shell and tube sides. The second available heat exchanger has a design pressure of 35 bar for the shell side, and it has a 10.2 bar as a maximum allowable pressure for the tube side. The first one will be used as a condenser in the steam loop and serve to condensate steam using chilled water or other water source. The second heat exchanger will be used as an intermediate heat exchanger since the steam pressure that enters the shell side is usually less than the working pressure of R-410A which may reach to 31 bar.

The experimental part of this work, will be conducted using a 10 kW refrigerant-cooled-steam condenser, developed in the MSU Turbomachinery Lab, shown in figure 5.4. Hence, the objectives of this study are:
1. To develop a 1-D & 2-D empirically-based theory for the thermal and fluid flow analyses, taking into consideration the operating conditions of the SPP cycle and environment

2. To model fluid flow and heat transfer into and out of cold storage system used for boosting steam power plant output during peak-load-periods

3. To create programmable algorithms and codes, established on the analyses, for predicting the refrigerant based cycle’s performance under design-level conditions and different operating conditions (off-design, day & night, seasonal, startup and shutdown), of the SPP cycle and environment; when using different refrigerants

4. To conduct experiments on the 10kW models of refrigerant-cooled-condensers, then to verify, supplement and complement the results obtained through the theoretical analysis and to demonstrate the feasibility of integrating the refrigeration based cycles into the condenser of the SPP

5. To conduct cost analyses for the refrigerant cooled condenser

6. To verify that the goals and mission of current SPP requirements are met

7. Using the knowledge gained, to be able to design and optimize a cost effective refrigerant-cooled-condenser unit for a specific SPP

8. To cooperate with power and refrigeration industries and thus to pave the way for the commercialization of the refrigerant based condenser cooling technology

4.2 Experimental System Description

An experimental model of a steam power plant is illustrated in the following figure. The system consists of three interacting main loops: steam loop, water loop, and VCRS loop. The steam loop interacts with water loop through the condenser (8). The water loop interact with the VCRS through the intermediate heat exchanger (16). In the steam loop, the steam will be generated
Figure 4.4 The experimental setup to model the refrigerant-cooled-condenser of the steam power plant
Table 4.1 The components of experimental test rig

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<th>Water loop</th>
<th>Refrigeration loop</th>
</tr>
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<td>(20) Refrigerant pressure relief valve</td>
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<td>(2) Water filter and ball valve</td>
<td>(13) Thermal expansion tank</td>
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<td></td>
<td></td>
<td>(29) Refrigerant high pressure switch</td>
</tr>
</tbody>
</table>
4.2.1 Steam Loop

The steam loop which is considered as an open loop represents the actual steam power plant. The steam loop has the following components.

4.2.1.1 Steam Generator

Steam generator (CMB- 9A, 6gal volume, 240 volt, and 3- phase) is responsible for providing 9 kW of the heat rate to generate steam at different selectable temperatures and steam pressures. Above the generator and beside the steam vent there is a pressure relief valve to prevent the pressure inside the boiler from exceeding 680 kPa.

Also, there is another valve to control the level of water inside the boiler called the float control valve. This valve works by maintaining the pressure difference between the water flows from the water source and the steam return from the boiler, 68 kPa. Moreover, T types of thermocouples and bourdon gauges are connected to the inlet and outlet of the boiler to measure the steam and water properties.

4.2.1.2 Steam Condenser

The condenser used in the steam loop contains a cylindrical steel shell with 16.7 W/m.k as a thermal conductivity, 110 mm as an inner diameter, 114 mm as an outer diameter and 737 mm as the length of shell. Around the shells, there are six copper u-tubes with 387 W/m.k as a thermal conductivity, 18.16 as inner diameter and 0.889 as a thickness of each tube. The steam that is generated in the boiler enters the condenser through the upper hole diameter 88 mm. Then, the steam passes through the shell and condensates due to the heat exchange with cooling water. The condensate drains through lower hole diameter 44 mm. A thermostatic check valve was installed at the exit of condensate to allow only the liquid to pass to the graduated tank. To find the mass flow rate of the condensate a stop watch is needed to calculate the volume of condensate in the
graduated tank at specific time. A T type thermocouple and a bourdon tube gauge are fixed near the exit of the condenser to measure the pressure and temperature of the condensate.

![Figure 4.5 Steam condenser](image)

The condenser was tested using R-22 at liquid phase and was circulated in the tube side. We find that the temperature of the refrigerant at the inlet and outlet of the condenser are 7.22 °C and 190 °C respectively, while the steam temperature at inlet was 103.6 kPa and has 170 bar as a burst pressure.
4.2.2 Water Loop

This loop is considered a closed loop and works as a link between the other two loops. It relates with the steam loop by the condenser, and it connects with the VCRS by the intermediate heat exchanger. The water loop consists of the following components.

4.2.2.1 Circulator Pump

The circulator pump’s function is to circulate and control the water flow rate in the water loop. The allowable working pressure inside the pump is 9.5 bar, while the temperature range is -10 °C to 110 °C.

Figure 4.6 Water circulator pump
4.2.2.2 Water Expansion Tank

The water expansion tank with diaphragm is fixed near the water inlet valve, and it helps to maintain the water working pressure at the required value. Also, the diaphragm can absorb the expanded water during the operation. The maximum working temperature and pressure are 115 °C and 6.8 bar.

Figure 4.7 Expansion tank with pressure valve

4.2.2.3 Water-Pressure Reducing Valve

The water-pressure reducing valve can control the pressure inside the water loop by filling the required amount of water in the piping system. The working pressure in the valve ranges between 68 and 170 kPa.
4.2.2.4 Water-Pressure Relief Valve

The water-pressure relief valve is considered a safety valve in the water loop. It is usually closed, but it opens when the pressure exceeds 204 kPa.

