“**Performance Analysis of 20 TR Air Cooled Chiller with Evaporative Cooling using Cooling Pad**

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I, ABHISHEK PUSHKAR, 2K21/THE/02, of M. Tech (Thermal Engineering), hereby declare that the project Dissertation titled "Performance Analysis of 20 TR Air Cooled Chiller With Evaporative Cooling using Cooling Pad" which is submitted by me to the Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfilment of the requirement for the award of the degree of Master of Technology, is original and not copied from any source without proper citation. This work has not previously formed the basis for the award of any Degree, Diploma Associateship, Fellowship, or other similar title or recognition.

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I hereby certify that the Project Dissertation titled "Performance Analysis of 20 TR Air Cooled Chiller With Evaporative Cooling using Cooling Pad" which is submitted by ABHISHEK PUSHKAR, 2K21/THE/02, Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfilment of the requirement for the award of the degree of Master of Technology, is a record of the project work carried out by the student under my supervision. To the best of my knowledge, this work has not been submitted in part or full for any Degree or Diploma to this University or elsewhere.

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ABSTRACT

Keywords –Heat Ventilation Air Conditioning (HVAC), Air cooled chiller, Condenser, Cooling Pads, Spraying, Evaporative cooling, Improvement, Coefficient of performance, Energy saving, Carbon dioxide, Environment

Energy is a basic necessity shared by all living things on the planet, much like water, food, and shelter so we need to save energy and protect our mother earth. The recent increase in awareness of global warming has rekindled the hunt for environment friendly refrigerants that have a lower impact on global warming and zero ozone depletion effect.

The aim of this study is to examine how a spray evaporative cooling system can improve the efficiency of an air-cooled scroll chiller in a tropical climate region. To achieve this, the study involves the design selection and simulation of a fin and tube type heat exchanger, as well as theoretical investigation that combines a spray evaporative cooling system with the air-cooled scroll chiller in the Fin and tube type heat exchanger. According to the theoretical results, the addition of the spray evaporative cooling system to the air-cooled chiller led to a reduction in compressor power consumption and an increase in the COP (Coefficient of Performance) of the chiller, ranging from 6.25% to 18.66%. Additionally, the use of the spray cooling system resulted in power savings of 6.36% to 17.26% in air-cooled chiller systems. To further improve the COP of the air-cooled chiller, it is suggested to use a more appropriate spray evaporative cooling mode, such as adjusting the water flows based on the cooling capacity and positioning the cooling pads at appropriate distance from condenser. Certain design constraints are addressed, as well as several methods for improving system functionality for future optimization.

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ABHISHEK PUSHKAR

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CHAPTER 1

INTRODUCTION

1.1 GENERAL

In the vapor compression cycle, the utilization of air-cooled condensers gives rise to concerns regarding power consumption, especially in tropical regions experiencing very hot weather conditions. This is due to the direct impact of the ambient air temperature on the temperature of the air-cooled condenser. In areas with exceedingly hot summers, there can be a substantial rise in the temperature and pressure of the condenser, leading to an increased pressure ratio and greater power consumption of the air-cooled scroll chillers. When the condenser temperature rises within the cycle of an air-cooled scroll chiller, it results in a decrease in the cooling capacity of the cycle as the evaporator's liquid content diminishes. This has a measurable adverse effect on the performance of the air-cooled scroll chiller. To address this situation and improve the performance of the air-cooled scroll chiller, reducing the condenser temperature is considered one of the most effective solutions. By lowering the condenser temperature, the pressure ratio across the compressor decreases, subsequently reducing power consumption. This action also enhances the refrigerant quality after the thermal expansion valve, thereby increasing the availability of liquid refrigerant in the evaporator and augmenting the mass flow rate of refrigerant and the cooling capacity. The most straightforward method to achieve a decrease in the condenser temperature is by incorporating a direct evaporative cooler in front of the condenser, which effectively cools the air temperature before it passes over the condenser. The utilization of an evaporative cooler is energy-efficient, environmentally friendly, and cost-effective, thereby significantly enhancing the performance of air-cooled scroll chillers. Given the widespread use of air-cooled scroll chillers in commercial and institutional buildings, as well as spot cooling in hot industrial environments like mills, foundries, and power plants, any substantial enhancement in the cycle's performance will have a tremendous impact on the power consumption of the entire network electricity storage.

1.2 REFRIGERATION CYCLE EMPLOYED IN AIR-COOLED CHILLERS

An air-cooled chiller operates using a refrigeration cycle that encompasses two important stages:

1. The refrigerant in the liquid phase undergoes evaporation in the evaporator, where it absorbs heat from the water and effectively cools the chiller-water system.
2. The refrigerant in vapor phase is condensed in the air-cooled fin and tube condenser, releasing heat to the atmosphere in the process.

During the refrigeration cycle of an air-cooled condenser chiller, water is directed to the evaporator, also known as the cooler. Inside the evaporator, the water is cooled by the refrigerant flowing through a separate circuit. The chilled water is then pumped to the building's coils to provide cooling. As the chilled water performs its cooling function, the warmer water returns to the evaporator to complete the cycle. Within evaporator, a combination of cold liquid phase refrigerant and flash gas phase flows through a circuit that runs counter to the flow of the water requiring the chilling. This causes the refrigerant to evaporate into a vapor state, which exits from the evaporator as a superheated vapor phase. The superheated vapor phase refrigerant then enters the compressor through the suction inlet, where it undergoes compression. After compression, the pressure and temperature of the refrigerant gas increase. Upon exiting into the compressor, the high-pressure and temperature refrigerant gas passes through the discharge line and enters the condenser. In the air-cooled condenser coil, the hot gas condenses into a liquid within the tubes and releases heat to the surrounding air. This is facilitated by the condenser fans drawing cooler air across the condenser coil. After the refrigerant in gaseous phase is condensed into a liquid phase in the air-cooled condenser, it leaves the condenser and goes to the expansion device. As the refrigerant flows through the expansion device, its pressure and temperature are reduced to a level that causes some of the liquid phase refrigerant to flash into vapor phase. The expansion device is mainly responsible for regulating the amount of flashing to maintain a specific superheat, ensuring that no liquid droplets enter the compressor suction. Following its passage through the expansion device, The refrigerant goes back into the evaporator, and the cycle continues.

Air-cooled chillers consist of primarily four components:

- Evaporator
- Condenser
- Compressor
- Expansion device.

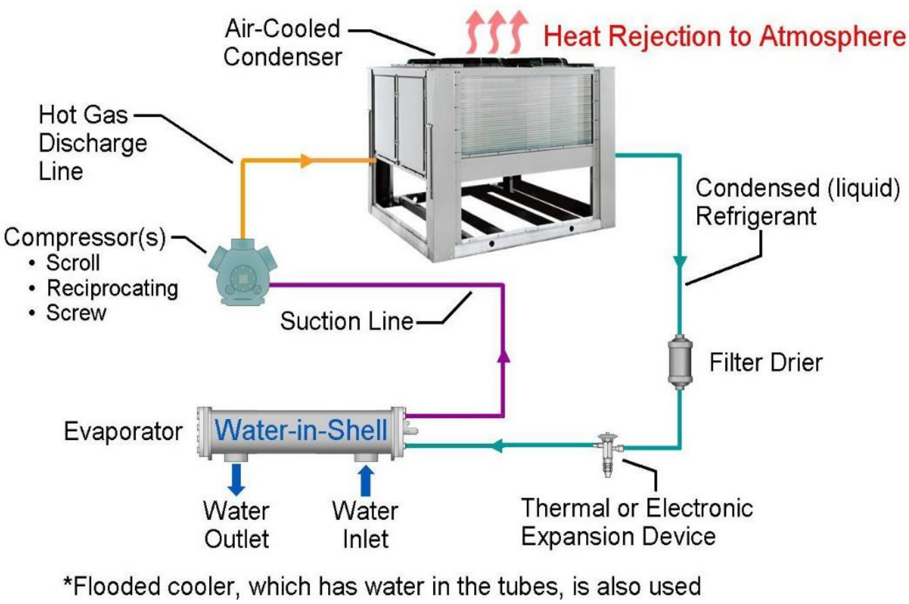

Figure 1.2.1 Refrigeration Cycle components of Air Cooled chiller]

1.3 EVAPORATOR

There are three types of evaporators used in air-cooled scroll chiller:

- Brazed-plate
- Direct expansion (DX) shell-and-tube
- Flooded shell-and-tube

Larger screw chillers commonly employ the shell-and-tube evaporator design, while smaller chillers often utilize brazed-plate evaporators. Shell-and-tube evaporators have widespread usage but are bulkier and larger compared to a bphe. Some distinct evaporator types incorporate internal and external grooving enhancements on the tubes to induce turbulence and enhance heat transfer between the fluid and the refrigerant. Turbulence is an effective mechanism for improving the heat transfer process by increasing the interaction between the fluid and the heat exchanger surface area. This increased contact results in improved heat transfer and higher efficiency ratings for both the refrigerant and water sides. To achieve this enhanced performance, the tubes are often rifled or finned during the manufacturing process to introduce the increased variation

1.4 BRAZED-PLATE

Air-cooled chillers with capacities up to 60 tons employ brazed-plate evaporators, which consist of a numbers of plates brazed together, with alternate plates rotated by 180 degrees. This arrangement creates two highly turbulent fluid channels flowing in counter flow, resulting in a greater surface area with a increase in heat transfer coefficient and excellent performance characteristics. The plates are in order of stacked manner to create a multi-layered pathway for the fluid to travel through, with manifolds at each end connecting the layers or circuits to the inlet and the outlet. Brazed-plate evaporators offer efficient heat transfer at a reasonable cost due to the fluid coming into contact with a large surface area of the heat exchanger. The smaller size and fluted construction of the plates induce turbulence in the fluid, further enhancing heat transfer efficiency. This turbulence persists even at low heat exchanger loadings, making it suitable for part-load applications. Compared to shell-and-tube heat exchangers, brazed-plate heat exchangers are significantly smaller, often only 25% of the size of an equivalent shell-and-tube heat exchanger. The compact size of brazed-plate heat exchangers offers several advantages. They require less space for installation and transportation, making them more convenient. Additionally, they require smaller volumes of refrigerant compared to the shell-and-tube systems due to their designed circuitry. However, the smaller circuitry also makes them more susceptible to clogging from debris. To prevent this, it is recommended to use a Y-type strainer to filter out foreign particles and contaminants from entering into the water circuit. It's important to note that the brazed-plate design cannot be mechanically cleaned on the inside.

Figure 1.4.1 Brazed-Plate Evaporator [19]

1.5 DIRECT EXPANSION (DX) SHELL AND TUBE

It comprises tubes through which the refrigerant flows, while the water circulates outside the tubes in the shell section and refrigerant in the tubes. Heat transfer occurs as the warm water or water/antifreeze mixture transfers heat to the colder refrigerant, resulting in the evaporation of the refrigerant in the tubes and the cooling of the water. Baffles positioned inside the shell guide the water flow over the tubes, generating turbulence that enhances heat transfer. Air-cooled chillers equipped with scroll compressors commonly employ a direct expansion evaporator. This type of evaporator enables the oil to travel alongside the refrigerant flows through the tubes to a specific location within the chiller. There, the oil can be separated from the refrigerant and get returned to the oil pump section, ensuring the necessary lubrication for scroll compressors

Figure 1.5.1 Direct Expansion Shell and Tube Evaporator

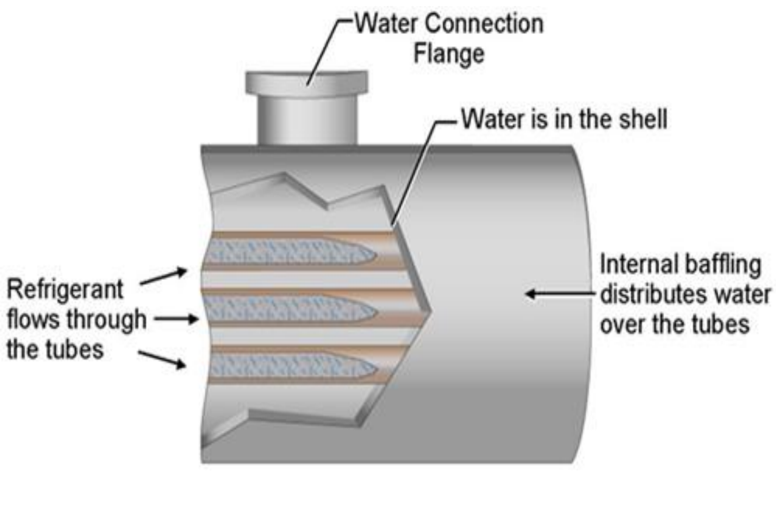

Figure 1.5.2 Direct Expansion Shell and Tube Evaporator

1.6 FLOODED-SHELL AND TUBE

Flooded shell-and-tube evaporators, also known as "flooded coolers," operate with water or a water/antifreeze mixture flowing through the tubes, while the refrigerant circulates through the outer shell. The liquid refrigerant is evenly distributed at the bottom of the cooler, and tube cleaning processes do not require the removal of refrigerant. While flooded evaporators may not necessarily be more efficient than other designs such as DX or brazed-plate evaporators, they are commonly used in chiller systems that incorporate more efficient compressors like centrifugal compressors. Air-cooled chillers that utilize flooded shell-and-tube evaporators can improve their efficiency by adjusting the number of tube passes in the evaporator vessel. This can be achieved by increasing tube velocity and turbulence through additional passes or reducing pressure drop by decreasing the number of passes. While higher pass arrangements can enhance heat transfer, they also result in a higher pressure drop, leading to increased pumping horsepower for the chilled-water circuit. Thus, an economic balance must be struck by considering the trade-off between higher pumping horsepower and lower compressor power. DX evaporators, on the other hand, do not offer options for "pass" arrangements since water is circulated on the outer shell. Instead, modifications can be made to the internal baffling of the shell to accommodate different water flow rates.

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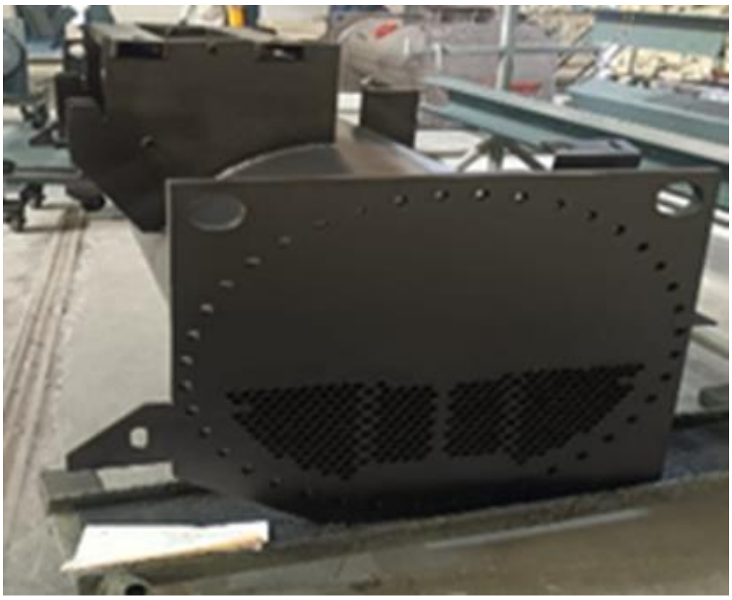

Figure 1.6.1 Flooded shell-and-tube evaporators [19]

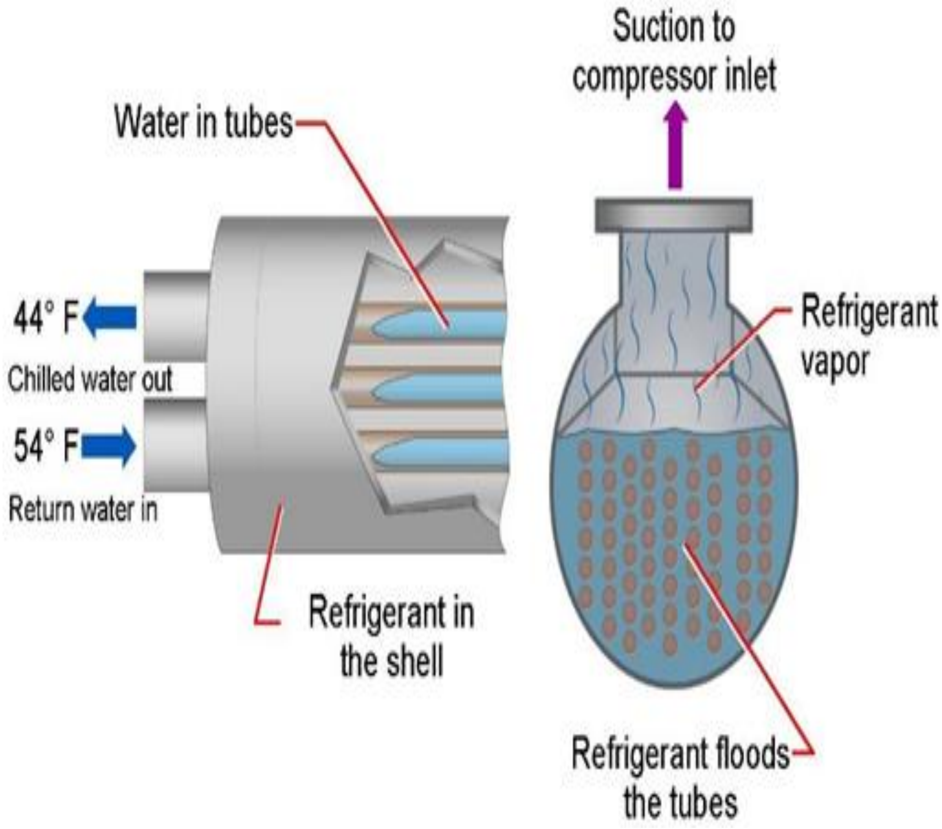

Figure 1.6.2 Flooded shell-and-tube evaporator

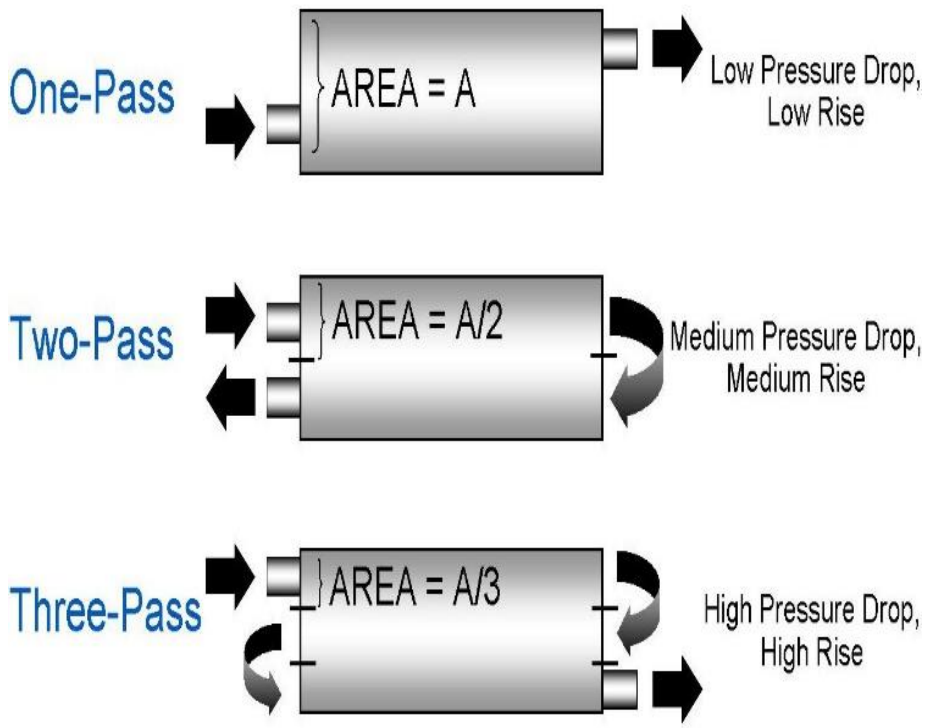

Figure 1.6.3 Flooded Evaporator Pass Arrangements

1.7 COMPRESSOR

The function of a compressor is to raise the pressure of a fluid by utilizing shaft work or power. The fluid enters the compressor at a low pressure and, through the rotation of blades or impellers driven by the shaft work, exits the compressor at a higher pressure and velocity. It is important to note that a compressor and a pump, while serving similar purposes, are not exactly the same. A compressor is specifically designed for handling gases, while a pump is used for liquids. Both devices are employed to increase fluid pressure, but they are designed to accommodate the different properties and characteristics of gases and liquids.

Compressors can be categorized into three main types: Open, Hermetically Sealed, and Semi-Hermetic (or semi-bolted).

1. Open Compressors: In this type, the compressor's shaft extends outside and is connected to an external prime mover, such as an electric motor or engine. The open design allows for easy access and maintenance of the compressor components.

2. Hermetically Sealed Compressors: The motor-compressor unit in hermetically sealed compressors is completely enclosed. Only the refrigerant lines and electrical connections extended outside the sealed unit. This design ensures that the motor and compressor are protected from external elements and do not require additional sealing.

3. Semi-Hermetic (Semi-Bolted) Compressors: Semi-hermetic compressors feature an encapsulated motor and compressor. However, the heads of the compressor can be removed to access and service the pistons and valves. Additionally, the motor can be repaired by removing a bolted plate, providing easier maintenance compared to hermetically sealed compressors.

These different types of compressors offer various advantages and are used in different applications depending on factors such as serviceability requirements, motor accessibility, and environmental considerations.

Both hermetic and semi-hermetic compressors facilitate direct contact between the refrigerant and the motor windings. Consequently, they are limited to employing halocarbon refrigerants that pose no threat to copper, whereas ammonia refrigerants are incompatible with these types of compressors. Only open-type compressors are capable of accommodating ammonia refrigerants.

In contrast to hermetic and semi-hermetic compressors, open type compressors are typically more effective in terms of efficiency. This is due to the fact that in hermetic compressors, the suction vapor flows over the motor, resulting in the vapor becoming superheated and demanding additional power for compression. Nevertheless, by implementing appropriate design practices in refrigeration systems, the impact on energy consumption can be minimized.

The utilization of Air-cooled chillers involves the employment of three distinct types of compressors: reciprocating compressors, scroll compressors, and the screw compressors

1.8 RECIPROCATING COMPRESSOR

It, much like reciprocating engines, comprise pistons, rods, and intake and discharge valves port. These valves port function based on suction and discharge pressure respectively to achieve compression by reducing a fixed quantity of refrigerant gas within a chamber. As a result, they fall under the category of positive displacement compressors, similar to scrolls and screws. Reciprocating compressors have served as the primary compressor type for small chillers for an extensive period and continue to be utilized in various designs today. In fact, in many cases, multiple staging reciprocating compressors were installed in a single chiller to enable chiller capacities of up to 400 tons. There are three distinct types of reciprocating compressors available: open, semi-hermetic, and hermetic. Open compressors employ an external driver and extends the shaft through a seal in the crankcase. Hermetic compressors encompass both the motor and compressor within the same housing, typically welded and sealed. On the other hand, semi-hermetic compressors utilize a bolted housing, facilitating convenient field repairs.

1.9 SCROLL COMPRESSOR

Scroll compressors utilize a distinct compression process in which a fixed scroll and an orbiting scroll collaborate to compress gas pockets from suction pressure to intermediate pressure and eventually discharge pressure, which exits through the discharge port. These compressors have gained popularity as an alternative to reciprocating compressors and are available in hermetic configurations with capacities up to 25 tons each for air-cooled chillers. To meet higher capacities, multiple scroll compressors are often employed in a single chiller design. When compared to reciprocating compressors, scroll compressors demonstrate approximately 10 to 15 percent higher efficiency and have proven to be highly reliable due to their reduced number of moving parts, approximately 60 percent fewer. Typically, reciprocating and scroll compressors are used individually or in multiples to deliver air-cooled chiller capacities ranging from approximately 10 to 400 tons.

Figure 1.9.1 Scroll Compressor [19]

1.10 SCREW COMPRESSOR

It is widely utilized into the medium-sized water chillers, either in one or in multiples, to achieve a chiller capacity range of approximately 70 to 500 tons. It is also known as "rotary" or helical-rotary type compressors. As the screw in the rotors turned by a motor which is in meshing with another screw these compressors compress the sucked gas. Similar to scroll compressors, screw compressors employ one or more rotors for compression. They, along with scroll and reciprocating compressors, fall into the positive displacement category of compressors, which means they compress a fixed amount of gas in a compression chamber. Screw compressors possess an advantage in terms of reliability compared to reciprocating compressors, owing to their fewer moving parts, similar to scroll compressors. Their exceptional reliability and good efficiency, particularly at part loads, have contributed to their growing popularity. Screw compressors are available in various designs, including those with one or more than one rotors. These compressors can unload by utilizing either a slide valve or solenoid valve of a mechanism, which adjusts the volume of gas is reduced whet it is being compressed and consequently reduces the compressor's capacity.

Figure 1.10.1 Screw Compressor [19]

1.11 CONDENSER

It is a heat exchanger that converts refrigerant gas into a liquid and dissipates its heat into the surrounding atmosphere. There are different types of condensers, including natural air draft cooled ones (typically used in small appliances like refrigerators), forced air draft cooled condensers, water-cooled shell and tube condensers, water-cooled plate heat exchangers, and evaporative condensers

1.12 WATER COOLED CONDENSER

Water-cooled condensers typically rely on cooling towers or spray ponds for water cooling. However, evaporative condensers provide an alternative solution. These systems utilize water sprayed directly onto coils, allowing heat to dissipate into the atmosphere without the need for a separate cooling tower or spray pond. The heat transfer coefficients of condensers may vary from 1400 to 11000 W/m2°K, depending on the design of the condenser and the refrigerant used. Most refrigerants tend to form a thin condensate film on the outer surface of the tube due to their low surface tension. The thickness of the condensate film directly affects heat transfer, with shorter, vertical fins on a horizontal tube resulting in a thinner film compared to a plain tube. Some condensers incorporate externally finned tubes, along with additional longitudinal or spiral grooves and ridges, to induce turbulence and enhance heat transfer on the water side. Nevertheless, finned tubes can trap condensate, thereby impeding their performance. To address this issue, enhanced surfaces with intricate geometries have been developed to facilitate surface tension-driven condensation.

1.13 AIR COOLED CONDENSER

The condenser of an air-cooled chiller functions as a heat exchanger that utilizes cooler outside air passing over a coil to condense the hot refrigerant gas. Propeller-type fans are commonly employed in air-cooled chillers to suck the outdoor air which flows over the coil. The coil tubes allow the hot refrigerant vapor to flow through and be converted into a liquid state. Towards the end of the condenser tubing, the condensed liquid refrigerant undergoes subcooling, which reduces its temperature below the saturated condensing temperature. This subcooling process enhances the specific refrigeration effect and increases the chiller system capacity by approximately 1% for every one degree of subcooling achieved. In the case of air-cooled condensers, the refrigerant condensing temperature and pressure are typically higher due to the limitations imposed by the dry bulb temperature of the air. Consequently, this leads to higher compression ratios and increased power consumption of compressors, often exceeding 20% compared to water-cooled condensers. However, in areas where water is scarce or proper treatment facilities are lacking, air-cooled condensers are the preferred choice. In such scenarios, evaporative cooling pads can be utilized prior to the condenser to lower the condensing temperature and pressure during dry weather conditions with low relative humidity. Additionally, air-cooled condensers can provide warm air for the required heating during cold chilled weather condition.

Figure 1.13.1 Air Cooled Condenser [19]

The performance of condensers is better when the temperature of the cooling medium is low.

• Evaporative condensers are more effective than shell and tube condensers, but they can be challenging to repair if the tubes get punctured. Additionally, evaporative condensers are more susceptible to corrosion because the tubes are constantly exposed to air.