4.2.2.5 Intermediate Heat Exchanger (IHX)

The intermediate heat exchanger (200A, shell and tube type) consists of straight tubes with epoxy-coated sheets, carbon steel shell, and heavy wall. The working pressure on the tube side is 10.2 bar and 30.6 bar on the shell side. The cooling capacity on the heat exchanger is 11.43 kW for R-22 and 11 kW for R-410A. Also, IHX has a pressure relief and works as a safety for the water loop. The operating temperature on the valve ranges between 29 °C and 93 °C. The intermediate heat exchanger task is releasing heat into the refrigerant loop.

Figure 4.8 Intermediate heat exchanger
4.2.3 Vapor Compression Refrigeration System Loop

The VCRS is a closed loop and considered the most important loop in the unit. The function of this loop is to dissipate heat from the intermediate heat exchanger to the atmosphere. The VCRS loop consists of the following components.

4.2.3.1 AC Condenser System

The air-cooled condenser system (3.0 Ton 14.5 SEER, 208/230 V, and single phase) dimensions are 91.44 cm for height and 73.66 cm width and depth. The vapor line size in the system is 19 mm, and the liquid line size is 9.5 mm. The system was charged with R-410A, and the maximum working pressure on ACCS is 34 bar. Above the air-cooled condenser there is a fan that helps in releasing heat from R-410A to the atmosphere by circulating air surrounding the condenser surface.

Figure 4.9 AC condenser unit
4.2.3.2 Pressure Regulator Valve

The outlet pressure regulator with pilot operated (A9 series range A) is made of cooper. Its fluid temperature range is -45 to 95 °C, while the working pressure range is between 34 and 816.33 kPa. The valve is installed in the vapor line outlet to control the pressure of steam.

4.2.3.3 Thermal Expansion Valve

The maximum working pressure in the valve is 45.5 bar, while the maximum fluid temperature is 100 °C. The expansion valve is put in the liquid line to do many tasks. The valve can control the R-410A flow rate and the superheating at the intermediate heat exchanger. Also, it controls the system’s cooling load.

Figure 4.10 Thermal expansion valve
4.2.3.4 Suction Line Accumulator

The accumulator consists of steel shell material and solid copper connections. The chosen accumulator has a height of 228 mm, a diameter of 127 mm, and a volume of 2.41 liter. The temperature range on the accumulator is -29 to 47 °C, and the maximum working pressure is 20.68 bar. The suction line accumulator is used to allow only the gas to pass into air cooled condenser system.

Figure 4.11 R-410A accumulator

4.2.3.5 Vapor and Liquid Lines Filter Drier

The vapor and liquid filter drier is installed close to the air-cooled condenser on the vapor and liquid line to drive out contamination and moisture from the refrigerant loop. The working
pressure for vapor filter drier is 34.5 bar and 49.6 bar for liquid filter drier, but the both filter
drier.

Figure 4.12 Filter driers and pressure regulator

4.2.4 Electrical Switches and Control

The experimental test rig contains four main electrical switches to control the operation and
safety inside the three loops. The main switches will be explained in details in the following
sections.
4.2.4.1 The High R-410A Pressure Switch

The high R-410A pressure switch is considered the primary safety switch for the vapor compression refrigeration side loop and the shell side in IHX. The pressure range on this pressure switch ranges between 3.4 bar and 34 bar.
4.2.4.2 The Low R-410A Pressure Switch

The low R-410A pressure switch is another safety switch that adjusts the pressure in the refrigerant loop to the required minimum pressure and hence can control its temperature. The pressure range for the low pressure switch is 3.4 bar to 34 bar.

4.2.4.3 The High Water Pressure Switch

The high pressure switch is used to control the water working pressure inside the water loop. The pressure switch consists of a valve with a spring return and valve actuator. The valve temperature range is 0 to 93 °C and the flow coefficient is 3.5. The valve working pressure range is 67.8 to 680 kPa.

4.2.4.4 The Low Water Temperature Switch

This switch is responsible for adjusting the water temperature at the intermediate heat exchanger outlet. Therefore, the switch is used to prevent freezing inside the IHX and can provide more safety for the tubes. The temperatures can range between -34 and 32 °C.

4.2.5 Measurement Devices

In this section different measurement devices that used to measure temperature, pressure, and flow rate in the three loops will be briefly explained.

4.2.5.1 Thermocouple

The thermocouples (T type and 316 SS) are used to measure the temperatures inside the experimental test rig loop. Thermocouples have probes made of stainless steel with a length of 17.78 cm and a diameter of 1.5 mm. They are able to measure temperature ranges between -185 and 370. All thermocouples are connected to a digital thermometer to do the digital reading.

4.2.5.2 Bourdon Tube Gauge

The pressure inside the loops are measured using the bourdon tubes’ gauges. These tubes’
gauges have different ranges according to their locations. The bourdon tube gauges in the liquid line have a higher range than these found in the vapor lines.

**4.2.5.3 Rotameter**

Rotameters are the devices that are used to measure the water flow rate and prevent the water flowing back to the water loop. The rotameter can measure up to 14.7 L/min. The maximum pressure on the rotameter is 13.8 bar, while the maximum temperature is 80.

**4.3 System Operational Challenges**

Many challenges may be met when using the vapor compression refrigeration system for cooling a steam power plant. In this section, the main challenges met during design, construction, and operation of the experiment will be clearly addressed and described.