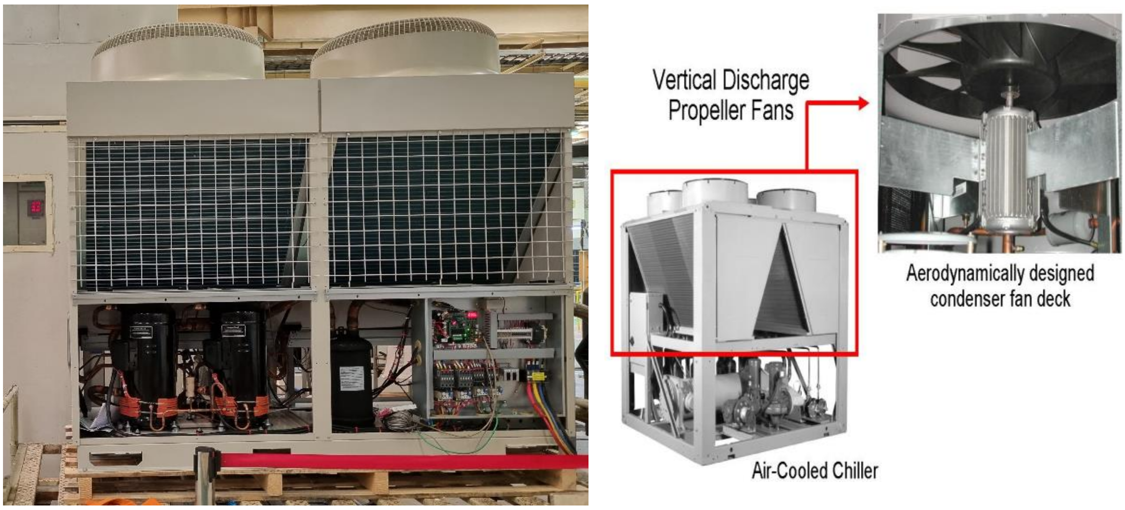

Figure 1.13.2 Air Cooled Chiller [19]

1.14 EXPANSION DEVICE

The expansion device plays a vital role in air-cooled chillers as it regulates the distinction in pressure between the high-pressure condenser and low-pressure evaporator sections of the refrigeration system. This pressure variation is necessary for the evaporator to absorb heat from the water being cooled, while simultaneously allowing the condenser's refrigerant to release heat to the outside air at an adequately high temperature. High-pressure liquid refrigerant flows from the condenser to the expansion device, where its pressure is reduced to match that of the evaporator. Consequently, a portion of the liquid boils off, cooling the remaining refrigerant to achieve the desired evaporator temperature. Besides regulating the pressure differential, the expansion device also ensures the appropriate suction superheat of the vapor entering the compressor, preventing any liquid from entering the compressor suction. This crucial process is known as superheat

1.15 THERMAL EXPANSION VALVE (TXV)

In order to regulate the refrigerant flow and ensure a desired level of superheat, an essential component known as a thermostatic expansion valve (TXV) is employed. The TXV consists of two main parts, namely a sensing bulb and a regulating valve. Placed on the suction line as it exits the evaporator, the sensing bulb detects the temperature of the refrigerant leaving the evaporator. Based on this information, the valve is able to make appropriate adjustments to the refrigerant flow..

Figure 1.15.1 Thermal expansion valve (TXV)[19]

1.16 ELECTRONIC EXPANSION VALVE (EXV)

An electronic expansion valve (EXV) utilizes external temperature and pressure sensors to determine the degree of superheat and subsequently regulates the refrigerant flow through the valve using a precisely controlled stepper motor. Although EXVs carry a higher price tag compared to thermostatic expansion valves (TXVs), their capacity to deliver precise refrigerant flow and rapidly adjust to varying conditions results in reduced compressor head pressure and enhanced chiller efficiency..

1.17 EVAPORATIVE CONDENSER

In areas characterized by dry summers, evaporative cooling can provide comfortable temperatures by adding moisture to the air through small desert coolers or central humidification plants. This technique consumes only a fraction of the energy used by a traditional air conditioning system equipped with HVAC chillers, typically ranging from 10% to 20%. The efficiency of air-cooled condensers is limited by the dry bulb temperature, but their performance can be improved in dry weather by introducing humidified air with a wet bulb temperature. This can be achieved through a pre-cooler that incorporates a cooling pad, which pulls air through while water trickles over it. Depending on the specific design, an additional booster fan may be necessary to counteract the increased air flow resistance. By implementing this approach during the dry summer months, energy savings of approximately 30% to 40% can be achieved..

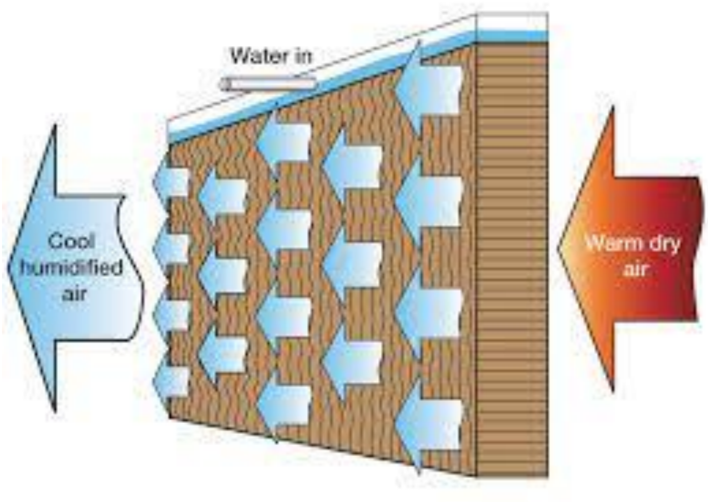

Figure 1.17.1 Evaporative Cooling Diagram [24]

When air comes into contact with water under adiabatic conditions, the heat from the air is utilized to cause the water to evaporate, thereby cooling the air and increasing its relative humidity. The air's lowest attainable temperature through this process is referred to as the wet-bulb temperature. The psychometric chart in Figure 3 illustrates the process of evaporative cooling and also provides a comparison with sensible cooling methods. The surface of the cooling pad is covered with a water film that directly interacts with the hot ambient air. The water absorbs heat from the air, undergoes evaporation, and gets carried along with the inlet air of the condenser. As a result of this contact, the ambient air is both cooled and humidified. The energy required for evaporation from the wetted cooling pad is drawn from the ambient air, resulting in a cooling effect. Thus, employing evaporative cooling to pre-cool the ambient air can effectively reduce energy consumption, particularly in hot weather conditions

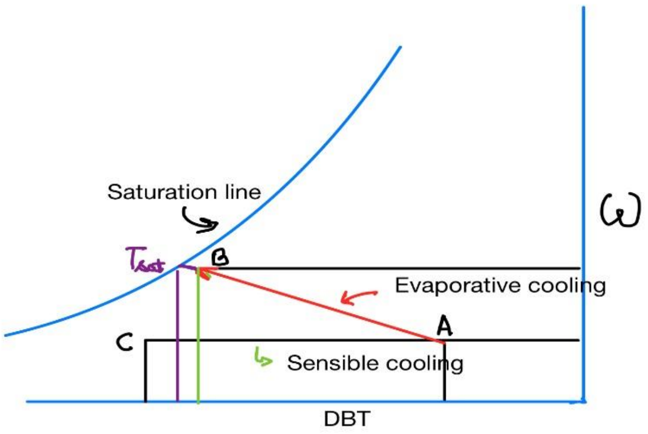

Figure 1.17.2 Psychometric Chart for Evaporative Cooling

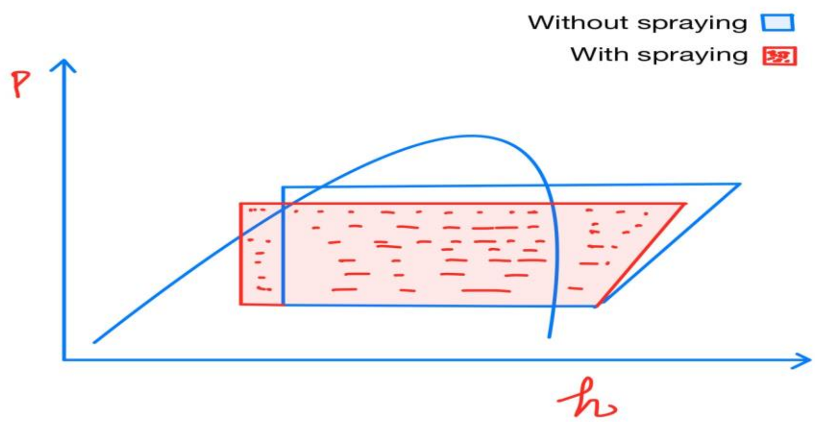

Figure 1.17.3 Qualitative refrigeration cycle with and without spraying

1.18 REFRIGERANT USED IN AIR-COOLED CHILLER

Previously, manufacturers of air-cooled chillers employed R-22 refrigerant in certain models, as it had been the preferred choice for many years. However, the scenario has now evolved due to the environmental repercussions of R-22's chlorine emissions when released into the atmosphere. In contrast, newer chiller designs have adopted R-134a, a chlorine-free refrigerant primarily used in larger centrifugal chillers. Moreover, innovative blends such as R-410A (Puron™) and R-407c are now utilized in air-cooled chillers

1.19 CONDENSER COIL CORROSION PROTECTION

Corrosion pertains to the deterioration or ruin of a metal or alloy due to chemical, physical, or electrochemical reactions with the surroundings. In air-cooled chillers, exposure to the environment can result in general or localized corrosion in the condenser coil. If the coils are not adequately safeguarded, corrosion can result in subpar performance, unappealing surface conditions, and, in severe cases, equipment failure, leading to increased operating, maintenance, and service costs. Fortunately, most corrosion concerns can be mitigated by implementing appropriate measures to shield the coil. It is feasible to identify potentially corrosive environments prior to equipment selection and apply suitable measures to safeguard the coil from harm. Corrosive atmospheres, such as those prevalent near coastal regions, industrial sites, densely populated urban areas, and certain rural locations, can cause premature equipment degradation and failure. Consequently, it is crucial to consider local factors like laundry facilities, diesel-burning devices, exhaust piping, sewer vents, and traffic when choosing equipment and coil protection. Adequate protection can minimize the impacts of these environments and ensure long-lasting equipment performance.

The subsequent enumeration presents the various coil structures, along with the frequently employed materials to counteract the impact of corrosive surroundings on condenser coils found in air-cooled chillers.

The subsequent enumeration presents the various coil structures, along with the frequently employed materials to counteract the impact of corrosive surroundings on condenser coils found in air-cooled chillers.

1.20 STANDARD COIL CONSTRUCTION

The standard coil design for air-cooled chillers typically comprises copper tubes with aluminum fins, making it ideal for non-corrosive surroundings like clean rural areas. Nevertheless, in environments that contain corrosive elements, this particular coil type is not advisable due to its susceptibility to visible deterioration caused by corrosion. However, the standard coil design is generally satisfactory for the majority of applications.

1.21 PRE-COATED ALUMINUM FIN COILS

Pre-coated aluminum fin coils are engineered to endure mild corrosive conditions, specifically in coastal regions. The aluminum fin stock undergoes a protective epoxy coating prior to the fin stamping procedure, which effectively mitigates galvanic action and corrosion arising from the dissimilar metals within the coil. The pre-coated fin material is subsequently stamped into a wavy pattern to maximize thermal efficiency. Nevertheless, it is advised against utilizing this coil type in harsh industrial or coastal settings where corrosion poses a substantial hazard..

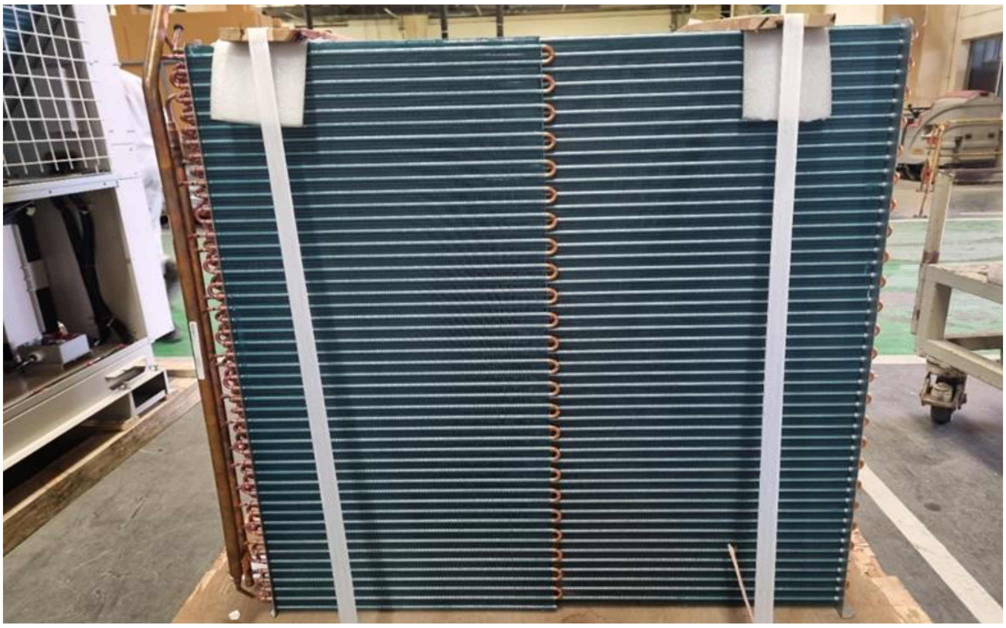

Figure 1.21.1 Pre-coated aluminum fin coils [19]

1.22 COPPER FIN COILS

The utilization of copper fin coils eliminates the bi-metallic connection found in conventional coil structures. A wavy copper fin design is mechanically bonded to the copper tube. To provide additional protection against galvanic corrosion, a Mylar strip is placed between the coil assembly and the sheet metal coil support pan. Compared to standard and pre-coated coil constructions, the use of copper fin coils can significantly improve durability in a coastal, unpolluted marine environment by eliminating the bi-metallic construction. In uncontaminated coastal regions, copper is generally resistant to corrosion due to the development of a natural protective film on its surfaces. However, in industrial settings, copper is susceptible to corrosion caused by different pollutants, which can harm both the tube and fin and weaken the bond between them. In such situations, alternative materials or protective coatings may be required to prevent corrosion.

CHAPTER 2

LITERATURE REVIEW

2.1 GENERAL

This chapter involves the previous studies done by researchers to understand the effect of evaporative cooling in air-conditioning system

Wang et al. (2015)[1] examined the effects of different amounts of water and angles of nozzles utilized in spray cooling on the temperature distribution of the adjacent air in close proximity to the condenser. The results revealed that augmenting the quantities of spray and adjusting the angles contributed to a uniform dispersion of air temperature surrounding the condenser, thereby enhancing heat dissipation.

Yu and Chan (2005)[2] performed an simulation on an air-cooled unit incorporating spray cooling equipment, and the results indicated that the equipment could effectively decrease the condensation temperature by a range of 2.1-6 degrees Celsius. Additionally, it was observed that the power usage of the refrigeration unit could be decresed by 1.4-14.4%, while simultaneously enhancing the refrigeration capacity by 1.3-4.6%.

Yan et al. (2010)[3] researchers carried out experiments on a closed circulating gas-assisted four-spray system that utilized R134a as the working fluid for electronic chip cooling. The results demonstrated that enhancing the flow rate led to enhancements in both the heat transfer capacity and temperature uniformity of the heating surface.

Liu et al. (2015)[4] proposed an idea for an air conditioning system that integrated two separate evaporative condensers. They performed experiments to analyze how various variables, such as the temperature of the water entering the evaporator and the dry-bulb temperature of the air, affected the system's coefficient of performance and cooling capacity.

According to the research conducted by Kabeel et al. (2017)[5], it was observed that the efficiency of the spray system increased with higher ambient air temperatures or lower relative humidity of the surrounding air.

In an experimental study conducted by Wang et al.(2014) [6], an air-conditioning system with an evaporative cooling condenser was investigated. The results showed an increase in the system's coefficient of performance ranging from 6.1% to 18%, accompanied by a decrement in the power consumption of the compressor by 14.3%.