During the system operation, the experimental test rig was exposed to a large number of vibrations due to the vapor compression refrigeration system. Therefore, it became necessary to place the vapor compression system on a separate cart instead of putting the whole system on one cart. Then, the two carts were connected through a vibration absorber.

Large thermal stress that can be found in the condenser tube wall is one challenge we met at startup. As the cold refrigerant passes directly to the steam power plant condenser, the temperature difference between the condensing steam and the cooling refrigerant increases. Hence, the thermal stress inside the condenser tube will increase, and it may lead to damage to the condenser material. To overcome this problem, an adjustable pressure ratio throttle valve were fitted in the vapor compression refrigeration system loop to control the refrigerant temperature before it left to the condenser.

The last critical challenge that we met is filling the vapor compression refrigeration system with R-410A and that is because there is no standard for this unique system. Several tests
were gotten during charging the unit with the refrigerator to fulfil the optimal charge which correspond with the working condition ranges.
5 Performance Evaluating of Steam Power Plant Condenser Cooled by a Vapor Compression Refrigeration System Using Aspen-HYSYS

5.1 Introduction

A steady-state simulation model of a steam power plant condenser cooled by a vapor compression refrigeration system is developed and validated using Aspen. First, an appropriate thermodynamic model should be selected to fit with the working fluid properties. Since the simulation of the steam and refrigerant cycle can be affected by the chosen thermodynamic properties method, it is essential to choose carefully a proper model to estimate the properties of the steam and refrigerants.

In this chapter, a steady-state simulation model of the engine is then developed. Components model of the steam power plant cooled with vapor compression refrigeration unit so local thermodynamic balances (energy, entropy and mass balance) are respected. Also, thermodynamic balances were used to determine the inlet and outlet states of each component.

The recent work the simulation model developed selecting the same conditions (temperatures, pressures, and duties) that used in the experimental part. The result was taken at different condenser pressures and different mass flow rates. The simulation results are found to be in perfect agreement with the experimental results.

After the simulation showed good concordance with the experimental results, another additional numerical studies for the same model with other refrigerants were used. The best model was then be chosen according to the coefficient of performance.
5.2 Aspen Model Description

A proper selection of the equivalent blocks for the experimental main components is very important in modeling the process in Aspen to allow a running model. The model consists of two cycles: steam cycle and refrigeration cycle. These two cycles are connected with an intermediate heat exchanger (IHX). The steam unit contains four components: boiler, condenser, intermediate heat exchanger, and pump. The boiler receives water and provides 9 kw to generate steam. Then, the steam condensates in the steam condenser with 8.5 kw as a heat load of this unit, water is used as a coolant for the steam that is generated in the boiler. The pump is used to pump more water to the condenser. The default values of the pump are assumed to be 100% since this parameter does not affect the overall cycle efficiency.

![Figure 5.1 Aspen-Hysys model of indirect cooling](image_url)
The compression refrigeration system also contains four components: compressor, expansion valve, evaporator, and air condenser. First, the water used for cooling in the steam cycle is cooled by refrigerant (R-410A, NH$_3$, R-134A) in the intermediate heat exchanger that works as an evaporator. Then, the refrigerant is cooled by using an air condenser. The previous model was then modified to cool the condenser directly by the refrigerant as shown in the Figure 5.2.
Table 5.1 Machine elements and their Aspen models with input data

<table>
<thead>
<tr>
<th>Machine Element</th>
<th>Aspen Block</th>
<th>Input Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam generator</td>
<td>Boiler/Thermal and phase</td>
<td>Water inlet temperature = 23 °C</td>
</tr>
<tr>
<td></td>
<td>change</td>
<td>Water mass flow rate = 3.5 g/s</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler duty = 9 kw</td>
</tr>
<tr>
<td>Condenser</td>
<td>Two-flow heat exchanger</td>
<td>Water inlet temperature (cold stream) = 9 to 32 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>water mass flow rate = 0.056, 0.11, 0.13 kg/s</td>
</tr>
<tr>
<td></td>
<td></td>
<td>condensate outlet = 83.5 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Condenser duty = 8.489 kw</td>
</tr>
<tr>
<td>Evaporator (IHX)</td>
<td>Two-flow heat exchanger</td>
<td>Refrigerant inlet temperature = -7.5 to -3.5 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Refrigerant inlet pressure = 650 kPa to 750 kPa</td>
</tr>
<tr>
<td>Pump</td>
<td>Pump/Pressure changer</td>
<td>Isentropic efficiency = 1</td>
</tr>
<tr>
<td>Compressor</td>
<td>Compressor/Pressure changer</td>
<td>Compressor duty = 4.5 kw</td>
</tr>
<tr>
<td>Expansions Valve</td>
<td>Control valve and pressure exchanger</td>
<td>Working pressure = 5.5 bar</td>
</tr>
<tr>
<td>AC condenser</td>
<td>Two-flow heat exchanger</td>
<td>Air inlet mass flow rate = 424.1 g/s</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Condenser duty = 13 kw</td>
</tr>
</tbody>
</table>

5.3 Result and Discussion

This section will show the results of Aspen simulation for a steam power plant condenser that is cooled by a compression refrigeration system.
(a) Experimental Results vs. Aspen model results