Martinez et al.(2016) [7] carried out a experiment to investigate the influence of various cooling pads on the overall performance of an air-conditioning system that utilizes an evaporative cooler. The experiments entailed using cooling pads of different thicknesses, and the researchers assessed their effect on power utilization and the coefficient of performance of the system. In recent times, numerous studies have been conducted to improve the performance of the evaporative condenser through a blend of experimental and numerical analysis.

Ketwong et al.(2021) [8] carried out an examination on evaporative cooled condensers in different hot-dry and hot-humid climates. The investigation discovered that the energy efficiency ratios (EER) for tropical hot and dry climates varied from 3.40 to 4.22, whereas for hot-humid climates, the range was 3.30 to 3.94. In contrast, a typical unit had an EER of 3.01..

Goswami et al. (1993)[9] team successfully integrated an evaporative cooler featuring a media pad onto an existing 2.5-ton air conditioning system. Through their research, it was discovered that this modified system achieved a remarkable 20% reduction in power consumption under ambient air temperatures of 34°C.

Zhang et al. (2000)[10] conducted a research on an evaporative cooler that made use of corrugated perforated aluminum foil. Correlations were established to calculate the efficiency, pressure drop, and final air temperature of the cooler. By comparing the final temperature of the evaporative cooler with the performance curve of an air-cooled chiller, the researchers predicted the potential enhancement. The findings revealed that the coefficient of performance of the chiller could be increased by approximately 39%.

Yu and Chan (2005)[11] performed a simulation of an air-cooled condenser, incorporating staging condenser fans along with a direct evaporative cooler. It was anticipated that the integration of the evaporative cooler could result in a potential decrease in power consumption by up to 14.4% and an enhancement in refrigeration capacity by up to 4.6%.

Nasr and Hassan (2009)[12] devised and employed an evaporative condenser model. Through the implementation of this enhanced system, they accomplished a 13% growth in capacity and a 55% improvement in COP (Coefficient of Performance).

Youbi-Idrissi et al. (2007) [13] developed and employed a unique design for an evaporative condenser. Through the integration of this enhanced system, a notable enhancement of 13% in capacity and an impressive 55% improvement in COP (Coefficient of Performance) were attained.

Pongsakorn and Thepa (2013) [14] conductead a study of performance of a 3.5 kW inverter air conditioning system equipped with an evaporative-cooled condenser was analyzed. A cellulosic media pad with a thickness of 150 mm was employed for this purpose. The results revealed a decrease of 35% in power consumption and an enhancement of 18% in COP when compared to a conventional air-cooled condenser.

Hajidavalloo (2007)[15] conducted investigation to examine the effects of integrating an evaporative cooler into a window air conditioner. The study involved introducing water onto the media pad positioned in front of the condenser. The findings demonstrated a reduction of 16% in power usage and a significant increment of 55% in overall performance.

Delfani et al. (2010)[16] evaluate the efficiency of a traditional air conditioning system equipped with a pre-cooling unit in four cities located in Iran. The findings revealed a significant decrease of 55% in the maximum power consumption.

Amrat et al. (2018)[17] performed a theoretical investigation on a combination of a window air conditioning system and a direct evaporative cooling system with the aim of reducing the size of the evaporative cooling unit. Their findings revealed that raising both dry and wet air temperatures led to increased energy savings in the system. Taking into account various operating and design conditions, they established that the most suitable thickness for the cooling pad fell within the range of 100 mm and 150 mm.

Hajidavalloo and Eghtedari (2010)[18] conducted investigation to analyze the influence of varying ambient temperatures, spanning from 35°C to 49°C, on the operational efficiency of a window air conditioning unit featuring a 50 mm cellulosic pad. The results indicated a noteworthy 20% reduction in energy consumption, coupled with a substantial

50% increase in the coefficient of performance (COP), in comparison to a system equipped with a standard air-cooled condenser.

2.2 STUDY GAP

From the literature survey, it was concluded that various studies have been conducted on evaporative cooling with cooling pads in room air conditioner condensers. However, there is very limited research done on air-cooled chillers with cooling pad, and none of the research work found specifically addresses evaporative cooling using cooling pads in an air-cooled 20TR chiller with an aluminum blue coil fin and copper tube heat exchanger.

2.3 OBJECTIVE OF STUDY

1. To demonstrate the impact of evaporative cooling using cooling pads in an air-cooled 20TR chiller, which features an aluminum blue coil fin and a copper tube heat exchanger,
2. To analyze the impact on mass flow rate, power consumption, and coefficient of performance due to evaporative cooling in air cooled chiller using a cooling pad.

CHAPTER 3

METHODOLOGY

3.1 GENERAL

An overview of an air-cooled chiller with an evaporative cooling in condenser has been discussed in the previous chapters. The basic objective of this study is to validate the stimulated model with existing experimental model, so that the various parameters can be studied at different input parameters. To achieve the stated objective the following two simulation software is used i.e. Hexim and Coolselector.

3.2 GENERATING CONDENSER LOOP

Using the Hexim software the generation of condenser loop is done by following Industry standards.

Input parameters:

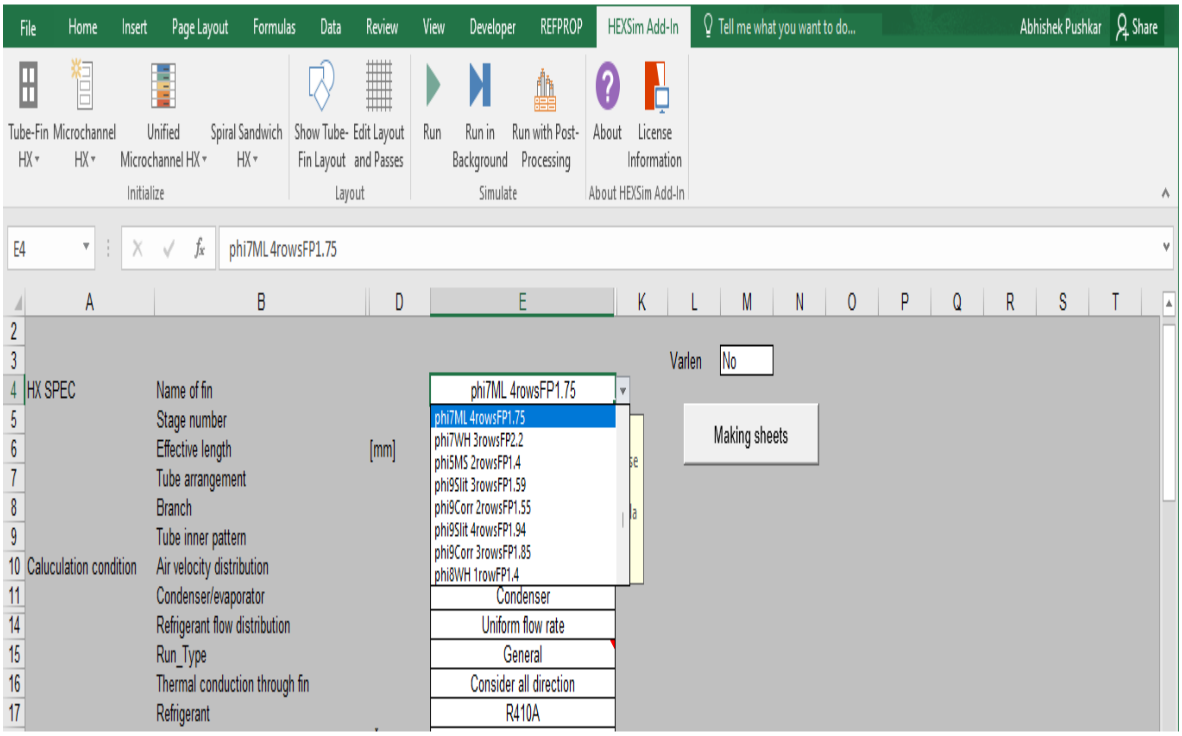

Figure 3.2.1 Input parameter in Hexim [20]

3.3 LOOP DRAWING

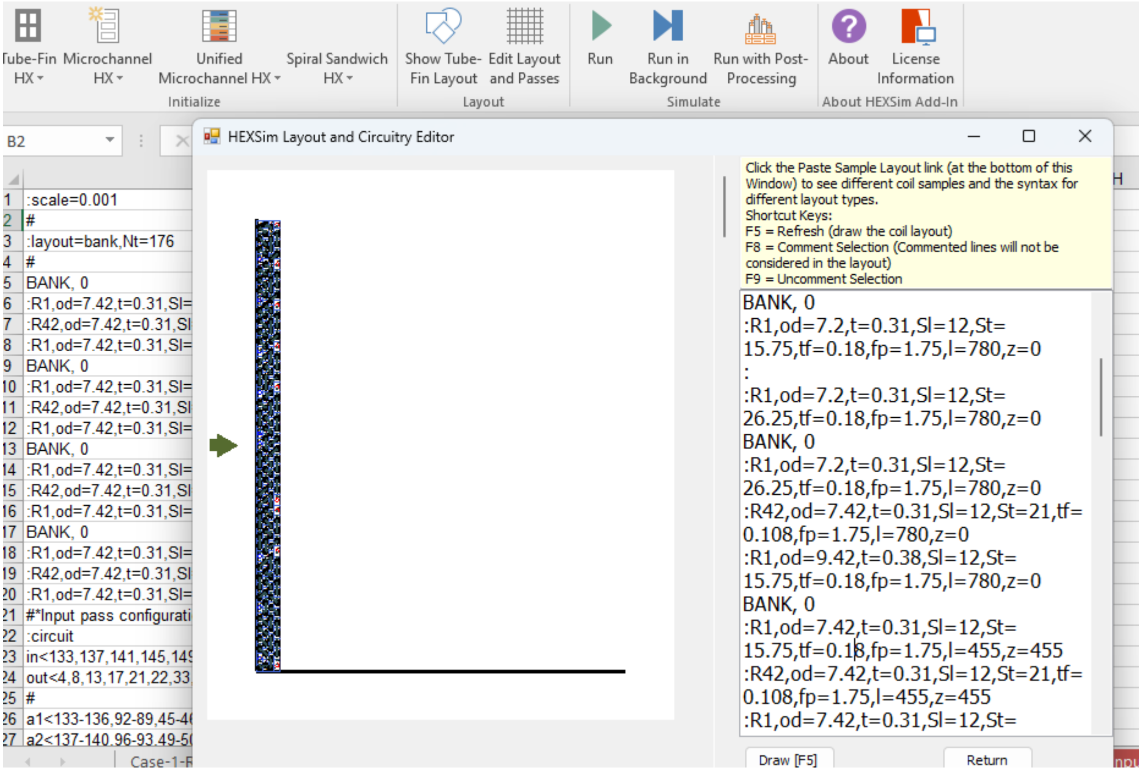

Figure 3.3.1 Loop drawing in Hexim [20]

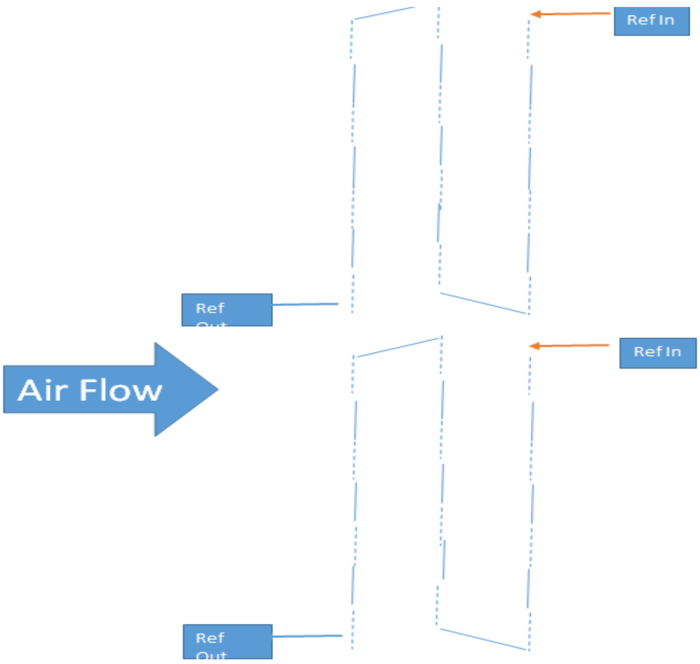

Figure 3.3.2 Flow of refrigerant in a pipe of a condenser

Figure 3.3.3 Fin and tube type Heat exchanger[19]

3.4 Code of drawing condenser loop

```
:scale=0.001
#
:layout=bank,Nt=176
#
BANK, 0
:R1,od=7.042,t=0.31,Sl=12,St=26.25,tf=0.108,fp=1.75,l=890,z=0
:R42,od=7.042,t=0.31,Sl=12,St=21,tf=0.108,fp=1.75,l=890,z=0
:R1,od=7.042,t=0.31,Sl=12,St=15.75,tf=0.108,fp=1.75,l=890,z=0
BANK, 0
:R1,od=7.042,t=0.31,Sl=12,St=15.75,tf=0.108,fp=1.75,l=890,z=0
:R42,od=7.042,t=0.31,Sl=12,St=21,tf=0.108,fp=1.75,l=890,z=0
:R1,od=7.042,t=0.31,Sl=12,St=26.25,tf=0.108,fp=1.75,l=890,z=0
BANK, 0
:R1,od=7.042,t=0.31,Sl=12,St=26.25,tf=0.108,fp=1.75,l=890,z=0
:R42,od=7.042,t=0.31,Sl=12,St=21,tf=0.108,fp=1.75,l=890,z=0
:R1,od=7.042,t=0.31,Sl=12,St=15.75,tf=0.108,fp=1.75,l=890,z=0
BANK, 0
:R1,od=7.042,t=0.31,Sl=12,St=15.75,tf=0.108,fp=1.75,l=455,z=455
:R42,od=7.042,t=0.31,Sl=12,St=21,tf=0.108,fp=1.75,l=455,z=455
:R1,od=7.042,t=0.31,Sl=12,St=26.25,tf=0.108,fp=1.75,l=455,z=455
#*Input pass configuration below.
:circuit
in<133,137,141,145,160,161,176>
out<4,8,17,22,33,38,43>
#
a1<133-136,92-89,45-46,1-2,47-48,3-4>
a2<137-140,96-93,49-50,5-6,51-52,7-8>
a3<141-144, ,53-54,9-10,55-56,11-12,57,13>
```

```
a4<145-148,104-105,62-60,103,59,102-101,58,14-17>
a5<149-154,110,67-65,109-108,64,107-106,63,18-21>
a6<160-155,111,68,112,69-70,114,71,27-22>
a7<161-164,120,77-75,118,74-73,117-115,72,28-33>
a8<165-170,125-124, ,123-121,78,34-37,81-83,38>
a9<176-171,126-128,84,39-40,85,129-130,86,41-42,131-132,88,44-43>
#
#b1<>
#b2<>
#
#<a1>-><b1,b2>
#
#In case of refrigenrat flow rate input with branch
#To divide 0.45:0.55 between a pass with top tube <1> and top tube <2>
#mdotratio<1>,0.45
#
#Example of input in case of Evapbpp
#instate<1>,x=0.15,mdot=30
#instate<2>,x=0.25,mdot=20
#instate<3>,x=0.35,mdot=10
#To specify the inlet condition for each pass (effective only for evaporator)
#instate<42>,P=600000,x=0.25,mdot=20
#
#To specify the length and inner diameter of capillary (effective only for evaporator)
#indist<>,l=200,id=2.2
#
#Example of input in case of EvapD
#instate<8,9,11>,x=0.25
#
#
#Example of input of pressure drop at branch.
#branchLosses
#[11, 0.002, 0.0043, 0.00287]
#11: Position of the branch (tube serial number before the branch).
#
#Example of input of pressure drop at valve.
#[11, 0.1]
#11: Position of the branch (tube serial number before the branch).
#0.1: Cv value of the valve.
```