Figure 5.3 Experimental COP results versus the Aspen model COP results at $m_{ci}=0.056$ kg/s
Figure 5.4 Experimental COP results versus the Aspen model COP results at $m_{cl} = 0.11 \text{ kg/s}$
Figure 5.5 Experimental COP result versus the Aspen model COP results at $m_{cl}=0.13 \text{ kg/s}$
Figure 5.6 Experimental $m_{co}$ results versus the Aspen model $m_{co}$ results at $m_{ct}=0.056$ kg/s
Figure 5.7 Experimental $m_{co}$ results versus the Aspen model $m_{co}$ results at $m_{ci}=0.11$ kg/s
Figure 5.8 Experimental $m_{co}$ results versus the Aspen model $m_{co}$ results at $m_{cl}=0.13$ kg/s
The Figure 5.3 shows the experimental COP results versus the Aspen model COP results as $m_{c1}=0.056$ kg/s at three different pressures. As regards the $P_c=101.325$ kPa, the maximum deviation is observed to be less than 1.4 %. However, the maximum deviation increased to 4.7% as the $P_c$ decreased to 84.382 kPa. The maximum deviation return to decreases to 2.6% at the least value of condenser pressure. From Figure 5.5 is shown that the difference between experimental results and the Aspen model results do surpass 5% and that happens at $P_c=70.825$ kPa. Finally, as the coolant mass flow rate increased to 0.13 kg/s, the maximum deviation in COP is 4.8% at $P_c=101.325$.

In Figures 5.6 through 5.8 the condensate mass flow rates of the experimental system are plotted versus the condensate mass flow rates in aspen model at three different coolant mass flow rates and condenser pressures. Figure 5.9 depicts the results at $m_{c1}=0.056$ kg/s and three different pressures. The maximum difference between experimental result and the Aspen result does not exceed 2%. Figure 5.6 shows that the deviation in the compression results are very small. Lastly, we can see that the maximum deviation happened at the maximum coolant mass flow rate is 1.3%.

(b) Model prediction of using various refrigerants for cooling the SPPC

In this second step, the Aspen- HYSYS model for the experimental is then modified to compare five different refrigerants. R-410A, R-404A, R-407C, R-134a, and NH$_3$ are used in the Aspen model to choose the most efficient that can produce the highest COP and condensate mass flow rate value. The compression between the five refrigerants is done at three different condenser pressures and coolant mass flow rates. The performance of the refrigerant cycle is quantified by using the coefficient of performance, so as the COP increases, the performance of cycle improves. From previous figures we can find that NH$_3$
provide the highest COP at all different condenser pressures and coolant mass flow rates. The higher the condensation rate, the better the condenser performance is. For all cases, \( \text{NH}_3 \) provides the highest condensation rates compared with the other four refrigerants.

![Figure 5.9a Coefficient of performance of SPP with different refrigerants at \( m_{cl}=0.056 \) kg/s and \( P_c = 101.325 \) kPa](image)

Figure 5.9a Coefficient of performance of SPP with different refrigerants at \( m_{cl}=0.056 \) kg/s and \( P_c = 101.325 \) kPa
Figure 5.9b Coefficient of performance of SPP with different refrigerants at $m_{cl}=0.11 \text{ kg/s}$ and $P_c = 101.325 \text{ kPa}$

Figure 5.9c Coefficient of performance of SPP with different refrigerants at $m_{ct}=0.131 \text{ kg/s}$ and $P_c = 101.325 \text{ kPa}$
Figure 5.10a Coefficient of performance of SPP with different refrigerants at $m_{cl} = 0.056$ kg/s and $P_c = 84.385$ kPa

Figure 5.10b Coefficient of performance of SPP with different refrigerants at $m_{cl} = 0.11$ kg/s and $P_c = 84.385$ kPa
Figure 5.10c Coefficient of performance of SPP with different refrigerants at $m_{cl}=0.13 \text{ kg/s}$ and $P_c = 84.385 \text{ kPa}$

Figure 5.11a Coefficient of performance of SPP with different refrigerants at $m_{cl}=0.056 \text{ kg/s}$ and $P_c = 70.825 \text{ kPa}$
Figure 5.11b Coefficient of performance of SPP with different refrigerants at $m_{cl}=0.11$ kg/s and $P_c=70.825$ kPa

Figure 5.11c Coefficient of performance of SPP with different refrigerants at $m_{cl}=0.13$ kg/s and $P_c=70.825$ kPa
Figure 5.12a Condensation rate in SPP with different refrigerants at $m_{cl}=0.05$ kg/s and $P_c = 101.325$ kPa

Figure 5.12b Condensation rate in SPP with different refrigerants at $m_{cl}=0.11$ kg/s and $P_c = 101.325$ kPa
Figure 5.12c Condensation rate in SPP with different refrigerants at $m_{cl} = 0.13$ kg/s and $P_c = 101.325$ kPa

Figure 5.13a Condensation rate in SPP with different refrigerants at $m_{cl} = 0.056$ kg/s and $P_c = 84.382$ kPa
Figure 5.13b Condensation rate in SPP with different refrigerants at $m_{ci}=0.11 \text{ kg/s}$ and $P_c = 84.382 \text{ kPa}$

Figure 5.13c Condensation rate in SPP with different refrigerants at $m_{ci}=0.13 \text{ kg/s}$ and $P_c = 84.382 \text{ kPa}$
Figure 5.14a Condensation rate in SPP with different refrigerants at $m_{cl}=0.056$ kg/s and $P_c=70.825$ kPa

Figure 5.14b Condensation rate in SPP with different refrigerants at $m_{cl}=0.11$ kg/s and $P_c=70.825$ kPa
Figure 5.14c Condensation rate in SPP with different refrigerants at $m_{cl}=0.13 \text{ kg/s}$ and $P_c=70.825 \text{ kPa}$

(C) Model prediction of comparison between direct cooling and indirect cooling

This section presents a comparative numerical study of using water and refrigerant to cool the SPPC (indirect cooling) and using only the refrigerant (direct cooling) for cooling the condenser.
Figure 15a Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.056$ kg/s and $P_c = 101.325$ kPa

Figure 5.15b Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.056$ kg/s and $P_c = 84.382$ kPa
Figure 5.15c Direct cooling COP results versus the indirect cooling COP results at $m_{ref} = 0.056$ kg/s and $P_c = 70.825$ kPa

Figure 5.16a Direct cooling COP results versus the indirect cooling COP results at $m_{ref} = 0.11$ kg/s and $P_c = 101.325$ kPa
Figure 5.16b Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.11$ kg/s and $P_c=84.382$ kPa

Figure 5.16c Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.11$ kg/s and $P_c=70.825$ kPa
Figure 5.17a Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.131$ kg/s and $P_c = 101.325$ kPa

Figure 5.17b Direct cooling COP results versus the indirect cooling COP results at $m_{ref}=0.131$ kg/s and $P_c = 84.382$ kPa
Figure 5.17c Direct cooling COP results versus the indirect cooling COP results at \( m_{\text{ref}} = 0.131 \) kg/s and \( P_c = 70.825 \) kPa

Figure 5.18a Condensation rate in direct cooling versus the condensation rate in indirect cooling at \( m_{\text{ref}} = 0.056 \) kg/s and \( P_c = 101.325 \) kPa
Figure 5.18b Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.056$ kg/s and $P_c = 84.382$ kPa

Figure 5.18c Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.056$ kg/s and $P_c = 70.825$ kPa
Figure 5.19a Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref} = 0.11$ kg/s and $P_c = 101.325$ kPa

Figure 5.19b Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref} = 0.11$ kg/s and $P_c = 84.382$ kPa
Figure 5.19c Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.11$ kg/s and $P_c = 70.825$ kPa

Figure 5.20a Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.131$ kg/s and $P_c = 101.325$ kPa
Figure 5.20b Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.131$ kg/s and $P_c = 84.382$ kPa.

Figure 5.20c Condensation rate in direct cooling versus the condensation rate in indirect cooling at $m_{ref}=0.131$ kg/s and $P_c = 70.825$ kPa.
Figure 5.21a Direct cooling COP results for different refrigerants at $P_c = 101.325$ kPa and $m_{ref} = 0.005$ kg/s

Figure 5.21b Direct cooling COP results for different refrigerants at $P_c = 101.325$ kPa and $m_{ref} = 0.01$ kg/s
Figure 5.22a Direct cooling COP results for different refrigerants at $P_c = 84.382$ kPa and $m_{ref} = 0.005$ kg/s.

Figure 5.22b Direct cooling COP results for different refrigerants at $P_c = 84.382$ kPa and $m_{ref} = 0.01$ kg/s.
Figure 5.23a Direct cooling COP results for different refrigerants at $P_c = 70.825$ kPa and $m_{ref}=0.005$ kg/s

Figure 5.23b Direct cooling COP results for different refrigerants at $P_c = 70.825$ kPa and $m_{ref}=0.01$ kg/s
Figure 5.24a Condensation rate in direct cooling for different refrigerant at $P_c = 101.325$ kPa and $m_{ref} = 0.005$ kg/s

Figure 5.24b Condensation rate in direct cooling for different refrigerant at $P_c = 101.325$ kPa and $m_{ref} = 0.01$ kg/s
Figure 5.25a Condensation rate in direct cooling for different refrigerant at $P_c = 84.382$ kPa and $m_{ref} = 0.005$ kg/s

Figure 5.25b Condensation rate in direct cooling for different refrigerant at $P_c = 84.382$ kPa and $m_{ref} = 0.01$ kg/s
Figure 5.26a Condensation rate in direct cooling for different refrigerant at $P_c = 70.825$ kPa and $m_{ref} = 0.005$ kg/s

Figure 5.26b Condensation rate in direct cooling for different refrigerant at $P_c = 70.825$ kPa and $m_{ref} = 0.01$ kg/s
Table 5.2 Percentage variation of condensation rate and COP with different pressures

<table>
<thead>
<tr>
<th>Coolant Name</th>
<th>Coolant Properties</th>
<th>Indirect cooling</th>
<th>Direct cooling</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Coolant inlet temp. (°C)</td>
<td>Condenser pressure (kPa)</td>
<td>COP%</td>
</tr>
<tr>
<td>R-134a</td>
<td>101.325</td>
<td>0.43</td>
<td>0.24</td>
</tr>
<tr>
<td></td>
<td>84.382</td>
<td>0.45</td>
<td>0.28</td>
</tr>
<tr>
<td></td>
<td>70.825</td>
<td>0.46</td>
<td>0.32</td>
</tr>
<tr>
<td></td>
<td>-11</td>
<td>101.325</td>
<td>0.11</td>
</tr>
<tr>
<td></td>
<td></td>
<td>84.382</td>
<td>0.13</td>
</tr>
<tr>
<td></td>
<td></td>
<td>70.825</td>
<td>0.14</td>
</tr>
<tr>
<td>R-410A</td>
<td>101.325</td>
<td>0.49</td>
<td>0.27</td>
</tr>
<tr>
<td></td>
<td>84.382</td>
<td>0.51</td>
<td>0.34</td>
</tr>
<tr>
<td></td>
<td>70.825</td>
<td>0.52</td>
<td>0.36</td>
</tr>
<tr>
<td></td>
<td>-11</td>
<td>101.325</td>
<td>0.17</td>
</tr>
<tr>
<td></td>
<td></td>
<td>84.382</td>
<td>0.18</td>
</tr>
<tr>
<td></td>
<td></td>
<td>70.825</td>
<td>0.19</td>
</tr>
<tr>
<td>NH₃</td>
<td>101.325</td>
<td>0.61</td>
<td>0.42</td>
</tr>
<tr>
<td></td>
<td>84.382</td>
<td>0.65</td>
<td>0.47</td>
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<td></td>
<td>70.825</td>
<td>0.67</td>
<td>0.49</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>84.382</td>
<td>0.24</td>
</tr>
<tr>
<td></td>
<td></td>
<td>70.825</td>
<td>0.25</td>
</tr>
</tbody>
</table>