3.5 MODEL VALIDATION

Table 3.5.1 Experimental and Simulated data

	Ambient 35°C			
	Experimental Data	Simulated Data	Difference	%
Condenser Inlet temperature (°C)	94.1	92.6	0.0159405	1.594049
Condenser outlet temperature (°C)	36.5	35.99	0.0139726	1.39726
Suction Pressure(bar)	8.5	8.485	0.0017647	0.176471
Suction Temperature(°C)	7.7	7	0.0909091	9.090909
Discharge Pressure(bar)	32.4	31.9	0.0154321	1.54321
Discharge Temperature((°C)	89.5	92.6	-0.0346369	-3.46369
	Ambient 30°C			
	Experimental Data	Simulated Data	Difference	%
Condenser Inlet temperature (°C)	87.3	83.3	0.045819	4.581901
Condenser outlet temperature (°C)	33.4	31.22	0.0652695	6.526946
Suction Pressure(bar)	8.51	8.485	0.0029377	0.293772
Suction Temperature(°C)	7.9	7	0.1139241	11.39241
Discharge Pressure(bar)	31.1	29.5	0.0514469	5.144695
Discharge Temperature((°C)	89.5	83.3	0.0692737	6.927374

In the above table 3.1 the stimulated data is compared with the experimental data at 30°C and 35°C ambient conditions and following are the results found:

1. Difference between the suction temperature in stimulated and experimental data is 0.9°C and 0.7°C at ambient 30°C and 35°C respectively which is very small.
2. Difference between the condenser inlet temperature in stimulated and experimental data is 4.5% and 1.5% at 30°C and 35°C ambient respectively.
3. Difference between the condenser inlet temperature in stimulated and experimental data is 4.5% and 1.6% at 30°C and 35°C ambient respectively.
4. Difference between the condenser outlet temperature in stimulated and experimental data is 6.5% and 1.4% at 30°C and 35°C ambient respectively.
5. Difference between the discharge pressure in stimulated and experimental data is 5.1% and 1.5% at 30°C and 35°C ambient respectively.

Figure 3.5.1 Experimental Testing [19]

3.6 FORMULA USED FOR CALCULATION

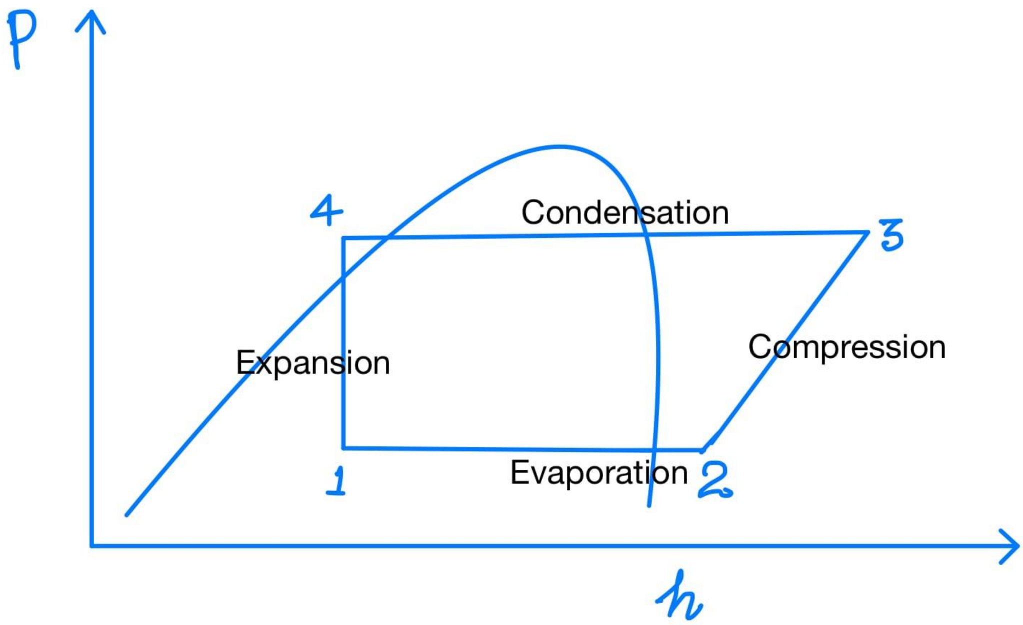

Figure 3.6.1 P-h diagram of VCRS for chillers

1. Heat addition in an evaporator

$q = \dot{m}(h_2 - h_1)$

q = refrigerating effect

$\dot{m}$ = mass flow rate of refrigerant

h_2 = enthalpy at compressor inlet

h_1 = enthalpy at evaporator inlet

2. Work done by compressor

$w_c = \dot{m}(h_3 - h_2)$

w_c = Compressor work

$\dot{m}$ = mass flow rate of refrigerant

h_2 = specific enthalpy at compressor inlet

h_3 = specific enthalpy at compressor outle

3.Total Power Consumption

Total Power Consumption = $w_c + w_p + w_F$

w_c = Compressor work

w_p = Pump work

w_F = Fan Work

4. **Coefficient of Performance**

Coefficient of Performance = $\frac{q}{w_c + w_p + w_F}$

3.7 SOFTWARE

3.7.1 COOLSELECTOR: VERSION 5.2.6

It is a Danfoss software used to design the system by selecting all the components is going to be used in a designed system. In this research work, the Coolselector selection software is utilized to perform the selection and calculation of components within a refrigeration cycle. The software's capabilities are illustrated through a graph and properties snippet. By utilizing this software, the compressor is carefully chosen, and the inlet condition for the condenser is determined, all while maintaining the integrity of the research.

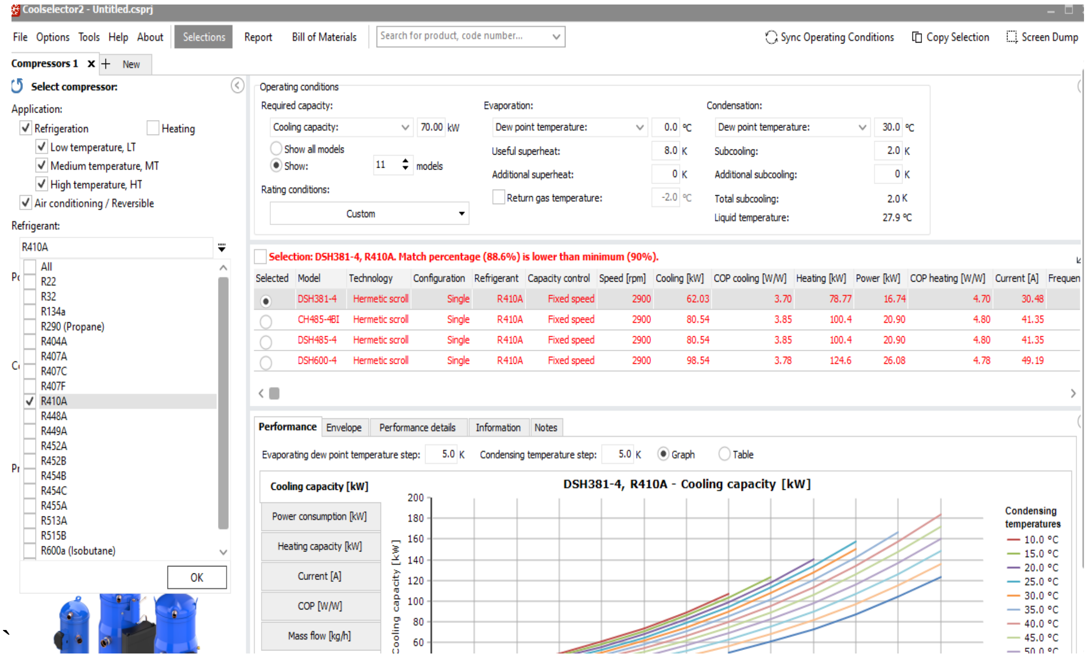

Figure 3.7.1 Coolselector software dashboard for input values [21]

Following are values we get the after the putting the input values in a Coolselector software.

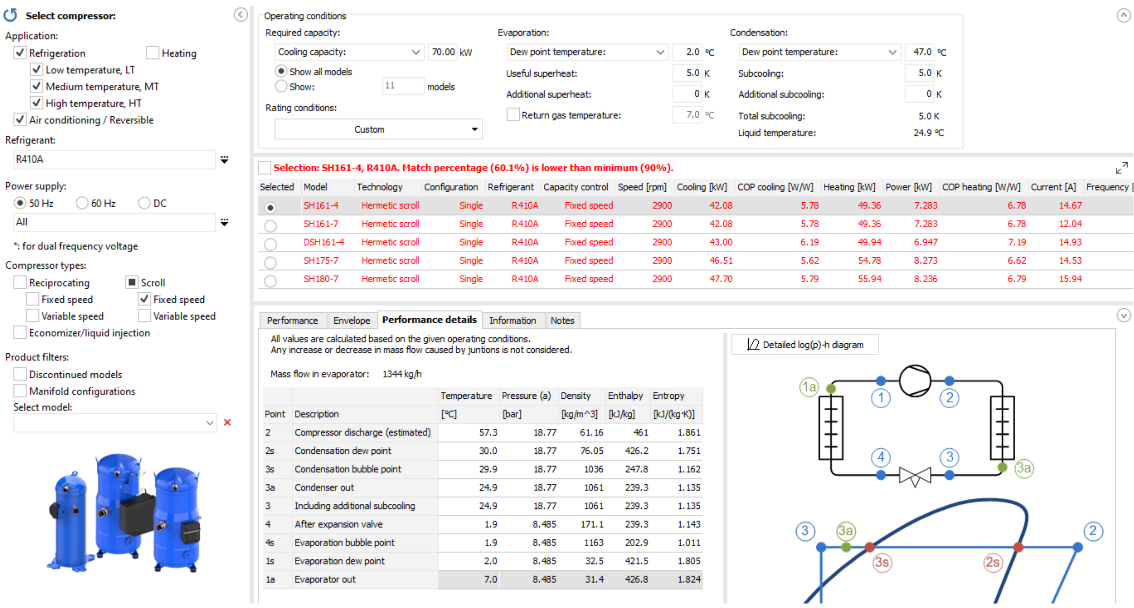

Figure 3.7.2 Coolselector software dashboard for input values [21]

After that we have also p-h diagram for the input values and the selected refrigerant for my work scroll compressor SH 161-4 of fixed speed is selected and refrigerant 410A which is a azeotropes, mixture of R125 and R32 with 50%-50% values is used for work.

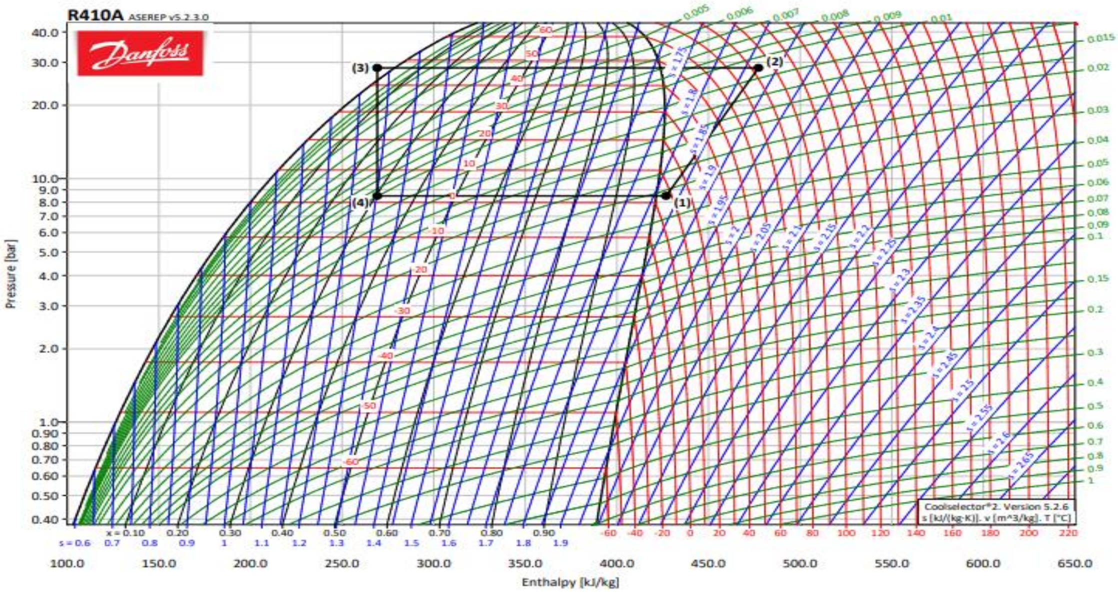

Figure 3.7.3 p-h diagram of R410A [21]

3.7.2 HEXIM : SOLVER VERSION 4.0.20194.1210

Hexim software is a simulation software used for the simulation of the condenser by taking the input values as shown below in the figure 3.4.2 then we design the condenser loop in the layout by writing the codes then we stimulate and get the condenser output parameters and the heat gained by the air.

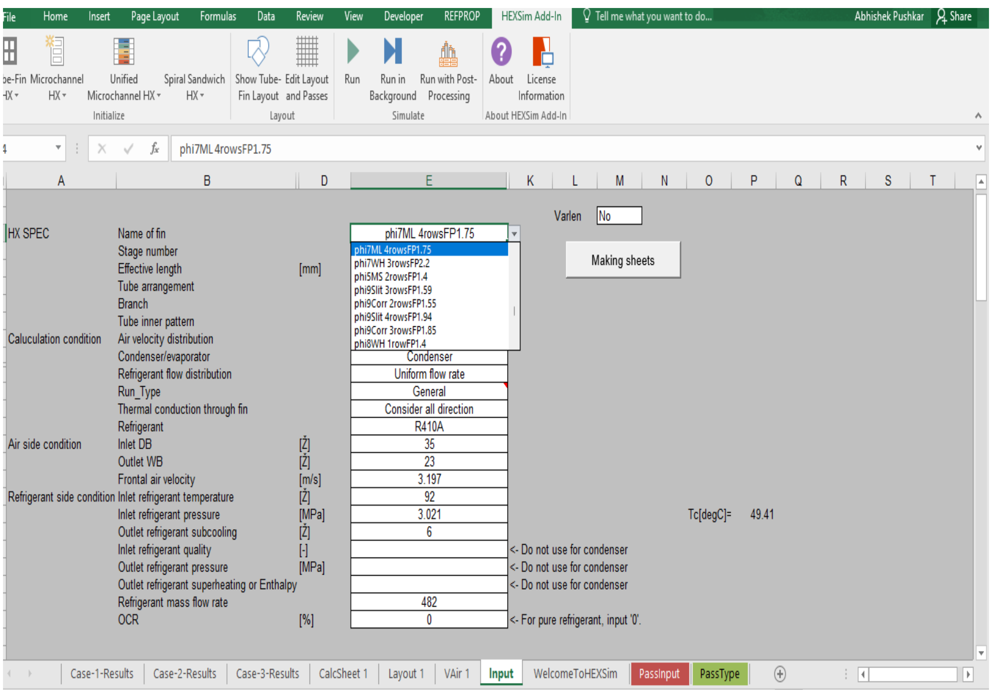

Figure 3.7.4Input Parameters [20]

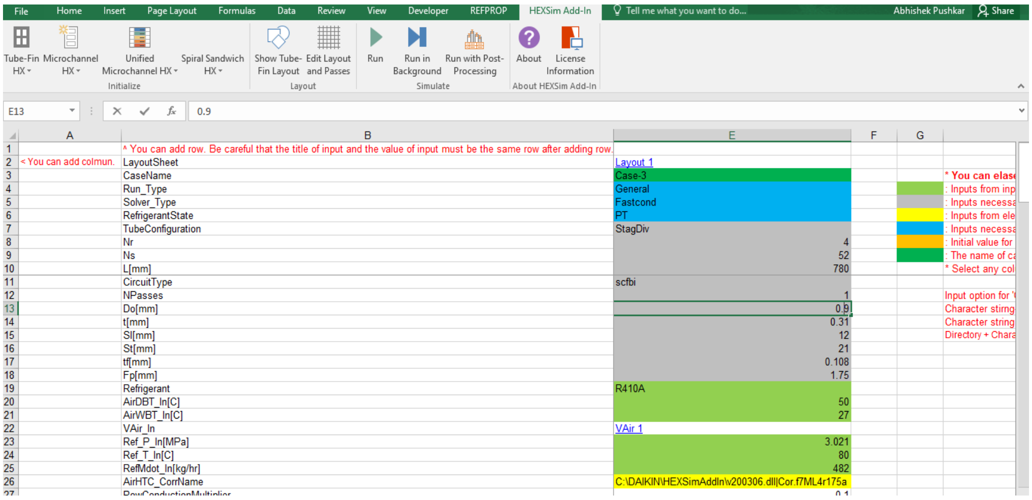

Figure 3.7.5Calculation Sheet [20]

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 SIMULATION RESULTS AND CALCULATION

On Hexim simulation software solver version 4.0.20194.1210, fin and tube type condenser is stimulated from which we get the output value and the Hexim input value we got from the coolsector selection software. There are total seven different cases of dry bulb temperature and wet bulb temperature are taken for this research work at different cooling pad efficiency varying from 90% to 65 %

Case 1: DBT 20°C and WBT 14°C

Table 4.1.1 Calculation and result at DBT 20°C and WBT 14°C

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 1	DBT = 20, WBT= 14	DBT = 14.6, WBT= 14	DBT = 15, WBT= 14	DBT = 15.2, WBT= 14	DBT = 15.5, WBT= 14	DBT = 15.8, WBT= 14	DBT = 16.1, WBT= 14
AirDBT_In[C]	20	14.6	15	15.2	15.5	15.8	16.1
AirWBT_In[C]	14	14	14	14	14	14	14
Air Out DBT [C]	29.4784581	24.09419447	24.49706374	24.70667754	25.0132272	25.31072183	25.60772559
Air Out WBT [C]	17.4266334	17.4340415	17.43486579	17.43806155	17.44017592	17.43920893	17.43807497
Air Out RH [%]	29.30852249	51.92636751	49.84137109	48.80275239	47.30273105	45.86773313	44.47164587
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	22.4	19.26	19.76	19.86	20.01	20.16	20.32
Compressor discharge Tempreature[C]	67	58.6	59.9	60	60.6	61	61.4
RefMdot_In[kg/hr]	805.4	807.8	807.7	807.7	807.6	807.5	807.4
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	466.4	461.4	462.4	462.5	462.5	462.9	463.2
Specific Refrigerating effect(kW)	29.02699633	29.11349346	29.10988941	29.10988941	29.10628537	29.10268132	29.09907727
Specific Compressor Work (kW)	8.8594	7.763855556	7.987255556	8.009691667	8.0087	8.097430556	8.163711111
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	9.9594	9.587255556	9.6087	9.763711111	9.363855556	9.609691667	9.697430556
Coefficient of Performance(COP)	2.914532635	3.109135258	3.036310991	3.029222	3.02915955	3.001071382	2.980329604

Case 2: DBT 25°C and WBT 18°C\

Table 4.1.2 Calculation and result at DBT 25°C and WBT 18°C

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 2	DBT = 25, WBT= 18	DBT = 18.7, WBT= 18	DBT = 19.05, WBT= 18	DBT = 19.4, WBT= 18	DBT = 19.75, WBT= 18	DBT = 20.1, WBT= 18	DBT = 20.45, WBT= 18
AirDBT_In[C]	25	18.7	19.05	19.4	19.75	20.1	20.45
AirWBT_In[C]	18	18	18	18	18	18	18
Air Out DBT [C]	34.44075304	28.23027031	28.60048376	28.9136688	29.26051925	29.60604831	29.95603891
Air Out WBT [C]	20.96876492	20.99894336	21.00476472	20.99360579	20.99247129	20.99094342	20.99074426
Air Out RH [%]	29.47515703	52.63836288	50.93712286	49.45033727	47.90942819	46.4161897	44.95404705
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	25.3	21.85	22.4	22.07	22.26	22.45	22.65
Compressor discharge Temprerature[C]	74.69999999	65.4	66.9	66	66.5	67	67.6
RefMdot_In[kg/hr]	799.9	806.1	805.4	805.8	805.6	805.3	805
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	471.3	465.5	466.4	465.8	466.1	466.5	466.8
Specific Refrigerating effect(kW)	28.82877373	29.05222466	29.02699633	29.04141252	29.03420442	29.02339228	29.01258014
Specific Compressor Work (kW)	9.887652778	8.665575	8.8594	8.7295	8.794466667	8.880669444	8.944444444
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	10.98765278	10.4594	10.39446667	10.54444444	10.265575	10.3295	10.48066944
Coefficient of Performance(COP)	2.623742697	2.830063066	2.77520664	2.811502252	2.79323657	2.769230767	2.751456494

Case 3: DBT 30°C and WBT 22°C

Table 4.1.3 Calculation and result at DBT 30°C and WBT 22°C

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 3	DBT = 30, WBT= 22	DBT = 22.8, WBT= 22	DBT = 23.2, WBT= 22	DBT = 23.6, WBT= 22	DBT = 24, WBT= 22	DBT = 24.4 WBT= 22	DBT = 24.8, WBT= 22
AirDBT_In[C]	30	22.8	23.2	23.6	24	24.4	24.8
AirWBT_In[C]	22	22	22	22	22	22	22
Air Out DBT [C]	39.42345005	32.32534941	32.70882329	33.11334114	33.51061951	33.90416283	34.30259079
Air Out WBT [C]	24.56054737	24.59173094	24.58718117	24.58808376	24.58711091	24.58517079	24.58449618
Air Out RH [%]	29.66771231	53.46853639	51.80118668	50.1246022	48.52018126	46.97498619	45.46463241
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	28.48	24.11	24.34	24.46	24.7	24.94	25.18
Compressor discharge Temprerature[C]	83.3	71.5	72.1	72.4	73.1	73.7	74.4
RefMdot_In[kg/hr]	790.9	802	802	801.8	801.2	800.7	800.2
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	477.3	469.2	469.6	469.8	470.2	470.7	471.1
Specific Refrigerating effect(kW)	28.50440948	28.90445872	28.90445872	28.89725063	28.87562634	28.85760611	28.83958587
Specific Compressor Work (kW)	11.09456944	9.445777778	9.534888889	9.577055556	9.658911111	9.764091667	9.846905556
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	12.19456944	11.13488889	11.25891111	11.44690556	11.04577778	11.17705556	11.36409167
Coefficient of Performance(COP)	2.337467478	2.616787998	2.595846174	2.585408159	2.564690853	2.539367594	2.519422016

Case 4 : DBT 35°C and WBT 23°C

Table 4.1.4 Calculation and result at DBT 35°C and WBT 23°C

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 4	DBT = 35, WBT= 23	DBT = 24.2, WBT= 23	DBT = 24.8, WBT= 23	DBT = 25.4, WBT= 23	DBT = 26, WBT= 23	DBT = 26.6, WBT= 23	DBT = 27.2, WBT= 23
AirDBT_In[C]	35	24.2	24.8	25.4	26	26.6	27.2
AirWBT_In[C]	23	23	23	23	23	23	23
Air Out DBT [C]	44.36219294	33.71302661	34.31992193	34.9029715	35.49610875	36.08764234	36.68410371
Air Out WBT [C]	25.44861342	25.4936361	25.49494715	25.49029844	25.48817178	25.48564496	25.48435009
Air Out RH [%]	21.8743419	52.12786116	49.67606826	47.39943372	45.19860764	43.10042077	41.08504565
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	31.97	24.7	25.3	25.54	25.91	26.29	26.67
Compressor discharge Temprerature[C]	92.6	73.1	74.7	75.3	76.3	77.3	78.4
RefMdot_In[kg/hr]	778.04	801.2	799.9	799.3	798.4	797.4	796.4
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	484.3	470.2	471.3	471.8	472.4	473.1	473.8
Specific Refrigerating effect(kW)	28.04092901	28.87562634	28.82877373	28.80714945	28.77471302	28.73867255	28.70263208
Specific Compressor Work (kW)	12.42702778	9.658911111	9.887652778	9.99125	10.11306667	10.25545	10.39744444
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	13.52702778	11.48765278	11.71306667	11.99744444	11.25891111	11.59125	11.85545
Coefficient of Performance(COP)	2.072955676	2.564690853	2.509544316	2.485249602	2.456633591	2.424089558	2.392395498

Case 5 : DBT 40°C and WBT 29°C

Table 4.1.5 Calculation and result at DBT 40°C and WBT 29°C

	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 5	DBT = 40, WBT= 29	DBT =30.1, WBT= 29	DBT = 30.65, WBT= 29	DBT =31.2, WBT= 29	DBT = 31.75, WBT= 29	DBT =32.3, WBT= 29	DBT =32.85, WBT= 29
AirDBT_In[C]	40	30.1	30.65	31.2	31.75	32.3	32.85
AirWBT_In[C]	29	29	29	29	29	29	29
Air Out DBT [C]	49.3131386	39.61099913	40.171032	40.70685744	41.25285278	41.81310009	42.33227899
Air Out WBT [C]	30.93453668	30.98130735	30.98291649	30.97967351	30.97846955	30.98012145	30.97354596
Air Out RH [%]	27.61778487	54.41722179	52.33826581	50.40538796	48.51911999	46.66977756	44.98913111
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	35	28.48	28.92	29.29	29.67	30.05	30.43
Compressor discharge Temprerature[C]	100.7	83.3	84.4	85.4	86.5	87.9	88.5
RefMdot_In[kg/hr]	765.3	790.9	790.9	788.2	786.9	785.5	784.1
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	487.3	477.3	478.1	478.8	479.6	480.3	481.6
Specific Refrigerating effect(kW)	27.58177339	28.50440948	28.50440948	28.40710021	28.36024759	28.30979093	28.25933427
Specific Compressor Work (kW)	12.86129167	11.09456944	11.270325	11.38511111	11.5412	11.67340278	11.93574444
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	13.52702778	11.48765278	11.71306667	11.99744444	11.25891111	11.59125	11.85545
Coefficient of Performance(COP)	1.975588939	2.245401831	2.214738904	2.187667088	2.158117036	2.132820905	2.087756191

Case 6 : DBT 45°C and WBT 32°C

Table 4.1.6 Calculation and result at DBT 45°C and WBT 32°C

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 6	DBT = 45, WBT= 32	DBT = 33.3, WBT= 32	DBT =33.95, WBT= 32	DBT = 34.6, WBT= 32	DBT =35.25, WBT= 32	DBT = 35.9, WBT= 32	DBT = 36.55, WBT= 32
AirDBT_In[C]	45	33.3	33.95	34.6	35.25	35.9	36.55
AirWBT_In[C]	32	32	32	32	32	32	32
Air Out DBT [C]	53.01443085	42.80809531	43.46711443	44.10648959	44.74169434	45.38046752	46.01898559
Air Out WBT [C]	33.48348553	33.7593715	33.76051516	33.75814622	33.75503286	33.75255869	33.75004003
Air Out RH [%]	27.37663157	54.56733445	52.25295928	50.09174912	48.03361125	46.05462978	44.16061755
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	39.95	30.54	31.25	31.68	32.15	32.63	33.12
Compressor discharge Temprerature[C]	113.8	88.8	90.7	91.9	93.1	94.4	95.7
RefMdot_In[kg/hr]	742	783	781	779.3	777.4	775.4	773.4
Ref. In h [kJ/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	495.2	481.3	482.8	483.7	484.6	485.6	486.6
Specific Refrigerating effect(kW)	26.74203039	28.21968975	28.14760881	28.08634	28.01786311	27.94578216	27.87370122
Specific Compressor Work (kW)	14.098	11.85375	12.14888889	12.31726944	12.48158889	12.66486667	12.84703333
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	15.92	13.74888889	14.08158889	14.44703333	13.45375	13.91726944	14.26486667
Coefficient of Performance(COP)	1.571212126	1.838549354	1.794946227	1.750251241	1.000370012	1.800947149	1.756945192