In direct cooling, the maximum ammonia mass flow rate can be used for cooling could not exceed 10 g/s. However the lowest value of cooling water in the direct cooling was 56 g/s, so it is not logical to compare indirect and direct cooling by using ammonia. In figures 21a -26b a compressions between three refrigerant were indicated using the direct cooling. These
compression were done at two different coolant mass flow rates (5 g/s and 10 g/s) and the same
direct cooling condenser pressures to show best refrigerant that can provide the highest COP and condensate mass flow rates. The previous figures showed using ammonia at the same condition in compression with R-410A and R-134a improves the COP in VCRS and increases the condensate mass flow rate.
In the current study, a numerical study has been developed for possible use of vapor compression refrigeration system to cool steam power plant condenser. First, the comparison showed good agreement between the Aspen- HYSYS model results and the experimental results. Therefore, it is concluded that the proposed Aspen-HYSYS model can be very useful tool for predicting the condensation rates and the coefficients performance of the steam power plant condenser cooled directly by VCRS.
6 Theoretical Study of Using a Vapor Compression Refrigeration System for Cooling the Condenser of a Steam Power Plant

6.1 Introduction
In this chapter, the analytical study of the thermodynamic of using R-410A as a coolant in a steam power plant condenser. As known, the refrigerant can generate higher thermal loads due to its high working pressure and low temperature. In addition, the refrigerant has a higher heat transfer rate than air and water. Engineering Equation Solver and MATLAB with the axillary function is used to analyze the thermodynamic properties of the combined systems.

6.2 The Steam Power Plant System (Reference System)
The reference steam power plant flows an idealized thermodynamic cycle is called a Rankine cycle. The reference steam power plant consists of turbine, steam generator (boiler), condenser, and feed water pump. In addition, high temperature and low temperature reservoirs are needed to complete the steam power plant cycle. The figure shows the arrangement of steam power plant components.

In the boiler high pressure steam is generated, and then the steam is expanded through the turbine at low pressure. Work will be produced by the turbine then will be converted to electricity and this is the function of the generator. The exhaust steam leaving the turbine will be condensed in the condenser (water- air condenser). The pump will move the condensate fluid again to the steam generator.
Figure 6.1 a schematic of a simple steam power plant with air-cooled condenser

a  air cooled steam condenser
b  steam turbine
c  steam boiler
d  pump

Figure 6.2 T-s diagram of the reference steam power plant
6.3 The Combined System of SPPS and VCRS (Studied System)

As we described the reference steam power plant in the previous section, here a new suggestion will be added to the new combined system. In the new system, the air-cooled system is replaced with a refrigerant-cooled condenser. The function of the vapor compression refrigeration system is to reject the heat to the surrounding atmosphere. The following figure shows the schematic of the new system.

Figure 6.3 a schematic of proposed integrated system

Water/steam          Refrigerant
<table>
<thead>
<tr>
<th></th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>steam condenser/refrigerant evaporator</td>
</tr>
<tr>
<td>b</td>
<td>feed water pump</td>
</tr>
<tr>
<td>c</td>
<td>refrigerant compressor I</td>
</tr>
<tr>
<td>d</td>
<td>refrigerant compressor II</td>
</tr>
<tr>
<td>e</td>
<td>water-cooled refrigerant condenser</td>
</tr>
<tr>
<td>f</td>
<td>air-cooled refrigerant con</td>
</tr>
<tr>
<td>g</td>
<td>throttle valve I</td>
</tr>
<tr>
<td>h</td>
<td>flash chamber</td>
</tr>
<tr>
<td>i</td>
<td>liquid suction HX</td>
</tr>
<tr>
<td>j</td>
<td>throttle valve II</td>
</tr>
<tr>
<td>k</td>
<td>steam boiler</td>
</tr>
<tr>
<td>l</td>
<td>steam turbine</td>
</tr>
</tbody>
</table>

Figure 6.4 T-s diagram of combined system
In the vapor compression refrigeration cycle, when the low pressure refrigerant vapor leaves, the liquid line heat exchanger is directly compressed to intermediate pressure throw the first compressor them feeding to the flash chamber where the two phases of the refrigerant will separate.

Then, some of compressed vapor leaving the first stage of compression goes through the second compressor, and then passes to the refrigerant compressor. The section line heat exchanger here works to dissipate the heat from the high pressure refrigerant that comes from the chamber. The two refrigerant condensers are responsible for removing heat that is absorbed by the heat of compression and the condensing steam. For the condensed refrigerant liquid and low pressure liquid refrigerant liquid, the first one passes through the expansion valve, and the other one crosses the condenser and evaporator tubes. The low pressure refrigerant absorbs the heat and becomes vapor. As this vapor and condensate water leave the cycle is completed.
6.4 Energy Analysis of the Reference System and the Studied System

Based in the thermodynamic analysis, the performance of cooling the steam power plant condenser with the vapor compression refrigeration system and the performance of cooling of the reference system at operating condition will be presented in this section. The two following figures show the T-s diagrams for the reference system corresponding to the Rankine cycle, also the diagram of studied system. In addition, there is a p-h diagram illustrating the vapor compression refrigeration system of the studied system.