Case 7 : DBT 47°C and WBT 36°C

Table 4.1.7 Calculation and result at DBT 47°C and WBT 36°

		Evaporative cooling					
	Normal	Efficiency = 90%	Efficiency = 85%	Efficiency = 80%	Efficiency = 75%	Efficiency = 70%	Efficiency = 65%
Case 7	DBT = 47, WBT= 36	DBT = 37.1, WBT= 36	DBT = 37.65, WBT= 36	DBT = 38.2, WBT= 36	DBT = 38.75, WBT= 36	DBT = 39.3, WBT= 36	DBT = 39.85, WBT= 36
AirDBT_In[C]	47	37.1	37.65	38.2	38.75	39.3	39.85
AirWBT_In[C]	36	36	36	36	36	36	36
Air Out DBT [C]	56.38094452	46.63670772	47.17523468	47.70380597	48.26052126	48.79772441	49.33873483
Air Out WBT [C]	37.471746	37.5023248	37.50018775	37.49652758	37.49717642	37.49483937	37.49308579
Air Out RH [%]	31.1382049	56.31072208	54.46534491	52.71053826	50.94345101	49.28735767	47.67917103
Evaporation T [C]	2	2	2	2	2	2	2
Superheat T [K]	5	5	5	5	5	5	5
Evaporating Pressure [bar]	8.485	8.485	8.485	8.485	8.485	8.485	8.485
Compressor Discharge Pressure [bar]	41.72	33.46	33.95	34.37	34.8	35.23	35.67
Compressor discharge Temprerature[C]	118.4	96.6	97.9	99.1	100.2	101.3	102.5
RefMdot_In[kg/hr]	736.3	772	770.2	768	766.2	764.3	762.3
Ref. In h [J/kg] Evap	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543	297.0543
Ref. In h [kJ/kg] Compressor Suction	426.8	426.8	426.8	426.8	426.8	426.8	426.8
Ref. out h [kJ/kg] Compressor Discharge	505.1	487.4	488.4	489.3	490.2	491.1	492.1
Specific Refrigerating effect(kW)	26.5365997	27.82324456	27.75837171	27.67908267	27.61420982	27.54573292	27.47365198
Specific Compressor Work (kW)	16.014525	12.99533333	13.17897778	13.33333333	13.49363333	13.65124722	13.827275
Fan Work(kw)	1.1	1.1	1.1	1.1	1.1	1.1	1.1
Pump Work(kW)	0	0.5	0.5	0.5	0.5	0.5	0.5
Total Power Consumption	17.114525	14.77897778	15.09363333	15.427275	14.59533333	14.93333333	15.25124722
Coefficient of Performance(COP)	1.550530891	1.906311005	1.878233537	1.85351	1.829527007	1.806129854	1.780849306

4.2 EFFECT OF EVAPORATIVE COOLING ON MASS FLOW RATE OF THE SYSTEM

Due to the direct evaporative cooling in an air-cooled chiller, the mass flow rate of the system is significantly influenced. A larger difference between the dry-bulb and wet-bulb temperatures can cause a reduction in the chiller's mass flow rate compared to a smaller temperature difference. Observations show that when the cooling pad efficiency is 90%, the mass flow rate of the chiller increases by 0.29%, 2.89%, 3.23%, and 4.6% at air temperatures of 20°C, 35°C, 40°C, and 47°C respectively. Similarly, when the cooling pad efficiency is 65%, the mass flow rate of the chiller also increases, with percentages of 0.25%, 2.34%, 2.42%, and 3.41% for the respective air temperatures. These findings suggest that a lower cooling pad efficiency leads to a decrease in the mass flow rate of the chiller.

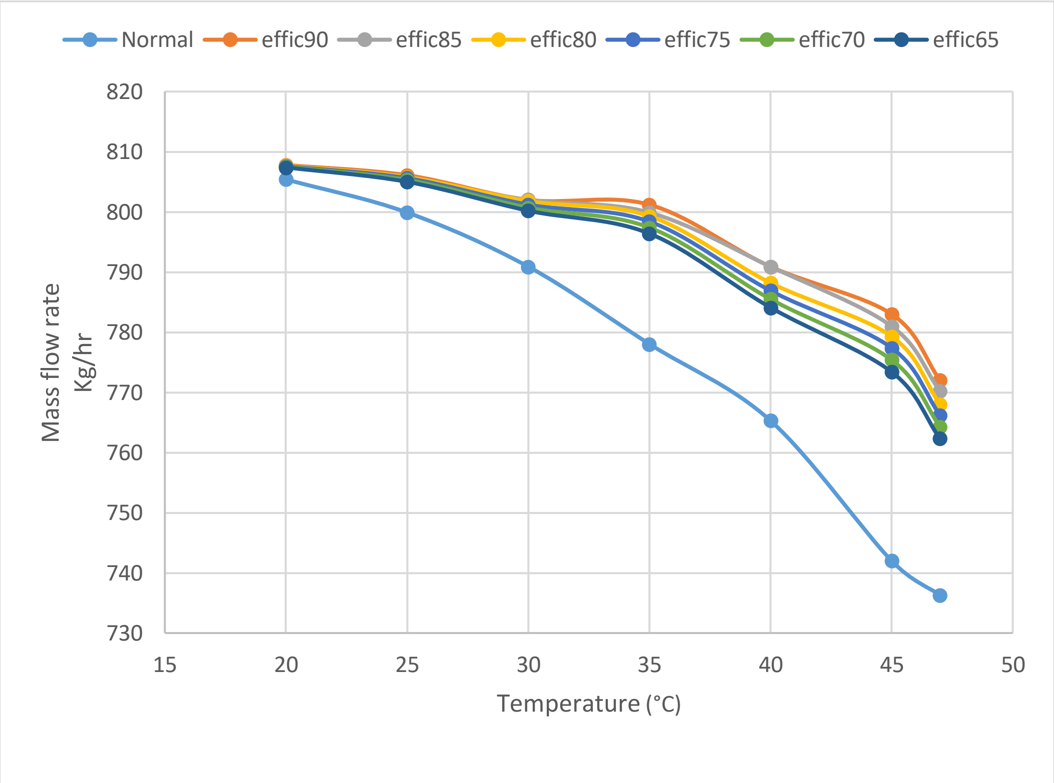

Figure 4.2.1 Effect of evaporative cooling on mass flow rate of the system

4.3 EFFECT OF EVAPORATIVE COOLING ON COMPRESSOR OUTLET TEMPERATURE OF THE CHILLER.

Due to direct evaporative cooling in an air-cooled chiller, there is a significant decrease in the compressor outlet temperature. A larger difference between the dry-bulb and wet-bulb temperatures can result in a greater reduction in the compressor outlet temperature compared to a smaller temperature difference. Observations indicate that with a cooling pad efficiency of 90%, the compressor outlet temperature decreases by 14.3% and 22.56% at air temperatures of 20°C and 47°C, respectively. Similarly, when the cooling pad efficiency is 65%, the compressor outlet temperature also decreases, with percentages of 9.12% and 15.52% for the respective air temperatures. These findings suggest that a lower cooling pad efficiency leads to an increase in the compressor outlet temperature of the chiller.

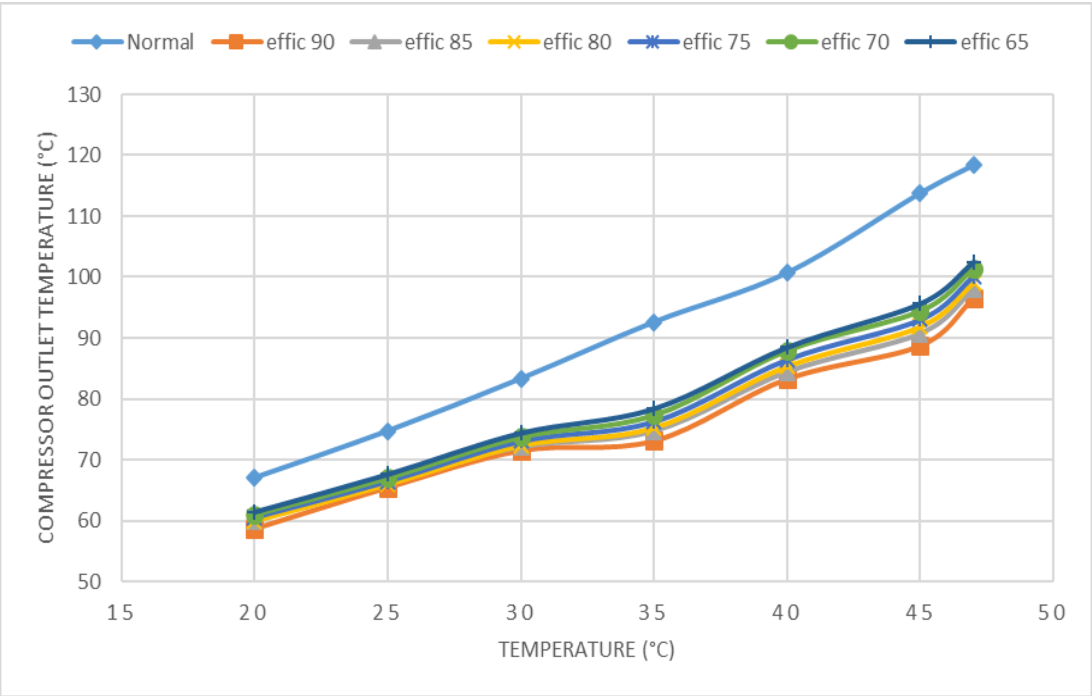

Figure 4.3.1 Effect of evaporative cooling on Compressor outlet temperature of the chiller.

4.4 EFFECT OF EVAPORATIVE COOLING ON REFRIGERATING CAPACITY OF THE SYSTEM

Figure 4.4.1 demonstrates that direct evaporative cooling has a significant impact on the specific refrigerating capacity. A larger difference between the dry-bulb and wet-bulb temperatures can result in a lower specific refrigerating effect compared to a smaller difference between these temperatures. It has been observed that with a cooling pad efficiency of 90%, there is an improvement of 0.29%, 2.89%, 3.24%, 5.24% and 4.62% in the specific refrigerating effect at air temperatures of 20°C, 35°C, 40°C, 45°C and 47°C, respectively. When the cooling pad efficiency is 70%, there is also an enhancement, albeit slightly lower, with percentages of 0.26%, 2.42%, 2.62%, 4.26% and 3.66% for the respective air temperatures. These findings indicate that a lower cooling pad efficiency leads to a decrease in the specific refrigerating effect.

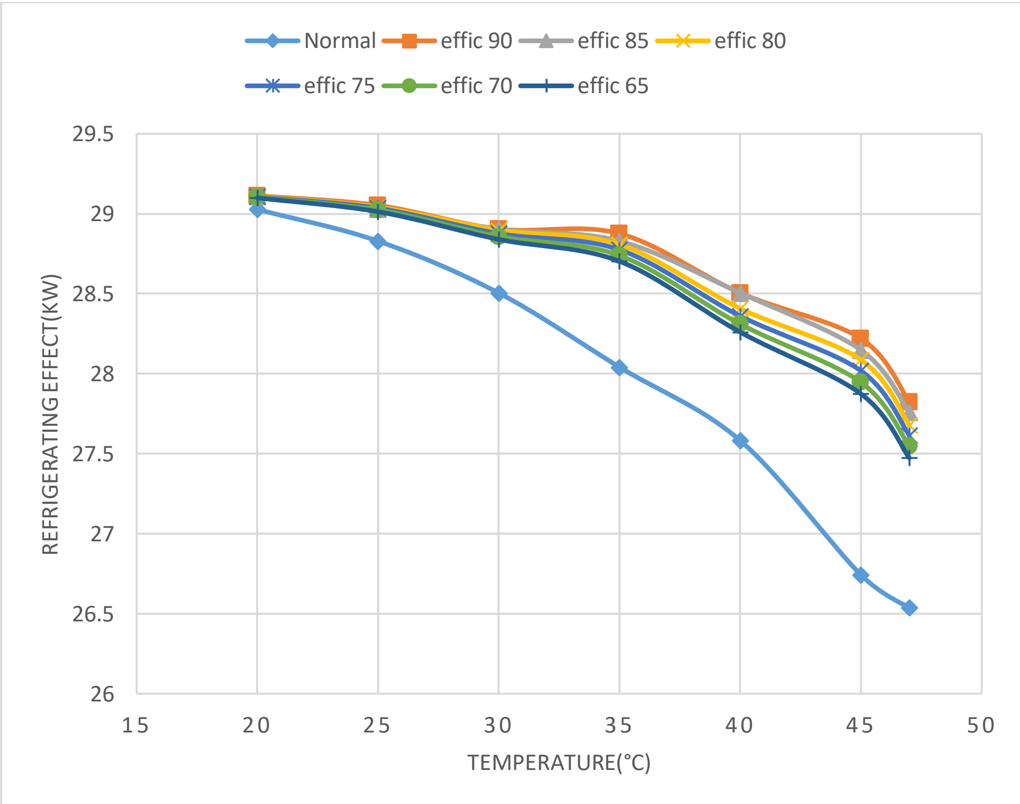

Figure 4.4.1 Effect of evaporative cooling on refrigerating capacity of the system

4.5 EFFECT OF EVAPORATIVE COOLING ON COMPRESSOR WORK

The data from Figure 4.5 highlights that direct evaporative cooling significantly impacts compressor work. When there is a larger difference between the dry-bulb and wet-bulb temperatures, the compressor work tends to increase, whereas a smaller difference leads to a relatively smaller reduction in compressor work. Simulation results demonstrate that at air temperatures of 20°C, 35°C, 40°C, 45°C and 47°C a cooling pad efficiency of 90% reduces the compressor work by 14.11%, 28.65%, 15.91%, 18.9% and 23.33% respectively. In contrast, when the cooling pad efficiency decreases to 65%, there is a slightly less decrement in compressor work, with respective percentage changes of 8.52%, 19.5%, 7.75%, 9.73% and 15.81% for the corresponding air temperatures. These findings underscore the inverse relationship between lower cooling pad efficiency and the decrease in compressor work within the system.