Many of the assumption are taken during the analysis study of these two systems. we assume the ΔT_{sc,rs} is higher than air cooling at the inlet of the condenser. Then, we can calculate the specific load of boiler q_{b,rs}, specific work of turbine w_{t,rs}, specific work of the pump w_{p,rs}, and the specific load of condenser q_{sc,rs} from the following equation:

\[ q_{b,rs} = h_1 - h_4 \]  \hspace{1cm} (6.1)
\[ q_{c,rs} = h_2 - h_3 \]  \hspace{1cm} (6.2)
\[ w_{t,rs} = \frac{(h_1 - h_2)}{\eta_{m,t,rs}} \]  \hspace{1cm} (6.3)
\[ w_{p,rs} = \frac{(h_1 - h_4)}{\eta_{m,p,rs}} \]  \hspace{1cm} (6.4)

Also, the thermal efficiency of the reference system can be calculated by

\[ \eta_{th,rs} = \frac{[(h_1 - h_2) - (h_1 - h_4)/\eta_{m,p,rs}]}{(h_1 - h_4)} \]  \hspace{1cm} (6.5)

On the other hand, for the studied system we assume that ΔT_{ac} is higher than the ambient temperature. The characteristic parameters that describe the performance of the studied system can then be calculated using the following equations:

\[ q_{b,ss} = h_{15} - h_{14} \]  \hspace{1cm} (6.6)
\[ q_{css} = h_{16} - h_{12} \quad (6.7) \]

\[ w_{t,ss} = (h_{13} - h_{12})/\eta_{m,t,ss} \quad (6.8) \]

\[ w_{p,ss} = (h_{13} - h_{12})/\eta_{m,p,ss} \quad (6.9) \]

\[ \eta_{th,ss} = \left[ \frac{(h_{15} - h_{16})}{\eta_{m,t,ss}} - \frac{(h_{13} - h_{12})}{\eta_{m,p,ss}} \right] / \left[ (h_1 - h_4) + w_{co,l} + w_{co,II} \right] \quad (6.10) \]

Here are some characteristic parameters that are used to calculate the refrigeration cycle:

\[ m_r = \frac{(h_{16} - h_{12})}{(h_{11} - h_{10})} \quad (6.11) \]

\[ q_{rc,wc} = \frac{m_r}{m_s} (h_5 - h_4) \quad (6.12) \]

\[ q_{rc,ac} = \frac{m_r}{m_s} (h_5 - h_6) \quad (6.13) \]

\[ w_{co,l} = \frac{m_r}{m_s \eta_{m,co,l}} (h_2 - h_1) \quad (6.14) \]

\[ w_{co,II} = \frac{m_r}{m_s \eta_{m,co,II}} (h_4 - h_3) \quad (6.15) \]

\[ q_e = \frac{m_r}{m_s} (h_{11} - h_{10}) \quad (6.16) \]

\[ \text{COP} = \frac{(h_{11} - h_{10})}{\left[ \left( \frac{(h_2 - h_1)}{\eta_{m,co,l}} \right) + \left( \frac{(h_4 - h_3)}{\eta_{m,co,II}} \right) \right]} \quad (6.17) \]
6.5 Result and Discussion

One advantage of using vapor compression refrigeration for cooling any steam power plant condenser is controlling the condensate fluid temperature, depending on the surroundings conditions. As the condensate fluid gets low, the turbine output power increases. Even though, the lower condensate temperature increases the turbine produced power, the coefficient of performance will decrease as the condensate temperature decreases. Thus, the vapor compression system needs more power to keep its running.

Depending on the previous thermodynamic analysis and using various operation conditions, the effects of decreasing the condensate temperatures on the reference and studied system will be described in detail in this section. Also, a compression study between the reference and studied systems will discuss bases on the vapor compression refrigeration system by using the above table.
Table 6.2 Design data for steam cycle of the studied and reference systems

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<td>the reference system condenser and cooling air inlet temperature</td>
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For the reference system, the difference in temperature between steam in air should be between 10 - 15 °C to enable heat to transfer effectively between these two fluid, so the $\Delta t_{sc,rs}$ is taken at 10 °C. In the studied system and during the condensation process, both the refrigerant and steam temperature remain constant, so the $\Delta T_{sc,ss}$ is taken at 4 °C.

Figure 6.6 The coefficient of performance of the VCRS on the temperature of the studied system condenser

\[ t_a=50^\circ C \quad \quad t_a=40^\circ C \quad \quad t_a=30^\circ C \]

The figure describes the effect of the steam power plant condenser for the studied system. As the figure shows the coefficient of performance of the vapor compression refrigerant increases as the ambient temperature decreases. Also, the coefficient of performance of vapor compression refrigeration is enhanced as the condensate temperature increases.
The figure depicts the effect of the ambient temperatures on the pump load. The pump loads for the reference and studied systems increase as the ambient temperatures increases. Also, the condensate temperature has no effect on the pump load for both the reference and studied systems.
It is seen from the figure that the specific load of the steam condenser of both reference and studied systems is an independent parameter of the steam condenser temperature. And that means the enthalpy of the steam that enters the condenser and condensate that is leaving the condenser do not depend on the temperature of the condenser.