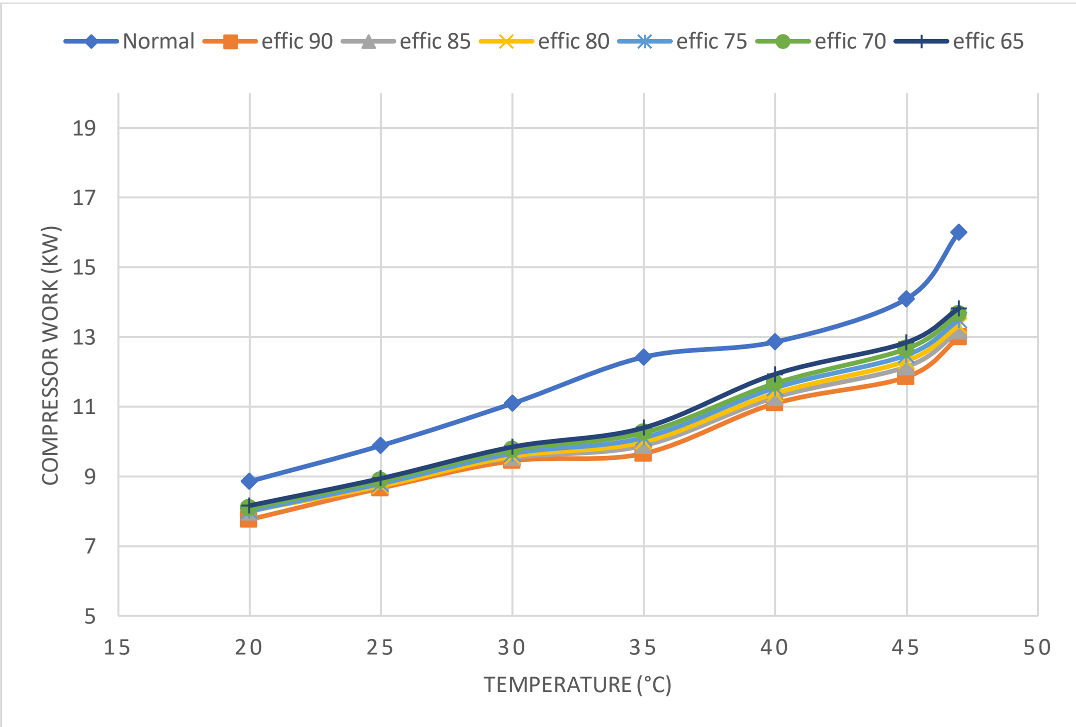

Figure 4.5.1 Effect of evaporative cooling on compressor work

4.6 EFFECT OF EVAPORATIVE COOLING ON TOTAL POWER CONSUMPTION OF THE SYSTEM

Due to direct evaporative cooling in an air-cooled chiller, there is a significant impact on the total power consumption of the chiller. A larger difference between the dry-bulb and wet-bulb temperatures can result in an increased total power consumption of the chiller compared to a smaller difference between these temperatures. It has been observed that the total power consumption of the chiller decreases by 6.36%, 20.12%, 12.13%, 18.33% and 17.26% at air temperatures of 20°C, 35°C, 40°C, 45°C and 47°C, respectively, when the cooling pad efficiency is 90%. When the cooling pad efficiency is 65%, there is also a decrease in power consumption, with percentages of 2.0%, 12.7%, 5.1%, 10.1% and 10.9%for the respective air temperatures. These findings indicate that a lower cooling pad efficiency leads to an increase in the total power consumption of the chiller.

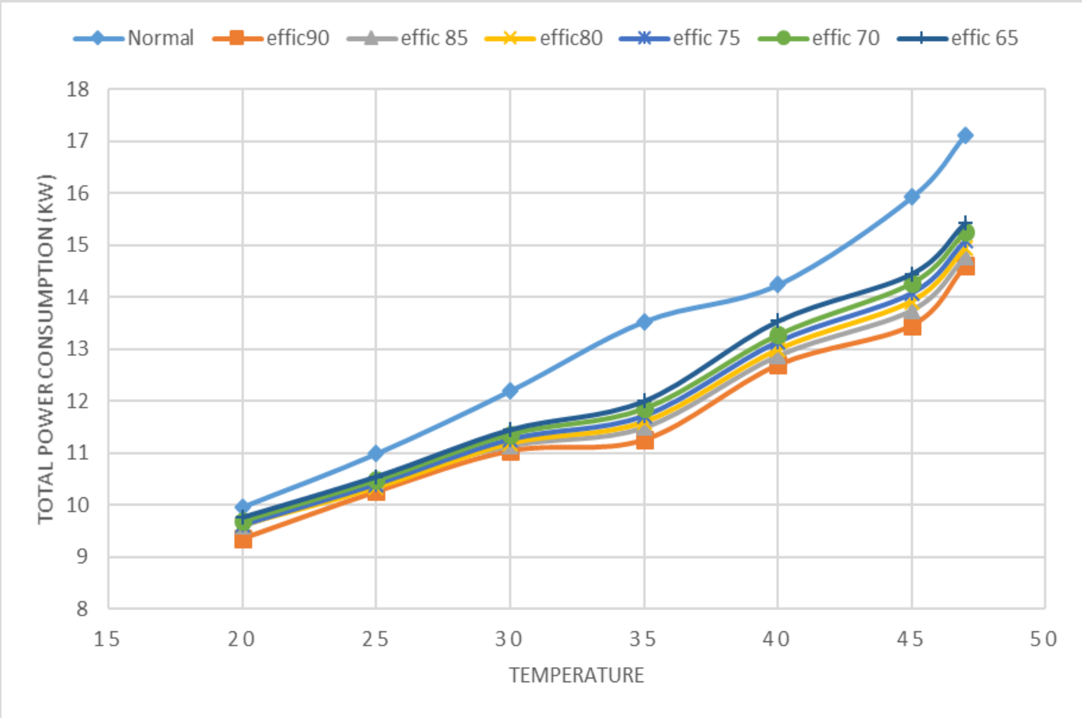

Figure 4.6.1 Effect of evaporative cooling on Total Power consumption of the system

4.7 EFFECT OF EVAPORATIVE COOLING ON COEFFICIENT OF PERFORMANCE OF SYSTEM

According to Figure 4.2, direct evaporative cooling has a significant effect on the system's coefficient of performance. A larger difference between the dry-bulb and wet-bulb temperatures can cause a decrease in the coefficient of performance, while a smaller difference yields better results. The simulation shows that at air temperatures of 20°C, 35°C, 40°C, 45°C and 47°C, a cooling pad efficiency of 90% leads to improvements of 6.25%, 19.17%, 12.02%, 16.11% and 18.66% in the coefficient of performance, respectively. When the cooling pad efficiency drops to 65%, the improvements are slightly lower, with percentages of 2.2%, 13.35%, 6.32%, 8.80% and 12.93% for the corresponding air temperatures. These findings highlight the relationship between lower cooling pad efficiency and a decrease in the system's coefficient of performance

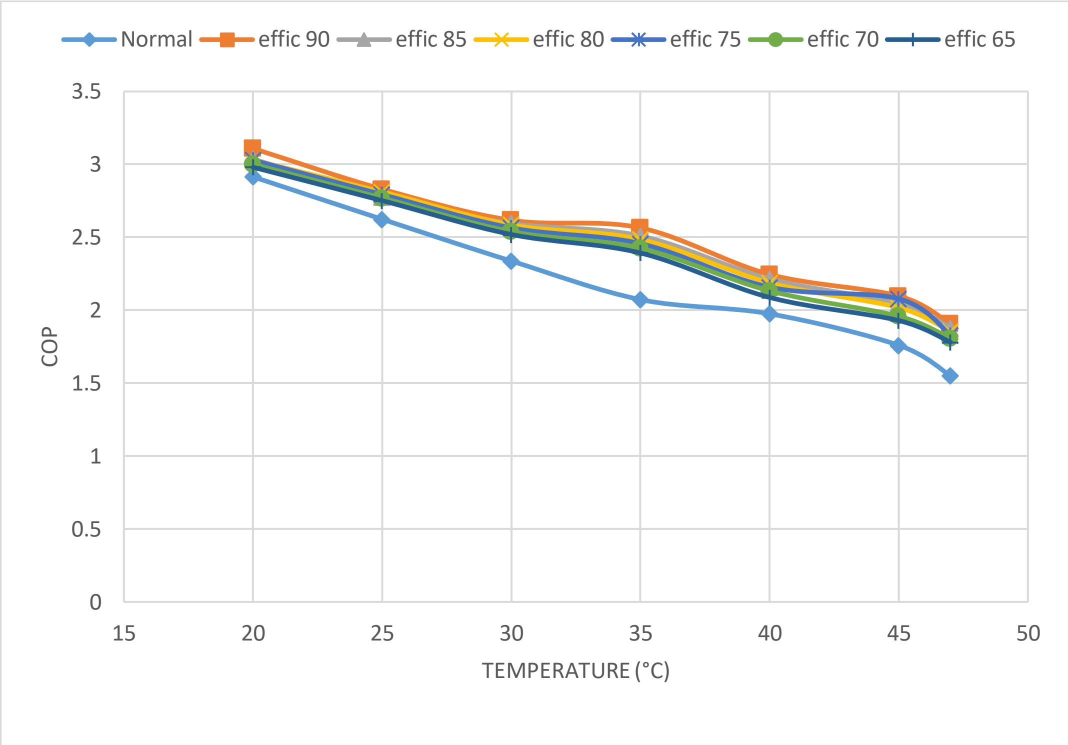

Figure 4.7.1 COP of an air-cooled chiller for different air temperatures

4.8 EFFECT OF EVAPORATIVE COOLING ON HEAT LOAD OF CONDENSER OF A CHILLER

Due to direct evaporative cooling in an air-cooled chiller, the heat load of the condenser is significantly increased. A larger difference between the dry-bulb and wet-bulb temperatures can cause a reduction in the chiller's heat load compared to a smaller temperature difference. Observations show that when the cooling pad efficiency is 90%, the condenser's heat load increases by 1.8% and 4.3% at air temperatures of 20°C and 47°C, respectively. Similarly, when the cooling pad efficiency is 65%, the heat load of the condenser also increases, with percentages of 1.51% and 3.07% for the respective air temperatures. These findings suggest that a lower cooling pad efficiency leads to a decrease in the heat load of the chiller's condenser.

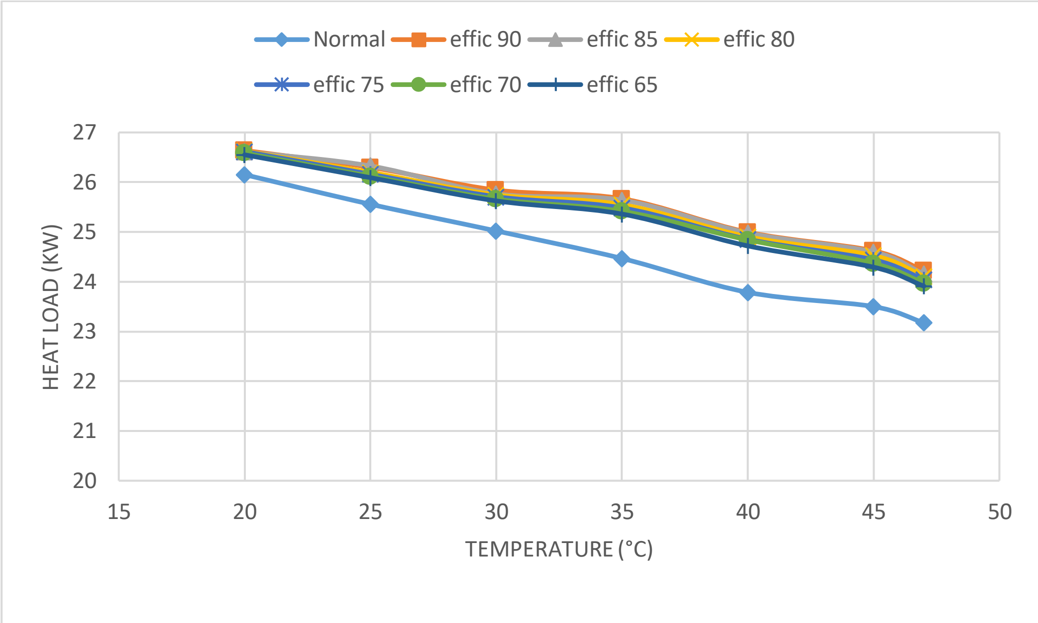

Figure 4.8.1 Effect of evaporative cooling on Heat load of condenser of a chiller

CHAPTER 5

CONCLUSION

Optimizing the utilization of energy resources requires enhancing the efficiency of air cooled chiller. Evaporative cooling pad attached with air cooled-chiller is a highly effective, reliable and cost-effective method to increase the coefficient of performance and save energy in any air-cooled refrigeration system.

Simulation and analysis of 20 TR Air-cooled chiller attached with evaporative cooling pad has been done and based on that the following conclusions are arrived:

1. Direct evaporative cooling has a significant impact on the specific refrigerating capacity. With a cooling pad efficiency of 90%, there is an improvement of 0.29% and 4.62% in the specific refrigerating effect at air temperatures of 20°C and 47°C respectively. When the cooling pad efficiency is 65%, there is also an improvement, but slightly lower, with percentages of 0.26% and 3.66% for the respective air temperatures. These findings indicate that a lower cooling pad efficiency leads to a decrease in the specific refrigerating effect.
2. Simulation results demonstrate that at air temperatures of 20°C and 47°C, cooling pad efficiency of 90% reduces the compressor work by 14.11% and 23.33% respectively. At 65% there is a slightly less decrement in compressor work of 8.52% and 15.21% for the respective air temperature.
3. When the cooling pad efficiency is 90%, the mass flow rate of the chiller increases by 0.29% and 4.6% at air temperatures of 20°C and 47°C respectively. Similarly, when the cooling pad efficiency is 65%, the mass flow rate of the chiller also increases but with slightly lower percentages of 0.25 and 3.41% for the respective air temperatures.
4. With a cooling pad efficiency of 90%, the compressor outlet temperature decreases by 14.3% and 22.56% at air temperatures of 20°C and 47°C, respectively. Similarly, when the cooling pad efficiency is 65%, the compressor outlet temperature also decreases, with percentages of 9.12% and 15.52% for the respective air temperatures.
5. The total energy consumed by the chiller decreases by 6.36% and 17.26% at air temperatures of 20°C and 47°C, respectively, when the cooling pad efficiency is

90%. When the cooling pad efficiency is 65%, there is also a decrease in power consumption, with percentages of 2.0%, and 10.9% for the respective air temperatures. The simulation shows that at air temperatures of 20°C and 47°C, a cooling pad efficiency of 90% leads to improvements of 6.25% and 18.66% in the coefficient of performance respectively. When the cooling pad efficiency drops to 65%, the improvements are slightly lower, with percentages of 2.2% and 12.93% for the corresponding air temperatures.

6. When the cooling pad efficiency is 90%, the condenser's heat load increases by 1.8% and 4.3% at air temperatures of 20°C and 47°C, respectively. Similarly, when the cooling pad efficiency is 65%, the heat load of the condenser also increases, with percentages of 1.51% and 3.07% for the respective air temperatures.

CHAPTER 6

SCOPE FOR FUTURE WORK

- Studies can be done on the different cooling pad material used for evaporative cooling in air cooled chiller.
- Work can be done on the cooling pad thickness used for evaporative cooling in air cooled chiller
- Studies can be done with different refrigerant in air cooled chiller

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19. Picture taken from the industry

20. Hexim software solver version 4.020194.121, Industry software

21. Coolesector software by Danfoss version 5.2.6,Selection software

22. Picture taken from Internet sources