In the figure 4.10 the specific work of the turbine is plotted versus the condensate temperature. The figure shows that as the condensate temperature and pressure do not change, the specific work of turbine in the studied system is an independent parameter of the ambient temperature. However, this specific work decreases with an increasing in the condensate temperature with condenser pressure increasing.
Figure 6.9 Specific load of the steam plant condenser against the temperature of the steam plant condenser of the studied system

Figure 6.10 Thermal efficiency of combined actual steam and refrigeration cycle versus temperature of the steam plant condenser of the studied system
Figure 6.11. Effect of the temperature of the steam plant condenser on the specific refrigerant mass for the studied system

Figure 6.12. Effect of the temperature of the steam plant condenser on the heat removed in the water cooled condenser of the refrigerant cycle of the studied system

- $t_a = 50^\circ$C
- $t_a = 40^\circ$C
- $t_a = 30^\circ$C
7 Conclusions and Recommendations for Future Work

7.1 Conclusions

Experimental, numerical, and theoretical studies of thermodynamics analysis in steam power plant condenser were performed. First, an experimental test rig was fabricated to study the possibility of using vapor compression refrigeration system for cooling the steam power plant condenser. Secondly, the numerical comparison between using direct cooling and indirect cooling of a steam power plant (also another comparison between using different refrigerant as cooling media for steam plant condenser) was presented in this thesis. This numerical study was carried out using the Aspen-HYSYS approach. Finally, an analytical study was done to evaluate the concept of cooling the steam power plant condenser using a vapor compression refrigeration system under different operating conditions. The main conclusions can be drawn as follows:

1. The experimental study showed that decreasing the coolant temperature and increasing the coolant mass flow rate has effects on increasing the condensation rate.
2. Higher COP values in the vapor compression refrigeration system can be occur at lower condenser pressure and coolant temperature.
3. As the coolant temperature decrease, the vacuum inside the condenser will rise up and then the power plant efficiency will increase.
4. The numerical study revealed that NH\textsubscript{3} has the best heat transfer characteristic and condensation rates.
5. The numerical compression proved that using refrigerants for steam condenser is better than using water as a coolant because it provides more condensation rates.
6. The analytical results indicated that using VCRS for cooling the steam power plant condenser can improve the thermal efficiency compared to the steam plant when the condenser is cooled using ambient temperature.

7. The studies showed that the maximum coefficient of heat transfer between the steam and the refrigerant is achieved and the condenser becomes compact.

8. The restrictions of using fresh, river, and sea water for a cooling steam power plant can be solved by using the VCRS.

7.2 Recommendations for Future Work

For future work, the following recommended.

1. Develop a 3D CFD model for condensation of water vapor in steam condenser.

2. Conduct cost analyses for the refrigerant cooled condenser.

3. Using the knowledge gained, design and optimize a cost effective refrigerant-cooled-condenser unit for a specific steam power plant.
APPENDICES
APPENDIX A

Experimental and Numerical Data

Table A.1 Aspen result of steam power plant cooled by R-410A at $m_{cl} = 0.056$ kg/s

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Table A.2 Aspen result of steam power plant cooled by R-410A at $m_{cl} = 0.11$ kg/s

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Table A.7 Aspen result of steam power plant cooled by NH$_3$ at m$_{cl}$ = 0.056 kg/s

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Table A.8 Aspen result of steam power plant cooled by NH$_3$ at $m_{\text{cl}} = 0.11$ kg/s

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APPENDIX B

Theoretical Study Data

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APPENDIX C

Experimental Conditions

• Coolant water conditions

\[ \dot{V}_{cl} = 0.00011 \pm 3.3 \times 10^{-6} \text{ m}^3/\text{s} \]

\[ \dot{m}_{cl} = 0.11 \text{ kg/s} \]

\[ T_{cl-in} = 9.6 \pm 0.3 \, ^\circ\text{C} \]

\[ T_{cl-out} = 26.8 \pm 0.3 \, ^\circ\text{C} \]

• Refrigerant conditions

\[ \dot{m}_{ref} = 35.83 \, \text{g/s} \]

\[ T_{ref-in} = -8.3 \pm 0.3 \, ^\circ\text{C} \]

\[ T_{ref-out} = 0.9 \pm 0.3 \, ^\circ\text{C} \]

• Condensation conditions

\[ \Delta V_{cs} = 0.912 \pm 0.05 \, \text{litt} \]

\[ \dot{m}_{cs} = 0.0038 \, \text{kg/s} \]

\[ \text{time} = 240 \pm 0.5 \, \text{s} \]

\[ T_{sat} = 101 \pm 0.35 \, ^\circ\text{C} \]
Electrical power conditions

\[ V = 209 \pm 0.7\% + 2 \text{ volt} \]

\[ I = 14.87 \pm 6\% \text{ amp} \]
APPENDIX D

Thermo-physical Properties

\[ C_{p-cl} = 4.190 \pm 0.001 \text{ kj/kg.k} \]
\[ C_{p-ref-in} = 1.48 \pm 0.0025 \text{ kj/kg.k} \]
\[ C_{p-ref-out} = 1 \pm 0.0035 \text{ kj/kg.k} \]
\[ \rho_{cs} = 990 \pm 2.5 \text{ kg/m}^3 \]
\[ \rho_{cl} = 990 \pm 2.5 \text{ kg/m}^3 \]
\[ \rho_{s} = 0.598 \pm 0.0015 \text{ kg/m}^3 \]
\[ h_{fg-s} = 2261 \pm 5.4 \text{ kj/kg} \]
\[ h_{fg-ref} = 228.3 \pm 1.5 \text{ kj/kg} \]
\[ \mu_{cs} = 591 \pm 16 \times 10^{-6} \text{ pa.s} \]
\[ k_{cs} = 0.638 \pm 5 \times 10^{-4} \text{ pa.s} \]
REFERENCES
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